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National Bureau of Standards
National Engineering Laboratory
Center for Building Technology
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Prepared for:
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EXPERIMENTAL AND ANALYTICAL INVESTIGATION OF A RESIDENTIAL HOT WATER BOILER WITH FINNED COPPER TUBE HEAT EXCHANGERS

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U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, *Secretary*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

ABSTRACT

In response to a request by a manufacturer of a nontypical boiler, the Department of Energy requested the National Bureau of Standards to perform laboratory measurements under controlled conditions of the effect on seasonal performance of several features (finned copper tube heat exchanger, water circulating pump delay, and gas valve modulation) of this boiler that might cause it to be unfairly treated by the existing test procedure. As a result of this study, recommended changes to the existing test procedure to allow rating tests with water circulating pump delay are presented. A recommended change to the assigned cyclic jacket loss factor and a simplified procedure for experimentally determining this factor are also presented. No change to the current test procedure treatment of gas valve modulation or flue gas mass flow as a function of temperature are recommended.

Key Words: Annual efficiency; annual operating costs; boilers; fossil fuel heating systems; jacket loss; modulating control gas fueled; part-load performance; rating procedures; seasonal efficiency

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

A	jacket surface area, ft^2 (m^2)
C_J	jacket loss factor
$C_{t,OFF}$	cool-down temperature profile correction factor for the effect of cycling
$C_{t,ON}$	heat-up temperature profile correction factor for the effect of cycling
$EFFY_A$	annual fuel utilization efficiency
h	jacket heat transfer coefficient Btu/ft^2h°F (W/m^2 °C)
h_c	heat transfer coefficient of convection, Btu/ft^2h°F (W/m^2 °C)
h_{rl}	black body heat transfer coefficient of radiation, Btu/ft^2h°F (W/m^2 °C)
$\dot{m}_F$	non-steady-state flue gas flow rate, lb/min (kg/min)
$\dot{m}_{F,SS}$	flue gas flow rate at steady-state operation, lb/min (kg/min)
Q_{IN}	fuel energy input rate at steady-state operation (including any pilot light input), Btu/h (W)
Q_{OUT}	fuel energy output rate at steady-state operation, the product of $Q_{IN} \times \eta_{ss}$, Btu/h (W)
t	time, min
t^+	delay time between burner shut-off and blower or circulating pump shut-off, min
t^-	delay time between burner start-up and blower or circulating pump start-up, min
$t_{B,OFF}$	burner off-time per cycle, min
$t_{B,ON}$	burner on-time per cycle, min
$t^*_{B,ON}$	burner on-time as a decimal fraction of cycle total cycle length
T_F	flue gas temperature while the system is in cyclic operation, °F (°C)
$T_{F,SS}$	flue gas temperature at steady-state, °F (°C)
T_j	jacket temperature while the system is in cyclic operation, °F (°C)

$T_{j,ss}$	jacket temperature at steady-state, °F (°C)
T_{RA}	laboratory room temperature, °F (°C)
X_{CO_2}	concentration by volume of CO_2 present in dry flue gas, %
y	ratio of blower or pump on-time to burner on-time
ε	emissivity
η_{SS}	steady-state efficiency, %
η_u	part-load fuel utilization efficiency, %
θ_j	temperature difference between steady-state and on-cycle jacket temperature, °F (°C)
$\theta_{J,O,X}$	temperature difference defined by equation 2 in Appendix A, °F (°C)
τ_{ON}	heat-up time constant, min
τ_{OFF}	cool-down time constant, min
ψ_J	temperature difference between steady-state and off-cycle jacket temperature, °F (°C)
$\psi_{J,O,X}$	temperature difference defined by equation 1 in Appendix A, °F (°C)

1. INTRODUCTION

The residential boiler under study incorporated several nontypical features. The most outstanding of these features were the use of a finned copper tube heat exchanger instead of the traditional non-finned cast iron shell, and the incorporation of a modulating gas valve and a temperature sensing time delay which controlled shut-off of the water-circulating pump.

Field data taken by the manufacturer indicated a seasonal efficiency substantially greater for boilers of this type than that calculated by the DoE/NBS Furnace Boiler Test Procedure. (Reference [1]*; referred to as the Test Procedure in the remainder of this report.)

In response to the manufacturer's request, the Department of Energy requested the National Bureau of Standards to perform laboratory measurements under controlled conditions of the effects of various innovative features in these boilers and to determine what changes, if any, should be made in the Test Procedure [1] to treat these units fairly. This report contains results of tests performed at NBS on a boiler supplied by the manufacturer as typical of their production units. Tests and calculations in this report were directed to the five following areas of concern:

a) Validity of DoE/NBS Test Procedure [1] Flue Gas Mass Flow Equation - The purpose of these tests was to determine if there was something different about the internal geometry of this boiler design that might alter the combustion air mass flow rate during heat-up and cool-down from that which is assumed in the Test Procedure [1].

*References listed on page 46.

b) Effect of Gas Valve Modulation on Seasonal Efficiency - The Test Procedure [1] assumes that there is no advantage in seasonal efficiency to be gained by modulating gas flow as opposed to cycling between a burner-off and full-fired condition.* Tests were performed at several reduced firing rates and compared to the efficiency resulting from this assumption.

c) Value of C_j in Test Procedure [1] - C_j is a multiplier on the measured steady-state jacket loss for calculating the jacket loss for units installed outdoors during cyclic operation in a 42°F ambient. This operating condition is assumed to result in an efficiency representative of the mean seasonal value. Jacket temperature measurements were made which would allow calculation of the cyclic portion of C_j for several assumed operating conditions.

d) Value of Delayed Pump Operation - Cool-down and heat-up tests were performed with the water pump running continuously to determine the maximum effect likely on seasonal efficiency due to water pump delay as opposed to the simultaneous cycling of the water pump with the gas burner as assumed by the Test Procedure [1]. It should be emphasized that continuous pump operation is not a recommended control strategy for actual field applications but was employed in this laboratory study as a limiting case.

* Work done in another study on development of a test procedure for modulating fuel controls has been completed for vented household heaters and furnaces [6]. Procedures developed in that program can be applied where applicable as the need arises.

e) Computer Simulation for Pump Delay - An analytical study was also performed to see if the current DoE/NBS Furnace/Boiler Test Procedure [1] could evaluate the performance of a residential hot water boiler system with a finned tube heat exchanger, and with a control device for the pump delay. Based upon the available information, computer simulations were made using an existing boiler simulation program. The NBS DEPAB program [5] was used for this analysis after some modifications were made. The analysis focused on the control strategy related to the pump operation.

Sensitivity analysis of the pump delay parameters and cost analysis of the boiler system were made. In the analysis, the annual costs in operation of the boiler were obtained using the annual operation cost analysis routine in the Test Procedure [1].

2. DESCRIPTION OF TEST BOILER

The name plate data of the tested boiler were:

Fuel: Natural Gas

Input Rate: 125,000 Btu/h (37 kW)

Output Rate: 100,000 Btu/h (29 kW)

Minimum Input Rate: 43,750 Btu/h (13 kW)

Maximum Water Pressure: 160 psi (1100 kpa)

Heating Surface Area: 11.4 ft^2 (1.06 m^2)

The boiler had five burners running the length of the combustion chamber. The boiler tubes were arranged in a single horizontal row, approximately 20 inches (51 cm) above the burners. The combustion chamber connecting the burners and the boiler tubes was insulated with a 2 inch (5 cm) thick cast refractory liner and was surrounded by fiber insulation approximately 9.5 inches (24 cm) wide and 13 inches (33 cm) long. The boiler bottom jacket surface was approximately 7 inches (18 cm) below and the top jacket surface 27 1/2 inches (24 cm) above the top surface of the burners. Top and bottom jacket surfaces were both uninsulated.

There were 8 transverse boiler tubes with a finned length of approximately 9 inches (23 cm) and possessing approximately 67 integral fins each of 5/64 inch (20 mm) thickness. The diameter of these circular fins was 1.5 inches (3.8 cm) and the outside diameter of the boiler tubes was 13/16 inches (2.1 cm). The 8 transverse boiler tubes were divided into two groups of four; half crossing from the water inlet of the boiler to a cast iron manifold and the other half recrossing from the manifold to the water outlet. The inlet and outlet (return and supply) fittings were combined in a single iron casting. Vee shaped flow guides were fitted on top of the finned boiler tubes.

The boiler stack was 5 inches (12.7 cm) in diameter and 13 1/2 inches (34.3 cm) in length from the boiler top to the relief opening of its integral draft hood.

In normal operation, gas modulation is provided by a 1/2 inch (1.3 cm), 130°F (54 °C) to 180°F (88°C), modulating valve with its sensing bulb in the hot water supply line. Modulation temperature range is adjustable using a dial with settings from 1 to 9. At the lowest setting (1) the valve would be at its maximum firing rate position for outlet water temperatures of 105°F (41°C) and below; at its minimum firing rate at a 120°F (49°C) outlet water temperature; and off at a outlet water temperature of 130°F (54°C). Similarly at its highest setting (9) the valve would be at its maximum firing rate position for outlet water temperature of 165°F (74°C) and below; at its minimum firing rate at a 180°F (82°C) outlet water temperature; and off at an outlet water temperature of 190°F (88°C). This modulating valve was disabled for all tests described in this report. For tests in which a reduced firing rate was desired, the gas supply to the boiler was throttled using a manually operated valve.

The water-circulating pump control of the boiler was designed to provide pump start-up at gas cut-on and delay on pump shut-down after gas cut-off. The pump cut-off delay switch senses outlet water temperature and had a factory-set differential of 15°F (8.3°C).

The manufacturer's literature recommended that the control be set at 20°F (11°C) to 30°F (17°C) below the outlet water temperature setting of the modulating valve, but not less than 120°F (49°C). This automatic pump delay feature was disabled for all tests described in this report. The factory "on" setting for the delay switch was at 120°F (49°C). Thus in operation, at the factory settings, the pump starts simultaneously with the burner by means of a relay in parallel with the time delay switch. If the outlet water temperature exceeds 120°F (49°C), the delay switch closes. When the burner is shut off this relay switch opens immediately, but the delay switch keeps the pump on until the outlet water falls to 105°F (41°C). For tests in which pump delay was desired, pump operation was controlled by a manually operated switch.

Ignition for this boiler was by a continuously burning pilot light.

3. DESCRIPTION OF TEST APPARATUS

Testing was conducted using the boiler test apparatus schematically shown in Figure 1. The test unit was mounted on four 7 3/4 inch (20 cm) legs allowing room air circulation under the boiler. An insulated 125 gallon (0.47 m^3) water tank, circulating pump, hot water dump, auxiliary boiler, and city water make-up comprised the major support equipment. In order to achieve repeatable water flow rates, 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 (25 kW) input unit, was used to raise the temperature of the water in the storage tank between tests. Pump and boiler operation was controlled manually.

A turbine meter in the inlet 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 inlet water temperatures from 150 to 180°F (66 to 82°C) Occasional calibration checks were made throughout the period of time that tests were being conducted.

A 32-junction thermopile made from 30 AWG type T (copper/constantan) thermocouple wire was used in stainless steel wells inserted in the inlet and outlet 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)

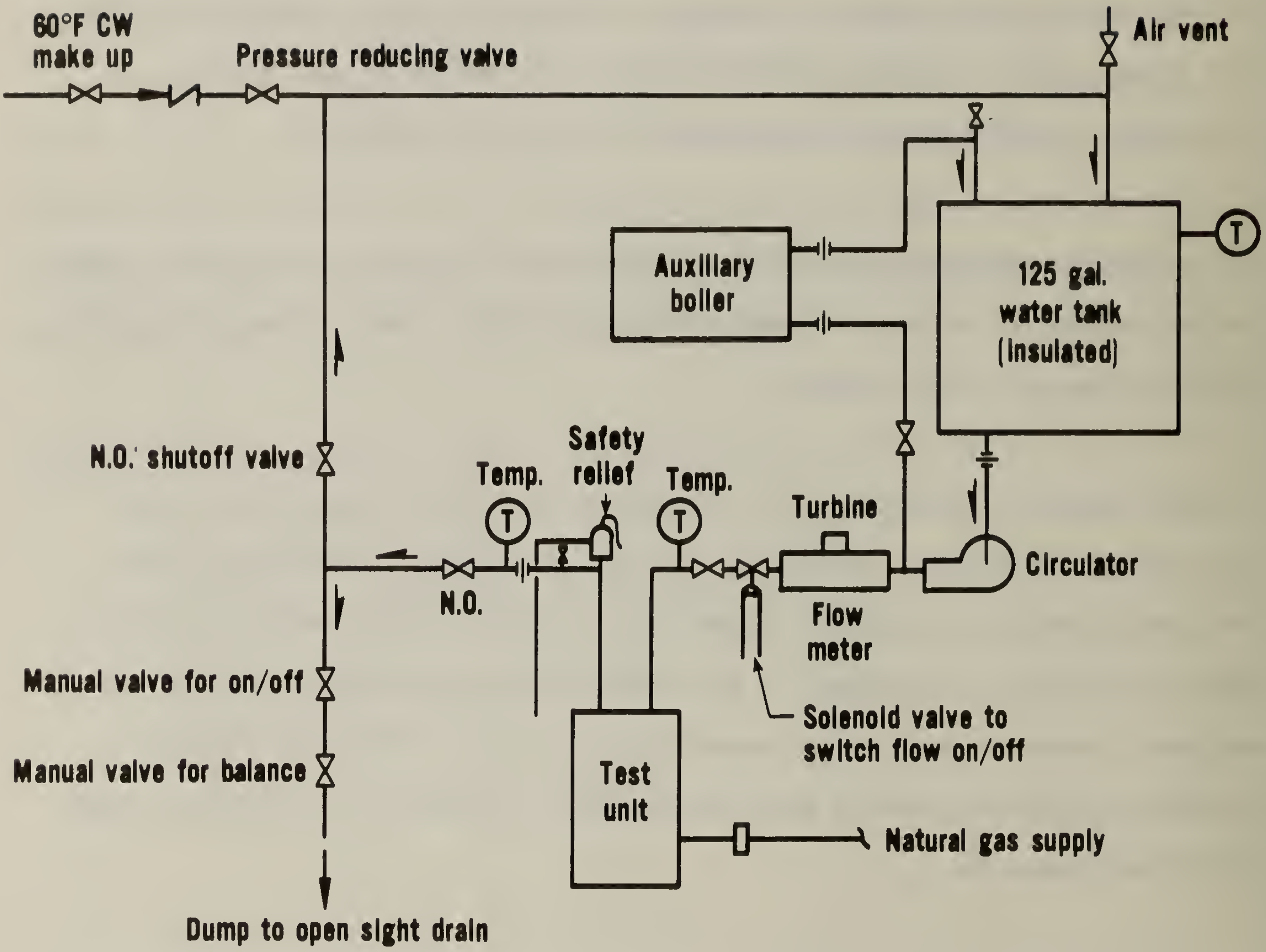

Figure 1. Test apparatus schematic

immersion length, as recommended in ASHRAE Std. 41.1-74 [4]. The flue gas temperature was measured by a 9-in-1 averaging grid of chromel-alumel, type K, thermocouples 11 inches (28 cm) from the boiler outlet. Additional type T thermocouples were provided to measure the test room ambient, dry bulb and wet bulb temperatures, the supply gas temperature, and the supply and return air temperatures.

Surface temperatures of the jacket were measured using 148 thermocouples bonded to the surface with epoxy and painted to match the unit. Fifteen thermocouples were on the top, fifteen on the bottom, and the remaining 118 on the sides. Thermocouples on similarly facing surfaces were connected in parallel to allow faster monitoring during transient tests.

The temperature data were recorded by an automatic data acquisition system having a programmable time interval capability between data scans.

Flue gas samples were taken at the center of the flue, 11 inches (28 cm) from the boiler outlet at the flue temperature measurement plane. The concentrations of carbon dioxide and methane tracer gas in the flue gas were measured using instruments of the infrared absorption type. The full scale reading of the carbon dioxide detector was 15 percent and, of the methane detector, 500 ppm. A dessicant column was used to remove moisture from the flue gas before it entered the detectors.

The methane tracer gas flow rate was measured with a 100 ml bubble meter. The tracer gas was injected through a manifold beneath the burner tubes. 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 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.

The flue from the furnace top to the bottom of the draft hood was wrapped in 2 inch (5 cm) thick R-7 glass fiber insulation.

A five foot (152 cm) stack was installed on top of the draft hood. Tests were performed both with the draft hood open and sealed. A vane anemometer was mounted 1 inch (2.5 cm) above the top of the stack for stack flow rate measurement during some cool-down tests.

The anemometer had been calibrated under a geometrically similar flow condition. Air velocity at the top of a calibration circular duct of the same diameter and height as the stack was measured at room temperature. A high capacity dry gas meter was installed between the air supply line and the calibration duct. The anemometer used here was 4 inches (10.2 cm) in diameter with an eight vane rotor. Its effective cross sectional area was 9.42 in^2 (60.8 cm^2).

4. LABORATORY TEST PROCEDURES

4.1 MASS FLOW

A steady-state test at full load following the guidelines of the DoE/NBS Furnace/Boiler Test Procedure [1] was first performed on the test boiler (Figure 2(a)). The draft hood was then sealed as seen in Figure 2(b) and the five-foot (152 cm) stack adjusted to produce the same flue CO_2 concentration as that measured during the first steady-state test. This second steady-state test, with the draft hood blocked, was used as the initial condition prior to cool-down for mass flow measurement.

It should be noted that the test condition of a sealed draft diverter and a five-foot stack with sufficient restriction to reproduce the unsealed draft diverter CO_2 concentration is the DoE/NBS Test Procedure [1] preliminary test condition for units which incorporate draft diverters instead of draft hoods. The Test Procedure [1] flow equations are designed to fit this sealed draft diverter test condition. For comparability in flow measurement, draft hood units should also have their relief openings sealed and stacks restricted.

A cool-down test was performed during which the stack mass flow was measured by two independent methods; vane anemometer mounted above the stack and methane tracer gas. After the boiler was allowed to return to steady-state idle conditions (the following day), an additional data point was taken representing the flow caused by pilot light operation. The stack flow data were normalized by division by steady-state values obtained during the pre-cool-down steady-state period. Vane anemometer steady-state data were used for normalizing the vane anemometer data. A mass flow rate value calculated from the stack CO_2

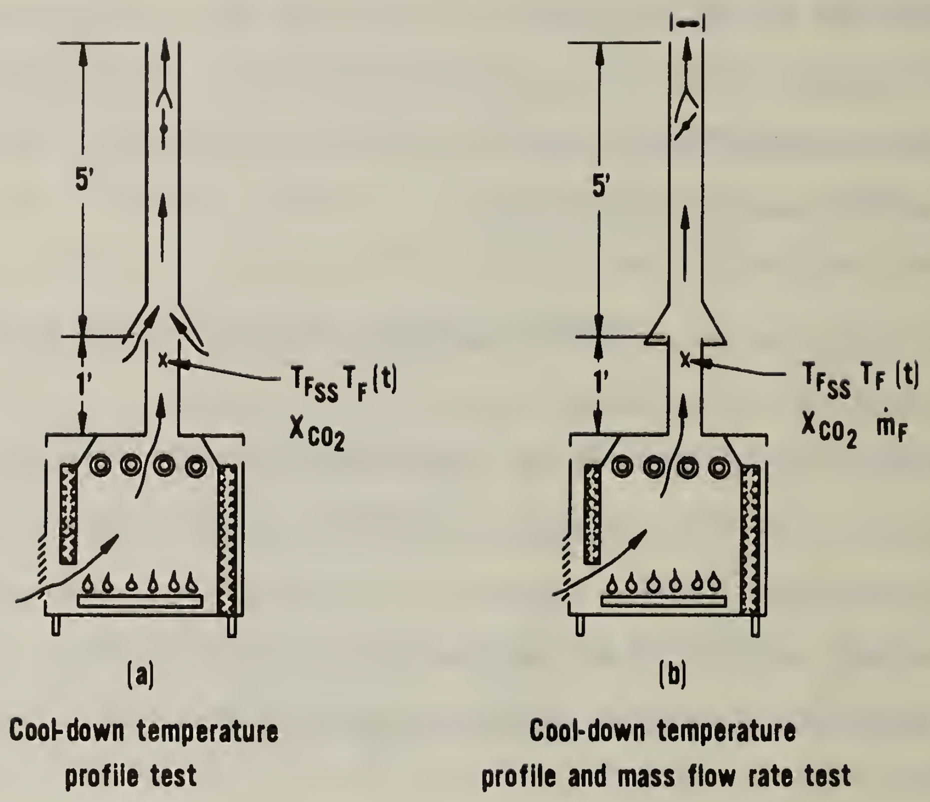

Figure 2. Test configuration

concentration and the fuel gas flow rate was used for the tracer gas cool-down data normalization.

The draft hood was then opened, and steady-state and cool-down tests were performed in accordance with the Test Procedure [1] to provide a cool-down flue temperature curve for comparison to that obtained during the sealed draft hood cool-down test (as in Figure 2(a)).

4.2 MODULATION

Steady-state cool-down and heat-up tests were performed according to the Test Procedure [1] at full load, and with the gas input rate reduced to 2/3 and to 1/3 of its rated value.

4.3 JACKET LOSS

The jacket temperatures were measured during all steady-state, cool-down, and heat-up tests.

A full-fired, cyclic test was also performed. For this test the boiler water was continuously circulated during both burner on-and off-cycles. The return water temperature was kept at a nominal 165°F (73.9°C) by a thermostatically controlled, auxiliary boiler in the test loop. These cycle periods were 9.68 minutes on and 33.26 minutes off as specified by the Test Procedure [1] for boilers.

Comparison of the jacket loss during the full fired, cyclic test and the full fired steady-state test would allow direct calculation of C_j (exclusive of ambient temperature effects). The heat-up and cool-down from steady-state jacket temperature data were used to see if a time constant method could be used to calculate the cyclic jacket temperatures (eliminating the need for cyclic testing).

4.4 PUMP DELAY

Full load steady-state, cool-down, and heat-up tests were performed with a 165°F (73.9°C) steady-state return water temperature with the water circulating pump cycling with the burner as specified by the Test Procedure [1], and were then repeated with the pump running continuously.

The above comparison was then repeated with a 105°F (40.6°C) steady-state return water temperature and, during steady-state operation, a proportionately reduced supply water temperature (i.e., constant water flow rate).

During the off-portion of the cyclic water flow tests the make-up water to the test loop was reduced to a minimum and the system insulation relied on to maintain the loop water temperature. Continuous water flow tests were conducted both by operating a thermostatically controlled auxiliary boiler in the test loop to maintain the return water at its steady-state value, and by allowing the inlet water temperature to decay at a rate given by leaving the system make-up water at the value used during steady-state operation and the auxiliary boiler off.

Continuous pump tests were used instead of actual pump delay tests in order to show the maximum effect possible due to pump delay and to eliminate the variable of length of delay.

5. LABORATORY TEST RESULTS AND DISCUSSION

a) Mass Flow - The measured stack mass flow with the draft diverter sealed divided by its full load value is shown as a function of normalized temperature in Figure 3. Points at 0.0, 0.0 are included by definition rather than measurement. The low temperature points are those resulting from pilot light operation. The conformance to the predictive equation of the DoE/NBS Furnace/Boiler Test Procedure [1] shown as a solid line is quite good, deviating by 8 percent at its worst.

In Figure 4 the flue temperature, as a function of time, measured during cool-down from steady-state with the draft hood open (Figure 2a) is shown. Also shown is the flue temperature measured during cool-down from steady-state with the same draft hood sealed (Figure 2b) and the flow restrictor adjusted to provide the same steady-state CO_2 concentration as in the open hood test. Both this agreement and that shown in Figure 3 are closer than would be expected for a unit employing a draft diverter instead of the draft hood with which this unit was equipped (see [2], [6]).

Only flue temperature profiles are normally measured during the cool-down test in the Test Procedure [1]. For ease of testing, draft hood units are allowed to be tested with the draft hood open on the assumption that in this configuration the cool-down temperature curve will not be significantly different from that which would have been measured with a sealed draft hood and the stack restricted to provide the same CO_2 value as the open hood condition. As can be seen from Figures 3 and 4 this assumption is valid for this unit. Therefore, a boiler of this internal geometry will receive a fair (consistent

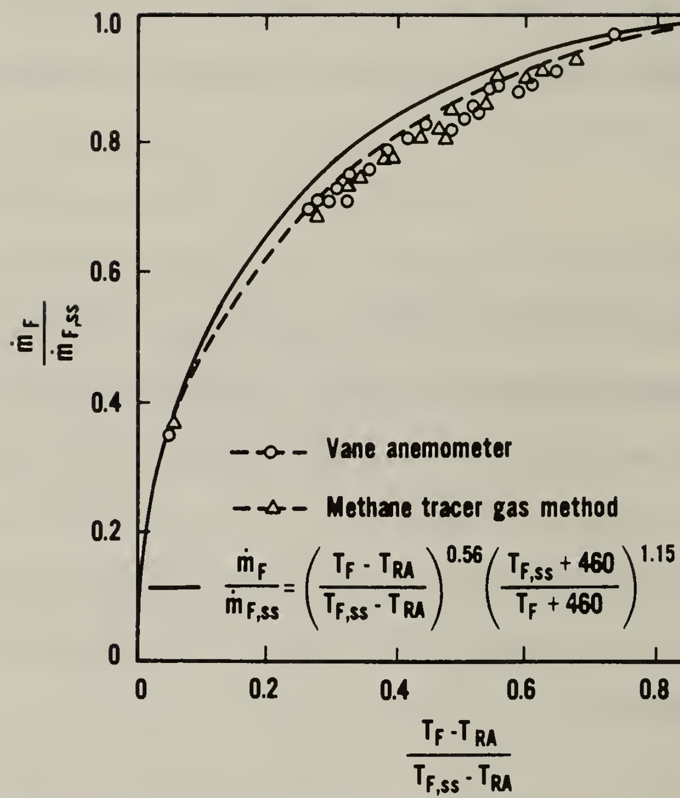

Figure 3. Mass flow rate of flue gas through the stack

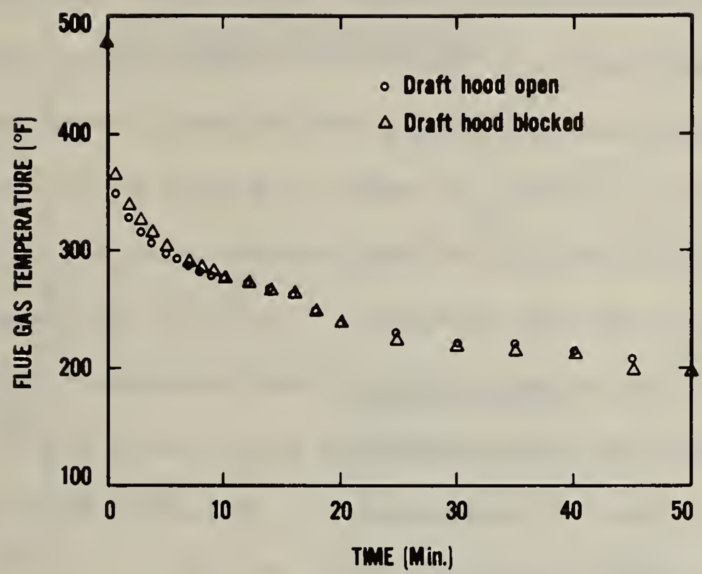

Figure 4. Flue gas temperature at cool-down test

with other designs) rating if tested in accordance with the current Test Procedure [1].

It should be noted that comparability between units in the Test Procedure [1] is based upon measurements with a 5 foot (152 cm) stack without draft relief. If mass flow measurements were made with a substantially shorter stack (as would be the case for the unit with the draft hood open) or in the flue and compared to the coincident temperatures, the results would not be comparable to units tested according to the Test Procedures [1], since the measured temperature would not be representative of the height averaged temperature causing flow. The Test Procedure [1] assumes that the flue flow is the result of convection in a stack at a uniform temperature, T_F, surrounded by an ambient atmosphere at a lower uniform temperature, T_{RA}.

b) Modulation - Steady-state tests were performed with the unit full fired and with the gas input reduced to 2/3 and to 1/3 of the full-fired input rate. The results of these tests are shown in Figure 5.

The non-cycling efficiency achieved by modulating firing rate is compared in Figure 6 to on/off cycling efficiency achieved by cycling with on- and off-times varied to match load at the full-fired gas input rate both with and without pump delay. The efficiency plotted in Figure 6 is modeled after the part load fuel utilization efficiency, η_u, of the Test Procedure [1] with the difference that the outdoor temperature was varied to produce a curve instead of a point value, and that cycle lengths and time constants consistent with the outdoor ambient temperature and the operational mode were used. The non-cycling efficiency is

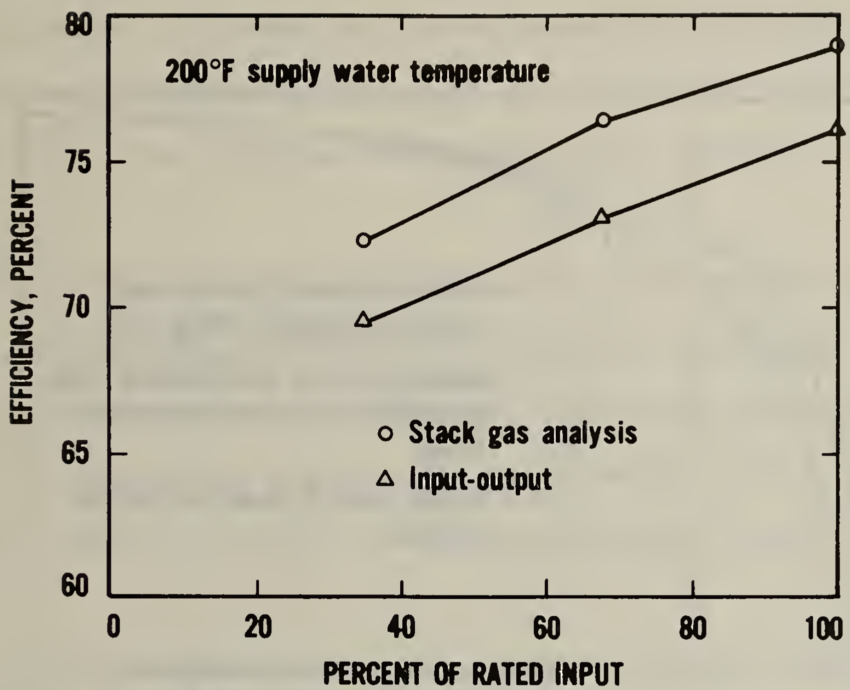

Figure 5. Reduced firing rate steady-state efficiency

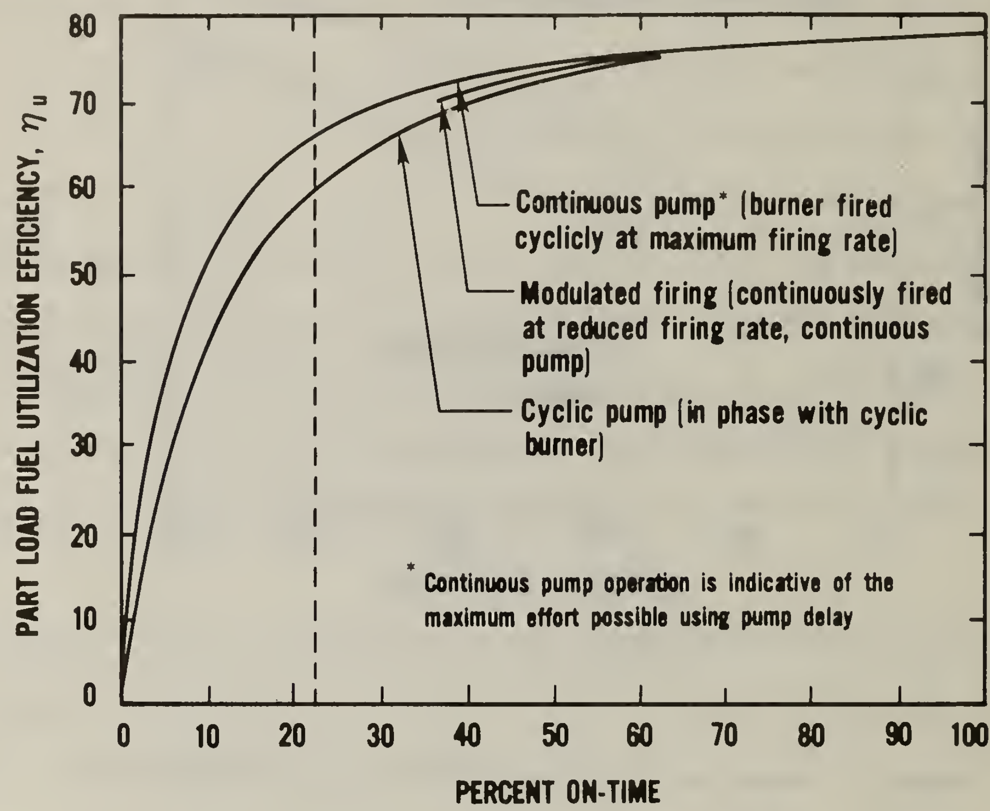

Figure 6. Comparison of modulated steady-state efficiency to full-fired cyclic efficiency with cyclic and with continuous pump operation

the steady-state efficiency reduced by the infiltration loss. This comparison indicates that while modulation would result in an efficiency improvement for a cyclic pump boiler, the incorporation of pump delay would result in a still greater improvement. It should be further noted that the improvement brought about by these two features is not additive. In fact, for this unit, modulation with pump delay is less efficient than cyclic burner operation with pump delay.

In an actual application it is presumed that this unit would not modulate at light loads, since firing rate reduction occurs only in response to near maximum outlet water temperatures. Such temperatures would only occur near the design outdoor temperature with the unit installed in a single loop system with a 70 percent oversizing factor as presumed by the Test Procedure [1].

The flue temperature elevation above the test room ambient temperature during cool-down from steady-state operation with the unit full fired is compared in Figure 7 to that occurring during cool-down from steady-state operation with the unit fired at 1/3 of its rated input. Time constants were calculated for the flue temperature heat-up and cool-down curves as follows:

	Full-fired	1/3 fired
τ_{on}	3.70 min.	4.13 min.
τ_{off}	18.02 min.	17.22 min.

In spite of the great difference between firing rates and steady-state flue temperatures at these two first conditions, the flue temperature time constants were changed very little.

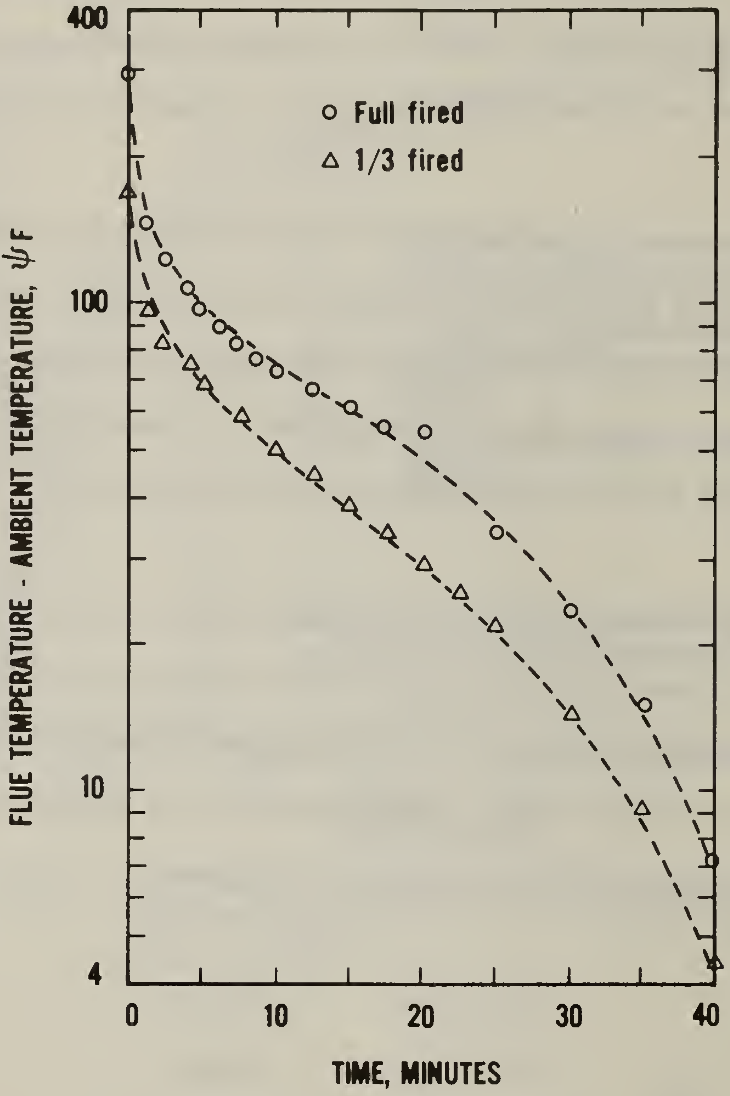

Figure 7. Comparison of flue temperatures during cool-down from full and 1/3 fired steady-state operation

c) Jacket Loss - The DoE/NBS Furnace/Boiler Test Procedure [1] calculations contain two correction factors for use with the laboratory measured jacket loss in calculating the part load efficiency and operating cost of units that are to be installed outdoors.

The first of these is C_j in step 27 of paragraph 4.1 [1] which is the ratio of the cycle jacket loss in a 42°F (6°C) ambient to the steady-state jacket loss measured in the laboratory. It accounts for the fact that the unit is assumed to be on only 22.5 percent of the time at this outdoor temperature and that the off-period (and on-period) jacket losses can significantly reduce the part load efficiency, η_u, of a furnace or boiler. Assigned values of C_j (step 27, paragraph 4.1) are 3.3 for furnaces and 4.7 for boilers.

The second of these factors is the value 3.3 in the equation for Q_{out} in section 4.2 which is the ratio between the steady-state jacket loss in a 5°F (-15°C) ambient and the steady-state jacket loss measured in the laboratory. It is pure coincidence that for furnaces the value of C_j happens to be numerically equal to the steady-state jacket loss factor used in calculating Q_{out}.

Laboratory tests of this boiler were directed only toward determining the cyclic component of C_j. It was assumed that, for a boiler of this type, the variation of jacket loss with ambient temperature would be slight. The heat exchanger covered only the top of the fire box, and the majority of the surface area, the sides and bottom, were exposed to the fire box temperature on one side and the ambient on the other. In a conventional boiler or furnace with large jacketing heat exchangers, the conditioned air or water temperature would more nearly govern the jacket loss than would the fire box temperature as in this boiler. It is reasonable to assume that if the surrounding ambient temperature were reduced, the combustion gas (and fire box) temperature would be reduced by the same amount, resulting in the temperature difference causing heat transfer to be nearly constant with ambient temperature variation. It was therefore assumed for the test plan and calculation procedures followed in this report that the jacket loss would be invariant with ambient temperature. The effects neglected in this assumption are the tendency of the top surface temperature to follow the water and flue temperatures and of the convective and radiative heat transfer coefficients to decrease with decreasing temperature. These two error sources would tend to be small and to cancel one another.

The tests performed to determine the cyclic portion of C_j were a steady-state, full load measurement of jacket loss, to serve as a reference, measurements of jacket temperature during cool-down and heat up tests, to determine time constants, and measurements of jacket temperature during repetitive cyclic operation at the on-and off-cycle lengths assumed by the Test Procedure [1] as typical for a boiler operating in a 42°F (6°C) ambient temperature (9.68 minutes on, 33.26 minutes off).

The average jacket temperature of the 118 thermocouples on the sides during cool-down from steady-state is compared in Figure 8 to a value calculated from the measured values 3.75 minutes and 25 minutes after shut-off, using assumptions of exponential decay and a jacket temperature at infinity equal to the room ambient temperature. The use of room air as an infinity value instead of the value 45 minutes after shut-off, as used by the Test Procedure [1] in flue and stack temperature calculations, was felt to be more accurate because of the slow rate of decay of jacket

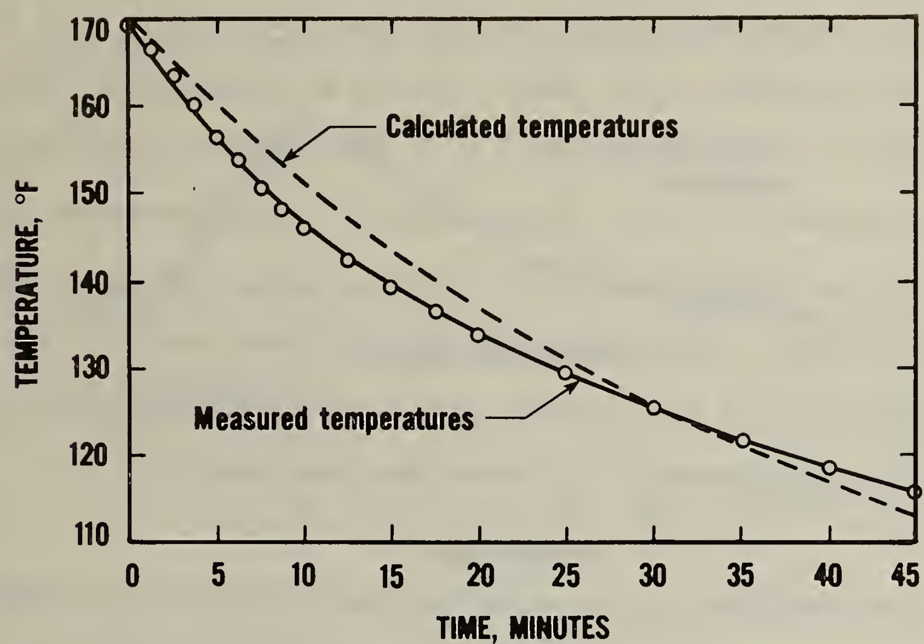

Figure 8. Side jacket temperature during cool-down

temperature as compared to the flue temperature, and because it was expected that the infinite time value would, in fact, be close to the room ambient temperature. Time constants for heat-up were calculated from experimental data taken 1 minute and 5.5 minutes after burner cut-on, which occurred at 50 minutes after burner cut-off as specified by the Test Procedure [1]. The resulting equations were not compared to the experimental heat-up data because of the assumption of the jacket temperature at infinite time being equal to the room ambient instead of that occurring 45 minutes after shut-off. These time constants were then used to predict the cyclic jacket temperature on each surface using a calculation procedure similar to that used for the flue temperatures in the Test Procedure [1].

A comparison of the predicted cyclic jacket temperatures for the unit side surfaces to the values measured in repetitive cyclic tests is shown in Figure 9. These jacket temperature profiles predicted from heat-up and cool-down time constants and steady-state temperatures were felt to be sufficiently close to the experimental values to justify their use in cyclic jacket loss calculations. This cyclic jacket loss calculation procedure is discussed in greater detail in Appendix A.

The ratio of the cyclic to the steady-state jacket loss was calculated for cyclic and continuous pump operation at nominal entering water temperatures of 165°F (74°C) and 105°F (41°C), using measurements taken of jacket temperature during steady-state, heat-up and cool-down tests in the normal laboratory ambient (approximately 80°F (27°C)) and using the calculation procedure of Appendix A with the following results:

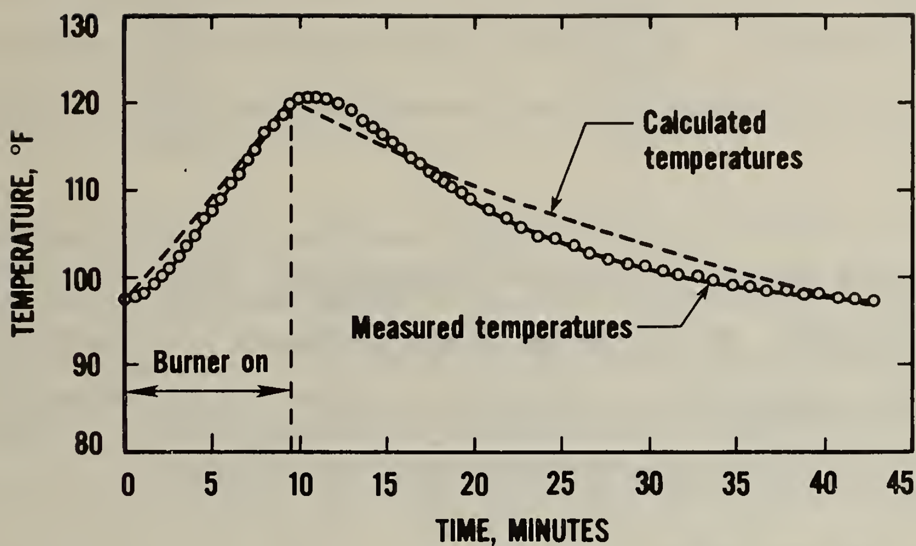

Figure 9. Side jacket temperature during cyclic operation

Pump Operation	Entering (Return) Water Temperature	Cyclic Jacket Loss / Steady-State Jacket Loss
Cyclic	167°F	1.21
Continuous	164°F	1.09
Cyclic	105°F	1.18
Continuous	105°F	1.02

Because of the small spread of these cyclic/steady-state jacket loss ratios and because the manufacturer of this design boiler recommends low return water temperatures for most applications, a pump delay value of 1.0 is recommended for C_j instead of the value of 4.7 specified by the Test Procedure.

The intuitive explanation for the near unity value of C_j is that the jacket of this boiler surrounds the fire box where a more conventional design would surround the fire box with heat exchanger surface and then the jacket. Thus this boiler has a high jacket temperature when operating which drops rapidly during the off-cycle. The conventional design, because of the surrounding heat exchangers, would have a lower on-cycle jacket temperature resulting in the off-cycle jacket loss being proportionately greater in comparison to on-cycle value. The lower value of C_j with continuous pump operation is the result of the circulating water recovering heat that would otherwise have been lost by the jacket.

d) Pump Delay - The maximum effect likely from water circulating pump delay was determined by comparing the results from a cool-down and heat-up test performed according to the Test Procedure [1] (pump cycled with burner) to tests performed identically except for continuous pump operation. The continuous pump tests were performed (1) with the inlet water temperature held constant by an auxiliary boiler in the test loop and (2) with the inlet water temperature allowed to decay at an arbitrary rate given by allowing makeup water to continue to replace test loop water during cool-down and heat-up The same water flow rate was used as during the steady-state period to balance the unit heat capacity.

It was observed during the continuous pump operation tests that the flue temperature during cool-down tended to approach the inlet water temperature as is shown in Figure 10. This means that in boilers which have pump delay the off-cycle loss is dependent upon the inlet water temperature. This is not the case in boilers without pump delay, in which case the off-cycle stack temperature approaches the test room ambient air temperature during off-cycle cool-down. Steady-state efficiency is affected by the mean (average of inlet and outlet) water temperature. The Test Procedure [1] specifies outlet water temperature and a temperature rise which may be conveniently obtained in a test laboratory, and which result in a mean water temperature typical of an actual application. The off-cycle loss dependency of pump delay boilers on inlet water temperature would require changing the allowable inlet water temperature of the Test Procedure [1] to values typical of an actual application if the scope is to be expanded to cover this design feature.

Because of the importance of this variable, a second set of tests was performed at a lower water temperature to clarify its effect. The test conditions and results for these continuous and cyclic pump tests are summarized in Table 1.

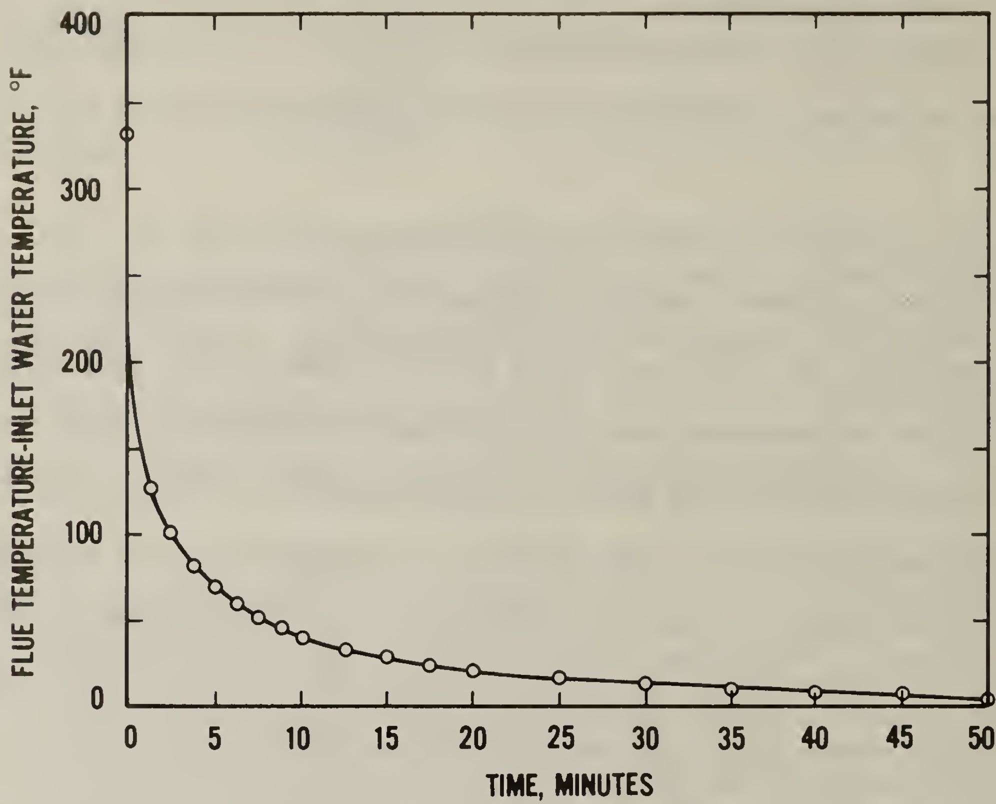

Figure 10. Flue temperature during continuous water pump operation cool-down test

Table 1. Comparison of Cyclic and Continuous Pump Heat-Up and Cool-Down Tests

	Cyclic Pump		Continuous Pump (Auxiliary Boiler on)		Continuous Pump (Decaying Water Inlet Temp.)	
Gas input, kBtuh	125.6	125.3	125.6	125.4	124.5	125.4
Steady-state flue temp., °F	497.9	467.1	496.1	468.4	492.6	468.2
Steady-state outlet water temp., °F	200.5	147.2	200.4	147.2	200.5	147.1
Steady-state inlet water temp., °F	167.1	105.2	163.7	105.0	159.2	105.1
Steady-state mean water temp., °F	183.8	126.2	182.1	126.1	179.7	126.1
Inlet water temp. at cut-on*, °F	156.5	108.6	163.9	104.6	125.7	85.9
Heat-up time constant, τ_{on}	4.24	4.09	3.91	3.70	5.02	4.38
Cool-down time constant, τ_{off}	20.9	19.3	11.0	10.3	13.2	12.0
Steady-state efficiency, %	78.5	79.4	78.5	79.2	78.7	79.2
Part load fuel utilization efficiency, η_u, %	60.9	62.5	66.8	71.0	68.6	71.7
Annual fuel utilization efficiency, $EFFY_A$, %	60.3	61.8	66.1	70.2	67.9	70.9

*The above listed inlet water temperature at cut-on was recorded one minute after cut-on to account for the time lag resulting from the mass of the thermocouple wells in the water circulating system.

The flue temperatures measured during the 165°F (74°C) return water temperature cyclic and continuous pump cool-down tests are compared in Figure 11 and the resulting efficiencies are compared in Figure 6.

The difference in part load efficiency between cyclic and continuous pump operation tests is seen to be quite significant. When continuous pump operation is employed, the effect of the inlet water temperature is significant. Decay of water temperature at the rates used in these tests had a comparatively small effect.

An additional test was performed for comparison to the above cyclic and continuous pump tests in which a 5 minute delay in pump shut-off after burner shut-off was employed. This test resulted in a flue temperature which followed the continuous pump curve until pump shut-off and which then rebounded and approached asymptotically from below the cyclic pump flue gas temperature decay curve (see Figure 11).

It should be emphasized that continuous pump operation was employed instead of actual pump delay because it served as an extreme case for showing the effects of pump delay. An actual continuous pump operation design in field operation may result in excessive pump power costs and, with sufficiently long boiler off-periods, would eventually result in heat from the house being transferred to the boiler and out the stack if not fitted with a stack damper. This latter occurrance would be most likely with intermittent ignition instead of the standing pilot employed on the test boiler since in the former case the flue temperature would approach the room air temperature, and in the latter case would

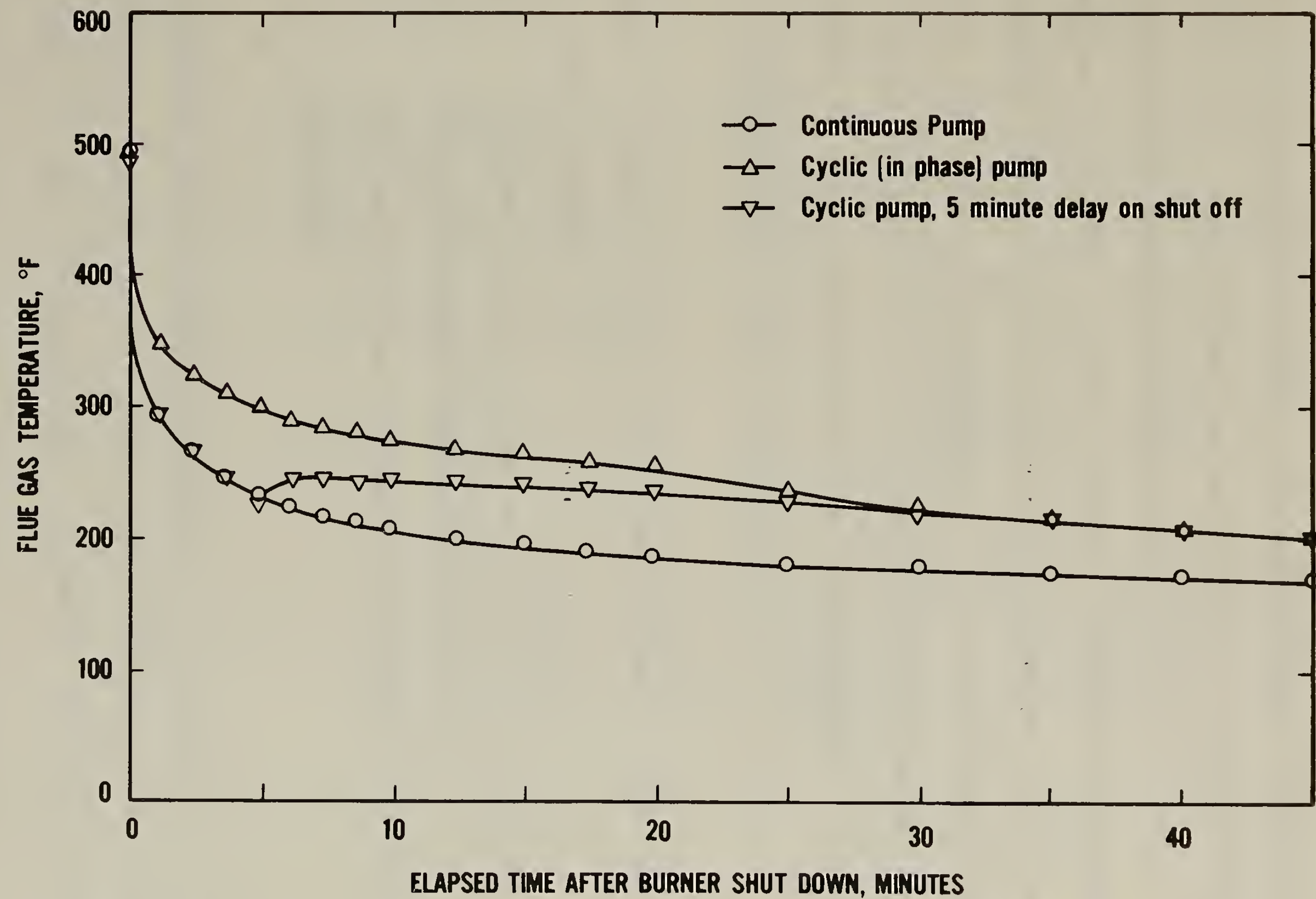

Figure 11. Flue temperatures during cool-down with cyclic and continuous pump and with 5 minute pump delay

approach a temperature approximately 20°F (11°C) to 30°F (17°C) above the room temperature.

6. COMPUTER SIMULATION RESULTS AND DISCUSSION ON PUMP DELAY

A control strategy analysis was made for general cases using the NBS DEPAB program [5]. In the simulation model, the heat exchanger was assumed to be a low mass shell type. Measured data was used as input to the simulation program. When the measured data was not sufficient, an estimation was made. Input data for the computer simulation is given in Table 2. The main difference of design between the actual boiler and the boiler under simulation was the heat exchanger. In addition to this, the circulating pump of the actual boiler operated by sensing the water temperature, but the simulated boiler operated by imposed time delay. This time delay feature was much easier to handle for sensitivity analysis of pump delay operation.

In the absence of field test data, three likely modes of operation were conceived as seen in Figure 12.

- Mode I: The pump operates in the same phase as the burner
- Mode II: The pump operates incorporating time delay
- Mode III: The pump operates continuously regardless of the burner operation

The Test Procedure calls for Mode I operation for all boiler performance tests. When the burner is turned on, the water circulating pump is also turned on. When the burner is off, the pump is off. In Mode I operation, no pump delay is allowed.

In Mode II, the water pump operates in a similar way to an air circulation fan of a forced warm air furnace. This mode was the focus of the study.

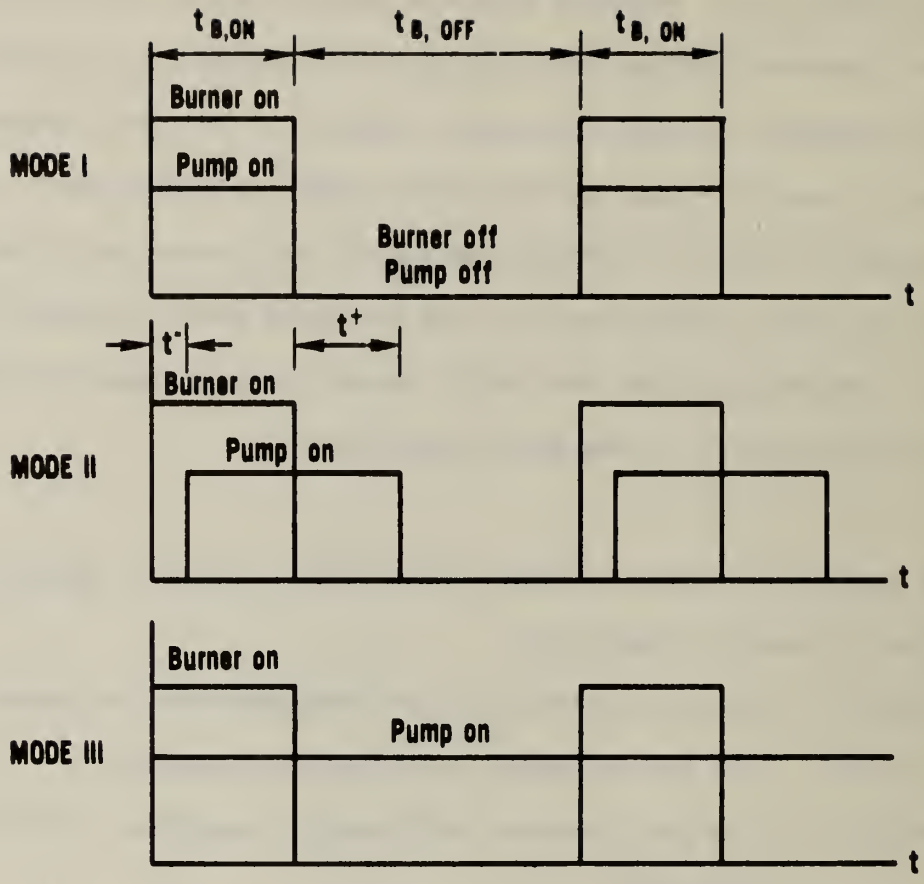

Figure 12. Three modes of boiler pump operation

Table 2. Input Data for Computer Simulation

Fuel:	Natural Gas
Burner Input Rate:	125.6 kBtu/h (36.79 kW)
Pilot Light, Input Rate:	426.0 Btu/h (124.8 W)
Pump Input Rate:	250 W
Steady-State Efficiency:	78%
Heat Exchanger Weight*:	60 lb (27.2 kg)
Type of Heat Exchanger:	Shell type
Water Flow Rate:	2,452 lbm/h (1,112.2 kg/h)
Water in the Heat Exchanger:	4.16 lb (1.89 kg)
Water in the Boiler*:	42.7 lb (19.37 kg)
Jacket Weight:	23.3 lb (10.57 kg)
Overall Heat Transfer Coefficient of Jacket:	0.5 $Btu/ft^2h°F$ (2.837 W/m^2K)
Room Air Temperature:	70°F (21.1°C)
Outdoor Air Temperature	42°F (5.6°C)
Inlet Water Temperature	105°F (40.6°C)
Outlet Water Temperature at Steady-State:	147°F (63.9°C)
Off-Cycle Draft Factor for Flue Gas Flow:	1.0
Off-Cycle Draft Factor for Stack Gas Flow:	1.0
Burner-On Time:	9.68 min.
Burner Off-Time:	33.26 min.

*More water in the boiler was added to consider the effect of heat capacity of the fire brick walls.

Computer simulations were performed for all three modes employing a constant inlet water temperature. Since the inlet water temperature at a field installation depends upon many factors such as heat losses through the building envelope, the flow rate of circulating water, outdoor temperature, infiltration loss, etc., the inlet water temperature was assumed to be constant at all times. Through the computer simulations, which were based on the heat loss method, fuel utilization efficiencies, η_u, were obtained at each mode in terms of fractional burner on-times, $t^*_{B,ON}$, as shown in Figure 13. As seen in the figure, higher efficiencies were obtained at Mode II and Mode III operation than at Mode I. In Figure 13, two parameters, t^- and t^+, which are a pump delay time during on-period and a pump delay time during off-period, respectively, were used. These parameters are shown in Figure 12 - Mode II.

When the water circulation continued during off-period, the flue gas temperature was lowered rapidly. The heat loss through the stack of the boiler was thus reduced, resulting in higher fuel utilization efficiency. The analysis was continued to obtain Figure 14 showing the effect of t^+ and t^- and η_u at 22.5 percent fractional burner on-time. The dimensionless pump on-time, y, is defined as in the revised version of the Test Procedure as:

$$y = 1 + \frac{t^+ - t^-}{t_{B,ON}}$$

The value of y in Figure 14 was calculated using a constant $t_{B,ON}$, equal to 9.68 min., as recommended in reference [1]. For a given y, the efficiency decreased as the time delay during on-cycle, t^-, increased. Pump operation

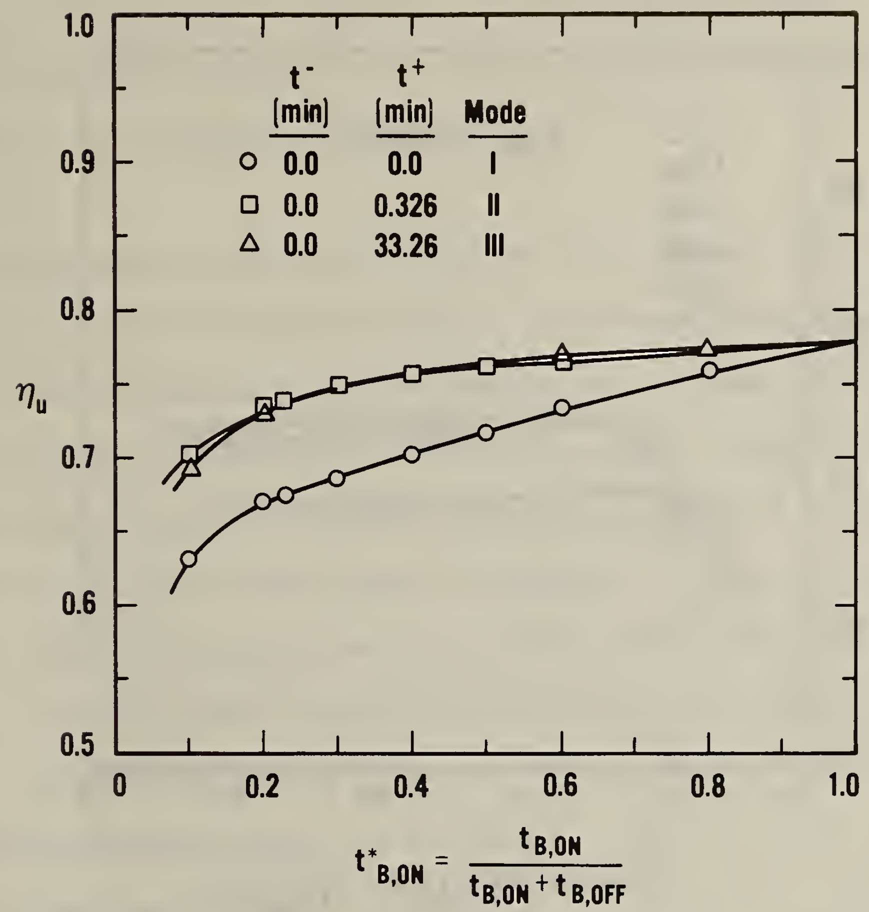

Figure 13. Fuel utilization efficiencies with respect to fractional burner on-times for three modes of pump operation

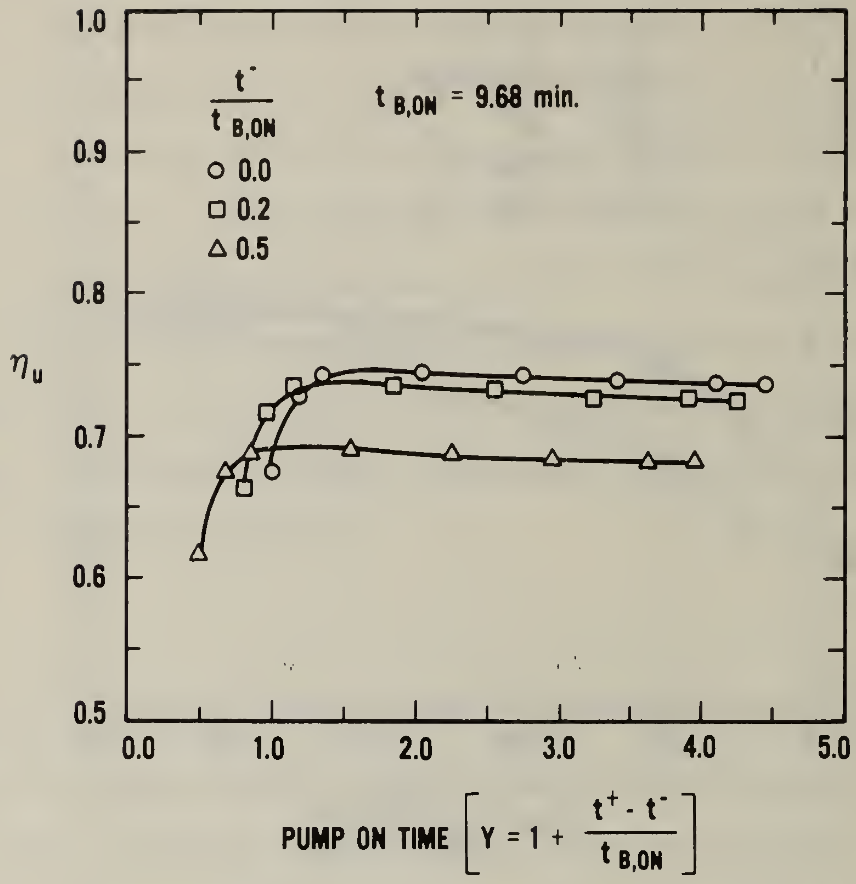

Figure 14. Fuel utilization efficiencies with respect to dimensionless pump on-times with a parameter of $t^-/t_{B,ON}$

without delay during on-period gave highest efficiency. Also, the prolonged pump operation did not improve the efficiency of the low thermal mass boiler (note that η_u did not show the effect of increased energy usage). A control strategy with proper choice of t^+, t^-, and y is needed to minimize the energy loss as well as the equipment decay rate.

Now consider the annual operating cost of the boiler system which includes the boiler itself and the circulating pump. Other associated devices like a thermostat, electromagnetic valves and sensors are excluded. Annual operating cost mainly depends on the fuel and electricity costs. Employing the annual cost calculation procedure in the Test Procedure and results of computer simulations given in Figure 14, annual costs in terms of the pump operating time, y, for selected values of $t^-/t_{B,ON}$ were calculated. The results are shown in Figure 15. To obtain Figure 15, the following numerical values were assumed in addition to the boiler data:

Design Heating Requirement	=	100 kBtu/h
Heating Load Hour (HLH)	=	2080 h
Fuel Cost Rate (Natural gas)	=	50¢/10^5 Btu
Electricity Cost Rate	=	7¢/kW-h

From Figure 15, there exists a minimum cost operation curve for a specific condition. The total operation costs for y > 1 and small value of $t^-/t_{B,ON}$ are always less than that for y = 1 and $t^-/t_{B,ON}$ = 0 (Mode I operation). One should realize that the annual operation cost figure is provided as an example. Under normal field operating conditions, the operation cost of a boiler system varies depending upon the boiler configuration, control strategy, heating load hour, design heating requirement, fuel cost and electricity cost.

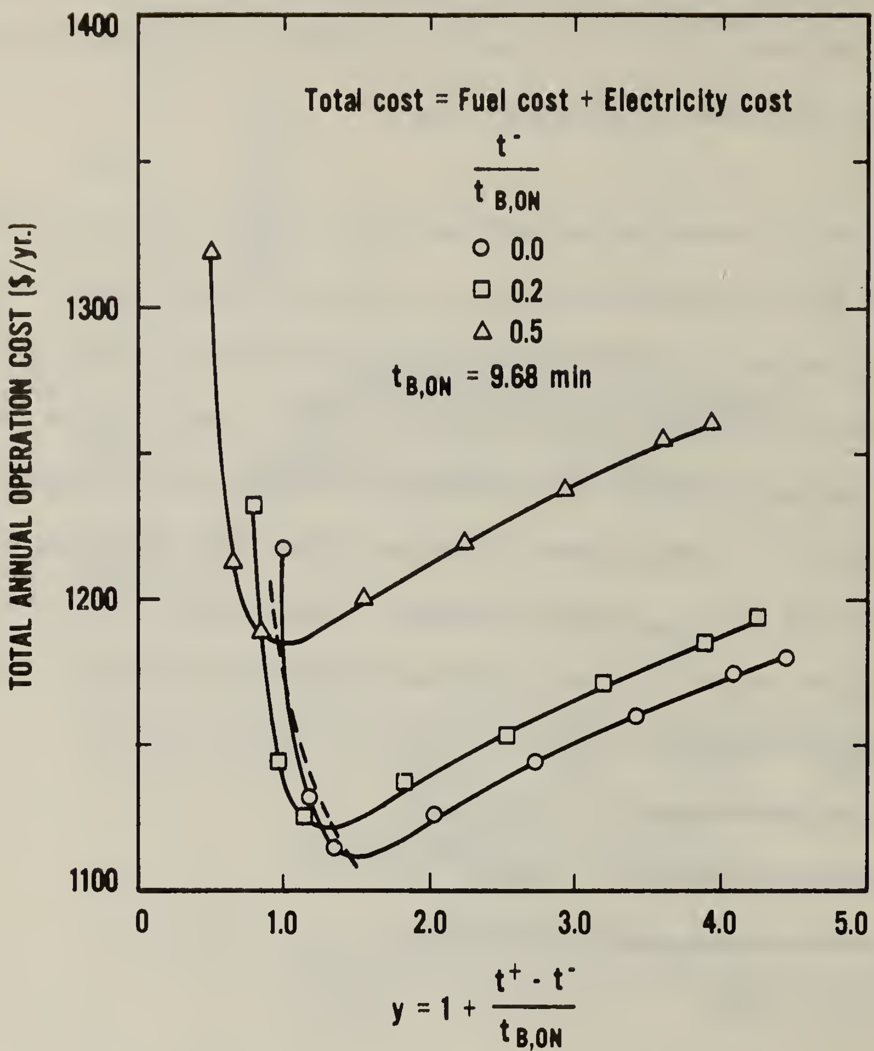

Figure 15. Total annual operation costs with respect to dimensionless pump on-times with a parameter of $t^-/t_{B,ON}$

The electricity cost for continuously running the circulating pump results in the increase of operation cost. Unless the pump delay at the start-up during on-period is considerably long, most cases of operation in Mode II ensure lower cost than in Mode I.

7. CONCLUSIONS

The unit tested was found to conform to the assumptions upon which off-cycle stack mass flow is calculated in the DoE/NBS Furnace Boiler Test Procedure [1]. Therefore, no change is recommended to the flue temperature measurement procedure or to the flue flow calculation for this unit.

Modulation of the firing rate of this boiler resulted in a part load efficiency that was greater than that resulting from full-fired cycling with in-phase cyclic pump operation but less than that resulting from full-fired cycling with continuous pump operation. Because (1) modulation reduced the efficiency of the boiler when operated with continuous pump; (2) it is questionable that much modulation would occur with the control system supplied by the manufacturer (maximum firing at minimum water temperature) in the 70 percent oversized single loop system assumed by the Test Procedure [1]; and (3) additional tests and calculation complexity are required for seasonal efficiency calculations for modulated equipment, it was decided not to recommend modification of the Test Procedure to incorporate modulation at this time for this boiler. It should be noted that continuous pump operation was used in these laboratory tests as a simplified control system that would show the maximum effect likely from pump delay. It is questionable that continuous pump operation would occur for any significant time in the home. Units which also modulate combustion air as well as the gas input would be expected to show substantial efficiency gain [6]. Possible amendment to the Test Procedure [1] to foster the introduction to the market of such air modulating designs should be considered. Parallel work covering the development of test procedures for modulating vented household heaters and furnaces [6] has been completed. Procedures developed in [6] may be applicable to such an amendment.

Laboratory measurements indicated that, for this class of a boiler, the value of C_j in the Test Procedure should be changed from 4.7 to 1.0. A procedure for calculating the cyclic portion of C_j from jacket time constants which could be measured during testing according to the current Test Procedure is described in Appendix A.

Water pump delay was found to produce a substantial energy saving when applied to this boiler. Changes to the Test Procedure [1] to allow testing of units with pump delay are given in Appendix B. These changes require testing at the same mean water temperature as the existing standard (thus producing comparable steady-state efficiencies) but at lower outlet and higher inlet water temperatures. It was felt necessary to test at inlet water temperatures near those actually occurring in the field because of the sensitivity of calculated annual efficiency to inlet water temperature for pump delay boilers.

It should be noted that the temporary exemption for this boiler to be tested at a smaller temperature rise than specified by the Test Procedure [1] results in an excessively high mean water temperature for comparability to other units. The above modification to the Test Procedure [1] rectifies this problem as well as allowing fair credit to be given for pump delay. These recommended changes do not increase the difficulty or time required for testing and will produce results for conventional boilers that will be consistent with previous tests.

REFERENCES

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2. Park, Cheol, Mulroy, William J., and Kelly, George E., "A Study of the Dynamic Flue-Gas Temperature and Off-Period Mass Flow Rate of a Residential Gas-Fired Furnace, NBS Technical Note 999, July 1979.

3. American National Standard for Gas-Fired Central Furnaces, ANSI Z 21.47 - 1979.

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5. Chi, J., "Computer Simulation of Fossil-Fuel-Fired Boilers," Conference on Improving Efficiency and Performance of HVAC Equipment and Systems for Commercial and Industrial Buildings, Purdue University, April 12-14, 1976.

6. Kweller, Esher, and Palla, Robert L., "A Test Method and Calculation Procedure for Determining Annual Efficiency for Vented Household Heaters and Furnaces Equipped with Modulating-Type Controls, NBSIR 82-2497, May 1982.

APPENDIX A. CYCLIC JACKET LOSS CALCULATION

Cyclic Jacket temperatures were calculated using a procedure modeled after that used by the DoE/NBS Test Procedure to calculate cyclic flue gas temperatures [1,2]. The derivation and nomenclature used here for the cyclic temperature calculation closely follows that given in greater detail in [2]. The primary difference between the calculation presented here for jacket temperatures and that of the Test Procedure for flue temperatures is the assumption that the infinite time value of jacket temperature with the unit off is equal to the room ambient temperature. This assumption was made instead of the Test Procedure assumption of the infinite time value being equal to that at 45 minutes after shut-off for cool-down from steady-state because the slow cool-down of the jacket resulted in the jacket temperature being well above its true infinite time value after 45 minutes and because, being relatively unaffected by the standing pilot, the infinite time value was close to the room ambient. A different assumption may, however, be necessary for a fully jacketed unit. As is shown in Figure 8, the jacket temperature during cool-down from steady-state closely conformed to the assumption of exponential decay to the room ambient temperature expressed as:

$$\Psi_J(t) = \Psi_{J,0,X}\, e^{-\frac{t}{\tau_{off}}} \tag{1}$$

where $\Psi_J(t) = T_J(t) - T_{RA}$, and $\Psi_{J,0,X} = T_{J,\,SS} - T_{RA}$

$T_{J,SS}$ is the steady-state and T_{RA} the room ambient temperature, and τ_{off} is to be determined from cool-down test data.

A similar expression can be written for jacket temperature during the heat-up period:

$$\Theta_J(t) = \Theta_{J,O,X}\, e^{-\frac{t}{\tau_{ON}}} \tag{2}$$

where $\Theta_J(t) = T_{J,SS} - T_J(t)$, and it is assumed that $\Theta_{J,O,X} = \Psi_{J,O,X} = T_{J,ss} - T_{RA}$.

As shown in [2], if the jacket temperatures at time t_1, and t_2 are experimentally known, the time constant during the heat-up period, τ_{ON}, can be obtained as:

$$\tau_{ON} = \frac{t_2 - t_1}{\ln\left[\dfrac{T_{J,SS} - T_J(t_1)}{T_{J,SS} - T_J(t_2)}\right]} \tag{3}$$

Similarly if the jacket temperature during the cool-down period is measured at times t_3 and t_4 after the furnace is turned off, τ_{OFF} can be obtained using:

$$\tau_{OFF} = \frac{t_4 - t_3}{\ln\left[\dfrac{T_J(t_3) - T_{RA}}{T_J(t_4) - T_{RA}}\right]} \tag{4}$$

When the cool-down and heat-up periods are finite, the pattern of the jacket temperature changes with respect to time. Due to the finite length of time for on- and off-cycle periods, the initial values of the exponential function, $\Theta_{J,O,X}$ and $\Psi_{J,O,X}$ become smaller due to the fact that the unit may never heat up to steady-state or cool down to equilibrium. It is therefore necessary to introduce correction factors such that for the heat-up period,

$$\Theta_J(t) = C_{t,ON}\Theta_{J,O,X}\, e^{-\frac{t}{\tau_{ON}}} \tag{5}$$

and for the cool-down period,

$$\Psi_J(t) = C_{t,OFF}\Psi_{J,O,X}\, e^{-\frac{t}{\tau_{OFF}}} \tag{6}$$

The quantities $C_{t,ON}$ and $C_{t,OFF}$, which are correction factors for the on- and off-periods respectively, are derived in [2]. If the assumption $\Theta_{J,O,X} = \Psi_{J,O,X} = T_{J,SS} - T_{RA}$ is substituted in the final formulae for $C_{t,ON}$ and $C_{t,OFF}$ given in [2] the resulting equations are:

$$C_{t,ON} = \frac{1 - e^{-\frac{t_{OFF}}{\tau_{OFF}}}}{1 - e^{-\left(\frac{t_{ON}}{\tau_{ON}} + \frac{t_{OFF}}{\tau_{OFF}}\right)}} \tag{7}$$

$$C_{t,OFF} = \frac{1 - e^{-\frac{t_{ON}}{\tau_{ON}}}}{1 - e^{-\left(\frac{t_{ON}}{\tau_{ON}} + \frac{t_{OFF}}{\tau_{OFF}}\right)}} \tag{8}$$

Figure 9 compares the experimental cyclic jacket temperature and the predicted jacket temperature given by

$$T_J(t) = T_{J,SS} - C_{t,ON}(T_{J,SS} - T_{RA})\, e^{-\frac{t}{\tau_{ON}}} \quad \text{for on-cycle, and} \tag{9}$$

$$T_J(t) = T_{RA} + C_{t,OFF}(T_{J,SS} - T_{RA})\, e^{-\frac{t}{\tau_{OFF}}} \quad \text{for off-cycle,} \tag{10}$$

with the values of $C_{t,ON}$ and $C_{t,OFF}$ obtained from equations (7) and (8), respectively.

The jacket loss at a given temperature may be calculated from the equation $Q = hA\Delta T$, where A is the surface area and ΔT is the temperature differential. Values of h given as a sum of convective and radiative components are given in [3] in graphical form for substitution into the equation:

$$h = (h_c + \varepsilon\, h_{r\ell}) \tag{11}$$

where h_c and $h_{r\ell}$ are convective and radiative heat transfer coefficients, respectively, and ε is the emissivity at the surface.

The following fits were prepared to the graphical presentations of [3].

$$h_c = 0.380\ (T_J - T_{RA})^{0.251} \quad \text{for the top surface} \tag{12}$$

$$h_c = 0.273\ (T_J - T_{RA})^{0.249} \quad \text{for the sides} \tag{13}$$

$$h_c = 0.200\ (T_J - T_{RA})^{0.251} \quad \text{for the bottom} \tag{14}$$

$$h_{r\ell} = (0.664 + 0.0026\ T_{RA})e^{0.00272\ T_J} \tag{15}$$

Recommended emissivity (ε) values are given in [3] for use in equation (11).

The cyclic jacket loss was calculated by using a subroutine to time integrate the product of the surface area, jacket temperature and h factor for top, side, and bottom surface using jacket temperature measured during steady-state, and cool-down and heat-up tests.

APPENDIX B

Recommended Changes to the DoE/NBS Test Procedure for Furnaces and Boilers to Accommodate Testing of Boilers with Pump Delay

The following recommended changes are indexed to the DoE/NBS Furnace and Boiler Test Procedure as presented in NBSIR 78-1543 titled "Recommended Testing and Calculation Procedures for Determining the Seasonal Performance of Residential Central Furnaces and Boilers." These changes affect boilers without pump delay only in specifying a mean water temperature for steady-state testing instead of a temperature rise and leaving (outlet) water temperature. This change increases the range of allowable test conditions to allow closer matching of the test conditions to the manufacturer's design values and greater flexibility in test cell design without sacrificing comparability. No changes have been made to the test procedure for furnaces. The recommended changes or additions are shown underlined in the following paragraphs:

pg. vi, Nomenclature:

t^{+} delay time between burner shut-off and blower <u>or circulating pump</u> shut-off, in minutes

t^{-} delay time between start-up and blower <u>or circulating pump</u> start-up, in minutes

pg. vii, Nomenclature:

y ratio of blower <u>or pump</u> on-time to burner on-time

pg. 4, Definitions:

<u>Mean Water Temperature - The arithmetic average of the inlet and outlet water temperatures of a boiler.</u>

pg. 8:

2.5.4 Gas- and Oil-Fueled Low-Pressure Steam and Hot Water Boilers (Including Direct-Vent Systems)

The water flow rate for hot water boilers shall be adjusted to produce a <u>mean water temperature (average of inlet and outlet), during steady-state operation, between 115.0°F (46.1°C) and 145°F (62.8°C) and a temperature rise greater than or equal to 20°F (11.1°C). The inlet water temperature for units which have pump delay after burner shut-off and/or which have counter-flow heat exchangers shall be greater than 120°F (48.0°C). The selected water temperatures shall be sufficiently high that condensation of the combustion products does not occur during steady-state operation</u>. For steam boilers, the steady-state performance test described in 3.1 shall be conducted at atmospheric pressure or at a pressure not exceeding 2 pounds per square inch gauge.

pg. 11:

3.1.2 Oil-Fueled Forced-Air Central Furnaces and Oil-Fueled Low-Pressure Steam and Hot Water Boilers (Including Direct-Vent Systems)

The furnace or boiler shall be set up and adjusted as specified in sections 2.1, 2.2 and 2.3.4. Begin the steady-state performance test by operating the burner and the circulating air blower or water pump with the adjustments specified by 2.4.2 and 2.5 until steady-state conditions are attained, as indicated by a temperature variation in three successive readings taken 15 minutes apart, of not more than: [1] 5°F (2.8°C) in the flue gas temperature and [2] 4°F (2.2°C) in the <u>mean</u> water temperature for hot water boilers.

. . . . *Remainder of Paragraphs Unchanged* . . . .

3.1.4 Electric Boilers

Flow conditions shall be as specified in section 2.5.6. Electrical supply shall be as specified in section 2.3.5. The boiler shall be operated until steady-state conditions are reached, as indicated by a temperature variation in three successive readings taken 15 minutes apart of not more than 4°F (2.2°C), in the mean water temperature for hot water boilers. Three measurements of the total power input to the boiler shall be made at 10-minute intervals and averaged to find the rated power input (E_{in}), in watts.

pg. 12:

3.2.2 Gas- and Oil-Fueldd Boilers (Including Direct-Vent Systems)

After steady-state testing has been completed, turn the main burner(s) off and measure the flue gas temperature at 3.75 ($T_{F,OFF}(t_3)$) and 22.5 ($T_{F,OFF}(t_4)$) minutes after the burner shuts off, using the nine thermocouples described above. During this off-period for units that do not have pump delay after shut-off, no water shall be allowed to circulate through the hot water boilers. For units that have pump delay on shut-off, the pump shall be stopped by the unit control and the time, t^+, between burner shut-off and pump shut-off shall be measured within one second accuracy. While the pump is operating the inlet water temperature and flow rate shall be maintained at the same values as used during the preceeding steady state test.

If temperature rebound after pump shut-off occurs such that $T_{F,OFF}(t_4)$ is equal to or greater than $T_{F,OFF}(t_3)$, or $T_{F,OFF}(\infty)$ is equal to or greater than $T_{F,OFF}(t_4)$, the time delay for pump shut-off after burner shut-off, t^+, shall be manually increased or the pump cycled so that $T_{F,OFF}(3)$ is greater than $T_{F,OFF}(t_4)$ and $T_{F,OFF}(t_4)$ is greater than $T_{F,OFF}(\infty)$. The time delay t+ shall be increased by this additional pump operation time.

. . . . *Remainder of Paragraphs Unchanged* . . . .

3.3.2 Gas- and Oil-Fueled Boilers (Including Direct-Vent Systems)

Fifty minutes after the main burner(s) is turned off for the cool-down test, the steam or hot water boilers shall be turned on and the flue gas temperature measured, using the nine thermocouples described above, at 1.0 $T_{F,ON}(t_1)$) and 5.5 ($T_{F,ON}(t_2)$) minutes after the main burner(s) comes on. <u>For units that do not have pump delay on start-up, the</u> pump circulating the water through the hot water boiler shall be started simultaneously with the main burner(s). <u>For units that have pump delay on started by the unit control and the time, t^{-}, between burner and pump start-up shall be measured within one second</u>. The water flow rate shall be the same as that maintained during the steady-state test described in section 3.1. During the heat-up test for oil-fired boilers, the draft in the flue pipe shall be maintained within ± 0.01 inches of water column of the manufacturer's recommended on-period draft. Record the measured temperatures.

pg. 13:

3.6 <u>Optional Procedure for Determining D_p, D_F and D_S for Systems Equipped Power Burners</u>

On power-burner systems not employing automatic stack dampers or power-burner systems with a stack damper and a draft diverter on draft hood, D_F shall be measured during the cool-down test described in section 3.2. On systems for which the flue or stack damper is to be closed during the cool-down test described in section 3.2, D_p shall be measured during a separate cool-down test. This separate cool-down test shall be conducted after the heat-up test described in section 3.3 is completed. It shall be conducted by letting the unit run after the heat-up test until steady-state conditions are reached, as indicated by temperature variation in three successive readings taken 15 minutes apart of not more than plus or minus 5°F (2.8°C) in the flue gas temperature and 4°F

(2.2°C) in the mean water temperature for hot water boilers, and then shutting the unit off with the stack or flue damper controls by-passed or adjusted so that the stack or flue damper remains open during the resulting cool-down period. If a draft was maintained on oil-fueled units in the flue pipe during the steady-state performance test described in section 3.1, the same draft (within -0.001 and +0.005 inches of water gauge of the average steady-state draft) shall be maintained during this cool-down period.

pg. 16:

22. Enter value of y equal to $1 + (t^+ - t^-)/3.87$ or furnaces, y equal to $1 + (t^+ - t^-)/9.68$ for boilers and y equal to 1.00 for boilers without pump delay or furnaces employing a single motor to drive a power burner and an indoor-air circulating blower.

pg. 23:

4.2 Recommended Procedure for Calbulating the Annual Cost of Operation of a Furnace or Boiler Located in Different Climatic Regions of the Country and in Buildings with Different Design Heating Requirements

The annual cost of operating a gas- or oil-fired furnace or boiler located in various geographic locations of the United States and in buildings with different design heating requirements shall be determined using the following three-step procedure:

Step 1. Determine the number of burner operating hours using the equation:

Burner Operating House = A (HLH) (C) (design heating requirement) - B (HLH)

where the number of heating load hours HLH, may be obtained from Figure 9 for the region of interest, the "design heating requirement" is the heating requirement to be met by the furnace or boiler in kBtu per hour at the 97 1/2 percent outdoor design temperature, and C = 0.77 is an "experience factor" which tends to improve the agreement between the average calculated burner operating hours and the average burner operating hours found in the field. It is strongly recommended that this "experience factor" be eliminated as soon as an improved method is available to more accurately estimate residential heating requirements. Typical values for the design heating requirement are given in Table for different furnace or boiler output capacities Q_{OUT}, where $Q_{OUT} \equiv \eta_{SS}$ (col. 30) X Q_{IN} (col. 4) rounded off to the nearest 1000 Btu/h for units intended for installation in a heated space and

$$Q_{OUT} = \frac{Q_{IN}\ (\text{col. } 4)}{100}\ (\eta_{SS}(\text{col. } 30) - 3.3\ L_J\ (\text{col. } 18))\ \text{rounded off}$$

rounded off to the nearest 1000 Btu/h for units intended for installation out of doors or in an unheated space. The constants A and B are unique to the unit under tests and may be calculated using information contained in the worksheet and the following expressions:

$$A = \frac{100,000}{341,300\ (PE + y\ BE) + (Q_{IN} - Q_P)\eta_u}$$

$$B = \frac{(2)\ (A)\ (Q_p)\ (\eta_u)}{100,000}$$

$$\text{where } y = \begin{cases} 1 + \left(\dfrac{t^+ - t^-}{3.87}\right) & \text{for furnaces} \\ 1 + \left(\dfrac{t^+ - t^-}{9.68}\right) & \text{for boilers} \end{cases}$$

1.00 for boilers without pump delay or furnaces employing a single motor to drive a power burner and an indoor-air circulating blower.

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[ ] Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)*

In response to a request by a manufacturer of a nontypical boiler, the Department of Energy requested the National Bureau of Standards to perform laboratory measurements under controlled conditions of the effect on seasonal performance of several features (finned copper tube heat exchanger, water circulating pump delay, and gas valve modulation) of this boiler that might cause it to be unfairly treated by the existing test procedure. As a result of this study, recommended changes to the existing test procedure to allow rating tests with water circulating pump delay are presented. A recommended change to the assigned cyclic jacket loss factor and a simplified procedure for experimentally determining this factor are also presented. No change to the current test procedure treatment of gas valve modulation or flue gas mass flow as a function of temperature are recommended.

12. KEY WORDS *(Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)*

Annual efficiency; annual operating costs; boilers; fossil fuel heating systems; jacket loss; modulating control gas fueled; part-load performance; rating procedures; seasonal efficiency

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