DIRECT MEASUREMENT OF THE OVERALL EFFICIENCY AND ANNUAL FUEL CONSUMPTION OF RESIDENTIAL OIL-FIRED BOILERS

ANNUAL REPORT
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JANUARY 1978

DEPARTMENT OF ENERGY AND ENVIRONMENT

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Abstract

A direct measurement procedure has provided accurate evaluation of the efficiency of residential heating units during full-load and part-load operation. Laboratory measured efficiency data for each heating unit are translated into a more useful form as annual fuel consumption and fuel-weighted seasonal efficiency values. The changes in fuel use and seasonal efficiency are evaluated for variations in operating conditions including: geographic location, design heat load of the building, domestic hot water use, and design fuel firing rate. The combination of direct, accurate efficiency measurement and calculation of annual fuel use provide a standard method for comparison of individual heating units and retrofit modifications on a common and realistic basis. The cost effectiveness and payback periods of equipment modifications can be quantitatively evaluated.

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I. INTRODUCTION

The Burner-Boiler/Furnace Efficiency Test Project at Brookhaven National Laboratory is a continuing program described in "Energy Management in Residential and Small Commercial Buildings, Fiscal Year 1976."[1] The project is funded through the Department of Energy and it is designed to provide the technical basis for reduced consumption of oil and gas for space heating of residential and small commercial buildings by identification of high efficiency heating equipment. Manufacturers and developers of heating equipment are encouraged to voluntarily submit heating units and refit devices to BNL for efficiency testing and evaluation. Public dissemination of test reports is used to promote the use of high efficiency equipment, and in addition, BNL provides technical presentations to industrial and educational groups to encourage utilization of energy conservation technology. The present system of voluntary participation in the program by equipment manufacturers can result in characterization of general design features that produce high efficiency performance. However, a cataloguing effort can provide individual brand name performance evaluations enabling consumers to select equipment more effectively, and manufacturers of less efficient heating units to upgrade the performance of their equipment.

The efficiency measurement technique utilized by BNL provides accurate experimental determination of the overall efficiency of heating equipment by considering heat loss incurred during continuous burner operation (steady state operation) and losses incurred during burner-off periods (cyclic operation). The project has an operating hydronic test stand including an electronic data acquisition/controller that allows approximately 24 tests per year, and a forced air heating equipment test stand is also in operation. The testing schedule is complete through the beginning of Fiscal Year 1979 and voluntary participation proposals are continually being received. To meet the demand for testing, an expansion

BNL VOLUNTARY OIL-FIRED EQUIPMENT TEST PROGRAM

	Participants	Tests Completed
Burner/Boiler (Matched/Available)	7	2
Burner/Boiler (Prototypes)	5	3
Burner/Furnaces (Matched/Available)	3	2
Burners (High Speed Ret. Head)	4	2
Burners (Prototypc)	3	0
Burners (Prototype) Low Input Variable Firing Rate	8	3
Vent Dampers	3	1
Off-Cycle Air Flow Control (Prototype)	1	0
Flue Heat Exchangers	6	1
Operating Controls	1	0
Domestic Water Heater	1	1
Oil-Water Emulsion (Prototype)	2	1
Humidified Combustion	4	1
Fuel Additives	2	0
	50	17

	Actions	Tests Completed
Refit and Special Projects	10	7

program was approved to include the construction of one additional hydronic test stand, and the performance of characterization testing in accordance with a proposed schedule.

Twenty-four tests were performed during Fiscal Year 1977, including the evaluation of seven refit modifications. Findings Reports and Annual Fuel Use and Efficiency evaluations were produced, and preliminary tests were performed on prototype burners to determine concept feasibility. Two Findings Reports files have been assembled. One file will be kept at BNL and the second at the DOE Branch Office in Washington. These files will be used by both project offices to disseminate test reports, and serve to maintain a record of both equipment Findings Reports and refit option topical reports.[2] Many oral presentations of the project work were provided by the program staff.

II. EFFICIENCY TEST RESULTS - OIL-FIRED RESIDENTIAL HEATING EQUIPMENT

A. Hydronic (Hot Water) Heating Equipment

This section provides efficiency test results performed at BNL on commercially available oil-fired hot water heating equipment. The various heating units tested will be categorized according to design features such as wet or dry base, and cast iron or steel construction. This does not imply that all units of similar design will perform with the same efficiency. Instead, this classification is intended to illustrate that there can be considerable variation in performance related to particular design features. As the testing program continues, we intend to identify those design characteristics that most frequently produce high levels of efficiency.

1. <u>Dry-Base, Single Pass, Vertical Fire-Tube Steel Boiler Fired by a Non-Retention Head Burner</u>

The dry base feature refers to the fact that the combustion chamber is not surrounded by water, and is separate from the heat transfer section. Generally, this design characteristic leads to increased heat loss and lower efficiency. The single pass, vertical fire-tube steel design, allows for only one pass of the hot combustion gases through the heat transfer section as opposed to a multiple-pass system.

The combination of dry base and single pass design is generally less costly to produce and, therefore, commonly encountered in the field. One purpose of the current project is to provide sufficient quantitative information to justify investment in highly efficient equipment and refit options in order to reduce operating costs by reduced fuel consumption. From the data to be presented, it is obvious that in many cases installation of high efficiency equipment is cost effective for the homeowner, with payback periods on the order of two years.

The test results for the dry-base, single pass boiler include the steady state efficiency as a function of boiler water outlet temperature as presented in Figure 1a. The steady state efficiency at 180°F outlet temperature is 69.2%, including jacket heat losses measured to be 10%. The effect of off-cycle heat losses can be seen in Figure 1b showing the overall efficiency as a function of burner fractional "on" time (heat load). Note that as the burner fractional "on" time is reduced (burner "off" time is increased), the overall efficiency decreases corresponding to increased heat loss during the burner off-cycle. The overall efficiency reaches zero at a burner fractional "on" time of approximately 4%, corresponding to boiler standby losses occurring when space heat and domestic hot water are not required, and the burner operates to replace the heat lost from the boiler during the burner off-period.

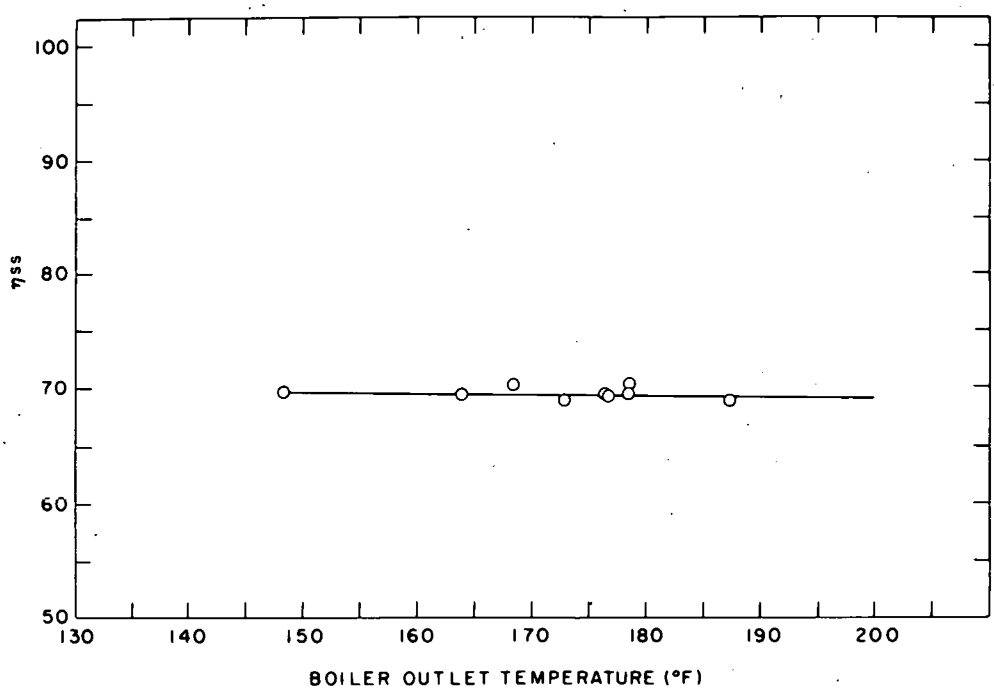


Figure 1a. Dry base single pass steel boiler steady state efficiency (η_{ss}) vs. boiler outlet temperature (°F)

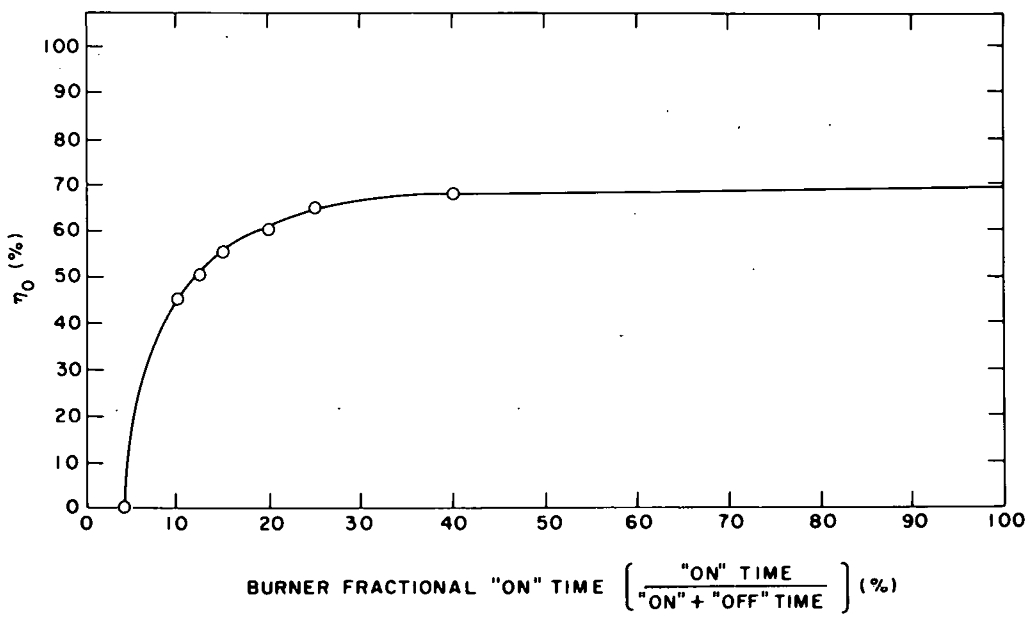


Figure 1b. Dry base single pass steel boiler overall efficiency (η_o) vs. burner fractional "on" time (%)

Table 1a. (Dry-base single pass steel boiler with non-retention head burner) 50,000 BTU per hour

HEATING UNIT: DRY-BASE SINGLE PASS STEEL BOILER
LOCATION: NEW YORK CITY
DESIGN HEAT LOAD: 50000.0 BTU PER HOUR
ROOM TEMP: 68.0°
OUTSIDE DESIGN TEMP: 0.0°

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.631	.555	.488	.435
	ANNUAL FUEL USAGE (GAL/YEAR)	1156.	1316.	1497.	1680.
	DESIGN OIL FLOW RATE (GPH)	.521	1.042	1.563	2.085
40.	SEASONAL EFFICIENCY	.639	.567	.502	.450
	ANNUAL FUEL USAGE (GAL/YEAR)	1274.	1437.	1623.	1812.
	DESIGN OIL FLOW RATE (GPH)	.536	1.071	1.607	2.143
80.	SEASONAL EFFICIENCY	.646	.577	.514	.463
	ANNUAL FUEL USAGE (GAL/YEAR)	1393.	1558.	1749.	1942.
	DESIGN OIL FLOW RATE (GPH)	.550	1.100	1.650	2.200
120.	SEASONAL EFFICIENCY	.652	.586	.525	.475
	ANNUAL FUEL USAGE (GAL/YEAR)	1511.	1679.	1874.	2073.
	DESIGN OIL FLOW RATE (GPH)	.565	1.129	1.694	2.258

Table 1b. (Dry-base single pass steel boiler with non-retention head burner) 25,000 BTU per hour

HEATING UNIT: DRY-BASE SINGLE PASS STEEL BOILER
LOCATION: NEW YORK CITY
DESIGN HEAT LOAD: 25000.0 BTU PER HOUR
ROOM TEMP: 68.0°
OUTSIDE DESIGN TEMP: 0.0°

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.631	.555	.488	.435
	ANNUAL FUEL USAGE (GAL/YEAR)	578.	658.	748.	840.
	DESIGN OIL FLOW RATE (GPH)	.261	.521	.782	1.042
40.	SEASONAL EFFICIENCY	.646	.577	.514	.463
	ANNUAL FUEL USAGE (GAL/YEAR)	696.	779.	874.	971.
	DESIGN OIL FLOW RATE (GPH)	.275	.550	.825	1.100
80.	SEASONAL EFFICIENCY	.656	.594	.535	.485
	ANNUAL FUEL USAGE (GAL/YEAR)	815.	900.	1000.	1102.
	DESIGN OIL FLOW RATE (GPH)	.290	.579	.869	1.158
120.	SEASONAL EFFICIENCY	.664	.607	.551	.503
	ANNUAL FUEL USAGE (GAL/YEAR)	934.	1020.	1125.	1232.
	DESIGN OIL FLOW RATE (GPH)	.304	.608	.912	1.216

The results of these efficiencies integrated over an entire heating season are presented in Tables 1a and 1b (see Section IV for details). The annual fuel consumption in gallons of oil per year and seasonal efficiency (S.E.) vary with many parameters including domestic hot water usage in gallons per day, and the nondimensional fuel firing rate, α. As the firing rate is increased (with all other parameters held constant), annual fuel use increases corresponding to smaller burner fractional "on" times, and increased heat loss during the off-cycle. For a design heat load of 50,000 Btu per hour in New York City, with an inside design temperature of 68^{o}F, 40 gallons per day of domestic hot water, and an overfiring ratio of 2 (100% overfired), the annual fuel use is 1440 gallons and the overall seasonal efficiency is 56.7%. Only 56.7% of the heat available from the fuel oil consumed is delivered to the heating load and the remaining 43.3% is lost during the burner "on" and "off" periods. In contrast, seasonal efficiencies as high as 75% have been measured for commercially available equipment of more efficient design under the same operating conditions, resulting in annual fuel savings of 24%.

2. <u>Dry-Base, Cast Iron Boiler Fired by a Retention Head Burner</u>

Test results for this boiler cannot be directly compared to those for the steel boiler (II-A-1) because a retention-head burner was used in the cast iron boiler and the steel boiler was fired by a low speed non-retention head burner. The use of retention-head burners can have a significant effect on the performance of a particular boiler (see Section II-C-2). The cast iron boiler and retention-head burner combination produced higher steady state and overall thermal efficiencies than the previous burner-boiler combination (II-A-1). The measured steady state efficiency curve of Figure 2a indicates a value of 74.8% at 180^{o}F, with jacket losses contributing 12%. The overall efficiency data of Figure 2b provide the test results for both the cast iron and steel boiler. Annual fuel consumption and seasonal efficiency are presented as a function of important variables in Table 2. For a design heat load of 50,000 Btu per hour in New York City, with an inside design temperature of 68°F, 40 gallons per day of domestic hot water, and an overfiring ratio of 2 (100% oversized), the annual fuel use is 1260 gallons and the seasonal efficiency is 64.7%. This corresponds to a 12% annual fuel savings over the burner-boiler test results of Section II-A-1.

3. Wet-Base, Two Pass, Horizontal Fire-Tube Steel Boiler Fired by an Induced Draft Burner

This unique boiler-burner combination is among the most efficient of all units tested at BNL. The combustion chamber is completely surrounded by boiler water, and combustion gases pass through the combustion chamber and turn back through horizontal steel fire-tubes. An exhaust fan located at the boiler flue gas outlet provides induced draft for the burner combustion air. The steady state efficiency of this boiler is 81.2%, with jacket heat losses of 1% (see Figure 3a). Overall efficiency data is plotted with the steel boiler of Section II-A-1 for comparison (Figure 3b). Note the high level of overall efficiency maintained by the two-pass boiler (upper curve) resulting from reduced off-cycle heat loss. Table 3 provides the results of the Annual Fuel Use and Efficiency computation. For a heat load of 50,000 Btu per hour in New York City, with an inside design temperature of 68°F, 40 gallons per day of domestic hot water, and an overfiring ratio of 2 (100% oversized), the annual fuel use is 1090 gallons, with a seasonal efficiency of 74.8%. This corresponds to a 24% reduction in fuel consumption compared to the boiler of Section II-A-1.

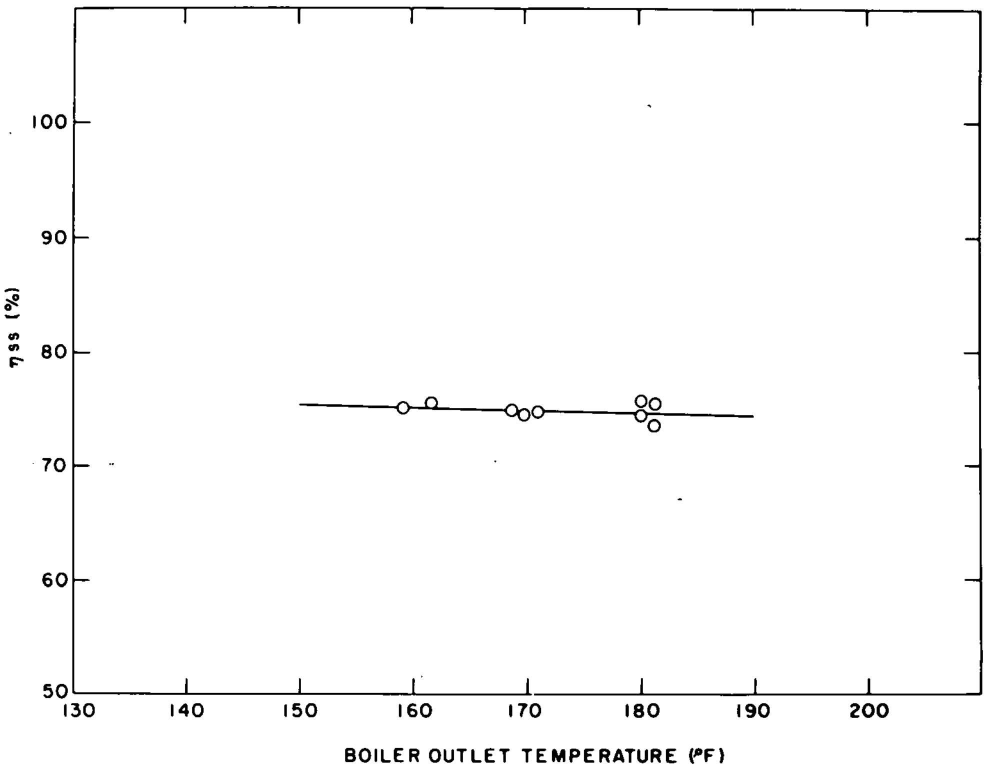


Figure 2a. Dry base cast iron boiler steady state efficiency (η_{ss}) vs. boiler outlet temperature (^{o}F)

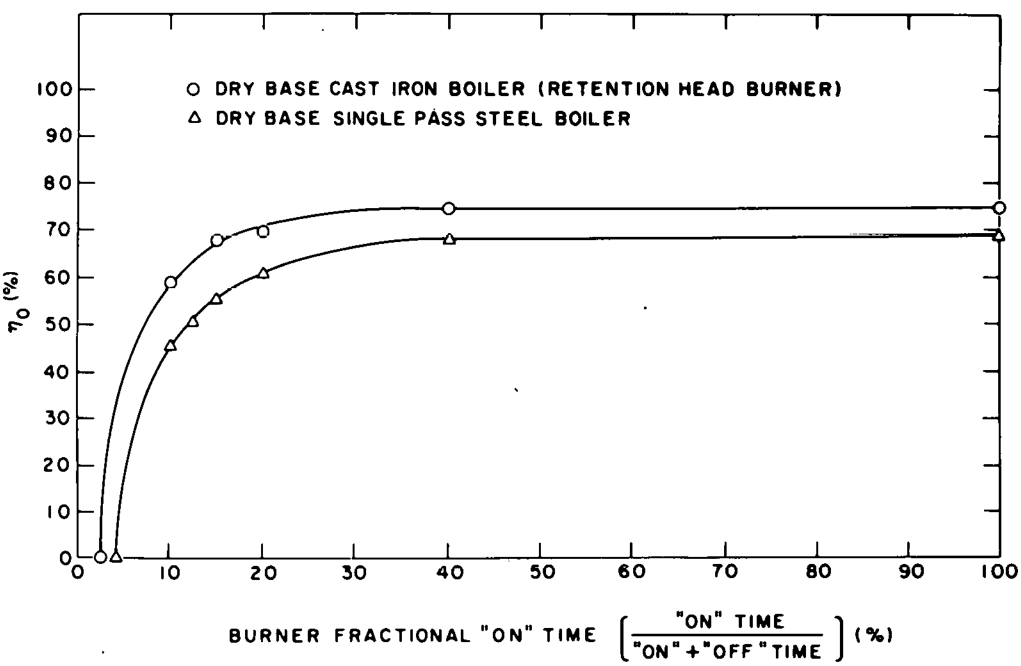


Figure 2b. Overall efficiency (η_o) vs. burner fractional "on" time (%)

Table 2. Boiler II-A-2 dry-base, cart iron boiler, fired by a retention head burner

HEATING UNIT: II-A-2 DRY-BASE, CAST IRON BOILER
LOCATION: NYC
DESIGN HEAT LOAD: 50,000 BTU
ROOM TEMP: 68.0
OUTSIDE DESIGN TEMP: 0.0

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.705	.642	.587	.542
	ANNUAL FUEL USAGE (GAL/YEAR)	1035.	1136.	1243.	1348.
	DESIGN OIL FLOW RATE (GPH)	.482	.964	1.446	1.929
40.	SEASONAL EFFICIENCY	.708	.647	.593	.547
	ANNUAL FUEL USAGE (GAL/YEAR)	1152.	1260.	1375.	1489.
	DESIGN OIL FLOW RATE (GPH)	.496	.991	1.487	1.982
80.	SEASONAL EFFICIENCY	.712	.655	.603	.559
	ANNUAL FUEL USAGE (GAL/YEAR)	1263.	1374.	1492.	1608.
	DESIGN OIL FLOW RATE (GPH)	.509	1.018	1.527	2.036
120.	SEASONAL EFFICIENCY	.716	.662	.612	.570
	ANNUAL FUEL USAGE (GAL/YEAR)	1374.	1487.	1609.	1728.
	DESIGN OIL FLOW RATE (GPH)	.522	1.045	1.567	2.089

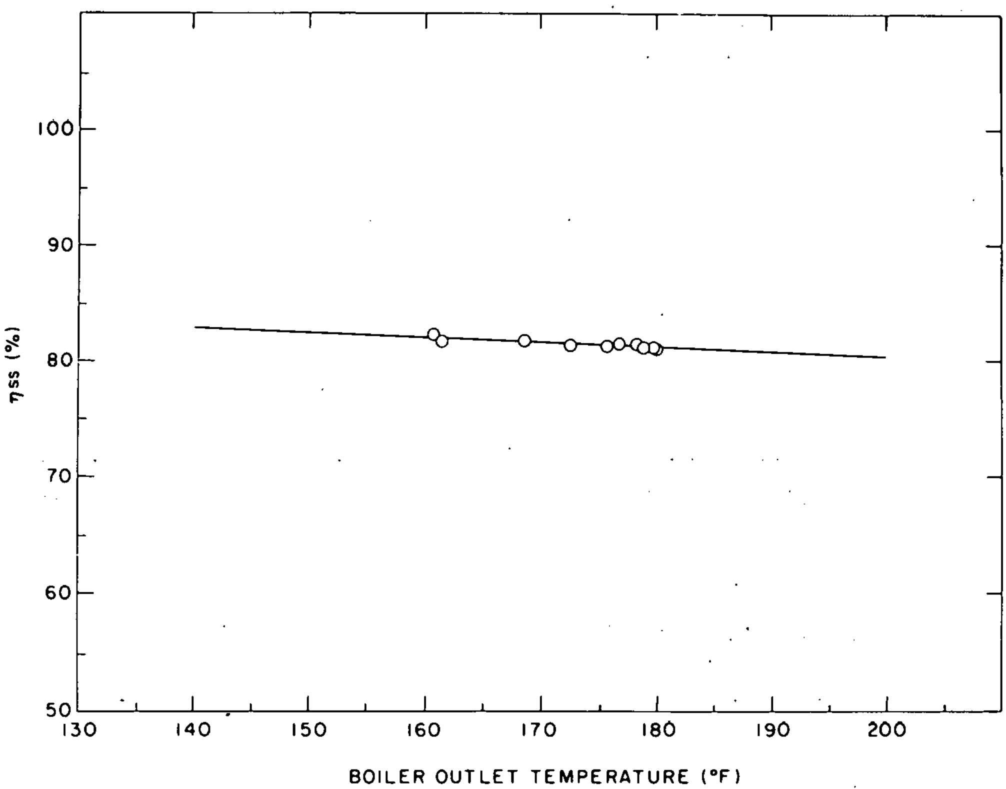


Figure 3a. Wet base two pass steel boiler steady state efficiency (η_{ss}) vs. boiler outlet temperature (^{O}F)

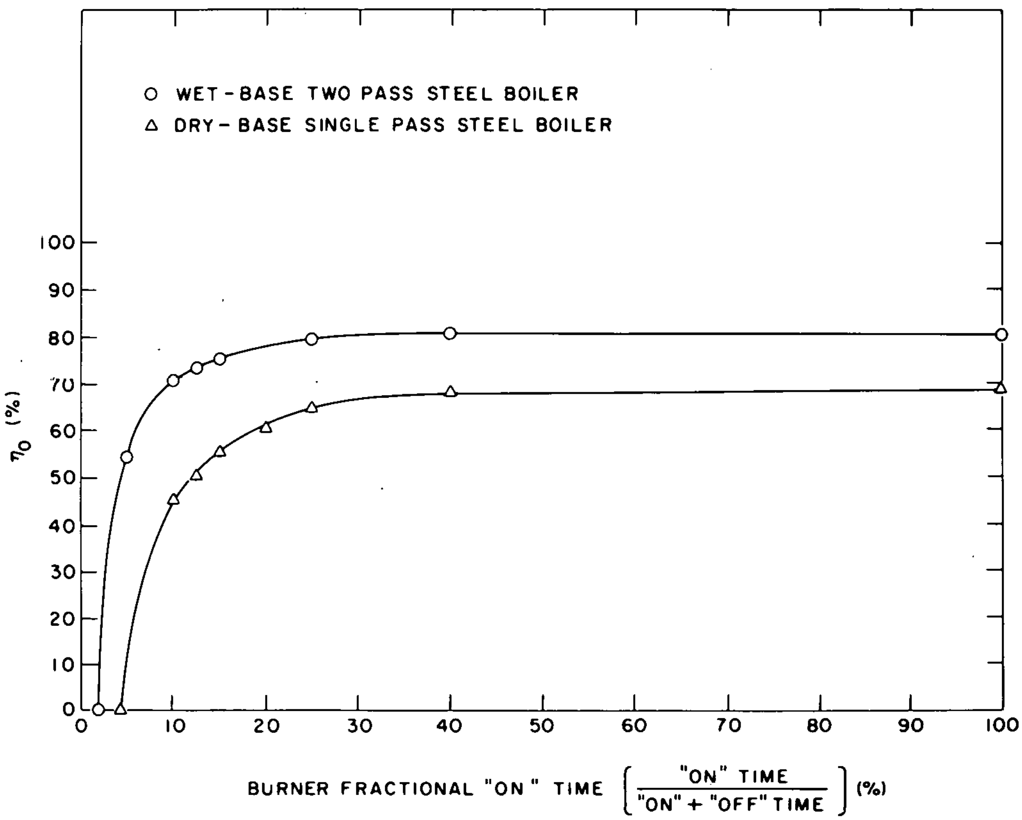


Figure 3b. Overall efficiency (η_o) vs. burner fractional "on" time (%)

Table 3. Boiler II-A-3 - wet-base, two pass, horizontal fire-tube steel boiler fired by an induced draft burner

HEATING UNIT: II-A-3 WET-BASE, TWO PASS, HORIZONTAL FIRE-TUBE STEEL BOILER
LOCATION: NYC
DESIGN HEAT LOAD: 50,000 BTU
ROOM TEMP: 68.0
OUTSIDE DESIGN TEMP: 0.0

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.785	.741	.697	.657
	ANNUAL FUEL USAGE (GAL/YEAR)	930.	985.	1848.	1112.
	DESIGN OIL FLOW RATE (GPH)	.444	.888	1.332	1.777
40.	SEASONAL EFFICIENCY	.789	.748	.706	.668
	ANNUAL FUEL USAGE (GAL/YEAR)	1033.	1089.	1154.	1220.
	DESIGN OIL FLOW RATE (GPH)	.456	.913	1.369	1.825
80.	SEASONAL EFFICIENCY	.792	.755	.715	.678
	ANNUAL FUEL USAGE (GAL/YEAR)	1135.	1192.	1259.	1326.
	DESIGN OIL FLOW RATE (GPH)	.469	.938	1.406	1.875
120.	SEASONAL EFFICIENCY	.795	.760	.722	.587
	ANNUAL FUEL USAGE (GAL/YEAR)	1238.	1295.	1364.	1433.
	DESIGN OIL FLOW RATE (GPH)	.481	.962	1.443	1.925

4. Wet-Base, Cast Iron Boiler, Fired by a Retention Head Burner

The overall efficiency of this boiler is similar to the efficiency of the wet-base, two pass steel boiler of Section II-A-3. The steady state efficiency at a $180^{o}F$ outlet temperature is 81.0% including a jacket heat loss of 1%. The overall efficiency data are presented in Figure 4b. The annual fuel consumption and seasonal efficiency for the same variables as the previous sections are 1084 gallons and 75.2%.

5. Conclusions

The results obtained for commercially available equipment must not be overgeneralized . There are many variables that will affect the efficiency of a particular boiler, including burner type, quality of installation and adjustment of controls. In the laboratory, all burners were adjusted to peak efficiency with a zero to one smoke number. In actual field installations, there can be a wide range of burner adjustments from the optimum setting. The test results are categorized in an attempt to identify design features that can contribute to improved efficiency. It is not stated or implied that all boilers of the type in Section II-A-4 are better than all boilers of the type described in Section II-A-1 and II-A-2. Each burner-boiler combination must be evaluated on an individual basis. However, as the test program continues and more efficiency information becomes available, it may be possible to generalize about the specific design characteristics that produce the highest operating efficiencies. For example, wet-base design appears to be a general feature that can produce higher efficiency than dry-base design. Further testing is required to substantiate this fact, and to develop other generalized results.

B. "New" Technology Burners

This section presents the results of tests performed on oil burners featuring design characteristics not found in

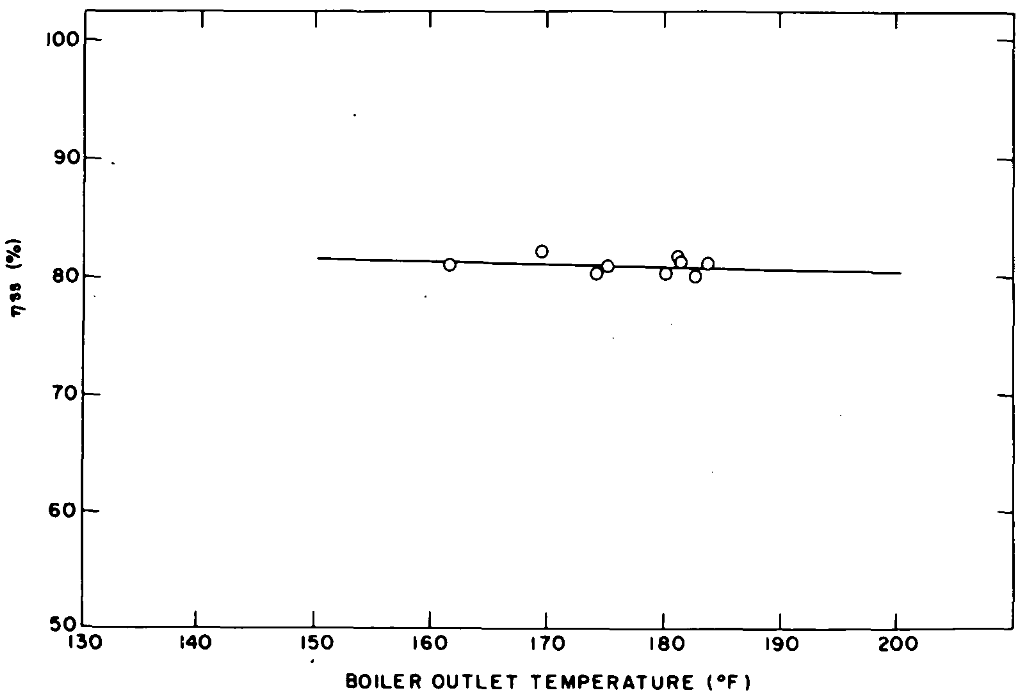


Figure 4a. Wet base cast iron boiler steady state efficiency (η_{ss}) vs. boiler outlet temperature (°F)

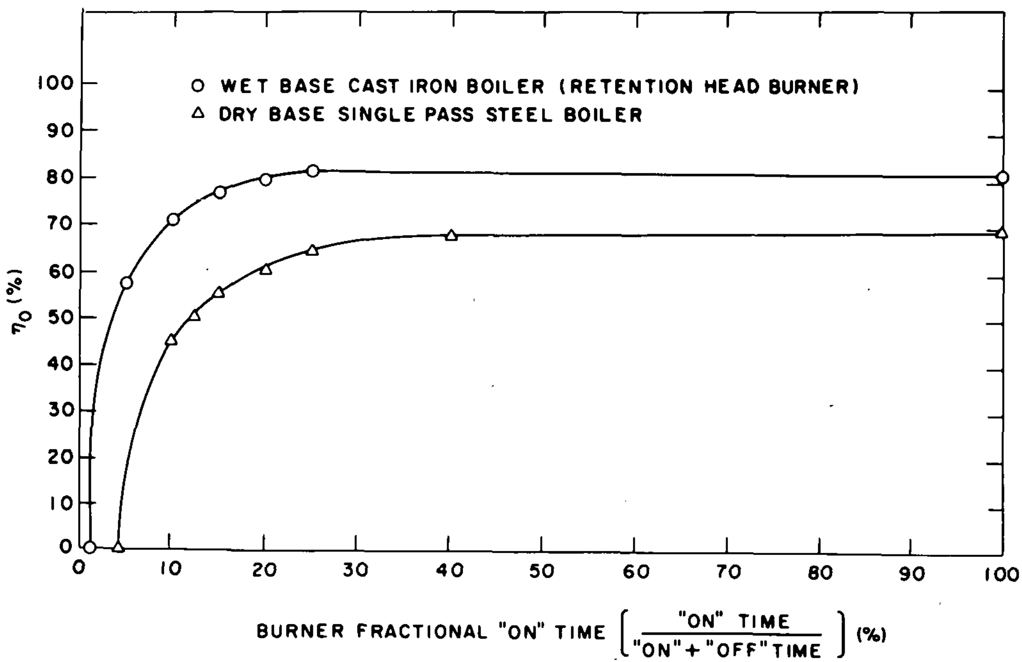


Figure 4b. Overall efficiency (η_o) vs. burner fractional "on" time (%)

Table 4. **Boiler II-A-4 - wet-base, cast iron boiler, fired by a retention head burner**

HEATING UNIT: II-A-4 WET-BASE, CAST IRON BOILER
LOCATION: NYC
DESIGN HEAT LOAD: 50,000 BTU
ROOM TEMP: 68.0
OUTSIDE DESIGN TEMP: 0.0

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.784	.744	.702	.662
	ANNUAL FUEL USAGE (GAL/YEAR)	931.	981.	1041.	1103.
	DESIGN OIL FLOW RATE (GPH)	.445	.890	1.336	1.781
40.	SEASONAL EFFICIENCY	.789	.752	.711	.673
	ANNUAL FUEL USAGE (GAL/YEAR)	1033.	1084.	1145.	1210.
	DESIGN OIL FLOW RATE (GPH)	.458	.915	1.373	1.830
80.	SEASONAL EFFICIENCY	.792	.758	.720	.684
	ANNUAL FUEL USAGE (GAL/YEAR)	1136.	1187.	1249.	1316.
	DESIGN OIL FLOW RATE (GPH)	.470	.940	1.410	1.880
120.	SEASONAL EFFICIENCY	.795	.763	.727	.693
	ANNUAL FUEL USAGE (GAL/YEAR)	1238.	1290.	1353.	1422.
	DESIGN OIL FLOW RATE (GPH)	.482	.965	1.447	1.929

conventional equipment. Included are blue flame (exhaust recirculating), air atomizing, and ultrasonic atomizing burners. The models that were tested included prototypes and feasibility models. In the case of the blue flame burner, both steady state and cycle efficiency tests were performed. In the cases of the air atomizing and ultrasonic burners, only steady state efficiency tests were performed and future tests with these burners are expected to include measurement of off-cycle heat loss.

1. <u>Blue Flame Burner, Fired into a Wet-Base, Two Pass, Horizontal Fire-Tube Boiler</u>

A prototype blue flame burner featuring internal combustion product recirculation was supplied to BNL in a wet-base, two pass steel boiler (similar to the boiler tested in Section II-A-3). Both steady state and cycle efficiency tests were performed, and the results are presented in Figure 5. Mild pulsations were observed during steady state operation, and their amplitude increased slightly as testing proceeded. It was observed that the combustion-related pulsations had a tendency to improve steady state efficiency by a small amount. The results of the seasonal efficiency computation indicated efficiency values slightly below those for the boiler of Section II-A-3 (see Table 5).

2. <u>Air-Atomizing Oil Burner, Fired into a Dry-Base, Single Pass Vertical Fire Tube Boiler</u>

A feasibility model air-atomizing oil burner was supplied to BNL for testing. The burner was installed in a dry-base, single pass, steel boiler, and steady state efficiency tests were performed. One important feature of air-atomizing burners is the ability to fire below 0.5 gallons per hour and maintain a stable and efficient combustion process. Unfortunately, the particular burner that was supplied could not be fired below 0.6 gph because of its oversized combustion air fan and ineffective

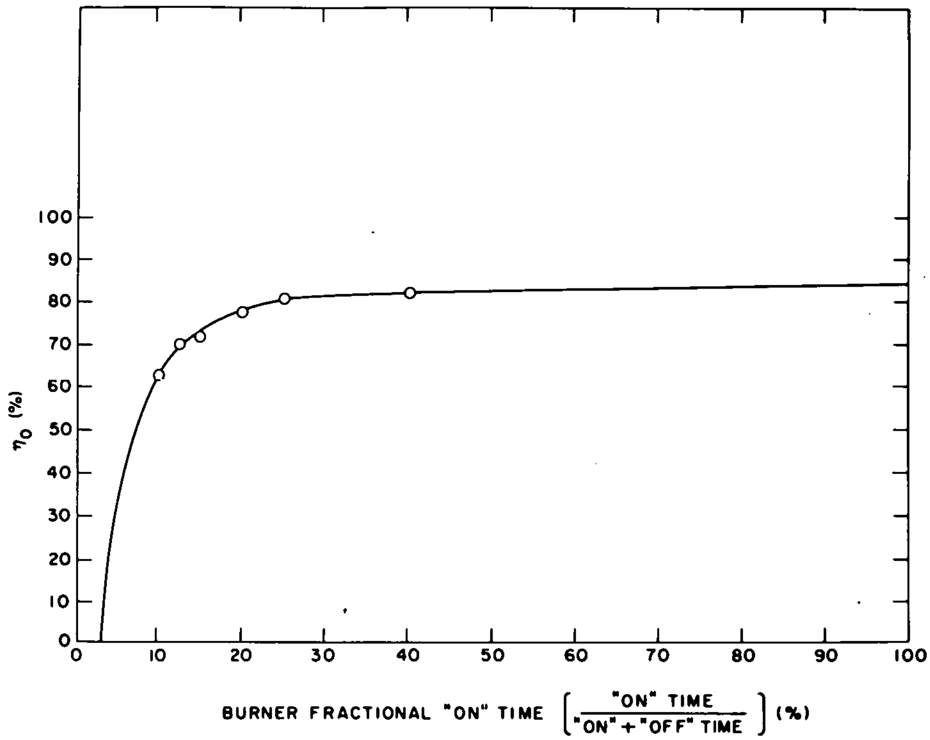


Figure 5. Blue flame burner in two pass steel boiler efficiency (η_o) vs. burner fractional "on" time (%)

Table 5. Boiler II-B-1 - blue flame burner, fired into a wet-base, two pass, horizontal fire-tube boiler

HEATING UNIT: II-B-1 BLUE FLAME BURNER
LOCATION: NYC
DESIGN HEAT LOAD: 50,000 BTU
ROOM TEMP: 68.0
OUTSIDE DESIGN TEMP: 0.0

DOMESTIC HOT WATER (GAL PER DAY)		1.	2.	3.	4.
0.	SEASONAL EFFICIENCY	.785	.710	.646	.593
	ANNUAL FUEL USAGE (GAL/YEAR)	930.	1028.	1129.	1231.
	DESIGN OIL FLOW RATE (GPH)	.423	.847	1.270	1.693
40.	SEASONAL EFFICIENCY	.791	.720	.659	.508
	ANNUAL FUEL USAGE (GAL/YEAR)	1030.	1131.	1236.	1340.
	DESIGN OIL FLOW RATE (GPH)	.435	.870	1.305	1.748
80.	SEASONAL EFFICIENCY	.797	.730	.672	.622
	ANNUAL FUEL USAGE (GAL/YEAR)	1129.	1232.	1340.	1447.
	DESIGN OIL FLOW RATE (GPH)	.447	.894	1.340	1.787
120.	SEASONAL EFFICIENCY	.802	.739	.682	.634
	ANNUAL FUEL USAGE (GAL/YEAR)	1228.	1333.	1443.	1553.
	DESIGN OIL FLOW RATE (GPH)	.459	.917	1.376	1.834

combustion air damper. In future tests, this operational constraint will be removed and firing below 0.5 gph will be evaluated.

The results of steady state efficiency tests are presented in Figure 6a as a function of fuel firing rate. As expected, the steady state efficiency increases as the fuel oil firing rate is reduced, corresponding to lower stack gas temperatures. This effect was observed during previous tests[1] using the same boiler fired by a conventional burner. Cycle efficiency tests could not be performed because of the pre-prototype nature of the burner supplied to BNL, and the seasonal efficiency could not be computed. Future tests will include both low fuel flow rates (below 0.5 gph) and cyclic performance, to better evaluate the total energy saving potential of this air-atomizing burner over conventional mechanical-atomizing burners.

3. Ultrasonic Atomizing Oil Burner, Fired into a Dry-Base, Cast Iron Boiler

A prototype ultrasonic atomizing oil burner was supplied to BNL in a dry-base, cast iron boiler. The requirement for manual burner operation limited the tests to steady state efficiency measurement only. The burner was capable of firing below 0.5 gph and the steady state efficiency is plotted as a function of firing rate in Figure 6b. With modification of the burner from manual to automatic start-up, future tests will include measurement of off-cycle heat loss and overall efficiency. The use of automatic variable firing rate burners provide the possibility for improved overall efficiency by use of modulating burners in place of conventional on-off burner systems, resulting in increased steady state and cyclic efficiencies.

C. Refit Modifications for Efficiency Improvement of Oil-Fired Boilers

An updated list of refit modifications for improving the efficiency of conventional heating equipment is presented in

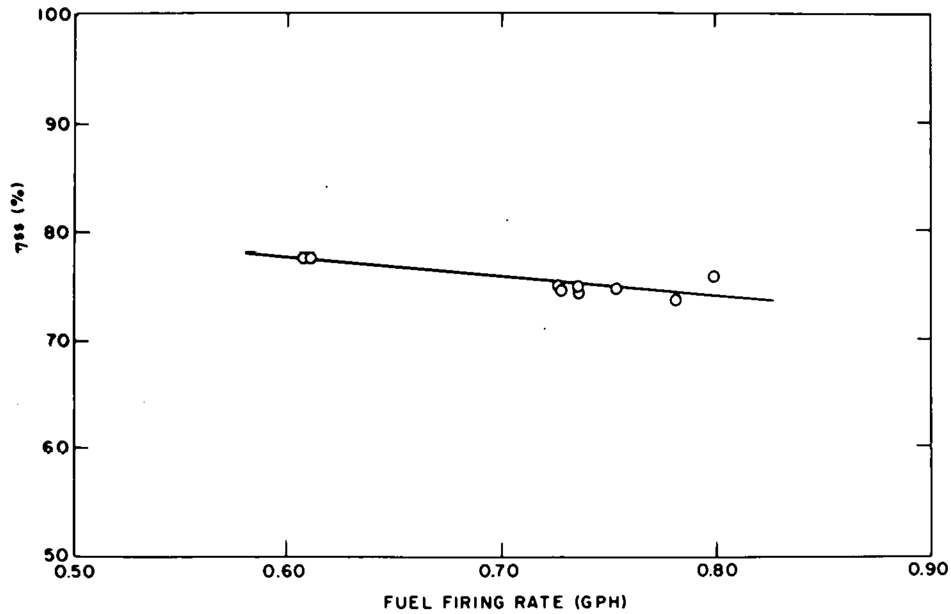


Figure 6a. Air atomizing burner in dry base single pass steel boiler steady state efficiency (η_{ss}) vs. fuel firing rate (GPH)

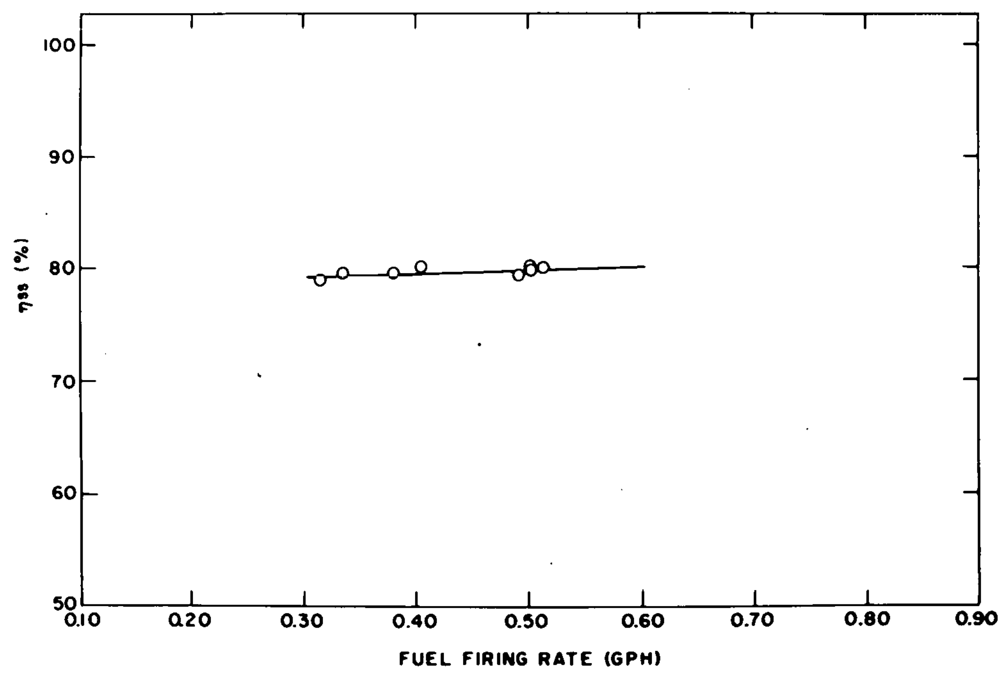


Figure 6b. Ultrasonic burner in dry base cast iron boiler steady state efficiency (η_{ss}) vs. fuel firing rate (GPH)

ERRATA

For BNL 50853, which should have appeared between pages 22 and 23.

Table 6

REFIT MODIFICATIONS FOR EFFICIENCY IMPROVEMENT
OF
OIL-FIRED BOILERS

	Refit Action	Estimated Fuel Saving (%)	Approx. Cost ($)	Payback Period (Years)
1.	Reduced Burner Firing Rate (by 25%)	8[a]	0[b] - 25	0 - 0.4
2.	Boiler Water Temperature Reduction (35°F)	5	0[c] - 20	0 - 0.5
3.	Thermostat Setback[d] - Manual Adjustment	8	0	0
4.	Thermostat Setback[d] - Automatic	8	80	1.3
5.	Burner Efficiency Adjustment	3	0[b] - 30	0 - 1.3
6.	Boiler Fire-Tube Turbulators	5[e]	50	1.3
7.	Retention-Head Burner[f]	16[a]	250	2.1
8.	Vent Damper[g]	10[a]	200	2.7
9.	Stack Heat Reclaimer[g] (Economizer)	15[a]	350	3.1
10.	Low Input/Variable Firing Rate Burners[h]	20	∿500	3.3
11.	Ducting Combustion Air from Outdoors[i]	0 - 3[j]	100	4.4
12.	Modern High Efficiency Burner-Boiler	24[a]	1500	8.3
13.	Blue Flame Burner-Boiler	21	1500	9.5
14.	Outdoor Boiler Installation	0 - 10[k]	-	-
15.	Combustion Air Humidification	1	200	26
16.	Water-Fuel Oil Emulsion	0	-	-

Notes: Savings from Refit Actions are not additive. Payback period is based on 1500 gallons per year fuel use at $.50 per gallon.

a. Based on dry-based steel boiler with non-retention head burner

b. May be included as part of annual servicing

c. Manual adjustment by homeowner

d. Setback of 10°F for 8 hours per day

e. Applicable only in boilers where turbulators are absent

f. Firing rate reduction should accompany burner installation

g. Possible safety hazard exists - long term testing required

h. Commercial equipment not available at this time

i. Including inlet air damper for burner off-cycle

j. Will vary depending on boiler location in structure

k. Will vary with boiler - testing required

Table 6. Many of these refit actions have been tested in the Burner-Boiler/Furnace Efficiency Test Facility at BNL as part of the ongoing measurement program. This section provides test results and discussions for the refit modifications that have been tested.

1. Reduced Fuel Firing Rate

The simplest and most cost-effective refit modification is reducing the firing rate of hot water heating units. It is not uncommon to find units in the field that are more than 100% overfired relative to the peak design heat load.[1,3] As shown in Table 1a of Section II-A-1, overfiring causes increased fuel consumption by reducing the burner fractional "on" time and increasing off-cycle heat losses. Further discussion of overfiring is found in Section IV-C. One method of minimizing off-cycle heat loss is to replace the fuel oil nozzle with one of smaller size followed by readjustment of the combustion air setting. The result is to operate for longer burner "on" periods at a lower fuel flow rate and decrease annual fuel consumption. Field verification of this procedure has been observed by several investigators.[3,4]

Data for the boiler of Section II-A-1 can be used to demonstrate fuel savings resulting from reduced overfiring. Assuming an initial overfiring of 100% ($\alpha = 2$), annual fuel oil consumption was observed to be 1440 gallons (Table 1a). Reducing the fuel flow rate 25% (from 1.07 gph to .80 gph) results in overfiring of 50% and reduction of annual fuel use by 80 gallons (5½%). Also, the steady state efficiency increases as the firing rate is reduced, and a 25% reduction of firing rate would result in an increase of steady state efficiency of 2%.[1] The total fuel savings resulting from reduced firing rate is 110 gallons of oil per year (7½%) or $50 per year for oil at 50¢ per gallon. Replacement fuel nozzles cost approximately $1.50, and nozzle replacement can be incorporated into the annual equipment tune-up. Obviously, this is a cost-effective refit modification.

The main disadvantage of reduce firing rate is that longer time periods are required to raise the boiler water temperature, and heating units utilizing "tankless coils" for supply of domestic hot water will not be able to satisfy the same peak load. The rate at which acceptable domestic hot water can be supplied will diminish. In cases where domestic hot water supply becomes a problem, hot water storage tanks can be used to satisfy the peak load. This problem, obviously, does not exist for heating units supplying only space heat or when domestic water is heated using a separate burner and storage unit.

2. Retention-Head Oil Burner Refit

Measurements were performed to determine the efficiency improvement of typical residential hot water boilers produced by replacement of oil burners of low speed, non-retention-head design with burners of modern design (high speed, retention head). It was found that refit retention head burners resulted in reduced annual fuel consumption (by 16.6%) for a single pass, dry-base, vertical fire-tube boiler.

Overall efficiency versus burner fractional "on" time is plotted in Figure 7 for the three burners tested. These curves illustrate the increase in overall efficiency due to the installation of retention-head burners. Burners (A) and (B) are of retention-head design, while Burner (C) is of conventional non-retention-head design. A tabular listing of the data obtained appears in Table 7.

From these data, the amount of fuel consumed over an entire year can be determined for each burner using the BNL Annual Fuel Use and Efficiency computation. Temperature data for New York City was used with a design heat load of 50,000 Btu per hour, and 40 gallons per day of domestic hot water. Based

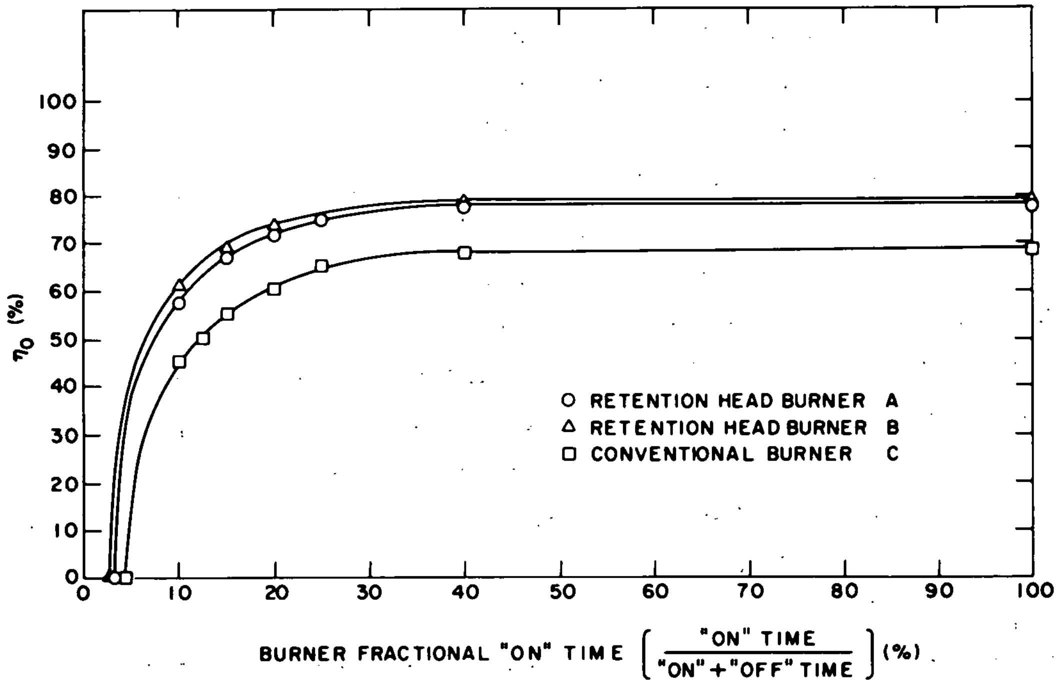


Figure 7. Retention-head burners refitted to a dry-base single pass vertical fire-tube boiler

Table 7

Burner Fractional "On" Time (%)	Overall Efficiency (%) Burner (A)	Burner (B)	Burner (C)
5	37.5	42.0	18.8
10	48.0	61.2	45.5
15	66.7	69.0	55.5
20	71.5	73.5	61.5
25	74.9	76.2	65.0
30	76.5	77.5	66.2
40	78.0	78.8	68.0
60	78.0	79.0	69.2
100	78.0	79.0	69.2

on field data, a fuel firing rate of approximately 1.1 gallon per hour can be assumed representing overfiring of about 100%. The results of this analysis are shown in Table 8a.

In Table 8b, a comparison of burners is presented, and it can be seen that for the particular boiler tested average fuel savings of 16.6% (or 240 gallons) can be achieved by installing a retention-head burner as a replacement for a low speed non-retention-head burner. Also, the overall efficiency (η_o) increase can be separated into two parts, improvement of steady state performance and improvement of cyclic performance. The steady state component accounts for two-thirds of the change in overall efficiency, while the cyclic efficiency improvement accounts for one-third of the total change. The improvement in steady state efficiency results from the ability of retention-head burners to operate with less excess air than conventional burners, thus decreasing the total stack heat loss. Excess air must be heated from ambient temperature to the elevated stack temperature producing a loss in efficiency. The increase in cyclic performance is due to added restriction to air flow through the burner during its off-cycle, reducing the total convective heat loss from the boiler.

It should be noted that the non-retention-head burner tested was new, in good condition, and well maintained. Conventional burners in field installations may not be in good condition, therefore, the quantity of fuel saved annually may be greater than computed. Also, boiler design is an important factor regarding optimized efficiency performance, and overall fuel savings is directly related to the combination of burner and boiler designs. In the field, it is recommended that accurate steady state efficiency tests be performed before and after burner replacement to ensure favorable results.

The heating equipment tested in the laboratory consisted of two flame retention-head burners, a non-retention-head burner,

Table 8a

Location:	New York City
Design Heat Load:	50,000 BTUH
Room Temperature:	$68^{o}F$
Outside Design Temp:	$0^{o}F$
Hot Water Usage:	40 Gallons Per Day
Overfiring Ratio:	2

	Burner (A)	Burner (B)	Burner (C)
Design Oil Firing Rate (GPH)	.95	.94	1.07
Annual Fuel Usage (Gal/Year)	1220	1180	1440
Seasonal Efficiency (η_o) (%)	67.1	68.9	56.7
Measure Steady State Efficiency (η_{ss}) (%)	78.0	79.0	69.2
Average Cycle Efficiency (η_c) (%)	86.0	87.2	81.9

Note: $\eta_{cycle} = \eta_{overall}/\eta_{steady\ state}$

Table 8b

	Percent Change: Burner (C) to (A)	Percent Change: Burner (C) to (B)	Average
Fuel Consumption	- 15.3	- 18.0	- 16.6
Steady State Efficiency	11.2	12.4	11.8
Cycle Efficiency	4.8	6.1	5.4

and a conventional residential boiler of single pass, vertical fire-tube, dry-base design. The non-retention-head burner (C) was tuned to provide the maximum CO_2 percentage using a Bacharach smoke spot number of 1 as a criterion for the fuel-air setting. The same procedure was followed for adjusting Burners (A) and (B). Details of the test procedure can be found in Section III-A.

3. Effect of Boiler Water Temperature on Overall Efficiency

Tests were performed on the boiler of Section II-A-1 to establish the relationship between off-cycle boiler heat loss and reduced boiler temperature. It was found that reducing the boiler water temperature during burner on-off cycling resulted in substantial fuel savings as a result of decreased burner-off-cycle heat losses. When the average boiler water outlet temperature was lowered from 185°F to 150°F, burner off-cycle heat loss decreased by 30%, corresponding to an increase in overall efficiency of 6%.

Overall efficiency versus burner fractional "on" time is plotted in Figure 8 for the two average boiler temperatures tested. From these curves it is evident that heat losses occurring during the burner "off" cycle are reduced by lowering the boiler temperature for all burner fractional "on" times. These data are presented in Table 9.

Table 9

Burner Fractional "On" Time (%)	Cycle Efficiency (%)	
	T = 185°F	T = 150°F
5	18.1	40.1
10	65.0	75.4
15	80.5	87.2
20	88.2	93.1
25	92.8	96.6
30	95.8	98.9
40	98.3	99.5
60	99.2	99.8
100	100.0	100.0

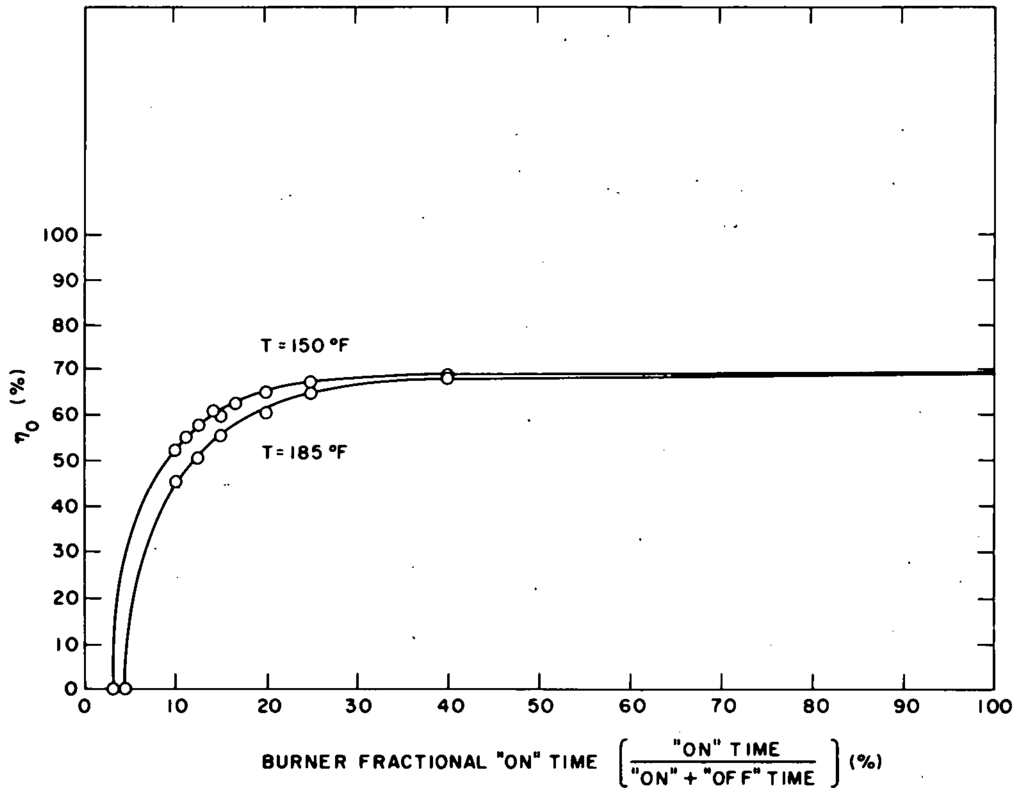


Figure 8. Effect of boiler water temperature on overall efficiency overall efficiency (η_o) vs. burner fractional "on" time (%)

Table 10 can be used to evaluate the reduction in burner off-cycle heat losses resulting from reduced boiler water temperature. The cycle efficiency (η_c) and the non-dimensional off-cycle heat loss (1-$_c$) are shown for the two boiler temperatures for fractional burner "on" times between 5 and 25%.

Table 10

	$\bar{T}$ = 185°F		$\bar{T}$ = 150°F		
Fractional Burner "On" Time	Cycle Efficiency % (η_c)	Non-dimensional "Off" Cycle Heat Loss ($1-\eta_c$)	Cycle Efficiency % (η_c)	Non-dimensional "Off" Cycle Heat Loss ($1-\eta_c$)	Percent Reduction "Off" Cycle Heat Loss (%)
5	18.1	0.812	40.1	0.599	26
10	65.0	0.350	75.4	0.246	30
15	80.5	0.195	87.2	0.128	34
20	88.2	0.118	93.1	0.069	42
25	92.8	0.072	96.6	0.034	53

Note: $\eta_{overall} = \eta_{steady\ state} \times \eta_{cycle}$

An Annual Fuel Use Efficiency analysis has been performed using the BNL computer program to evaluate the actual reduction in annual fuel consumption resulting from lower boiler operating temperature. Temperature data for New York City were used with a design heat load of 50,000 Btu per hour, and 40 gallons of domestic hot water per day. Based on field data, a fuel firing rate of 1.1 gallons per hour can be assumed, representating an overfiring rate of about 100%. The lower boiler water temperature resulted in a reduction of off-cycle heat loss by 30%, producing a 6% increase of overall efficiency (from 57.6% to 61.4%). The total quantity of fuel consumed dropped by 90 gallons per year (from 1440 to 1350) for identical space heat and domestic hot water loads, as a direct result of reduced boiler operating temperature.

A simple illustrative heat flow calculation can be used to approximate the above results. If we assume that the air mass flow rate through the boiler remains constant for the two boiler temperatures (a reasonable approximation), we find that the off-cycle heat loss rates ($\dot{H}$) vary with the boiler water temperature. The off-cycle heat losses are dominated by two processes: off-cycle jacket losses and off-cycle stack losses. Although the precise heat transfer characteristics involved cannot be generalized, we know that both these processes are driven by the difference in temperature between the boiler and the ambient air. Thus, the off-cycle heat losses are the sum of stack and jacket losses.

$$\dot{H} = \dot{m}_{air}\, C_p\, \Delta T + h_j\, \Delta T$$

where:

$\dot{H}$ - is the boiler off-cycle heat loss rate (BTUH)

$\dot{m}_{air}$ - is the air mass flow rate through the boiler during the "off" period (lbs/hr)

C_p - is the heat capacity of the air $\left(\frac{\text{Btu}}{\text{pound -}^{\circ}\text{F}}\right)$

ΔT - is the temperature difference between the boiler and the ambient air ($^{\circ}$F)

h_j - is the heat transfer coefficient characteristic of jacket losses for a specific system (BTUH/$^{\circ}$F)

and: $$\dot{H} = \dot{m}_{air}\, C_p\, \Delta T + h_j\, \Delta T = (\dot{m}_{air} C_p + h_j)\, \Delta T$$

If (h_j) and ($\dot{m}_{air}$) are approximated to be constant, then

$$\Delta\dot{H} = \frac{\dot{H}_2 - \dot{H}_1}{\dot{H}_1} = \frac{\Delta T_2 - \Delta T_1}{\Delta T_1}$$

For the present case:

$$\dot{H}_1 = \left(\dot{m}_{air}C_p+h_j\right)(185-70) = K(115)$$

$$\dot{H}_2 = \left(\dot{m}_{air}C_p+h_j\right)(150-70) = K(80)$$

The reduction in off-cycle heat loss rate for lower boiler temperature can be calculated from the above (assuming K = constant):

$$\Delta\dot{H} = \frac{115-80}{115} = 30\%$$

This agrees closely with the measured efficiency improvement. Therefore, for relatively constant off-cycle air flow rates through the boiler we can approximate the temperature of the air leaving the boiler to be equal to the average boiler water outlet temperature. Additional measurements are required to substantiate and further refine this relationship.

The quantitative relationship between boiler temperature and off-cycle losses have been established by these tests. It has been shown experimentally that reducing boiler water outlet temperature from 185°F to 150°F resulted in substantial reduction of burner off-cycle heat losses. While the above evaluation was carried out for a fixed boiler temperature of 150°F, in an actual application the boiler water temperature would be expected to be variable (perhaps from 100 to 200°F) to accommodate varying heat loads.

This could be accomplished by an automatic boiler temperature controller that would sense the outside temperature and select the minimum boiler water temperature that is sufficient to meet the heating load of the house. Another approach would be for the homeowner to adjust the boiler water temperature manually, in each climatic season (or more often), to a minimum acceptable temperature. By varying the boiler temperature automatically (or manually), the resulting fuel savings would be expected to be larger than 6%. A more detailed analysis would be required to evaluate fuel savings resulting from variable aquastat operation.

This concept is directly applicable to all hydronic heating boilers that maintain a minimum temperature in order to supply domestic hot water and/or space heat. In the case of boilers used to supply domestic hot water and space heat, the boiler temperature must be maintained at a level sufficient to produce hot water at an acceptable temperature, and satisfy the space heat load. "Acceptable" temperature settings can vary, depending on individual preferences of residents, and the condition of the boiler system.

4. Vent Dampers for Reduced Off-Cycle Heat Loss

(a) Reduction of Boiler Off-Cycle Heat Loss

The effect of vent dampers on annual fuel consumption was measured for two boilers, with widely differing results. Installation of a vent damper on the boiler-burner combination of Section II-A-1 (dry-base, single pass steel boiler with non-retention-head burner) produced a 10% reduction in annual fuel consumption. In contrast, the same vent damper provided less than 2% reduction of fuel use for the boiler-burner combination of Section II-A-2 (dry-base cast iron boiler with retention-head burner). Apparently, for some boiler-burner combinations, vent dampers can save substantial amounts of fuel, while only negligible savings can be attained for other combinations of boilers and burners. Additional testing is required to establish recommendations for the proper use of vent dampers to maximize fuel savings.

A schematic of vent damper operation is provided in Figures 9a and 9b. The damper remains in the "open" position (Fig. 9a) during burner operation, and closes (Fig. 9b) after termination of burner firing. The device used for the tests, provides a

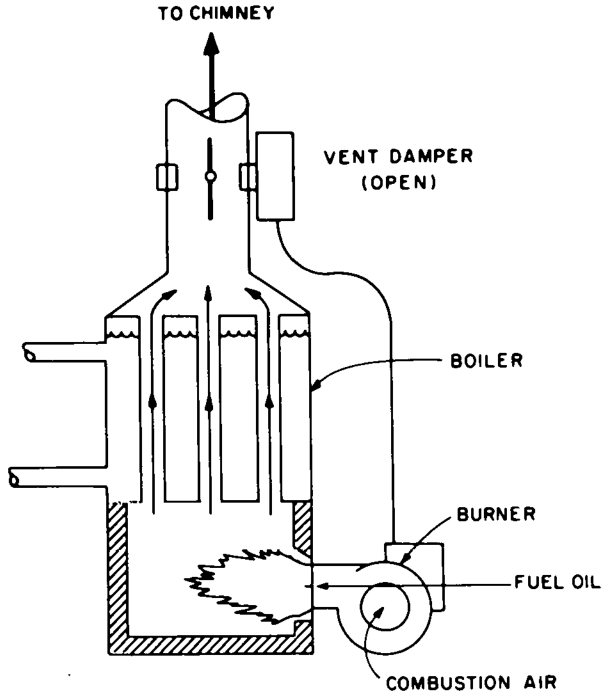


Figure 9a. Burner operating - vent damper open

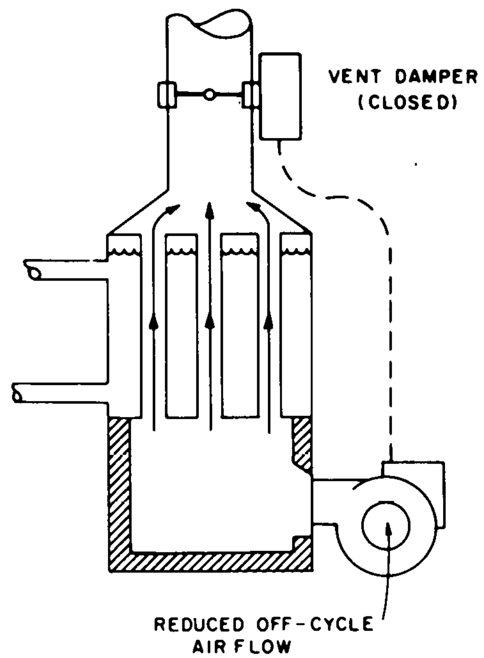


Figure 9b. Burner not operating vent damper closed

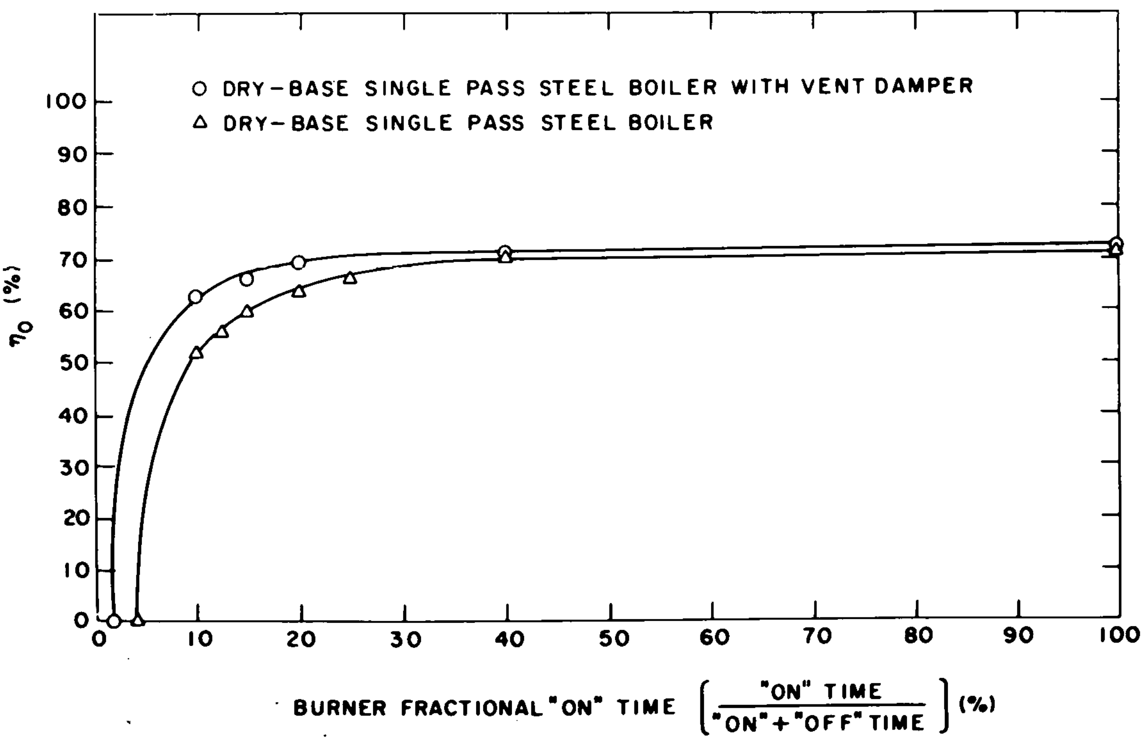


Figure 10. Overall efficiency (η_o) vs. burner fractional "on" time (%)

three-minute delay following termination of burner firing, to allow for post-purging of combustion products. Additional tests were performed with bypasses cut into the damper (instead of the three-minute delay) and reduced fuel savings were measured. These preliminary tests indicate that time delays are better than damper bypasses for obtaining maximum fuel savings.

The overall efficiency over the entire range of heating loads is presented in Figure 10 for boiler II-A-1 with and without the vent damper installed. The efficiency of operation is significantly improved by use of the damper. The Annual Fuel Use and Efficiency computation (see Table 11) for New York, 50,000 Btu per hour design heat load, 40 gallons per day domestic hot water, and a firing rate of 1.07 gallons per hour, provided an annual fuel consumption of 1290 gallons for boiler II-A-1 using the vent damper. Without the damper the consumption was 1440 gallons, corresponding to a 10% difference in fuel use. For this particular boiler-burner combination, the vent damper is capable of providing significant, cost-effective reduction of fuel consumption.

Less favorable results were obtained for use of the vent damper with boiler II-A-2. Annual fuel savings of less than 2% was determined. The use of retention-head burners can be an important factor to consider in conjunction with vent dampers. Boiler II-A-1 was fired with a non-retention-head burner, while a retention-head burner was used with boiler II-A-2. Section II-C-2 concerned with the effect of retention-head burners indicates that a portion of the fuel savings attributed to retention-head burner refit results from the reduction of burner off-cycle heat loss. It is clear that these refit modifications are not additive in a linear fashion, and consideration must be given to each refit candidate on an individual basis. Total energy savings resulting from refit actions depend on the initial overall efficiency of the system to be refitted. Again, additional investigation into the proper utilization of vent dampers is required.

(b) Vent Damper Reduction of Air Infiltration

Vent dampers can provide additional fuel savings by reducing the flow of cold outside air into the heated structure. Fuel-fired heating units consume warm room air through the burner and draft regulator during the burner "on" and "off" periods. Room air passes through the heating unit and out of the building via the chimney, and this warm air must be replaced by unheated outside air. The heating requirements of the building are increased by this heating-unit-induced air infiltration. However, for the case of oil-fired hot water boilers, the component of fuel savings produced by reduced air infiltration (by use of vent dampers) is small. The most significant savings resulting from vent damper installation on oil-fired hot water boilers is the component attributed to decreased heat loss from the boiler during the burner off-cycle.

A simple calculation can be used to estimate increased building heating requirements attributed to boiler-induced air infiltration during the "off" period. Boiler II-A-1 (for previously discussed conditions) operated at an average <u>overall efficiency</u> of 56.7% with a <u>steady state efficiency</u> of 69.2%. The average annual <u>cycle efficiency</u> for this case is the ratio of these values equal to 81.9%, corresponding to an average off-cycle heat loss (non-dimensional) of 1.00 minus 0.819 equal to 0.181. The annual heat loss during the off-cycle can be calculated to be the corresponding fraction of total fuel use per year.

$$\text{Annual Off-Cycle Heat Loss} =$$

$$(.181)\left(1440\ \frac{\text{gal. of oil}}{\text{year}}\right)\left(7.11\ \frac{\text{lbs}}{\text{gal.}}\right)\left(19{,}500\ \frac{\text{Btu}}{\text{lb}}\right) = 3.61 \times 10^{7}\ \frac{\text{Btu}}{\text{year}}$$

The average fraction burner "on" time for the above case is 0.15 corresponding to a fractional burner "off" time of 0.85.

The total hours per year of burner "off" time is:

$$\text{Burner Off Time} = \left(365\ \frac{\text{days}}{\text{year}}\right)\left(24\ \frac{\text{hrs}}{\text{day}}\right)\left(.85\right) = 7450\ \frac{\text{hours}}{\text{year}}$$

Therefore, the average heat loss rate during the burner off-cycle can be calculated to be:

$$\dot{H}_{\text{Off}} = \frac{3.61 \times 10^7\ \text{Btu/year}}{7450\ \frac{\text{hours}}{\text{year}}} = 4840\ \frac{\text{Btu}}{\text{hour}}$$

Knowing the total heat loss rate from the boiler during the off-cycle, it is possible to determine the convective flow rate of air through the boiler. The total heat loss rate ($\dot{H}_{\text{Off}}$) is the sum of convective heat flow through the boiler and off-cycle jacket heat loss. Installation of a vent damper on the boiler of this example resulted in reduction of off-cycle heat loss by 50%. Therefore, it can be assumed that the convective component of heat loss represents 50% of the total, and the convective heat loss rate is equal to 0.5 $\dot{H}_{\text{Off}}$. It is possible to write an equation for the convective heat loss through the boiler.

$$\dot{H}_{\text{Conv}} = \left(\dot{V}_{\text{air}}\right)\left(C\right)\left(\Delta T\right)$$

where

$\dot{H}_{\text{Conv}}$ is the convective vent heat loss rate (Btu per hour)

$\dot{V}_{\text{air}}$ is the volumetric air flow rate through the boiler during the off-cycle (cubic feet per minute)

C is the heat capacity of the air per unit volume

$$\left(\frac{\text{Btu - Min}}{\text{cubic feet - deg. F - hour}}\right)$$

ΔT is the temperature rise of the air passing through the boiler (deg. F)

Solving for the air flow rate $\dot{V}_{air}$:

$$\dot{V}_{air} = \frac{\dot{H}_{Conv}}{(C)\ (\Delta T)}$$

The temperature of the boiler water is maintained at an average value of $180^{\circ}F$ during tests, and from Section II-C-3, we can assume that the outlet air temperature is $180^{\circ}F$. Therefore, the temperature rise of the air passing through the boiler is $180^{\circ}F$ minus $68^{\circ}F$ which equals $112^{\circ}F$. We have found experimentally that $\dot{H}_{Conv} = 0.5\ \dot{H}_{Off}$. Using these values to compute $\dot{V}_{air}$:

$$\dot{V}_{air} = \frac{\dot{H}_{Conv}}{(C)\ (\Delta T)} = \frac{(0.5)\ (4840)}{(1.08)\ (112)} = 20 \text{ CFM}$$

The air flow rate through the boiler during the burner off-cycle is 20 cubic feet per minute.

It has been estimated by others that between 50 to 70% of the total air flow through the heating units contributes directly to increased infiltration of cold outside air into the heated space.[5] (The remaining 30 to 50% reduces exfiltration.) Also, it has been assumed that a properly functioning barometric damper provides no significant air flow during the burner off-cycle because it remains in the "closed" position.[5] It is now possible to evaluate the overall effect of convective air flow through the boiler on the total heat load of the structure. The heating load increase resulting from boiler-induced infiltration can be represented by ΔHL:

$$\Delta HL = (.70)\ (20 \text{ CFM}) \left(1.08 \frac{\text{Btu-Min}}{\text{CF-Deg. F - Hr}}\right) \left(68\text{-}40 \text{ Deg. F}\right) \quad (.85) =$$

$$360 \text{ Btu/Hr}$$

Where: the maximum value is assumed for the infiltration factor (0.70); an average winter temperature of $40^{\circ}F$ is used; and the seasonal average burner "off" period is 0.85. For a 40°F outside temperature and a design heat load of 50,000 Btu per hour (at zero degrees F), the total heat load of the structure is 22,000 Btu per hour (including 40 gallons per day of domestic hot water). The heat load reduction from vent damper decrease of air infiltration corresponds to:

$$\frac{360}{22,000} = 1.6\%$$

For an average winter temperature in the New York area, the building heat load is reduced 1.6% by a vent damper, and the annual fuel consumption decreases by 1.5%. This fuel saving is small compared to the 10% savings provided by reduced off-cycle boiler heat loss discussed earlier.

The above calculation is an estimate of the maximum expected fuel savings (1.5%) for an oil-fired boiler located in the heated space. Boiler location in the structure can reduce this savings (see Section V-A). The above calculation is not intended to provide a rigorous and complete evaluation. However, the essential facts are clear and indicate that boiler-induced infiltration losses are quite small. Our analysis provides an estimate of fuel savings that can be attributed to reduction of boiler-induced air infiltration by use of a vent damper. Specific oil-fired boilers operate over a wide range of design and installation conditions and the actual fuel savings for individual installations is expected to vary. Also, the effect of vent dampers on gas-fired boilers, and oil and gas-fired furnaces is expected to be different from the case of oil-fired boilers.

(c) Safety of Vent Dampers

Consideration must be given to the possible safety hazard created by placing a vent damper in the exhaust ducting of a fuel-fired heating unit. The damper must open before the burner is energized to prevent combustion products and fumes from escaping into the building. The damper is required to operate in a hot environment (between 400°F and 900°F) over thousands of cycles without malfunctioning. Improper installation of a vent damper could result in a potentially dangerous situation. While dampers can provide significant reduction of annual fuel use, the potential safety hazards produced require further examination, particularly for the case of vent damper refit on existing heating units.

5. Economizers for Reduced Stack Heat Loss

Tests were performed to evaluate annual fuel savings from installation of a stack economizer, designed to reclaim heat from the boiler exhaust gases. A schematic of the economizer is shown in Figure 11a. Water is pumped from the boiler across heat exchanger surfaces located in the stack economizer, and the heated water is returned to the boiler for distribution to the heating load. The economizer reduces heat loss during burner operation by decreasing the temperature of exhaust gases vented through the chimney. Energy savings are directly related to stack temperature reduction, represented in Figure 11a by T_{flue} minus T_{exit}. Obviously, higher values of T_{flue}, corresponding to larger stack heat losses, provide the potential for larger energy savings with an economizer. Actual fuel savings in field installations will vary over a wide range, depending on the stack temperatures of individual heating units.

The dry-base, single pass steel boiler with non-retention-head burner (Section II-A-1) was used to evaluate economizer performance. The stack temperature of this boiler is 600°F compared to an average value of 690°F for 100 oil-fired heating units tested on Long Island.[6] Therefore, the energy savings for the boiler under investigation may be less than the average savings for heating units in the field. Test results are shown

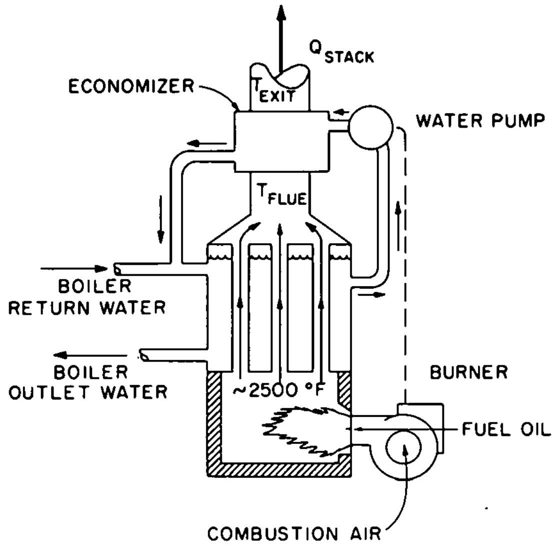


Figure 11a. Economizer schematic

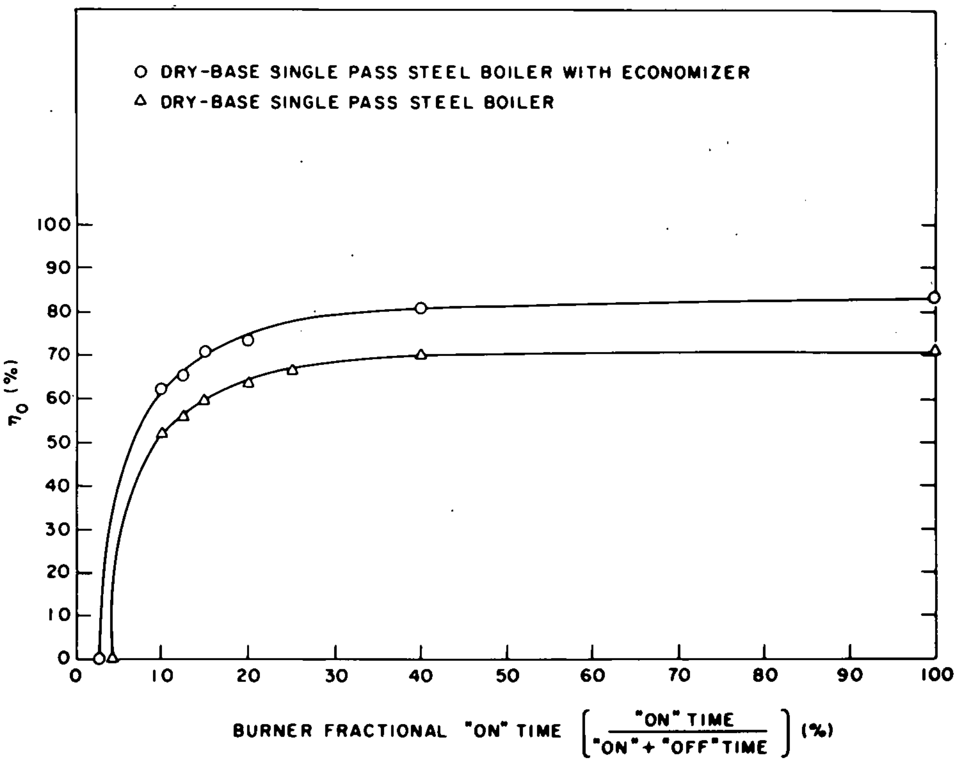


Figure 11b. Overall efficiency (η_o) vs. burner fractional "on" time (%)

in Figure 11b, for the same boiler before and after stack economizer installation. As expected, the steady state efficiency was significantly improved, while a small change was observed for cyclic efficiency. Annual fuel savings and efficiencies can be compared for the same boiler without the economizer. For New York City, a design heat load of 50,000 Btu per hour, 40 gallons of domestic hot water per day, and $\alpha = 2$ (100% overfired), annual fuel consumption is reduced 18% (from 1440 to 1180 gallons per year) by use of the stack economizer. Note that the steady state efficiency is increased to 83.6% resulting in a reduction of the design firing rate from 1.07 gph to 0.881 gph for the above set of parameters.

One possible problem area observed during economizer operation is the tendency for soot to collect on heat exchanger surfaces. The smoke number was maintained at a low level (#1) and soot accumulation was observed after two weeks of testing. The average smoke number observed in a field study[6] was #3, which could contribute to substantially increased fouling of stack economizer surfaces. Eventually, soot deposition will degrade economizer performance, and could result in reduced draft at the fire-box as the flue passages become restricted. Another area that requires further investigation is the possibility of equipment damage from flue gas condensation after economizer installation. For the test boiler, stack temperatures were reduced from 600°F to 260°F. This lower temperature could create condensation damage in some heating systems.

6. Fuel Oil - Water Emulsion Burner

A mechanical fuel oil-water emulsifier was tested in two retention-head burners supplied by the equipment developer. Both burners were fired into the dry-base, single pass steel boiler of Section II-A-1, and the emulsion composition was varied from 0 to 33% water by volume. The oil firing rate was maintained at 0.85 gallons per hour for all tests, while the amount of water added was increased.

Steady state efficiency tests were performed and an efficiency decrease proportional to the amount of water added was observed. The efficiency reduction can be attributed to the additional energy required to vaporize and heat the water contained in the emulsion. Test results are plotted in Figure 12 and it can be seen that Burner (A) operated with a steady state efficiency of 76.7% without the addition of water to the fuel. The efficiency decreased to 73.8% as the percent of water in the fuel was increased to 33%. Similar results were found for Burner (B). The particular fuel oil-water emulsifier that was tested produced an increase in annual fuel consumption.

7. Combustion Air Humidification

A combustion air humidification device was tested on a non-retention-head burner (supplied by the manufacturer) in a conventional single pass, dry-base vertical fire-tube steel boiler. Base line steady state efficiency tests provided an average value of 68.3% without the humidification process in operation. After the humidification device was installed and the combustion air flow was readjusted to a Bacharach smoke number of one, the steady state efficiency was observed to be 69.4% (see Table 12).

Stack gas analyses were performed during the steady state tests before and after installation of the device. No change was observed in the percentage of CO_2 (8%) or Bacharach smoke number (#1). A slight reduction of stack gas temperature (from $600^{\circ}F$ to $590^{\circ}F$) was observed, corresponding to a 1/2% increase of steady state efficiency. Because of the small efficiency improvement and the possibility that the change could have resulted from readjustment of the combustion air setting, it can be concluded that the combustion air humidification device that was tested had no significant effect on the efficiency of the test boiler.

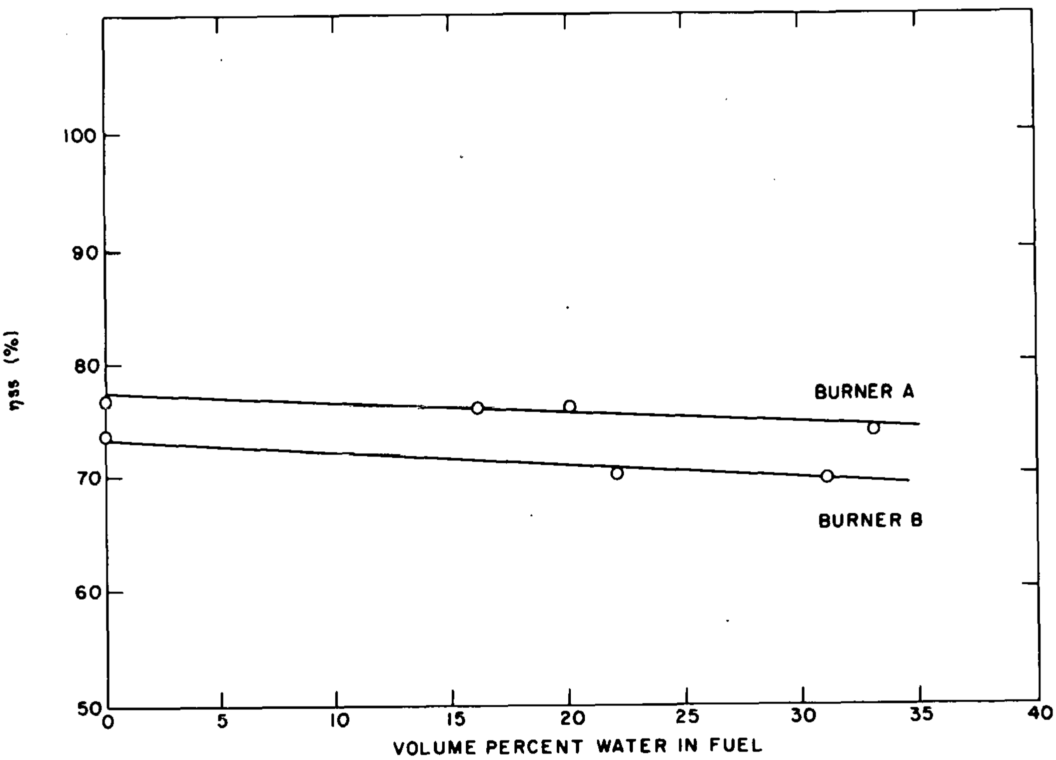


Figure 12. Effect of water-fuel oil emulsion steady state efficiency (η_{ss}) vs. percent water in fuel

Table 12

STEADY STATE EFFICIENCY DATA FOR
COMBUSTION AIR HUMIDIFICATION TESTS

WITHOUT HUMIDIFICATION

Steady State Efficiency (%)	Boiler Outlet Temperature ($^{\circ}F$)
68.5	176.9
68.2	178.3
68.6	184.0
68.0	181.2

Average Steady State Efficiency at 180°F Outlet Temperature:	68.3%
Average Steady State Efficiency by Stack Gas Sampling (Neglecting Jacket Losses):	71.8%
Percent CO_2	8%
Net Stack Temp.	600°F

WITH HUMIDIFICATION

Steady State Efficiency (%)	Boiler Outlet Temperature ($^{\circ}F$)
69.2	183.1
69.4	183.9
69.6	186.3

Average Steady State Efficiency at 180°F Outlet Temperature:	69.4%
Average Steady State Efficiency by Stack Gas Sampling (Neglecting Jacket Losses):	72.2%
Percent CO_2	8%
Net Stack Temp.	590°F

D. Warm Air Heating Equipment

This section presents efficiency test results for a commercially available warm air furnace fired by a blue flame oil burner. At the present time base line efficiency data for "conventional" warm air heaters are not available for comparison. Also, the electronic data acquisition and controller system is not yet operational and the manual mode of data collection prevented performance of detailed cycle efficiency tests. Instead, steady state tests were conducted, and measurements were performed to determine the quantity of heat stored in the furnace following termination of burner firing. The furnace heat storage capacity can be used to place an upper limit on the quantity of heat subject to loss during the burner-off period. In addition to laboratory tests, a field investigation was performed to evaluate long-term efficiency of the heating units.

The steady state efficiency was measured over a range of outlet air temperatures from 145 to 175 degrees F. An average efficiency of 85% was observed following the test procedures outlined in Section III-B.

Heat storage measurements were observed to be a function of several factors including: the temperature rise of the air passing through the heater during steady state (continuous) operation, the length of the burner firing period, and the air blower low limit (shut-down) temperature. In Table 13, the temperature rise across the furnace, ΔT, (above ambient temperature) is presented as a function of the time period that the air blower operates following the end of the burner firing period. The quantity of heat stored in the furnace corresponding to each ΔT is provided, together with the percentage of stored heat that is recovered. Obviously, the most efficient mode of operation requires cooling the furnace to low temperatures following burner operation. For example, if the furnace is cooled by air blower operation for 2.7 minutes

Table 13

EQUIPMENT COOLDOWN HEAT STORAGE

Air Circulating Fan Operation After Burner Shutdown (Min.)	ΔT (^{o}F)	Percent of Steady State ΔT	Maximum Heat Stored (Btu)	Percent of Total Stored Heat Recovered
0.0	87.6	100	3960	0
.4	80	91.3	3310	16.5
.8	70	79.9	2920	26.4
1.1	60	68.5	2500	36.8
1.5	50	57.1	2110	46.8
2.0	40	45.7	1700	57.1
2.7	30	34.2	1300	67.1
3.4	20	22.8	940	76.4
4.7	10	11.4	530	86.7
17.0	0	0	0	100

Note:

Steady State ΔT: $87.6^{o}F$

Air Flow Rate: 976 SCFM

Total Heat Storage Capacity: 3960 Btu

following burner shut-down to ΔT equal to $30^{o}F$, then 1300 Btu will be stored while 2660 Btu will be recovered. Similarly, operation of the air blower for 1.1 minutes following termination of burner firing would result in ΔT of $60^{o}F$ and stored heat equal to 2500 Btu. Heat storage in the furnace following cessation of blower operation represents the maximum amount of heat that can be lost during the burner off-cycle.

Actual off-cycle heat losses of field-installed furnaces depends on the operating set-points and installation characteristics for specific heating units. Heating units with reduced air blower low limit settings would be expected to operate more efficiently than those with higher set-points. Also, units operating with fewer on-off cycles would be expected to operate more efficiently.

A field investigation of blue flame furnaces was conducted to evaluate the long-term performance capability of field-installed equipment. Stack gas sampling procedures were used to determine the steady state efficiency, and ten furnaces were tested (see Table 14) with an average efficiency of 83.3% observed. The average stack temperature was 440 degrees F, the carbon dioxide concentration was 12.9%, and all units operated with a smoke number (Bacharach) of zero. Most of the units tested had been operating for between 1-1/2 and 2 years, and it can be concluded from the low stack temperatures and high carbon dioxide concentrations, that a high level of efficiency was maintained by these heating units.

Future laboratory tests will include more detailed evaluation of off-cycle heat loss from which seasonal efficiencies and fuel use can be calculated, and base line equipment will be evaluated for comparison. As this information becomes available, care must be taken to avoid fuel use comparisons between warm air furnaces and hot water boilers. While furnaces can supply only space

Table 14

RESULTS OF BLUE FLAME FURNACE FIELD TEST

Field Unit	Net Stack Temp (°F)	Percent CO_2 (%)	Smoke Number	Steady State Efficiency
1	375	12.5	0	84.5
2	450	13.5	0	83.5
3	455	14.5	0	83.9
4	455	12.0	0	82.4
5	460	13.5	0	83.5
6	478	13.0	0	82.6
7	435	12.5	0	83.3
8	405	11.8	0	83.5
9	435	13.0	0	83.6
10	475	12.5	0	82.2
Average	442	12.9	0	83.3

heating, hot water boilers are often used for both space heating and production of domestic hot water. Comparison on an equal basis would require consideration of fuel use by separate domestic hot water heaters in conjunction with warm air furnaces. It should be noted that the seasonal efficiency of many hot water boilers can be substantially increased when used for the sole purpose of space heating.

III. METHODOLOGY AND TEST FACILITIES

A. Hot Water Heating Equipment

1. Efficiency Measurement Technique

The enthalpy flow technique provides a direct, fundamental, physical measurement of heating equipment efficiency during steady state (continuous) and cyclic (intermittent) burner operation. The quantity of heat transferred to the boiler water is measured directly and compared to the corresponding heat content of the fuel consumed. The efficiency of operation is defined to be the ratio of heat transferred to the water to the total heat available from the fuel consumed. The enthalpy flow measurement technique does not depend on any simplifying assumptions but results from basic physical laws including the definition of enthalpy. Consequently, the accuracy of the test results is limited only by the accuracy of measurement of the physical quantities of interest (mass flow rates and temperatures).

The quantities that are measured include: (see Figure 13)

(a) $\dot{m}_w$ - Water flow rate through boiler (pounds per minute)

(b) ΔT - Temperature rise across the boiler $\Delta T = T_{outlet} - T_{inlet}$ (degrees F)

(c) $\dot{m}_{oil}$ - Fuel flow rate to burner (pounds per minute)

(d) h_{oil} - Heating value of fuel (Btu per pound)

From these values the efficiency of operation can be determined:

$$\eta = \frac{\text{Heat transferred to boiler water}}{\text{Total heat available from combustion of fuel oil}}$$

The heat transfer rate to the water is simply the product of the water mass flow rate and temperature rise.

$$\dot{H} = \dot{m}_w \times C_p \times \Delta T$$

Similarly, the heat available from the combustion of fuel is the product of the fuel mass flow rate and the heating value of the fuel.

$$\dot{H}_{total} = \dot{m}_{oil} \times h_{oil}$$

The efficiency is the ratio of these two quantities

$$\eta = \frac{\dot{m}_w \times T}{\dot{m}_{oil} \times h_{oil}}$$

The enthalpy flow measurement technique is a fundamental method for direct determination of the efficiency of heating equipment. It can be considered a "standard" method (traceable to basic laws) whose accuracy depends solely on the accurate measurement of the above mentioned quantities.

2. Physical Meaning of Steady State, Cycle and Overall Efficiencies

Steady State Efficiency: During continuous burner operation, a fraction of the heat released by combustion of the fuel is lost as combustion products vented through the stack and as radiative and convective boiler jacket losses. The steady state efficiency represents the fraction of total heat that is transferred to the boiler water for continuous burner operation.

Cycle Efficiency: During burner on-off cycling, heat is lost as off-cycle boiler losses. The cycle efficiency indicates the fraction of the useful steady state heat that is availalbe during intermittent burner operation. The cycle efficiency varies with the fractional burner "on" time. As the burner percentage "on" decreases, the "off" time increases, and a larger fraction of heat is subject to standby losses. Therefore, the cycle efficiency decreases as the fractional burner "on" time is reduced.

Overall Efficiency: The total fraction of useful heat available at the boiler outlet is the product of the steady state and cycle efficiencies.

$$\eta_{overall} = \eta_{steady\ state} \times \eta_{cycle}$$

For example, if $\eta_{ss} = .70$ and $\eta_{cycle} = .70$, then the overall efficiency is equal to $(.70) \times (.70) = .49$, or 49% of the heat released by the combustion process is available as useful heat in the boiler water. For continuous burner operation the cycle efficiency equals unity ($\eta_c = 1.0$) and the overall efficiency equals the steady state efficiency.

3. Test Facilities

The test facilities at BNL were planned and fabricated in order to produce maximum measurement accuracy and reliable performance. A brief description of the instrumentation follows:

(a) $\dot{m}_w$ - A turbine type flow meter is used to measure the water flow rate through the boiler. A thermocouple measurement is provided to enable the conversion of volumetric flows to mass flow rates. The meter is calibrated to within 0.2% each day by use of a 1000 pound digital scale and digital timer. A timed sample is collected and the pound per minute value is compared to the mass flow rate indicated by the flow meter.

$\dot{m}_w$ Measurement Uncertainty $\leq$ 0.5%

(b) ΔT - A thermopile is used for precise measurement of the temperature rise of the boiler water. This device consists of two 2" pipes (one for inlet water and one for outlet water) containing an array of ten thermocouples in each pipe to provide an accurate average temperature value. Two thermocouples (one inlet and one outlet) are used to determine the average boiler temperature for conversion of the thermopile output from millivolts to degrees F.

ΔT Measurement Uncertainty $\sim$ 0.5%

(c) $\dot{m}_{oil}$ - A turbine type flow meter is used to measure the fuel flow rate. A thermocouple is used to convert volume flow rates to mass flow rates. Calibration procedure is similar to the method in 3(a).

$\dot{m}_{oil}$ Measurement Uncertainty $\leq$ 0.5%

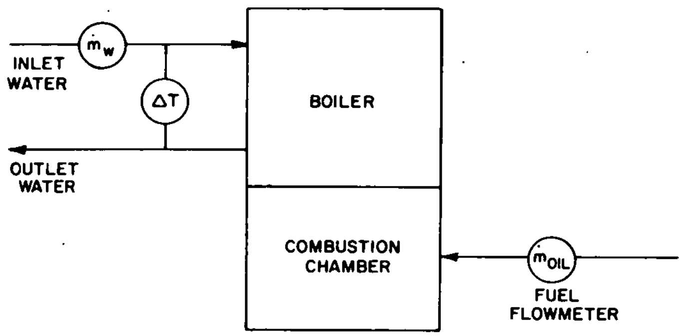
(d) h_{oil} - The fuel oil heating value is measured by use of an oxygen bomb calorimeter following the appropriate ASTM procedure.

h_{oil} Measurement Uncertainty $\leq$ 0.5%

(e) Data Processing Equipment (see Figure 14) - An analog to digital "computer" has been built to provide for instantaneous automatic data reduction to facilitate more rapid operation and to eliminate human error as much as possible. The computer receives the analog signals from the various flow and temperature transducers and produces heat and efficiency values by the method outlined in Figure 13. The quantities of interest are converted to digital values and displayed on readouts and four 4-channel printers. The most powerful feature of the computer is the ability to perform integration over time for the heat input and heat output values. During cyclic operation the mass flow rates and temperatures vary with time and the computer provides an instant-by-instant accumulation of the heat values. The only alternative method requires manual integration of areas under curves (e.g., the ΔT - time curve) which are time consuming and a large source of error. The unique combination of accurate flow and temperature measurement and electronic data reduction provide for system accuracy of within 2% of the absolute efficiency value and system precision and reproducibility within 1%.

4. Comparison of Efficiencies Determined by Enthalpy Flow and Stack Gas Sampling Methods

The enthalpy flow technique provides a standard basis from which to compare other methods of efficiency measurement. Stack gas sampling is an example of a less accurate field measurement technique. The stack gas is sampled to determine the percentage of CO_2 (or O_2) and the stack temperature. The percentage of CO_2 indicates the amount of excess combustion air passing through the heating unit and provides an indication of the total



MEASURED QUANTITIES

$\dot{m}_w$ = WATER FLOW RATE

ΔT = TEMPERATURE RISE ACROSS BOILER

$\dot{m}_{OIL}$ = FUEL FLOW RATE

h = HEATING VALUE OF FUEL

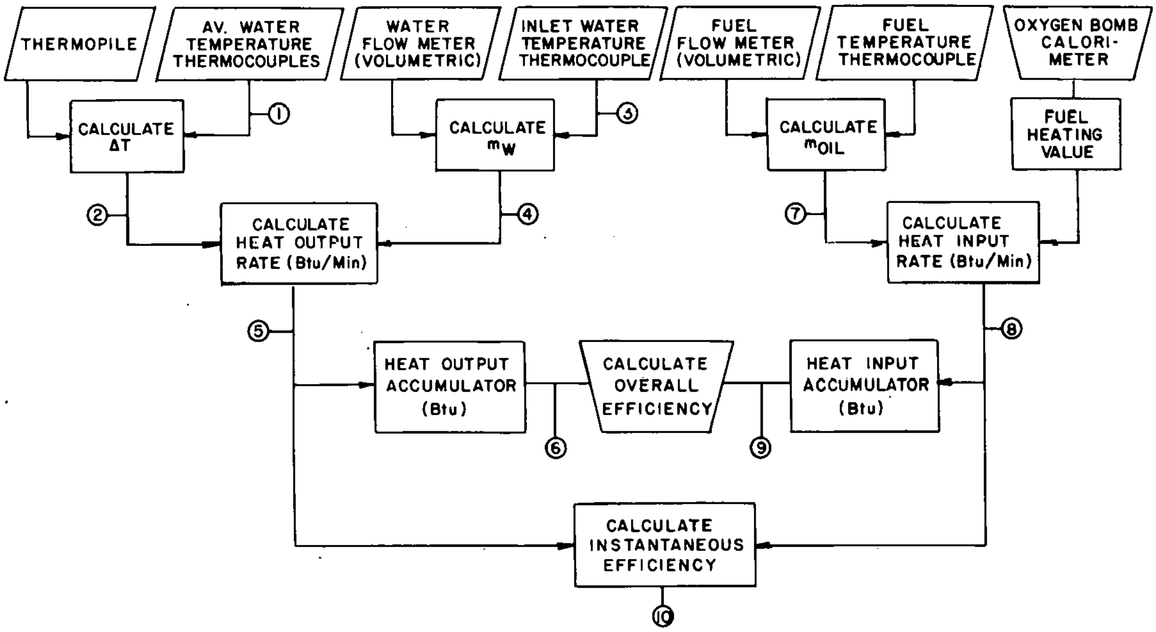
EFFICIENCY DETERMINATION

HEAT INTO WATER = $\dot{m}_w \, \Delta T$

TOTAL HEAT FROM FUEL = $\dot{m}_{OIL} \, h$

EFFICIENCY = $\dfrac{\dot{m}_w \, \Delta T}{\dot{m}_{OIL} \, h}$

Figure 13. Brookhaven National Laboratory boiler efficiency measurement schematic



○ INDICATES PRINTED OUTPUTS

(1) AVERAGE WATER TEMPERATURE
(2) ΔT ACROSS BOILER
(3) INLET WATER TEMPERATURE
(4) WATER MASS FLOW RATE
(5) HEAT OUTPUT RATE

(6) ACCUMULATED HEAT OUTPUT
(7) FUEL MASS FLOW RATE
(8) HEAT INPUT RATE
(9) ACCUMULATED HEAT INPUT
(10) INSTANTANEOUS EFFICIENCY

○ STACK TEMPERATURE (°F)
○ FLUE GAS CO_2 PERCENTAGE
○ BURNER ELAPSED TIME
○ WATER FLOW ELAPSED TIME
○ TOTAL ELAPSED TIME

Figure 14. Flow chart of digital processing equipment

flow of combustion products. The stack temperature together with the stack gas flow rate represents the quantity of heat lost through the stack. From this quantity an efficiency can be calculated. The accuracy of this calculation is subject to a number of experimental uncertainties as well as computational and analytic difficulties. These are discussed below. It should be noted that stack gas sampling is only applicable for measurement of steady state efficiency.

There are several sources of error inherent to efficiency measurement by stack gas methods:

1) Complete combustion is assumed. This need not be the case.
2) A constant factor is assumed for jacket losses for all heating units ($\sim$2%). This is generally a serious error (see experimental results cited earlier in this report).
3) Air leaks downstream of the heat exchanger (boiler or furnace) can lead to substantial error in both stack temperature and composition measurement.
4) Stack gas composition measurements involve sampling and reading errors. (Uncertainty $\sim$3%)
5) Stack gas temperature errors can result from thermometer stem losses, calibration error and nonuniformity of stack gas temperature. (Minimum uncertainty $\sim$2%)
6) Errors are possible from inaccuracy in thermodynamic calculation of efficiency from stack gas composition and temperature. These derive from uncertainties in the heat of combustion of the fuel as well as from the computational methods employed.

The measurement, precision and accuracy of stack gas sampling techniques is about an order of magnitude inferior to the enthalpy flow methods used at BNL, and is on the order of 10%. The actual percent deviation from the enthalpy flow method will usually vary between 0 and 12%, depending on the magnitude of jacket losses for individual heating units.

5. Test Procedures

Steady State Efficiency Tests: For steady state efficiency tests the burner firing rate and water flow rate through the boiler are maintained at constant values. The water flow rate is adjusted to give a 180°F outlet boiler temperature (within 5°F). The system is allowed to "warm up" for at least one hour until the temperature rise across the boiler reaches a constant value. Data is collected for one hour of operation and the average steady state efficiency value is determined and recorded. Flue gas efficiency tests are performed during steady state enthalpy flow tests.

Cycle Efficiency Tests: A digital programmer is used to control the "on" and "off" times of the burner and water flow through the boiler. The burner "on" time is maintained at 5.0 minutes for all cycle tests while the burner "off" period is varied. For a 20% fractional burner "on" time, the burner "off" time becomes 20.0 minutes for a total cycle length of 25.0 minutes. The water flow rate is maintained at the steady state value and heat removal from the boiler is controlled by digital programmers which operate solenoid valves at the inlet and outlet of the boiler. After the burner firing is terminated, the heating unit remains idle until water flow is initiated at which time the heat flow from the boiler is measured by the enthalpy flow method. At the end of the heat removal period, the output of the data acquisition system provides an integrated value for the total heat input and heat removal for that cycle. The water flow "on" time is adjusted to produce an average outlet water temperature of approximately 180°F during cycle efficiency tests. The water flow "off" time is adjusted to equal the total cycle time of the burner (25.0 minutes for a 20% fractional burner "on" time).

Once the cycle is programmed, the system is allowed to operate for many cycles until "steady cycles" are achieved. Maximum variations of heat input and output of 1% are observed between cycles throughout this steady cycle period. The overall efficiency for each cycle is determined by dividing the heat removed by the heat input and the average value is calculated. The cycle efficiency is simply the overall efficiency divided by the steady state efficiency (see III-A-2). Cycle tests are conducted over the entire range of burner fractional "on" times from standby to 100%. The data points generally include 10%, 15%, 20%, 25%, 40%, and 100%, but additional data points may be included.

6. Test Results

Steady State Efficiency: The results of the steady state efficiency tests are presented in tabular form with an average value indicated, and in graphical form as a function of boiler outlet temperature or fuel firing rate. The results of stack gas measurements are included when available.

Cycle Efficiency: The results of cycle efficiency tests are presented graphically as overall efficiency versus fractional burner "on" time. Tabular cycle efficiency values are provided for the range of tests in 5% intervals of fractional burner "on" time.

Seasonal Efficiency and Fuel Use: From the steady state and cycle data, annual efficiency and fuel use are calculated. The seasonal efficiency is a measure of the overall efficiency of a particular heating unit averaged over the heating season for a particular design heating load and weather pattern for a variety of operating conditions. For each seasonal efficiency a corresponding value for annual fuel use (in gallons of oil) is determined.

B. Warm Air Furnace Test Facility

1. Efficiency Measurement Technique

Again, the enthalpy flow technique provides direct measurement of heating equipment efficiency. At present the control and data acquisition "computer" for the Furnace Test Facility is not yet operational. As a result, only steady state efficiency tests can be completed. The basic techniques used in testing warm air furnace equipment are similar to those used in testing hydronic equipment and reference will be made to the previous Section (III-A) for the sake of brevity. The major difference between the two procedures is that the circulating heat transfer fluid used in furnaces is air rather than water (used in hydronic systems). This, of course, means that domestic hot water must be generated by a hot water heater when a home is equipped with a warm air furnace heating system.

2. Physical Meaning of Steady State, Cycle and Overall Efficiencies

Similar to hydronic system facility. See Section III-A-2 for detailed description of η_{ss}, η_c and η_o.

3. Test Facilities

The test facilities at BNL were planned and fabricated in order to produce maximum measurement accuracy and reliable performance. A brief description of the instrumentation follows:

(a) $\dot{m}_A$ - A vortex shedding flow meter is used to measure the circulating air flow rate through the furnace. A built-in compensator unit automatically corrects flow measurements for temperature and pressure changes, resulting in standard cubic feet per minute flow rate outputs. Temperature and pressures are measured by a pressure transducer and resistance thermometer.

The use of the vortex flow meter necessitated the use of an external blower to force air through the system. The air intake system provides for heating and cooling to maintain similar conditions for all tests

performed over the various seasons.
In heating units under test, the internal blowers are disconnected and the air flow rate is set by adjusting the external blower to the desired value.

$\dot{m}_A$ - Measurement Uncertainty $\leq$ 1%

(b) ΔT - A set of thermocouple arrays are used for precise measurement of the temperature rise of the circulating air passing through the furnace. The two probe units (one inlet and one outlet) are placed adjacent to the furnace inlet and outlet connections.

ΔT - Measurement Uncertainty $\leq$ 1%

(c) $\dot{m}_{oil}$ - A turbine type flow meter is used to measure the fuel flow rate. A thermocouple is used to convert volume flow rates to mass flow rates. The meter is calibrated to within 0.2% each day by use of a digital scale and a timer. A timed sample is collected and the pound per minute value is compared to the mass flow rate indicated by the flow meter.

$\dot{m}_{oil}$ - Measurement Uncertainty $\leq$ 0.5%

(d) h_{oil} - The fuel oil heating value is measured by use of an oxygen bomb calorimeter following the appropriate ASTM procedure.

h_{oil} - Measurement Uncertainty $\leq$ 0.5%

(e) Data Processing Equipment - As stated earlier, the warm air "computer" is not presently in its final form. Upon completion, the system will be similar to the hydronic test facility (see Section III-A-3-c). Again, the most important feature of the system will be the ability to perform integration over time for heat inputs and outputs. The results are then used in direct calculation of overall efficiency.

Overall System Accuracy $\sim 3\%$ Reproducibility 1%

4. Calibration Procedure

Due to the difficulty involved with calibrating air mass flow rates ($\dot{m}_A$), an enthalpy calibration procedure is used to determine the accuracy of the measurement system. With this procedure, the input to an electric furnace is measured using a digital watt meter, and the total heat input is compared to the heat output measured using the enthalpy flow technique. The electric furnace is well insulated and the total energy input (measured by watt meter) is transferred to the air flowing through the furnace. The heat output (measured by enthalpy flow) should be equal to the electrical input. In actuality, the difference between input and output energy is uncertainty in the enthalpy flow technique and uncertainty in the measurement of energy input (watt meter). This procedure provides for calibration of the enthalpy flow measurement facility within the accuracy of the watt meter.

5. Comparison of Efficiencies Determined by Enthalpy Flow and Stack Gas Sampling Method

Similar to hydronic facility. See Section III-A-4 for details.

6. Test Procedure

Steady State Efficiency: During continuous burner operation, a fraction of the heat released by combustion of the fuel is lost as combustion products vented through the stack and as radiative and convective furnace jacket losses. The steady state efficiency represents the fraction of total heat that is transferred to the circulating furnace air for continuous burner operation.

Steady State Efficiency Tests: For steady state efficiency tests the burner firing rate and air flow rate through the furnace are maintained at constant values. The air flow rate is adjusted to a constant rate (which will limit ΔT). The system is allowed to "warm up" for at least one hour until the temperature rise across the furnace reaches a constant value. Data are collected for one hour of operation and the average steady state efficiency value is determined and recorded. Flue gas efficiency tests are performed during steady state enthalpy flow tests.

7. Test Results

As stated, only steady state results are currently available. Since cycle efficiency tests are required to determine seasonal efficiency, AFUE results are not available at this time.

Steady State Efficiency: The results of the steady state efficiency tests are presented in tabular form with an average value indicated, and in graphical form in cases including steady state measurements at various furnace outlet temperatures. The results of stack gas measurements are included when available.

IV. ANNUAL FUEL USE AND EFFICIENCY

A. General Discussion

A simple computational procedure has been devised at Brookhaven National Laboratory to evaluate annual fuel consumption for hydronic heating equipment tested in the laboratory. The computer-based calculation relies on precise measurement of the "intrinsic merits" of a particular heating unit expressed in terms of steady state efficiency and cycle efficiencies over the entire range of heating loads. The annual fuel consumption for the heating unit under investigation varies as a function of several key parameters including: geographic location, building design heat load, outside design temperature, room temperature, domestic hot water use, and design fuel firing rate. Each of these parameters can be varied over a range of practical values in order to determine the resulting change in overall seasonal efficiency and fuel consumption for a particular heating unit. The output of these calculations is expressed as a matrix of overall seasonal efficiency and fuel consumption values (gallons of fuel oil per year) as a function of the important variables (see Figure 15).

The Annual Fuel Use and Efficiency calculation utilizes hourly outside temperature information averaged over a ten-year period for a particular geographic location, and the overall efficiencies measured in the laboratory (including burner on-cycle and off-cycle heat losses) are related to the outside temperatures. Fuel consumption at each outside temperature is calculated, and the total quantity of fuel consumed per year is evaluated by summing fuel usage over all outside temperatures during the heating season. Hourly temperature calculations are considered to be superior to methods utilizing degree-day temperature data because of the large temperature variations possible during a 24-hour period. The averaging process inherent in degree-day calculations can introduce unnecessary error in the determination of fuel use. Hourly information and calculations can follow the thermal response of conventional residential structures, and the corresponding change in heating system overall efficiency.

In the computational program for boilers, the design heat load of the building (Btu per hour), the outside design temperature, and the inside design temperature can all be varied by changing the corresponding input values. Each computer output presents values of seasonal efficiency and gallons of fuel consumed per year for a range of domestic hot water loads and design

HEATING UNIT	
LOCATION	
DESIGN HEAT LOAD (BTUH)	
OUTSIDE DESIGN TEMP (°F)	
ROOM TEMP (°F)	

HOT WATER \ α		1	2	3	4
0 GAL PER DAY	SEAS EFF				
	GAL PER YEAR				
	GPH				
40 GAL PER DAY	SEAS EFF				
	GAL PER YEAR				
	GPH				
80 GAL PER DAY	SEAS EFF				
	GAL PER YEAR				
	GPH				
120 GAL PER DAY	SEAS EFF				
	GAL PER YEAR				
	GPH				

Figure 15. AFUE Output Format

fuel firing rates. The non-dimensional fuel firing rate is expressed in terms of α (the overfiring ratio) such that α equal to one is a properly sized unit, α equal to two is 100% oversized, and so on. The domestic hot water load is an average value, and it is assumed that there is sufficient storage capacity in the system to satisfy peak instantaneous hot water needs. In boilers using "tankless coils" to supply domestic hot water, fuel firing rates generally above 1.1 gallons per hour must be maintained to satisfy the peak domestic hot water demand.

The laboratory efficiency measurements and BNL computational procedures provide the technical bases by which the annual fuel consumption of heating units can be quantitatively compared, assuming a structure with known thermal features and a properly installed and serviced heating unit. The actual annual fuel consumption of field installed heating units cannot be predicted on the basis laboratory measurements and computer programs alone. It is ultimately influenced by many parameters affecting the heat load of a specific house, in a specific city, during a specific heating season, and subject to an infinitude of variables including: house size, shape, quality of insultation, wind exposure, air-tightness, internal heat sources, site topography, landscaping effects, and conditioned room air venting (through exhaust fans and chimneys via barometric dampers or draft diverters, and fireplace flues). Additional variables include those pertaining to heating system installation such as distribution losses, design firing rate, location of heating equipment in the building, (use/non-use of conditioned air for combustion) and servicing (cleanliness and adjustment of boiler and burner components). The BNL measurements and calculational program provide an accurate comparison of the "intrinsic merits" of heating units operating within any prescribed standard structure. Additionally, it provides a highly accurate basis from which to begin further systems comparisons where the myriad of non-standard individual practices may serve to modify the so-determined standard results.

For example, the design heat load of a well-studied structure could be significantly increased by unwise use of bathroom and kitchen exhaust fans that are dependent on the individual life styles of residents. The warm air exhausted by these fans increase the air infiltration and heat load of the building. Also, long uninsulated runs of piping (or ducting) of boiler (or furnace) heat transfer fluid in unheated portions of the building could lead to substantial heat loss and degradation of operating efficiency. This factor is solely dependent on the wisdom of the equipment installer. The list of possible schemes by which one may degrade efficiency performance of an individual building is almost without limit.

Another feature of building structures that has received considerable attention is the contribution of fuel-fired (air-consuming) heating equipment to the building infiltration heat load. There are many variables that will affect this factor including the precise heating unit location within the structure, and local building air-tightness in the vicinity of the heating unit. A heater located within the conditioned space in a central area of the structure would have a larger impact on infiltration than a unit located in a laundry room on an outside wall, in a "drafty" unheated basement, or in an unheated garage. The total impact of oil-fired hydronic heating units on building infiltration must be considered to be small, and vary by a factor between zero and 100% depending on the specifics of the individual installation (see Section II-C-4-(b)).

One additional feature that should be mentioned is the degree to which jacket losses from the boiler contribute to the heating load of the structure. The BNL computational program assumes that jacket losses are <u>not</u> available for space heating. As previously mentioned, many heating units are located in unheated and drafty portions of the building where boiler heat loss cannot directly contribute to heating the structure. Also, while jacket losses may produce locally elevated temperatures in the boiler room, it is unlikely that the majority of this heat is distributed evenly throughout the structure. Combustion air and draft diverter air flows incurred by the heating unit continually exhaust the heat from jacket losses up-the-stack during both burner "on" and "off" cycles. In addition, conduction losses through building walls can increase in the vicinity of the heating unit in response to localized high temperature. Of course, there are cases in which jacket losses may supply useful heat to the building. The BNL laboratory measurement procedure includes the measurement of jacket losses, and it is possible through the computation program to evaluate the effect of this heat source on seasonal efficiency for specific cases. It is particularly important to note that jacket losses for specific boilers may vary substantially from one to another. Accordingly, no "rule of thumb" can be reliably used to estimate these losses. The shortcomings of stack gas analysis for determination of steady state efficiency is in part due to the requirement that jacket losses be included in the calculation. By measuring the steady state efficiency by the BNL enthalpy method, jacket losses can be deduced from the differences between these values and the apparent corresponding stack gas results.

B. Annual Fuel Use and Efficiency Calculation Procedure

1. Introduction

The AFUE calculation provides the procedure by which detailed laboratory efficiency measurements for a particular heating unit can be expressed in terms of annual fuel use and seasonal efficiency over a range of operating conditions. Annual fuel use is determined directly by totaling hourly fuel consumption over the entire year. The calculation depends on precise measurement of burner on-cycle and burner off-cycle heat losses for the heating unit to be evaluated. Annual fuel use and seasonal efficiency results can be presented for a variety of design firing rates, domestic hot water loads, building design heat loads, and geographic locations.

AFUE results for a specified heating unit can be applied to field situations, provided that the *in situ* design heat load of the structure is known. Presently, precise deisgn heat loads of structures are not easily determined under field conditions. However, work is being performed at BNL in conjunction with the State University of New York at Stony Brook to develop a procedure for accurate field measurement of building heat loads and this will enable AFUE results to be applied to specific field-installed heating equipment. In all cases, the results of AFUE analysis can be used to compare the relative merits of various heating units on a common basis. Precise efficiency measurements that are performed in the laboratory are translated to annual fuel consumption data for identical operating conditions. The AFUE calculation is useful as both an absolute and relative measure of heating equipment efficiency performance.

2. Heat Balances and Computer Calculation of Annual Fuel Use[7]

Conventional residential heating units generally feature on-off control modes. That is, the burner is capable of firing at only one fuel flow rate and the burner firing period reduces to satisfy reduced heating requirements, resulting in burner on-off cycling. Accordingly, thermal efficiency can be divided into two distinct parts. Heat losses arising during burner operation are generally characterized by a steady state efficiency (η_s). Heat losses occurring during the burner off-period are accounted for by a cycle efficiency (η_c). Total heat losses during both "on" and "off" periods can be accounted for by an overall efficiency (η_o) which is the product of steady state and cyclic components.

$$\eta_o = \eta_s \eta_c$$

For example, with a steady state efficiency of 70% and cyclic efficiency of 80%, the overall efficiency is 56%.

To determine annual fuel use it is necessary to specify the design firing rate (nozzle size) of the heating unit. Optimally, at design conditions (as the outside temperature reaches the design temperature) the burner should operate continuously for properly sized equipment as the total heat supply is balanced by the total heating requirement. Heat supplied by the burner is the product of the design oil flow rate, $\dot{m}_{oil}$ (nozzle size), steady state efficiency ($\eta_c = 1$ for continuous operation) and the heating value of the fuel, h. The total design heating requirement is the sum of the design load, L_d, and the domestic hot water load, H_o. An equation can be written to equate these two quantities, where steady state design load conditions are exactly met by the steady state heat output of the heating unit.

$$\dot{m}_{oil} \eta_s h = L_d + H_o \tag{1}$$

It has been observed by field investigators that many heating units are installed with firing rates in excess of the design value. We can define an overfiring ratio (α) to account for the factor by which the heating equipment is overdesigned beyond peak conditions:

$$\dot{m}_{oil} \eta_s h = (L_d + H_o)\alpha \tag{2}$$

In field installations, α equal to two is not uncommon, corresponding to systems overfired by 100%. One factor contributing to overfiring is the <u>rate</u> at which domestic hot water is used. The total daily heat load resulting from domestic hot water use is small compared to the instantaneous rate at which the hot water is utilized. Larger nozzles are often installed to satisfy the peak usage. An obvious solution to this problem is the use of domestic hot water storage tanks used in conjunction with reduced firing rate.

After the design fuel flow rate has been determined, fuel use can be calculated for each hour of the year based on hourly outside temperature data. Each outside temperature (provided in one degree F intervals) corresponds to a specific space heat load that is less than the design heat load, L_d. Therefore, the burner will operate for less than 100% of the time, and burner off-cycle heat loss must be incorporated into the heat balance. The laboratory cycle efficiency data provide the functional relationship between burner fractional on-time (F) and cycle efficiency (η_c).

$$\eta_c = f\ (F)$$

where f is a non-linear function measured in the laboratory, (e.g., see Figure 1b).

We can write a heat balance for the system at various outside temperatures by including the burner cycle efficiency. Cycle efficiency, η_c, is a function of the burner fractional on-time, F, which can be related to the outside temperature. As the outside temperature increases, the heat load decreases producing smaller values of F and smaller values of η_c. The heat balance can be expressed as:

$$\dot{m}_{oil}\ F\ \eta_s\ \eta_c\ h = L_d \left[\frac{T_{in} - T}{T_{in} - T_d} \right] + H_o \qquad (3)$$

where: T is the variable outside temperature

T_{in} is the inside temperature

T_d is the outside design temperature

The quantity in brackets is a non-dimensional temperature, corresponding to the fraction of design-inside temperature differential. Note that for T greater than T_d, F (the burner fractional on-time) becomes smaller than 1, and $\eta_c = f(F)$ also becomes smaller than one. At design conditions, F and η_c equal one, the quantity in brackets goes to one, and equation (3) reduces to equation (1).

The overfiring ratio can be incorporated by solving equation (2) for L_d and combining the results with equation (3):

$$\dot{m}_{oil}\, F\, \eta_s\, \eta_c\, h = \left[\frac{\dot{m}_{oil}\, \eta_s\, h}{\alpha} - H_o\right]\left[\frac{T_{in} - T}{T_{in} - T_d}\right] + H_o$$

solving for the product of F and η_c:

$$F\, \eta_c = \left[\frac{1}{\alpha} - \frac{H_o}{\dot{m}_{oil}\, \eta_s\, h}\right]\left[\frac{T_{in} - T}{T_{in} - T_d}\right] + \frac{H_o}{\dot{m}_{oil}\, \eta_s\, h}$$

For each outside temperature (T_i) there is a unique burner fractional on-time (F_i) corresponding to a unique cycle efficiency ($\eta_{c,i}$).

$$F_i\, \eta_{c,i} = \left[\frac{1}{\alpha} - \frac{H_o}{\dot{m}_{oil}\, \eta_s\, h}\right]\left[\frac{T_{in} - T_i}{T_{in} - T_d}\right] + \frac{H_o}{\dot{m}_{oil}\, \eta_s\, h} \qquad (4)$$

Thus, we have obtained an equation in which each outside temperature (T_i) can be used to solve the right-hand side of the equation. H_o α, $\dot{m}_{oil}$, η_s h, T_{in} and T_d are all known quantities. The left-hand side of the equation is the product of laboratory measured quantities.

Solving the right-hand side of equation (4) for a specific outside temperature (T_i) determines a unique value for the product $F_i\ \eta_{c,i}$ which corresponds to a unique value for the burner fractional on-time, F_i. Once F_i is determined, fuel consumption in gallons (M_i) can be calculated for that outside temperature by multiplying F_i by the design fuel flow rate, $\dot{m}_{oil}$.

$$M_i = F_i\ \dot{m}_{oil}$$

For each outside temperature T_i there is a corresponding fuel consumption M_i. The total fuel consumed at temperature T_i is the product of M_i and N_i, the number of hours per year during the heating season at temperature T_i.

$$M_{i,t} = N_i\ F_i\ \dot{m}_{oil}$$

To obtain the total fuel use for the heating season, M, we sum over all outside temperatures.

$$M = \sum_i M_{i,t} = \sum_i N_i\ F_i\ \dot{m}_{oil}$$

For example, with an outside design temperature of $0^{o}F$ and an inside temperature of $68^{o}F$, the summation would be:

$$M = \sum_{i=o}^{68} M_{i,t} \tag{5}$$

where M is the total volume (in gallons) of fuel used during the heating season for space heating and domestic hot water generation.

Fuel use during the non-heating season (summer months) for production of domestic hot water can be calculated by:

$$F' \eta_c = \frac{H_o}{\dot{m}_{oil} \eta_s h}$$

Once F' is determined fuel use during the non-heating season, M', can be found by summation:

$$M' = \sum_{\substack{\text{summer} \\ \text{hours}}} F' \dot{m}_{oil} \qquad (6)$$

Total annual fuel consumption M_{annual} is equal to the sum of heating season and non-heating season fuel use:

$$M_{annual} = M + M' \qquad (7)$$

The AFUE Computer Program uses equation (2) to calculate the design fuel flow rate ($\dot{m}_{oil}$) once L_d has been chosen, for a variety of α and H_o values. η_s is determined by laboratory measurement. During the computational process, the overfiring ration (α) and the domestic hot water load (H_o) are varied to monitor equipment performance under various operating conditions. The AFUE program calculates M_{annual} by use of equations (4), (5), (6) and (7) for each value of α (corresponding to a specific design fuel flow rate), and for each value of domestic hot water load (H_o). Ten-year averaged hour-by-hour weather data for each specified geographic location provide the values for N_i, the number of hours per year at each outside temperature.

The seasonal efficiency is calculated based on the total heat available from the quantity of fuel that is consumed. The "ideal" annual fuel use is recalculated for a "perfect system" in which both steady state and cycle efficiencies are taken to equal 100 percent. The resulting annual fuel use is divided by the actual fuel use to provide a value for seasonal efficiency.

$$\text{Seasonal Efficiency} = \frac{\text{Perfect System Fuel Use}}{\text{Actual Fuel Use}}$$

Manual fuel use calculations could be used instead of a computer to obtain the same fuel use results. A computer program is used because of the substantial savings in time compared to manual methods.

C. Example of Laboratory Test Results and Annual Fuel Use and Efficiency Computation

Results of laboratory tests performed on a dry-base, single pass, vertical fire-tube boiler, equipped with a conventional non-retention head burner are presented in Figures 1a and 1b of Section II-A-1. Steady state efficiency data are plotted as a function of outlet water temperature in Figure 1a and the effect of burner on-off cycling is presented in Figure 1b. As the burner fractional "on" time is reduced, (burner "off" time is increased) the overall efficiency decreases corresponding to larger off-cycle boiler heat losses. The overall efficiency includes burner on-cycle and off-cycle heat losses and is plotted for the entire range of heating unit loads from stand-by to full load (steady state operation). Annual fuel consumption and seasonal efficiencies have been calculated by the AFUE procedure and the results are provided in matrix form (see Table 1a and 1b) for a variety of domestic hot water loads and design fuel firing rates.

Table 1a provides the computed fuel use and efficiency for a building located in the New York City area with a design heat load of 50,000 Btu per hour, while Table 1b is calculated for the same location and a design load of 25,000 Btu per hour. Each of the parameters at the top of the output page can be varied. The seasonal efficiency is abbreviated S.E., and is presented together with the annual fuel use as a function of the domestic hot water load (gallons of water per day) over a range of overfiring ratios (α). Each α corresponds to a unique design oil flow rate (firing rate in gallons of oil per hour), and as the overfiring ratio is increased, the seasonal efficiency drops and annual fuel use increases. For example, in Table 1a at 40 gallons per day of domestic hot water, increasing α from 2 to 3 corresponds to increasing the design firing rate from 1.07 to 1.61 gph, and a resulting decrease in seasonal efficiency from .567 to .502. The corresponding increase in fuel consumption is 186 gallons per year from 1437 to 1623 - a 13% increase in annual fuel use. This example demonstrates the significant effect of overfiring on the quantity of fuel required to satisfy the same heat load for the particular heating unit being considered. All equipment that has been tested at BNL substantiate this result with varying degrees of performance degradation depending on the particular heating unit, and its part-load efficiency performance. It has been observed that both steady state and cyclic efficiencies vary over a wide range of values for commercially available equipment of varying design.

D. Summary

The BNL Annual Fuel Use Efficiency calculation in conjunction with precise laboratory measurements, provide the basis by which the intrinsic merits of individual heating units can be compared under a full range of "standard" conditions. "Standard" conditions include any given design heat load, any given hour-by-hour weather pattern for the heating season, any given range of domestic hot water requirements, and any given cycle characteristics which may be employed in a home heating strategy. AFUE analysis cannot be used to evaluate the annual fuel consumption of a particular building/heating unit system, unless <u>all</u> specific building heat sources and losses can be incorporated into a total heat load. Instead, the strength of the procedure is its simplicity and ability to provide a quantitative measure by which equipment of various designs can be compared on a realistic and common basis.

V. SUMMARY AND CONCLUSIONS

A. Direct and Indirect Efficiency Measurement Techniques

The efficiency test facility at Brookhaven provides quantitative evaluation of oil-fired heating equipment efficiency. The Absolute Enthalpy Flow Technique is a direct and accurate measure of the useful heat produced by heating units, and the Annual Fuel Use and Efficiency calculation permits translation of test results to seasonal efficiency and annual fuel consumption values. Together, the direct measurement technique and AFUE computation provide a precise and common basis for comparing the performance of specific heating units.

In contrast, efficiency measurement procedures used by other organizations are often based on indirect measurement techniques. A commonly employed method consists of flue gas analysis to deduce steady state (full load) efficiency only, and assumption of a constant "power burner draft factor" to characterize burner off-cycle heat loss. It has been demonstrated that steady state efficiency can be overestimated by as much as ten percent (or more) by use of flue gas analysis. In addition, using a constant "draft factor" for all power burners assumes that the cycle efficiency of all these burners is the same. In fact, this is not true, and part-load efficiency of heating equipment has been shown to vary over a wide range of values.

An example of annual fuel uses for two boilers tested at Brookhaven, compared to the expected fuel use by indirect testing methods, can provide quantitative evidence of the inherent weakness of the indirect method. Using the two boilers of Figure 4b in Section II-A-4, it can be observed that a significant difference exists in the efficiency of these two boilers during both steady state and cyclic operation. Recall that the upper curve represents test results for a wet-base cast iron boiler with a retention-head burner. The steady state efficiency by direct measurement is 81.0%, and the average overall (seasonal) efficiency for α equal to two, and 40 gallons per day of domestic hot water, is 75.2%. Therefore, the cyclic efficiency is 92.8%. Annual fuel use (see Table 4) is 1080 gallons of oil per year for space heating and supply of domestic hot water under the above conditions. In contrast, the dry-base, single pass steel boiler, represented in the lower curve of Figure 4b, operates with a steady state efficiency of 69.2% and an average overall efficiency of 56.7% for the same conditions imposed on the wet-base cast iron boiler. The average cycle efficiency for the dry-base steel boiler is 81.9%, and the annual fuel use is 1440 gallons of oil for the same heating load satisfied by the wet-base cast iron boiler. There exists a 25% difference in the quantity of fuel consumed annually by these two boilers (equivalent to 360 gallons of oil) in order to satisfy identical space heat and domestic hot water requirements.

Similarly, we can calculate the expected variation in fuel use for the same two boilers by utilizing stack gas analyses and <u>assuming</u> a "power burner draft factor." The steady state efficiency of the wet-base cast iron boiler is 82% by stack gas sampling which is close to the value measured by enthalpy flow. However, the stack gas steady state efficiency for the dry-base steel boiler is 77%, compared to 69.2% measured directly by the enthalpy flow method. In this case, the stack gas method overestimates the steady state efficiency by more than 10%. At the same time, the cyclic efficiencies of the two boilers are assumed to be equal by use of a constant "power burner draft factor."

Therefore, the total annual fuel use is assumed to be a function of the steady state efficiency only, and the stack gas sampling procedure predicts a difference in annual fuel use of only 6% for the two boilers.

The indirect measurement technique relies on many simplifying assumptions, and annual fuel consumption predicted by this method indicates a small variation of only 6% for the two boilers presented in Figure 4b. In contrast, direct measurement of the useful heat produced by the same two boilers under identical operating conditions provides a difference in annual fuel use of 25%. The indirect method of analysis cannot successfully differentiate between highly efficient and inefficient heating equipment. Therefore, any economic analyses of payback periods, or the economic feasibility of increased capital expenditures for equipment of high efficiency are in error based on indirect measurement procedures.

B. Field-Installed Heating Equipment

A study performed to determine the efficiency of 100 field-installed oil-fired heating units indicated an average steady state efficiency (by stack gas sampling) of 72%,[6] and subsequent studies provided similar results.[3,4] The average stack temperature was 690°F and the average carbon dioxide concentration was 8½%. Unfortunately, while cycle efficiencies were not measured, these data indicate that the performance of typical heating units currently encountered in the field are more closely represented by the boiler of Section II-A-1, the dry-base steel boiler with non-retention-head burner. It can be concluded that there exists considerable potential for efficiency improvement through new equipment installation and implementing refit modifications as described in Section II of this report. Also, it should be noted that indirect efficiency measurement techniques, which are used by field studies, tend to overestimate equipment efficiency as discussed earlier, and it is possible that potential fuel savings will be underestimated.

Proper installation of heating equipment in the field can have a significant impact on the efficiency of these units. There exists an endless list of improper installation practices that will degrade the efficiency of heating units. Failure to insulate hot water piping for boiler systems and warm air ducting for furnace systems is an example of unwise installation practice. Long uninsulated runs of hot water piping from the boiler through unheated basements and crawl spaces can reduce the amount of useful boiler heat delivered to the heated space, and reduce operating efficiency. Similarly, uninsulated warm air ducting passing through cold attic areas can degrade the efficiency of furnace systems. Careful attention must be placed on these and all other equipment installation procedures in order to maximize the efficiency of the heating unit/house system.

C. Information Dissemination

The results of the testing program are made available to the public in several ways. Findings Reports are prepared for each unit tested at Brookhaven, providing detailed performance and annual fuel use information. Topical reports are prepared for general types of equipment and refit modifications that are tested, and an annual report is prepared summarizing the testing accomplishments for each fiscal year.

Detailed verbal presentations of project work are provided on site at Brookhaven and other locations throughout the country by program staff members. Groups that were given presentations include:

Manufacturing and Oil Supply Related Groups

Empire State Petroleum Association (in New York City and Syracuse, New York)

Oil Heat Institute of Long Island
National Old Timers Association (of Energy Related Businesses)
National Oil Jobbers Council
Blueray Systems, Inc.
Thermodynamics Corp.
Fuel Oil News
Fuel Oil and Oil Heat

Federal and Other Governmental Organizations

FEA (Washington and New York representatives)
National Bureau of Standards
Office of Research, New York State Public Service Commission
ERDA Public Meetings
Congressman T. J. Downey
Office of Research & Development, U.S. Coast Guard
Bureau of Energy Resources, Nassau County, N. Y.
Department of Environmental Affairs, Suffolk County, N. Y.
Nassau County Service Organization

Other Groups

NAHB
Commonwealth Gas Company of Massachusetts
ASHRAE Combustion Committee TC3-7
ABC Radio, Washington (R. Peterson)
Newsday (New York area newspaper)
IGT
Booz-Allen & Hamilton
Department of Engineering, Ontario Research Foundation
Long Island Lighting Company
Scientific Energy Systems, Inc., Massachusetts
Lehman Brothers (major stock investigation group)
Energy Systems, Inc., New York, N. Y.

Battelle Laboratories

Adelphi University, Long Island, N. Y.

State University of New York at Stony Brook,
Mechanical Engineering Department

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