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Energy Conservation Potential of Modular Gas-Fired Boiler Systems

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The modular concept of boiler operation was examined in a laboratory test of five gas-fired, cast iron, hydronic boilers. Four of the boilers, each having an input rating of 85,000 Btu per hour, were arranged so that they could either be operated like a single boiler (i.e., all of the boilers either on or off) or as a modular installation in which the boilers are sequentially fired to match the number in operation with the heating load. The fifth boiler had an input rating of 300,000 Btu per hour and was operated as a single boiler installation. Efficiency versus heating load curves were obtained for the single boiler installation, the four small boilers run like a single boiler and the modular installation operated with and without water flowing through the "idle" modules. These efficiency curves were then used to theoretically predict the effect of the modular concept and boiler oversizing on the seasonal efficiency of gas-fired heating plants. It was found that under certain conditions the use of a gas-fired modular boiler installation instead of a single large boiler could result in considerable energy savings.

Key words: Boiler oversizing; efficiency versus heating load; modular boilers; modular concept; seasonal efficiency.

1. Introduction

Interest in the efficient heating of buildings has been increasing as this country heads into an energy shortage. One type of heating plant which has been advocated as being more efficient than many others is the modular boiler installation. Proponents of modular boilers base their claim for higher efficiency upon the "modular concept" of equipment operation. This concept, which has also been applied to compressors, engines, fans, pumps, etc., is that many advantages can be gained by employing several small machines (modules) rather than a single large machine. These advantages include easier installation and repair, standby capacity (if one of the modules should become inoperative) and increased system efficiency. It is this potential for increased efficiency and thus energy conservation of the modular boiler installation that will be considered in this paper.

In the modular boiler heating plant, the modules are sequentially fired to match the heat output with the heating load. This differs from a conventional plant employing a single large boiler, which would normally handle heating loads that are less than the maximum heating capacity of the boiler by intermittent operation or "cycling". It is generally believed that the larger the percentage of time off, the greater are the off-cycle losses. Such an effect will result in a drop in efficiency of the large boiler as the heating load decreases. If this drop is very rapid, the seasonal efficiency of the single large boiler could be considerably lower than its efficiency at maximum load because a heating system rarely operates at full capacity during the heating season. In the modular system, on the other hand, only a single small boiler need cycle, thereby operating at less than full load efficiency. The other modules that are in operation, the number depending upon the heating load, will run continuously at their maximum efficiency. Whether or not the seasonal efficiency of a modular system is indeed better than the seasonal efficiency of a single large boiler will be determined by how the efficiency of each system behaves as a function of heating load. The functional dependence of efficiency on load for the modular system will in turn depend upon whether controls are employed to prevent water from flowing through the idle modules.

To test the modular concept of boiler operation, five gas-fired, cast iron, hydronic boilers were obtained from a leading manufacturer. Four of the boilers had input ratings of 85,000 Btu per hour*, while the fifth had a rating of 300,000 Btu per hour. The design and construction of the large boiler was similar to that of the small units. The boilers were installed in an experimental setup in which either the single large boiler or a modular system consisting of the four small boilers could be tested at various loads. Since the manufacturer of the boilers normally used the 300,000 Btu unit as the basic module in modular installations, the modular system tested represented a scaled down version of a modular heating plant that might be installed in a commercial building. Neglecting the valving and instrumentation, which are discussed in detail in the next section, the experimental setup consisted of a closed loop of pipe through which water, heated by the boiler or boilers being tested, was circulated. The heating load was simulated by removing hot water from this loop and replacing it with cold water. By changing the rate at which this hot water was replaced by cold, different heating loads could be achieved. The efficiencies of various boiler systems were measured at different loads and the results plotted to give efficiency versus load curves. The systems for which this was done are:

1. The modular system with "primary pumping" in which water was allowed to flow through all the modules.

2. The same system as #1, except that the pilot lights were turned off in the modules that were not in operation (idle modules) at a given test load.

*Adherence to the International System of Units has not been followed in the interest of effective communication with the expected readers of this paper.

3. The same system as #2, except that "secondary valving" allowed water to flow only through the modules which were either operating continuously or cycling.

4. The single large boiler.

5. The modular boilers run together as if they were a single boiler, i.e. all of the modules either on or off.

System #2 was tested to determine if the pilot lights on the idle modules amounted to a significant energy waste in the modular system with "primary pumping" (water passing through all modules). It was possible that, if this turned out to be the case, the pilot lights in system #1 could be penalizing the scaled down modular system considerably more than they would an installation made up of larger boilers. The efficiency versus heating load curves for systems #4 and #5 were compared with those for system #1, the modular system with "primary pumping", and with system #3, the modular system with "secondary valving", in order to evaluate the modular concept of boiler operation.

The data obtained from testing these five gas-fired boilers are contained in the section on Experimental Results. The Discussion deals with the implication of these results and the potential effect of the modular concept and oversizing on the seasonal efficiency of a gas-fired heating plant.

2. Apparatus

Boiler Setup

Schematics of the experimental setup are shown in Figures 1,2 and 3, while Figures 4 and 5 are pictures of the actual installation. As can be seen from the piping diagram in Figure 1, when the single large boiler was being tested, water could be shut off to the small boilers by closing valves a,b,c,d, and e. Similarly when the modular installation consisting of the four 85,000 Btu/hr modules was operating, water could be prevented from circulating through the large boiler by closing valves f and g. In this paper, a test of the modular system with "primary pumping" shall mean that water was continuously circulated through all the modules. A test of the modular system with "secondary valving" shall indicate that the valve between the return header (a,b,c, or d) and an "idle" boiler (i.e., one which is neither operating continuously or cycling) was closed to prevent water from circulating through the boiler. In addition, during the secondary valving tests, the pilot lights on the idle modules were also turned off. This last configuration was examined to determine the effect of the losses from the idle boilers on the efficiency of the modular system.

The supply and return headers were covered with a one-half inch thick layer of foamed plastic insulation in order to reduce the heat loss. Three lengths of three-inch pipe were employed to increase the water capacity of the single boiler test setup and the modular boiler test setup to approximately 41 and 43 gallons, respectively. This was done so that the test setup would have approximately the same water capacity as similar heating plants installed in a real building. The purpose of this was to obtain the proper on - off cycling rate.

Natural gas was supplied to all five boilers by a single header which can be seen in Figure 2. A combined pressure regulator - gas valve on each boiler allowed for the reduction of gas pressure from 8 or 9 inches of water column in the header to the 3 1/2 inches of water column in each boiler's manifold that was recommended by the manufacturer. This valve also allowed for manual on-off control of the gas to the pilot light and manual and automatic on-off control of gas to the main burners. It was this valve that was manually operated to turn off the gas to the idle modules in test configurations 2 and 3.

A schematic of the breeching and duct work are shown in Figure 3. The draft diverters were standard factory equipment and were supplied with the boilers. No mechanical draft inducer was used, although two cast-iron dampers, D_1 and D_2, were installed to block out the draft to the system (the single large boiler or the modular setup) which was not in operation.

Heating Load Simulation

The heating load consisted of piping losses between S and R and the heat removed from the system by dumping hot water at I and replacing it with cold water at J. (see Figure 1).

Since the maximum heating load which a system can handle equals its maximum output under steady state conditions, simulating the maximum heating load for the single boiler or the modular system corresponded to removing, by the above means, as much heat as was being transfered to the circulating water by continuous operation of the single large boiler or the four small boilers, respectively. As will be explained in the section on experimental procedure, the flow rate was adjusted by means of valve h to give a 20 F degrees temperature rise to the water passing through the heating plant under maximum heating load conditions. Thus simulating the maximum heating load of either the single boiler or modular boiler system was equivalent to replacing hot water with cold water at such a rate that the temperature of the circulating water was lowered by 20 F degrees as it traveled from the end of supply header S to the entrance of the return header R. This resulted in a constant temperature distribution throughout the system and steady state operation. To achieve this 20 F degree temperature drop, a motorized valve K on the hot water discharge pipe I and an electronic proportional controller were employed. The signal from a two junction thermopile, which measured the temperature difference between points L and N, was fed into the controller which adjusted the motorized valve to maintain a given temperature difference between L and N. As an example, 180°F supply water underwent a temperature drop of about 1.5 F degrees in passing through the piping between the end of the supply header and point L. Thus the controller would be adjusted to provide an additional 18.5 F degree decrease between L and N in order to obtain a total temperature drop between S and R of 20 F degrees.

The primary reason for employing such a complicated control system however, was not to simulate the maximum heating load, since the application described above could have been handled using a manually operated valve.

The real function of this system was to simulate a heating load when a boiler or boilers was cycling on and off. It was decided to accomplish this for a load equal to X% of the maximum heating load by replacing hot water with cold water in such a manner that the temperature of a volume of water entering the return header would be $\left[\frac{X}{100} \cdot (20)\right]$F degrees lower than its temperature upon leaving the supply header. This would have the effect of removing the same amount of heat from each volume of circulating water and would result in the instantaneous rate at which heat was extracted from the system (the instantaneous load) being constant with time. Since the supply temperature would, however, be changing with time, it was necessary to vary the rate at which hot water would be replaced by cold to achieve this goal. Thus if the temperature of the water at L increased (decreased) the controller would cause the motorized valve to decrease (increase) the rate at which cold water replaced hot water, since less (more) cold water was now needed to lower the temperature of a volume of circulating water by the same number of degrees. The effect of this was that, in simulating a load equal to X% of the maximum heating load, the temperature of a volume of water would first be decreased approximately 1.5 F degrees in passing through the pipe between S and L and then by another $\left[\left(\frac{X}{100}\right) (20) - 1.5\right]$ F degrees between L and R.

If the above process for simulating the heating load worked exactly as described, a plot of the temperature of the water entering the return header against time would have the same profile as a plot of the temperature of the water leaving the supply header against time, except that the former would be displaced by the time it took a volume of water to travel from S to R and would be $\left[\frac{X}{100} \cdot (20)\right]$ F degrees lower in temperature. In actual practice, this turned out to be approximately the case, although in tests involving cycling, the instantaneous load did tend to vary slightly with time in a cyclic manner. The result was a slight flattening out of the high temperature peaks and a slight filling in of the low temperature valleys. Figure 6 is a plot of a typical supply and return temperature and an ideal return temperature corresponding to an instantaneous heating load which is independent of time.

Control

A solid state sequence controller supplied by the manufacturer was used to cycle on and off both the single boiler and the modules in the modular boiler installation. As used in a normal modular heating plant, the controller would accept signals from a temperature sensor in the supply header at S and from an outside temperature sensor. It would then sequentially turn the modules off as the supply temperature increased or sequentially turn the modules on as the supply temperature decreased. In addition, the controller used the signal from an outside temperature sensor to in effect raise the supply temperature setting as the outside temperature decreased in order to avoid the undesirable effect of having lower supply temperatures at higher loads.

To avoid the questions of how the heating load was related to the outside temperature and what effect lowering the outside temperature had on the supply temperature, all tests on both the modular system and the single large boiler were run with the cycling unit having approximately the same on and off supply temperature settings. This was done by bypassing the outdoor sensor and using a single step on the controller. The normal temperature control differential was 10 F degrees, with the on-setting at 175°F and the off setting at 185°F. However, the performance of the controller was such that while fairly repeatable control was achieved during any single efficiency test, the on and off set points, and to a lesser extent the control differential, tended to vary by ±2F degree from day to day. Thus for a 10 F degree control differential, the cycling boiler might go on (off) at 173°F (183°F) on one test and on (off) at 177°F (187°F) on another test. In addition, while most tests had a control differential of approximately 10 F degrees, several tests were run at control differentials of 8 and 12 F degrees. This day to day variation of the controller, however, did not seem to have a large effect and repeating a test always gave a result that was within the estimated experimental error.

3. Instrumentation

The temperature rise of the circulating water across the boilers was measured with an eight junction copper-constantan thermopile and a potentiometer-type, strip chart recorder. The range spans used on the recorder during these tests were 1, 2 and 5 mv and each had an accuracy of 0.25% of span. The accuracy of these spans was verified with a laboratory type potentiometer prior to the start of testing. The ends of the thermopile were inserted in stainless steel wells located at R and S (see Figure 1). Each well had an outside diameter of three-eights of an inch, extended approximately 6.5 inches into the water pipe and was filled with light machine oil. Right angle bends were located one-half foot upstream of R and S in order to mix the circulating water and thereby obtain a more average water temperature at R and S. (The bend upstream of R can be seen at N in Figure 1; the one upstream of S is, however, not shown in this schematic.)

The supply and return temperatures at S and R respectively, and the temperature of the natural gas at the gas meter were also measured using copper-constantan thermocouples. The signals from these thermopiles were fed into a multi-pole switch which was used to select which temperature was to be displayed on a dial-type indicating potentiometer. This instrument was checked prior to the start of testing and was found to be accurate to 0.5 F degrees in the temperature range of interest. In order to have a continuous record of the supply and return temperature, other thermopiles at S and R were used in conjunction with an ice bath and a two pen, strip chart recorder. Although the accuracy of the readings obtained on this recorder were only 1°F, this was sufficient for monitoring the behavior of the supply and return temperatures with time.

The meter used to measure the amount of water passing through the boiler was a 1 1/2 inch turbine meter having a linear response in the flow range from 17.4 to 174 gallons per minute. The flow rates encountered in the various efficiency tests ranged from 21 to 27 gallons per minute and were thus well within this linear response region. Magnetically generated voltage pulses from the turbine meter were fed into a flow totalizer. The meter was factory calibrated and had a reported accuracy over its entire range of ±0.4% and a reported repeatability of ±0.1%. Although these figures provided by the manufacturer will be assumed correct for this experiment, a check was made to determine if the turbine meter and totalizer were working correctly. This was done by passing water through the turbine meter and measuring the length of time required to fill a small tank.

An electric clock was used to measure the duration of a test. The clock, the chart motors on the two strip chart recorders and flow totalizer were started and stopped simultaneously by means of a single switch and several relays.

The meter employed to measure the gas used was an ordinary household gas meter which was donated to NBS by a local gas company. The meter's register had 0.5 and 5 cubic foot proofing dials and 1,000, 10,000, 100,000 and 1,000,000 cubic foot metering dials. It was calibrated in the Fluid Meter Section of NBS and was found to have the ratios of actual to indicated volumes shown in Table 1. In all except two efficiency tests, the amount of gas used by the system being tested was determined by counting the number of revolutions made by the 0.5 and 5 cubic foot proofing dials. In the two exceptions, the tests were run for the period of time required for the 1,000 cubic foot metering dial to make one revolution. This latter method, however, required too much time, especially at low heating loads. The gas pressure at the meter was measured using a U-tube manometer and an aneroid barometer. The manometer and the meter had minimum scale divisions of 0.05 inches of water and 0.02 inches of mercury, respectively. A check was made on the aneroid barometer by comparing it with a mercury barometer; the former was found to agree with the latter to within 0.05 inches Hg.

A MSA Lira Infared Analyzer, Model 300*, was used to measure the percentage by volume of CO_2 in the flue gas from each boiler. The analyzer was calibrated using sample gases having a known concentration of CO_2. Measurements were taken just below the draft diverter and the flue gases were passed through a condensate trap and a float-type flow meter before entering the analyzer. The flow meter assured that the flow rate remained within the limits recommended by the manufacturer. After the flue gas had passed through the analyzer it was returned to the breeching.

The test setup was located in a fully air-conditioned laboratory which could easily handle the heat lost by the boiler jackets, breeching and piping. Because of this, the room temperature was measured using only a single chromel-alumel thermocouple in all tests and a sling psychrometer in a limited number of tests. The thermocouple was located approximately three feet from the back of the large boiler and was shielded against radiation. The sling psychrometer was used to measure the wet and dry bulb temperature at different locations around the test setup. In all measurements taken with the thermocouple and the sling psychrometer, the dry bulb temperature was always within ± 4 degrees F of 79°F and the wet bulb temperature was always within ± 3 degrees F of 61°F.

The flue gas temperature of each boiler was measured with a single chromel-alumel thermocouple having a brazed junction. It was located approximately in the center of the flue and about two inches from the bottom of the draft diverter. The signals from these thermocouples, along with the signal from the thermocouple used to measure the room temperature, were fed into a multi-pole switch. This switch was used to select the temperature which was to be displayed on a dial-type indicating potentiometer. Prior to the start of testing, the indicating

*Trade names are used in this report as a means of clear identification and neither constitute nor imply endorsement by the National Bureau of Standards.

potentiometer was checked using a more sensitive laboratory-type potentiometer. The flue gas temperature measured in this manner was found to be within 2% of the average flue gas temperature as determined by using a bead-type thermocouple and taking transverse readings across the duct.[1]

During the period of testing, the higher heating value of the gas varied from an absolute maximum of 1033 BTU per standard cubic foot* to an absolute minimum of 1004 BTU per standard cubic foot, as determined from hourly measurements taken by the local gas company at their central plant. This agreed well with the value of 1011 BTU per standard cubic foot which was obtained by the Air Pollution Analysis Section of NBS in a single test of the gas higher heating value. This test, however, revealed that the gas, which came from the NBS mains was exceedingly dry, having a water vapor content which was less than 0.1% by volume. Because the energy content per cubic foot of dry gas at standard conditions** is higher by a factor of $\left(\frac{30}{30 - 0.517}\right)$ than the energy content of a standard cubic foot of gas [2], the following higher heating value was used in calculating the efficiency results present in this paper:

$$(1018 \pm 15) \left(\frac{30}{30 - 0.517}\right) = 1036 \pm 15 \qquad (1)$$

BTU per cubic foot of dry gas at standard conditions.

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*A standard cubic foot of gas is 1 cubic foot of gas saturated with water vapor and measured at a temperature of 60°F and a pressure of 30 inches of Mercury [2].

**Standard conditions are defined to be a temperature of 60°F and a pressure of 30 inches of Mercury.

4. Experimental Procedure

Before the first efficiency test was performed, the gas regulator on each boiler was adjusted so that the heat input to each boiler was within 2% of its rated input and the pressure regulator on the cold water feed line was adjusted to obtain a water pressure inside the boilers of approximately 35 psig. In addition, since the manufacturer recommended that the water passing through each boiler have a 20 F degree temperature rise under steady state operation, the flow rate of the circulating water was adjusted to give this recommended temperature rise in the modular system under maximum heating load simulation. This was accomplished by allowing the four small modules to run continuously and adjusting valve h and the electronic proportional controller (and thus the motorized valve K) until: (1) a steady state supply temperature of 180°F was achieved and (2) the thermopile across the heating plant indicated a temperature difference of 20 F degrees. Without changing valve h, the single boiler system was then run with the proportional controller adjusted to also give a steady state supply temperature of about 180°F. It was found that the larger pressure drop across the single large boiler reduced the flow rate. This was, however, exactly compensated by the decreased heating capacity of the single boiler to again give a 20 F degree temperature rise to the water passing through the single boiler. Thus a single setting of valve h resulted in roughly a 20 F degree temperature rise in both the modular boiler system and the single boiler system at their respective maximum heating levels. This valve setting was therefore maintained throughout the entire testing period.

The experimental procedure employed in determining the efficiency of the various systems at different heating loads is most easily described by dividing the tests into two categories: steady state tests and non-steady state or cycling tests.

Steady State Tests

In the steady state tests, the simulated heating load was such that the boiler or boilers which were operating ran continuously at their rated input. The result was that the temperature rise through the heating plant was exactly cancelled by the temperature drop caused by the simulated heating load and thus the temperature distribution throughout the system remained constant. This occurred at the maximum heating load for each of the five configurations discussed in the introduction. In addition, it occurred at approximately three-quarters, one-half and one-quarter of the maximum heating load for configurations 1, 2 and 3; that is, it occurred whenever the heating load exactly cancelled the heat input from the steady state operation of three modules, two modules and one module, respectively.

In preparing for a steady state test, the boiler or boilers that were to be run at their maximum heating capacity, were turned on and the supply temperature was allowed to reach approximately 180°F. The electronic proportional controller was then adjusted so that the simulated heating load equalled the heat output of the heating plant and a constant supply temperature between 180°F and 183°F was achieved. The system was then run for at least an hour in order to reach equilibrium. During this period, fine adjustments were made, when necessary, to the controller to keep the supply temperature within the above range. When the supply temperature no longer appeared to be changing, its value was noted and the test was begun. During the test, the system was required to maintain the supply temperature within 3 F degree of this initial value, without further adjustment of the electronic controller. In the majority of these tests, the supply temperature actually varied less than one F degree.

The shortest steady state test lasted 53 minutes, while the majority of them ran for a period of 1 hour. The supply and return temperatures, the temperature rise across the boilers, the duration of the test, the amount of water passing through the boilers during the test, the barometric pressure, the gauge pressure and temperature of the gas at the gas meter, and the volume of gas used as indicated by the gas meter were measured in accordance with the discussion in the Instrumentation Section. This information was used to determine the efficiency of the boiler configuration by the calorimetric method [1]. A discussion of this method and the calculations involved are given in the next section. In addition, the stack temperatures, the percentage of CO_2 in the flue gas, the room's dry bulb temperature, and, in some cases, the room's wet bulb temperature were also measured as described in the section on Instrumentation. Although this was normally done for the purpose of monitoring the installation, information obtained on two tests was used as a check on the experimental procedure. This was accomplished by calculating the stack losses for the modular system and the single large boiler, when both were being run at their respective maximum heating loads. By using these losses plus estimated jacket losses to perform heat balances [3], an independent determination of the maximum efficiency for each system was made which was then compared with the efficiency determined by the calorimetric method. The calculations involved in determining the efficiency of each system by the heat balance method are given in Appendix A.

Non-Steady State Tests

Whenever the output of the heating plant was not instantaneously cancelled by the heating load, the supply temperature varied with time, and cycling occurred. In systems 1, 2 and 3, a single module would cycle on and off while in systems 4 and 5, the single large boiler and all four of the modules, respectively, would cycle.

In tests involving cycling or non-steady state operation, the boiler or boilers that were to be tested at a specific load were run until the supply temperature reached 180°F. The electronic proportional controller was then adjusted to give the desired heating load and the system was allowed to operate until a repeating pattern was observed in the curve giving the temperature rise between R and S with time. This warmup period was usually at least one hour. After it was completed, the clock, recorders, and flow totalizer were started and the efficiency test begun as the cycling boiler came on (or went off). During the test, the amount of gas used, as indicated by the gas meter, was determined as discussed in the Instrumentation Section. With the exception of two long tests which were run until 1000 cubic feet of gas was consumed, the efficiency tests lasted between 53 minutes and 111 minutes. The number of cycles completed by the cycling boiler during the different tests, varied from 3 to 15. A test was ended by turning off the clock, recorders and flow totalizer when the cycling boiler again came on (or went off), after having completed a certain number of cycles. At the start of each test and just before the end of each test, the barometric pressure, the gauge pressure of the gas at the gas meter and the temperature of the gas at the meter were measured. These were then averaged to obtain a mean value of barometric pressure, gauge pressure and gas temperature for each test. This information, along with the results from the gas meter, was used to determine the number of standard cubic feet of gas used during a test. This calculation, as well as the calculations involved in determining the efficiency of a given configuration at a specific load, are discussed in the next section.

The room temperature, stack temperatures and percentage of CO_2 in the flue gas of each boiler were also measured. This information, however, was only used to monitor the behavior of the installation. The supply temperature and return temperature were continuously recorded throughout the test by means of the two pen recorder mentioned in the section on Instrumentation.

5. Calculations

When the calorimetric method is used to determine the efficiency of a hot water boiler installation, the efficiency may be defined as heat transferred to the circulating water during the testing period (Q_{out}) divided by the energy input to the installation in the same period of time (Q_{in}).

In the case of steady state operation, the only constraint on the testing period is that it be long enough to obtain an accurate measurement of Q_{out} and Q_{in}. In addition, since the supply temperature, return temperature and flow rate are constant with time, the heat transferred to the circulating water may be written:

$$Q_{out} = M \, \overline{C} \, \Delta T \quad , \tag{2}$$

where $\overline{C}$ is the average thermal capacity of the water as it passed through the boiler installation, M is the total mass of water flowing through the boiler or boilers during the testing period, and ΔT is the temperature rise of the water. Since the thermal capacity of water is, for all practical purposes, linear over the range of water temperature encountered in this study, one can write:

$$\overline{C} = C\,(\overline{T}) \quad ,$$

where $C(\overline{T})$ is the thermal capacity of water at the temperature $\overline{T}$, and

$$\overline{T} \equiv \frac{\text{supply temperature + return temperature}}{2}$$

is the average temperature of the water as it passes through the boiler installation. Substituting the above equation into (2), we obtain:

$$Q_{out} = M \, C(\overline{T}) \, \Delta T \tag{3}$$

This equation was used to determine the heat transferred to the water during the steady state tests.

With tests involving non-steady state operation, Q_{out} is more difficult to determine. This is true even if the heating system is cycling in a repeating pattern (i.e., with a cyclic supply and return temperature as illustrated in Figure 6), which was the case in the non-steady state efficiency tests. Again using the fact that the thermal capacity of water is very nearly linear over the range of temperatures encountered in the tests, the heat tranferred to the circulating water during the testing period is:

$$Q_{out} = \dot{M} \int_{t_o}^{t_f} C\left(\frac{T_R(t) + T_S(t + \tau)}{2}\right) \left[T_S(t + \tau) - T_R(t)\right] dt \tag{4}$$

where $\dot{M}$ is the mass flow rate and is a constant for any one test,

τ is time required for a mass of water to flow from R to S,

$T_R(t)$ is the temperature of the water at R at the time t,

$T_S(t)$ is the temperature of the water at S at the time t,

t_o is the time at the start of the test; and

t_f is the time at the end of the test.

Equation (4) can be simplified by observing that while

$$\left[T_S\,(t+\tau) - T_R\,(t)\right]$$

varies over a wide range,

$$C\left(\frac{T_R\,(t) + T_S(t+\tau)}{2}\right)$$

changes very little. In fact, in every test

$$C\left(\frac{T_R(t) + T_S(t+\tau)}{2}\right)$$

was always within $\pm$ 0.4% of $C(\overline{T})$, where $\overline{T}$ can now be defined as:

$$\overline{T} = \frac{1}{t_f - t_o} \int_{t_o}^{t_f} \left[\frac{T_R(t) + T_S(t+\tau)}{2}\right] dt \tag{5}$$

If,

$$C\left(\frac{T_R(t) + T_S(t+\tau)}{2}\right)$$

in equation (4) is replaced by $C(\overline{T})$, only a very small error is introduced (see Appendix B) and a considerable simplification is achieved. Making this substitution,

$$Q_{out} \simeq \dot{M}C(\overline{T})\left[\int_{t_o}^{t_f} T_S(t+\tau)dt - \int_{t_o}^{t_f} T_R(t)dt\right] \tag{6}$$

which can be transposed to:

$$\simeq \dot{M}C(\overline{T})\left[\int_{t_o+\tau}^{t_f} T_S(t')dt' + \int_{t_f}^{t_f+\tau} T_S(t')dt' - \int_{t_o}^{t_f} T_R(t)dt\right] \tag{7}$$

If the test period is taken to be the time between the occurrence of two identical events in two separate cycles.

$$\int_{t_f}^{t_f + \tau} T_S(t)\,dt = \int_{t_o}^{t_o + \tau} T_S(t)\,dt \qquad (8)$$

because of the cyclic nature of the supply temperature. Equation (7) then becomes:

$$Q_{out} \simeq \dot{M} C(\overline{T}) \int_{t_o}^{t_f} \left[T_S(t) - T_R(t) \right] dt \qquad (9)$$

$$\simeq MC(\overline{T})\ \overline{\Delta T} \qquad (10a)$$

where
$$\overline{\Delta T} \equiv \frac{1}{t_f - t_o} \int_{t_o}^{t_f} [T_S(t) - T_R(t)]\,dt \qquad (10b)$$

and use is made of the fact that $(\dot{M})\ (t_f - t_o) = M$.
Equation (10a), which is very similar to (3), was used to determine Q_{out} in all efficiency tests involving cycling. In almost all cases, the test was begun when the cycling boiler(s) came on and ended several cycles later when the same boiler(s) again cycled on. The quantities $\overline{T}$ and $\overline{\Delta T}$ were found by evaluating the integrals in (5) and (10b) by determining the area under the curves:

$$\frac{T_R(t) + T_S(t + \tau)}{2}$$

and $(T_S(t) - T_R(t))$, respectively, and then dividing by the time $(t_f - t_o)$. The curves were obtained from the strip chart recorders and the area under them was approximately determined by counting squares.

The energy input during the testing period (Q_{in}) was found by determining the number of cubic feet of gas at standard conditions used during the test and multiplying this by the higher heating value in (1). The number of cubic feet of gas at standard conditions that were **burned** were found by using the ideal gas law to relate the volume of gas which the meter indicated was used to the volume the gas would occupy at standard conditions. The formula used was:

$$V_s = V_m \, R \left(\frac{520}{T_m}\right)\left(\frac{P_m}{30}\right) \tag{11}$$

where V_s is the volume of gas at standard conditions which was used,

- V_m is the volume of gas used according to the meter in cubic feet,
- R is the appropriate ratio of actual to indicated volume (see Table 1),
- T_m is the gas temperature at the meter in degrees R, and
- P_m is the gas pressure at the meter in inches of Hg.

In the steady state tests, the instantaneous heating load equaled the rate at which heat was transferred to the water by the boiler or boilers being tested. This was not, however, true in the non-steady state efficiency tests. In these tests, which involved a repeating pattern of operation, the heat removed from the water equaled the heat transferred to the water only over a period of time equal to one or more complete cycles. Although the rate at which heat was removed was fairly constant with time, there was a tendency for it to have a slight cyclic fluctuation, as mentioned in the section on apparatus. This was primarily because the electronic controller was often a little slow in responding to a very rapid change in the supply temperature, such as occurred when a cycling boiler first came on. As a result of this, an average heating load, $<\ell>(t)>$, was defined, which is given by:

$$<\ell(t)> \equiv \frac{1}{(t_f - t_o)} \int_{t_o}^{t_f} \ell(t)\,dt = \frac{Q_{out}}{(t_f - t_o)} \tag{12}$$

where $\ell(t)$ is the instantaneous heating load at time t, and the last equality follows from the fact that the heat removed during the test

$$\int_{t_o}^{t_f} \ell(t)dt$$ must equal the heat output, Q_{out}, since $(t_f - t_o)$ is the time required for several complete cycles. Since equation (12) is true for the steady-state tests as well, $\langle \ell(t) \rangle$ will be used in plotting the results for both cycling and non-cycling tests.

6. Experimental Results

The experimental results which were obtained by the calorimetric method are shown in Figure 7,8, and 9. The dotted lines in these figures represent an extrapolation of the data down to zero load. The curves are not normalized since both the single boiler and the modular system had approximately the same maximum efficiency. Figure 7 gives the efficiency versus load curves for systems 1 and 2. As mentioned in the Introduction, system 2 was tested to determine if the pilot lights were penalizing the scaled down modular system in system #1 more than they would an installation made up of larger boilers. This turned out not to be the case since the two curves in Figure 7 are nearly identical.

Figure 8 contains a plot of efficiency versus heating load for system #3, i.e. the modular system with secondary valving. In this test, the flue losses were minimized by shutting off the water and gas to the modules which were neither operating continuously or cycling (idle modules). The unusual drop in the curves at a heating load approximately equal to 25% of the maximum heating load, is due to the fact that when the load on the second module is small, the flue losses, jacket losses and lower efficiency of this module tend to decrease the efficiency of the system. Similar although smaller drops in efficiency can be expected to occur when the heating load is approximately equal to 50% and 75% of the maximum heating load. These were not, however, experimentally investigated because the decreases in efficiency were expected to be less than the experimental error. In addition, these discontinuities will become smaller as the number of modules in an installation increase. The efficiency versus load curves for system #4, the single large boiler, and system #5, the modular system run like a single boiler, are shown in Figure 9. There was very little difference in the results obtained for these two systems.

The efficiency versus load curves for systems #1, #3 and #4 and #5 are compared in Figure 10. It can be seen that the curve obtained for the modular system with primary pumping is almost identical to the curve for the single boiler and the curve for the modular system run as a single boiler. There is however considerable difference between the results from these three systems and those from the modular system with secondary valving. The latter had a higher efficiency over the entire range of heating loads tested.

The efficiencies of the single boiler and modular boiler installations at their maximum heating loads were respectively found to be .765 and .775 by the heat balance method described in Appendix A. This agrees well with the efficiencies which were obtained at the same loads by the calorimetric method (see Appendix A).

7. Discussion

Since nearly identical efficiency versus heating load curves were obtained for the modular system with primary pumping (system #1) and the modular system run like a single boiler (system #5), the modular concept of boiler operation proved no better, in our test setup with the water flowing through the idle boilers, than the conventional method of operation (i.e. all on or all off). This result is also expected to hold for an installation containing more than four of the 85,000 BTU/hr gas-fired modules tested. The reason for this is that while the losses, at a given percentage of the maximum load, in a modular installation with primary pumping increase as the number of modules in the installation increase, the heat output tends to rise proportionally. For example, at approximately 25% of the maximum load, a modular installation with primary pumping containing 16 boilers will have losses from twelve idle modules and output from four operating ones. This is a ratio of three idle modules per operating one, which is the same as that found in our test setup at 25% of the maximum heating load.

A comparison of the efficiency versus load curves for system #1 and system #4 indicates that, for this boiler design and this size range, there was practically no difference in efficiency between the modular system with primary pumping and the single boiler tested. Whether or not a modular installation containing a larger number of 85,000 BTU/hr gas-fired units and having primary pumping would be better than a single large boiler would depend on the efficiency versus load curve of the single large boiler choosen for comparison. If the efficiency of this single boiler dropped slower than the efficiency of the 85,000 BTU/hr module as the percentage of the maximum heating load decreased, the single boiler would be more efficient. If it dropped faster, the modular system with primary pumping would be more efficient. In the latter case, however, the increase in system efficiency is not attributable to the modular concept since operating the modular system like a single boiler would achieve the same result.

In our test setup, the modular concept of boiler operation resulted in an improved system efficiency when the modules operated independently of each other. This occurred in system #3, the modular system with secondary valving, since in this case the idle boiler losses were minimized and did not greatly affect the efficiency of the operating units. The purpose of testing this system was to simulate the employment of self ignitors, automatic valves, and pumps, which would light the gas on a module and circulate water through it only when the output from that module was needed. The heating system could be arranged as in the NBS test setup or as shown in Figure 11. The latter arrangement is usually referred to as "primary-secondary pumping" [4,5], and differs from system #3 in that; 1) the boilers, in our experimental setup were in series with the primary water loop whereas in Figure 11 they are in parallel with the primary loop, 2) there was only a single pump in the NBS test setup and consequently the flow rate through the operating modules increased as the water to more and more idle boilers was cut off, and 3) in system

#3 the pilot lights on the cycling module operated continuously and the water circulated through such a boiler even when the unit was off. It is, however, felt that these differences will not have an appreciable effect on efficiency and therefore the tests results from system #3 are also representative of an installation employing primary-secondary pumping.

As discussed under Experimental Results, the efficiency **versus** load curve obtained from system #3 is higher than the curve obtained from the single boiler (system #4) and the modular system run like a single boiler (system #5). In the region where data was obtained the difference in system efficiencies varied from about 1% at 60% of the maximum load to between 10% and 14% at 10% of the maximum load (see Figure 10). To illustrate how this might affect the seasonal efficiency of each system, test data from a report [6] published by the Engineering Experiment Station, The Ohio State University, Columbus has been used. These data, which are shown in Table 2, give the percentage of time that a given heating load (or cooling load) existed in the Legal Aids building and the Devonshire School building during the testing period. Both of these buildings were located in Columbus, Ohio. A plot of their load distribution function, $f(\ell)$, is contained in Figure 12, where $f(\ell)\Delta\ell$ is the fraction of time the load was between ℓ and $\ell + \Delta\ell$. These graphs show that the boiler in the Legal Aids building was fairly well sized for the maximum required heating load and that the one in the Devonshire School building was slightly oversized. The load pattern of the latter building was rather unique in that, while heating loads up to 90% of the boiler capacity were occasionally required, during 68.8% of the testing period the heating load was between 0.0 and 30%. In contrast, the Legal Aids building had a heating load between 30 and 60% of its boiler's capacity for 50.8% of its testing period.

Although the boilers in both buildings had a considerably larger heating capacity than our modular boiler installation, it was assumed that the curves in Figure 10, for the modular system with secondary valving and the modular system run like a single boiler, are respectively representative of:

1) a modular system with primary-secondary pumping containing four large gas-fired modules and 2) a single gas-fired boiler system, either of which might be installed in each of these two buildings. The seasonal efficiencies of these two hypothetical systems were calculated by multiplying the efficiency of each system at a given heating load, by the percentage of time at that heating load, summing over all heating loads and dividing by the total percentage of time that a heating requirement existed. Five cases were investigated: 1) both heating systems installed in the Legal Aids building and both having the same capacity as the existing boiler in that building, 2) both systems installed in the Legal Aids building but both having been oversized by 50%, 3) both systems installed in the Legal Aids building but both having been oversized by 100%, 4) both systems installed in the Legal Aids building but both having been oversized by 200%, and 5) both heating systems installed in the Devonshire School building and both having the same capacity as the existing boiler in that building. Table 3 summarizes the results obtained. For the Legal Aids building, the difference in seasonal efficiency between the two systems ranged from approximately 4% when the systems were well sized to roughly 10% when the systems were 200% oversized. It should be noted in the latter case, however if many more smaller modules had been used in the modular installations, its seasonal efficiency could probably have been increased from 65.5% to between 72 and 75%. This would have resulted in a difference in seasonal efficiency between the modular boiler system with primary-secondary pumping and the single boiler of between 16 and 19%. For the Devonshire School building, the two systems differed in seasonal efficiency by approximately 9%. Again if more modules had been used, this difference could have been increased to between 11 and 15%. Comparing the calculated seasonal efficiencies in Table 3 for cases 1 and 5, it is apparent that the skewed load pattern in the Devonshire building caused about a 10% lower seasonal efficiency for the single boiler and about a 5% lower seasonal efficiency for the modular system containing four modules.

The above calculations indicate that employing a gas-fired modular boiler installation with primary-secondary pumping instead of a single large boiler could result in considerable energy savings - especially when there is a chance that the system might be oversized or the building to be heated has a load distribution function whose mean value occurs at a small percentage of the maximum heating requirement. It must be emphasized however, that this example does not recommend or condone the installation of oversized heating plants, but only illustrates that in the unfortunate case where oversizing does occur its effect is less detrimental to the modular system with primary-secondary pumping than to the single boiler system.

Acknowledgements

The author would like to express his appreciation to Mr. James D. Allen for his assistance in reducing data and in performing several of the efficiency tests. The help of Messrs. Walter M. Ellis and John W. Grimes in setting up the instrumentation and the support and encouragement during this investigation of Dr. James E. Hill are gratefully acknowledged.

8. SI Conversion Units

In view of the present accepted practice in this country for building technology, common U.S. units of measurement have been used throughout this paper. In recognition of the position of the United States as a signatory to the General Conference on Weights and Measures, which gave official status to the metric SI system of units in 1960, assistance is given to the reader interested in making use of the coherent system of SI units by giving conversion factors applicable to U.S. units used in this paper.

1 inch = 0.0254 meter (exactly)

1 inch of water at 60°F = 248.8 pascal

1 psi = 6894.1 pascal

1 cubic foot = 0.02832 cubic meter

1 gallon = 0.003785 $meter^3$

1 BTU = 1055 joule

1 pound = 0.4536 kilograms

Temperature, °C = (°F - 32) 1/8 (exactly)

9. References

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[4] Sotirious Contos, Economics of Primary - Secondary Pumping of Modular Heating Plants, ASHRAE Journal, December 1972, p.34.

[5] C. G. Segeler, Match Heating Loads With Modular Boilers, Power, June 1972, p.102.

[6] A Study of Commercial Buildings Heated with Gas, Report EES 269x, Engineering Experiment Station, The Ohio State University, Columbus (Project Supervisor: Charles F. Sepsy, Department of Mechanical Engineering).

10. Appendix A

In this appendix, the steady state efficiencies of the single large boiler and the modular system, operating at their respective maximum heating load, are determined by performing heat balances on the two systems. The heat balance method sets the heat input per cubic foot of gas at standard conditions equal to the heat output per cubic foot of gas at standard conditions. Since the latter consists of useful heat (heat transferred to the circulating water) and losses, the following equation is obtained:

useful heat output per cubic ft. of gas at standard conditions =

= higher heating value of gas at standard conditions - losses per cubic ft. of gas at standard conditions. (A1)

Dividing both sides of (A1) by the higher heating value of the gas, the efficiency of the heating plant is found to be:

$$\text{efficiency} = 1 - \frac{\text{Losses per cubic ft. of gas at standard conditions}}{\text{higher heating value of gas at standard conditions}} \quad \text{(A2)}$$

The losses in a boiler may be broken down as follows [3]:

1. dry flue gas loss,
2. loss due to evaporation of water formed from burning hydrogen in the fuel,
3. radiation and convection losses from the boiler jacket,
4. loss due to heating the moisture in the entering combustion air,
5. loss due to moisture in fuel,
6. loss due to CO and other combustibles in the flue gases,
7. loss due to unconsumed carbon in refuse.

In our tests, the major losses were items 1, 2 and 3. The losses from item 4 will be neglected here, since for a dry bulb temperature of 79°F and a wet bulb temperature of 61°F, the amount of water vapor which is present in the air has only a very small effect on the efficiency. The loss due to the presence of water vapor in the fuel need not be considered because, as mentioned in the section on Instrumentation, the gas was extremely dry. In addition, gas burns very clean and produces little CO when there is excess air present, as was the case. Thus the losses from items 6 and 7 are also neglegible and will not be considered.

Although the single large boiler and the modular system will be discussed separately, the following information is common to both systems and will be used in determining the losses from items 1, 2 and 3:

a. the heating value of the gas is 1036 BTU per cu ft of gas at standard conditions,

b. the rooms' dry bulb temperature is approximately 79°F,

c. approximately 0.10 lb of water is formed upon burning one cubic ft of gas at standard conditions,

d. radiation and convection losses from the boiler jacket for a modular heating plant are approximately 1.5% of the energy input [4].

The uncertainity associated with these values and the flue gas temperatures and CO_2 concentrations to be given below will not be considered.

The Single Large Boiler

In the steady state test of the single large boiler, the flue gas temperature was found to be 525°F and the percent by volume of CO_2 in the flue gas was 8.8%. From Figures 13 and 14, which are nomographs obtained from reference [3], we find that the dry flue gas loss and loss due to hydrogen in the fuel are respectively 102 and 125 BTU per cubic foot of gas at standard conditions. The loss due to radiation and convection from the boiler jacket equals (.015) (1036) or 16 BTU per cubic foot of gas at standard condition consumed. Inserting these losses into equation (A2) we obtain:

$$\text{efficiency} \cong 1 - \frac{(102 + 125 + 16)}{1036} = 0.765$$

This efficiency is in excellent agreement with the value of .761 which was obtained by the calorimetric method.

The Modular System

In the steady state test of the modular system, modules 1, 2, and 3 were found to have a flue gas temperature of approximately 425°F and a percent by volume of CO_2 in their flue gas of 7.6%. Module 4, however, was slightly less efficient, having a flue gas temperature of 462°F and a concentration of CO_2 in its flue gas of 6.9% by volume. From Figures 13 and 14 we find that the dry flue loss and loss due to hydrogen in the gas were respectively 92 and 120 BTU per cubic foot of gas at standard conditions for modules 1, 2 and 3. For module 4, these losses were respectively 110 and 122 BTU per cubic foot of gas at standard conditions. As in the case of the single large boiler, the loss due to radiation and convection from each module jacket will be taken to be 16 BTU per cubic foot of gas at standard conditions. Inserting the losses for each module into (A2), we find for modules 1, 2 and 3:

$$\text{efficiency} \cong 1 - \frac{(92 + 120 + 16)}{1036} = 0.780$$

and for module 4:

$$\text{efficiency} \cong 1 - \frac{(110 + 122 + 16)}{1036} = 0.761$$

The average efficiency for the modular system as obtained by the heat balance method is then:

$$\frac{[3(0.780) + 0.761]}{4} = 0.775$$

The steady state efficiency which was obtained by the calorimetric method for the modular system was .759. Considering that there is an error associated with both methods, these two efficiency values are felt to be in fairly good agreement. The difference in efficiencies obtained by the two methods could easily be the result of underestimating the jacket loss from the small modules in the heat balance method.

11. Appendix B

Error Analysis

The maximum possible uncertainty associated with the efficiency result obtained for configuration 1 through 5 can be estimated by examining the potential errors in Q_{out} and Q_{in} for non-steady state operation at small loads.

The energy transferred to the circulating water, in case of a non-steady state test, is given by equation (10a). This equation may be rewritten as:

$$Q_{out} = (1 \pm 0.004)MC(\overline{T})\overline{\Delta T} \ , \tag{B1}$$

making use of the fact that $C\left(\frac{T_{R(t)} + T_{S(t+\tau)}}{2}\right)$ had to be approximated by $(1. \pm .004)C(\overline{T})$ in order to obtain equation (10a) (see section on Calculations). Since the error involved in determining the total mass of water flowing through the boiler or boilers during the test period is primarily due to the inaccuracy of the turbine meter and the totalizer, the error in M is $\pm 0.5\%$. The error in $C(\overline{T})$, which results from the uncertainty in $\overline{T}$, is very small and may be neglected. The error in the value of $\overline{\Delta T}$ arises from three sources: the error associated with the thermopile used to measure the temperature difference, the inaccuracy of the recording device, and the error resulting from counting the boxes under the curve $(T_S(t) - T_R(t))$ in order to evaluate the integral in (10b). The maximum error from the thermopile* is estimated to be $\pm 1.6\%$. The accuracy of the recorder used in conjunction with the thermopile is $\pm 0.25\%$ of span. Since a full span was rarely used, a better estimate is $\pm 0.5\%$. Estimating the error resulting from counting the boxes under the curve $(T_S(t) - (T_R(t))$ is exceedingly difficult, since overestimating the area in one region will compensate for underestimating it in another. It will be assumed

*The question of whether there is an error resulting from measuring the average temperature of a circulating mass of water using a thermopile inserted in two stainless steel wells has not been considered because of the extreme complexity of this problem.

that this process contributed an error of $\pm 0.5\%$. Combining these errors, equation (B1) may be written:

$$Q_{out} = (1. \pm 0.004 \pm 0.005 \pm 0.016 \pm 0.005 \pm 0.005)MC(\bar{T})\overline{\Delta T}$$
$$= (1. \pm 0.035)MC(\bar{T})\overline{\Delta T} \qquad \text{(B2)}$$

where it is understood that M, $C(\bar{T})$, and $\overline{\Delta T}$ are the measured values.

The error in Q_{in}, the energy input during a test, is the sum of errors in the number of cubic feet of gas at standard conditions used during the test (V_s) and the potential error in the higher heating value of the gas (see equation (1)). Examining equation (11), we find that the uncertainty in V_s is due to any error which might arise in determining V_m by reading the gas meter, the uncertainty in R and any measurement errors associated with T_m and p_m.

The uncertainty in Vm is estimated to be approximately 0.1% in the worst case. The values of R, which are given in Table 1, will be assumed to be sufficiently accurate so that they may be disregarded as a source of error. The temperature T_m and pressure p_m have respective maximum errors of approximately 0.2% and 0.3%. Combining these results, V_s is found to be known to within $\pm$ 0.6%. The higher heating value of the gas is known to within $\pm$ 1.5% if we assume that the gas used in any one test could have had a heating value anywhere between the maximum and minimum values reported by the gas company during the period of testing. The error in Q_{in} is thus found to be $\pm$ 2.1%.

Since efficiency is defined as the ratio Q_{out}/Q_{in}, the maximum error in the measured efficiencies is ($\pm$ 3.5% $\pm$ 2.1%) or $\pm$ 5.6%. As an example, system #5 had an efficiency of 0.66 at 8% load. Multiplying 0.66 by 5.6%, we find that the efficiency of this system at 8% of the maximum heating load could have been higher or lower than the measured result by a maximum value of 0.04 (i.e. efficiency = 0.66 $\pm$ 0.04). It should be understood, however, that the error obtained in this manner is to be considered the maximum possible error. The actual error is expected to be considerably smaller than this, due to the high probability that the various errors will not be additive.

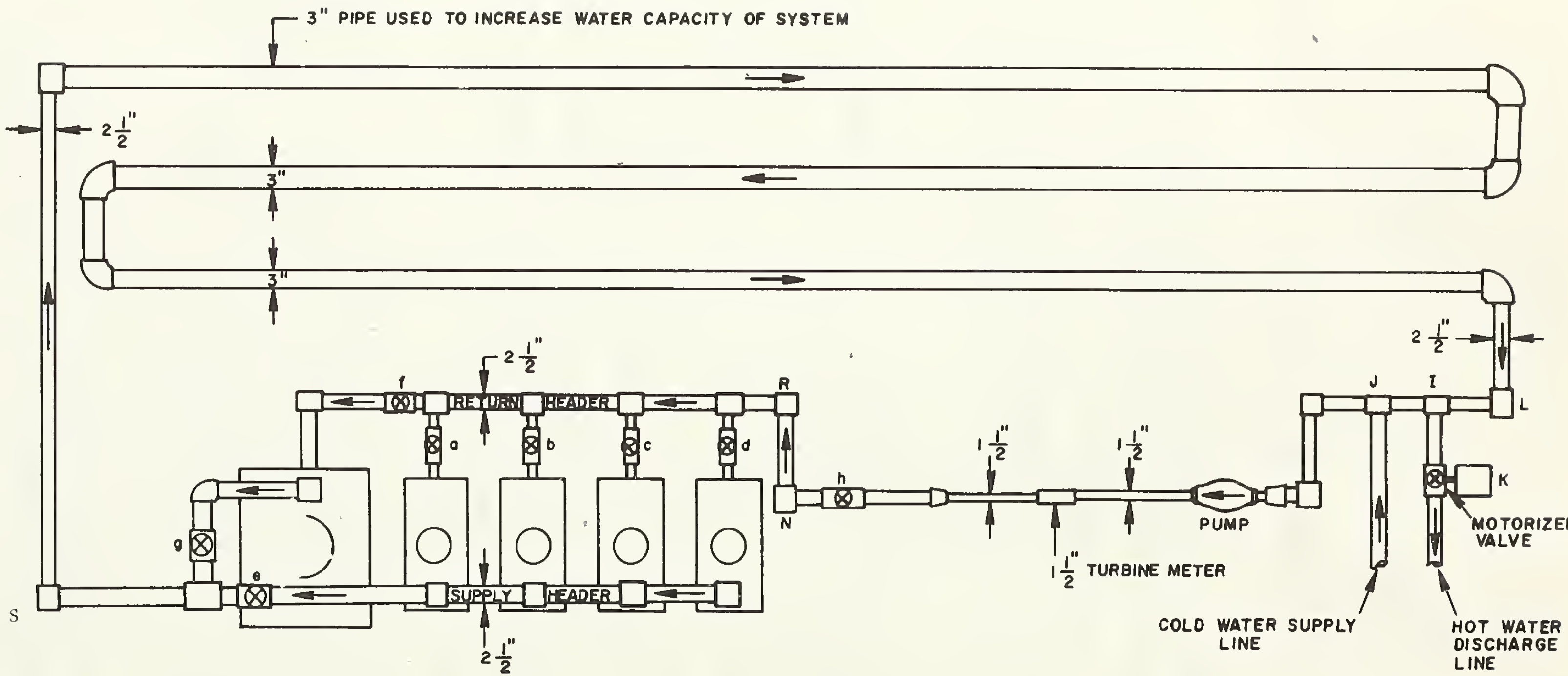

Figure 1. Piping schematic of the test installation.

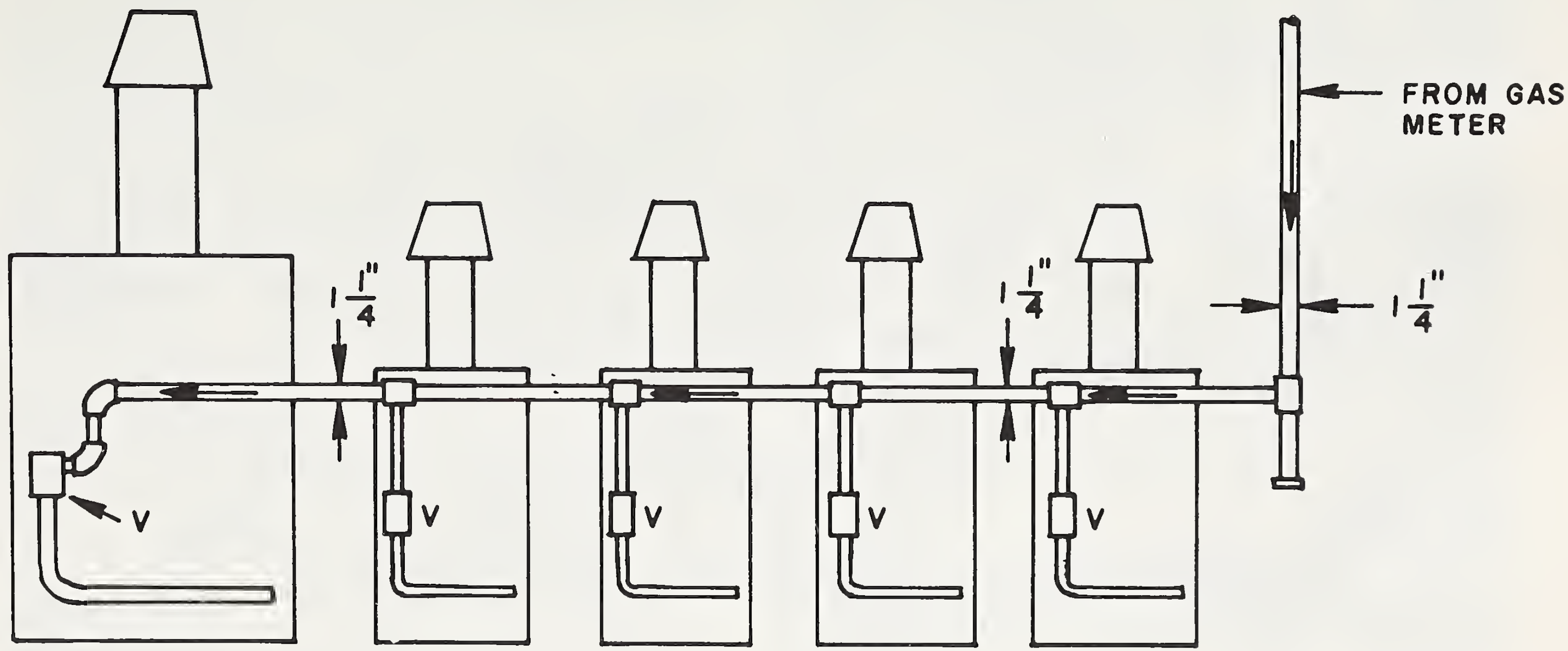

Figure 2. Gas header to all five boilers.

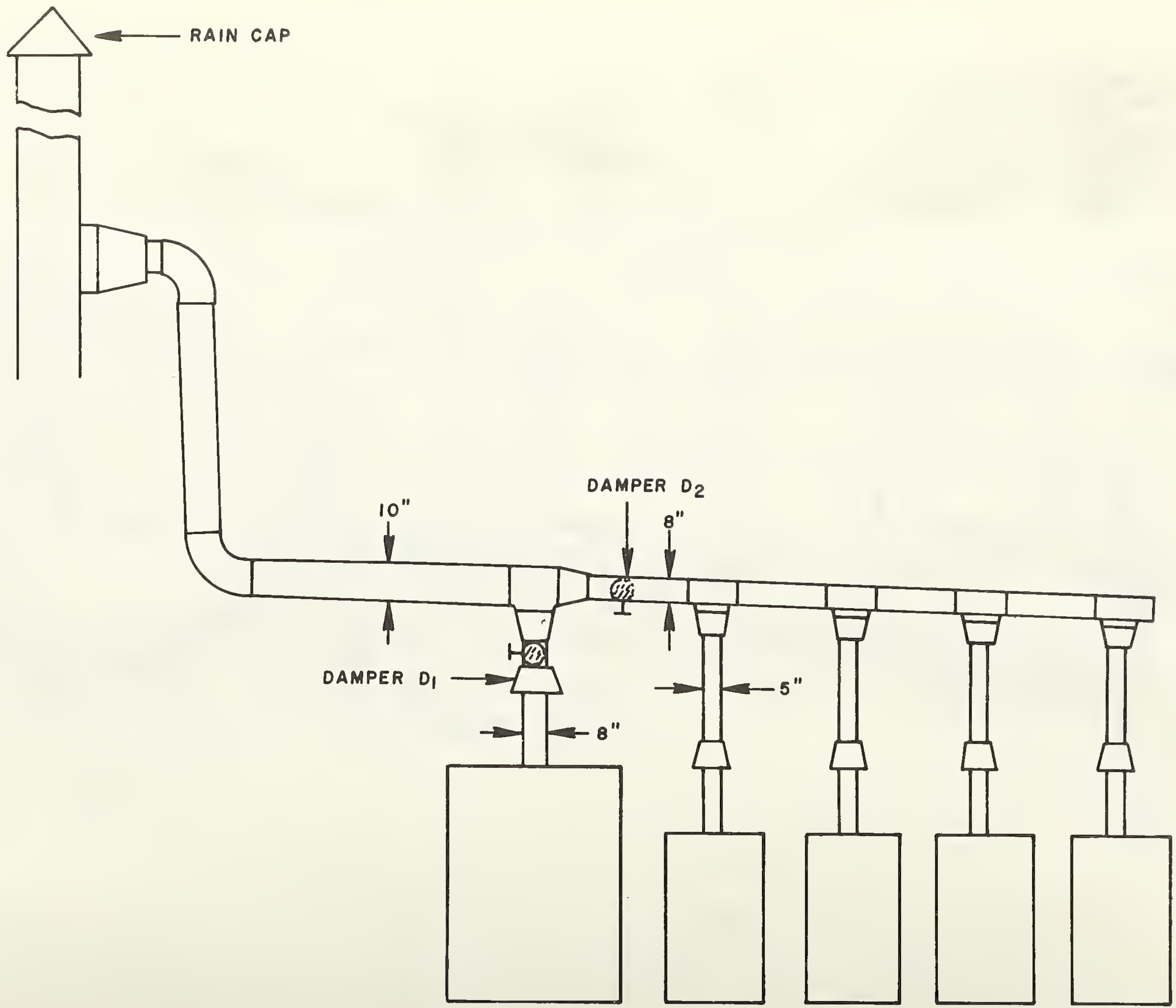

Figure 3. A schematic showing the breeching arrangement.

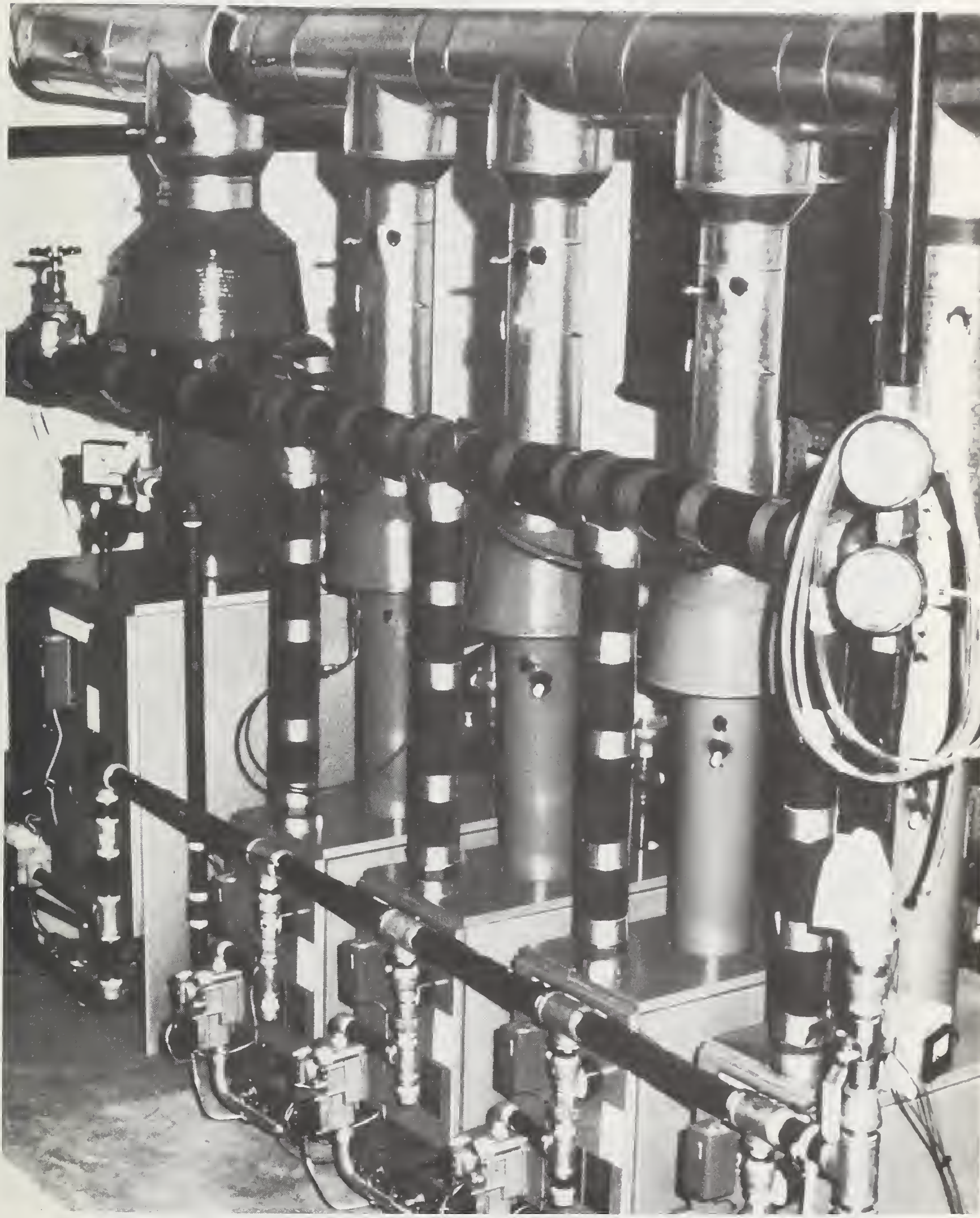

Figure 4. Front view of experimental setup.

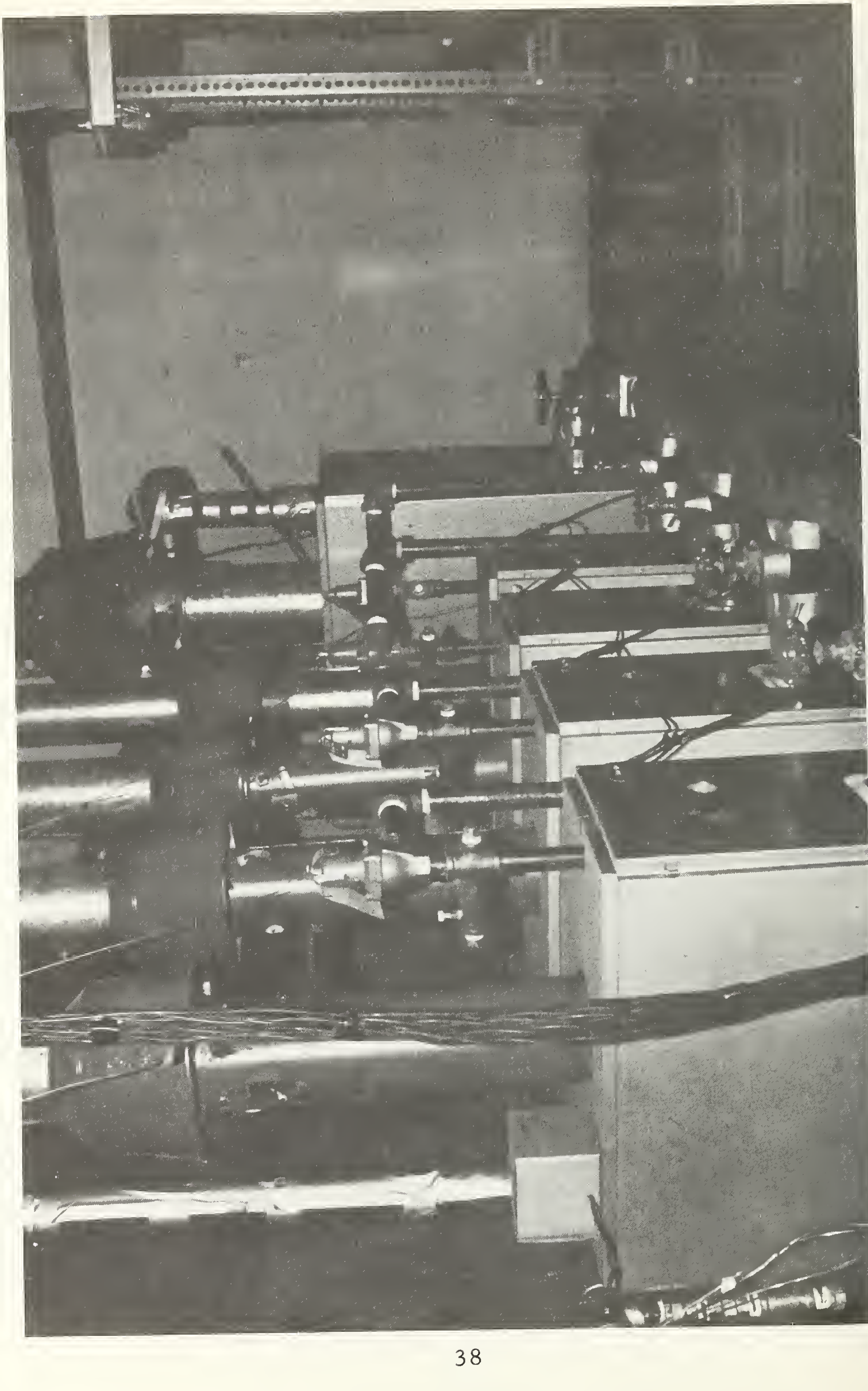

Figure 5. Rear view of experimental setup.

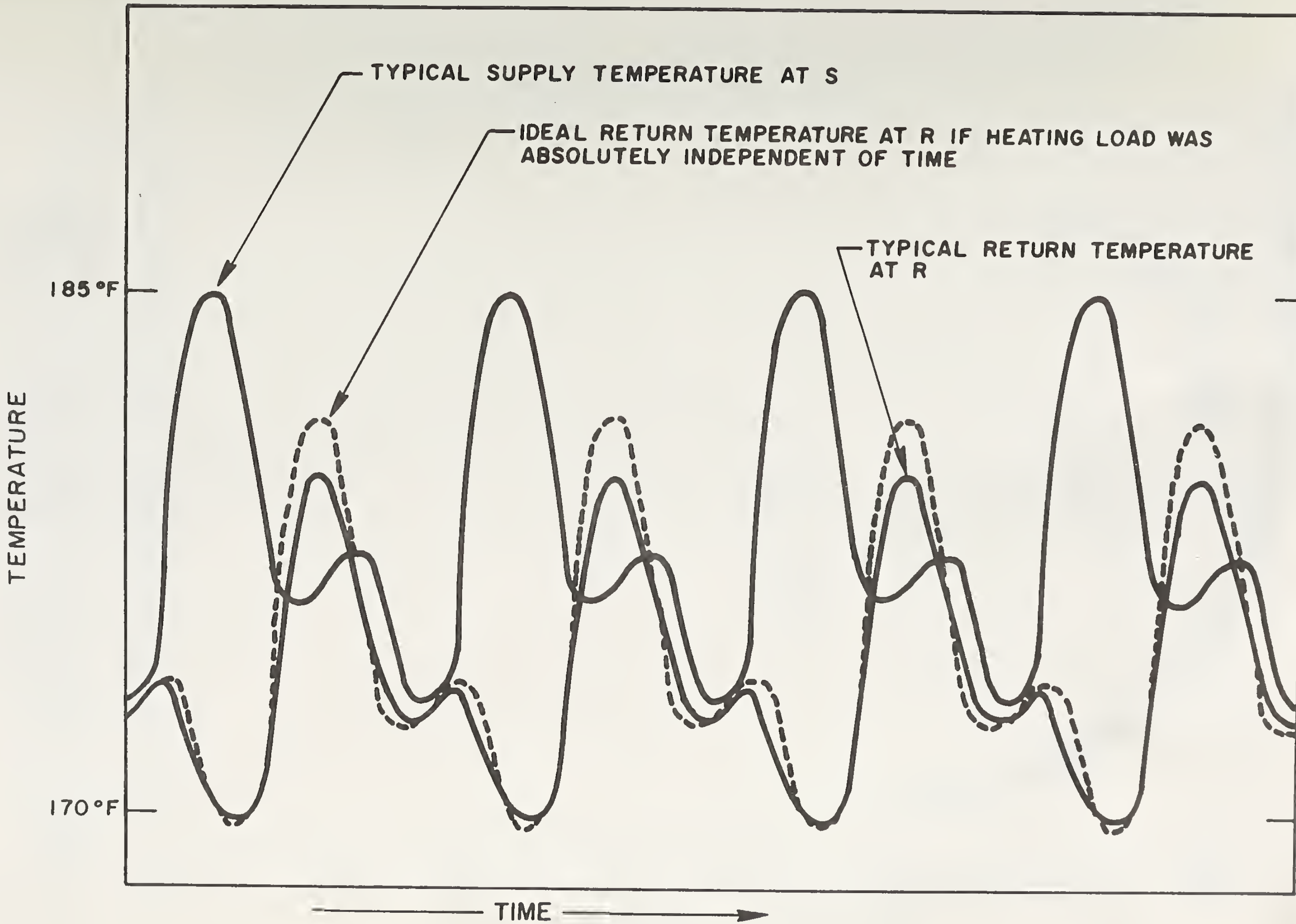

Figure 6. A plot of a typical supply and return temperature along with an ideal return temperature.

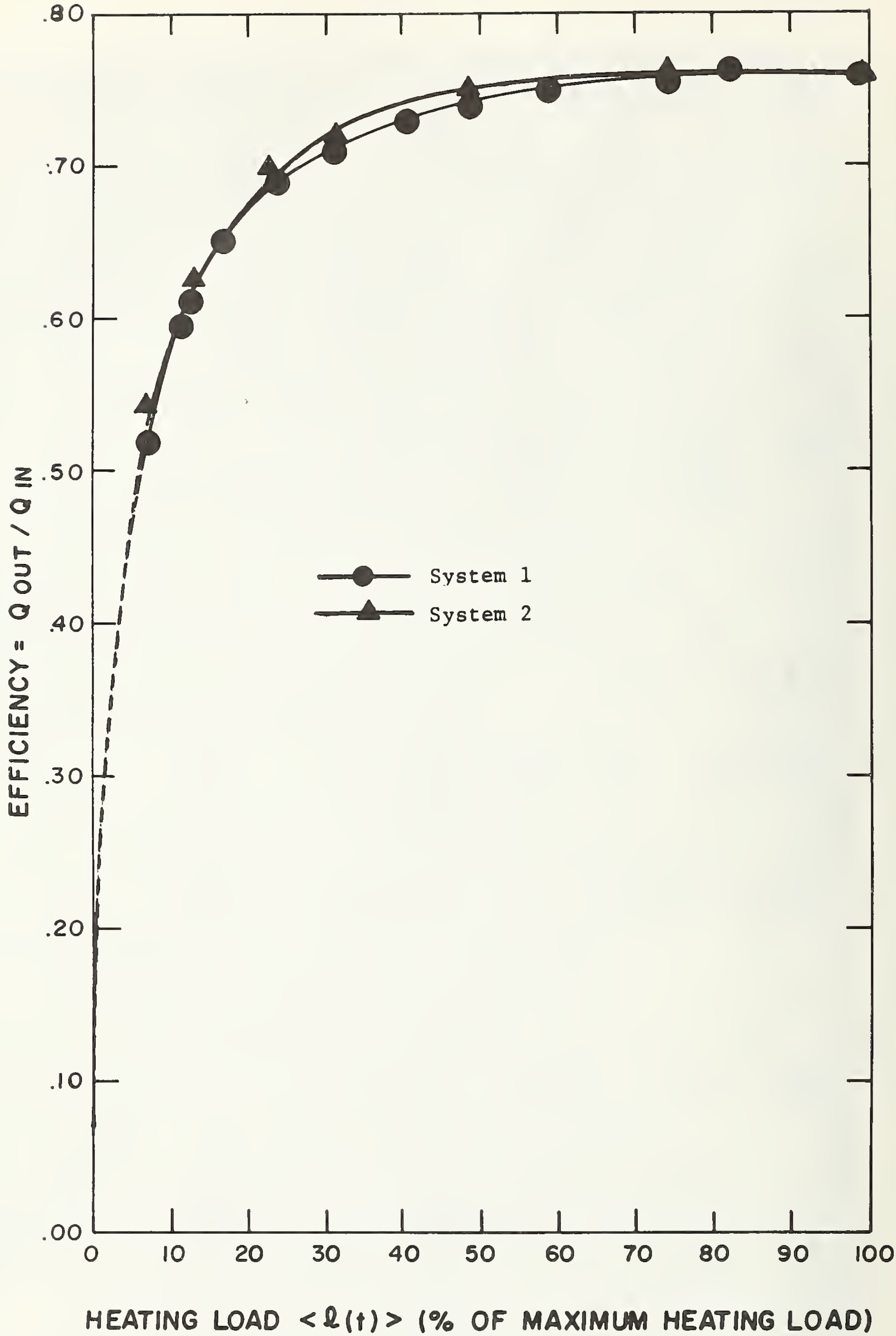

Figure 7. Efficiency versus heating load curves for systems 1 and 2.

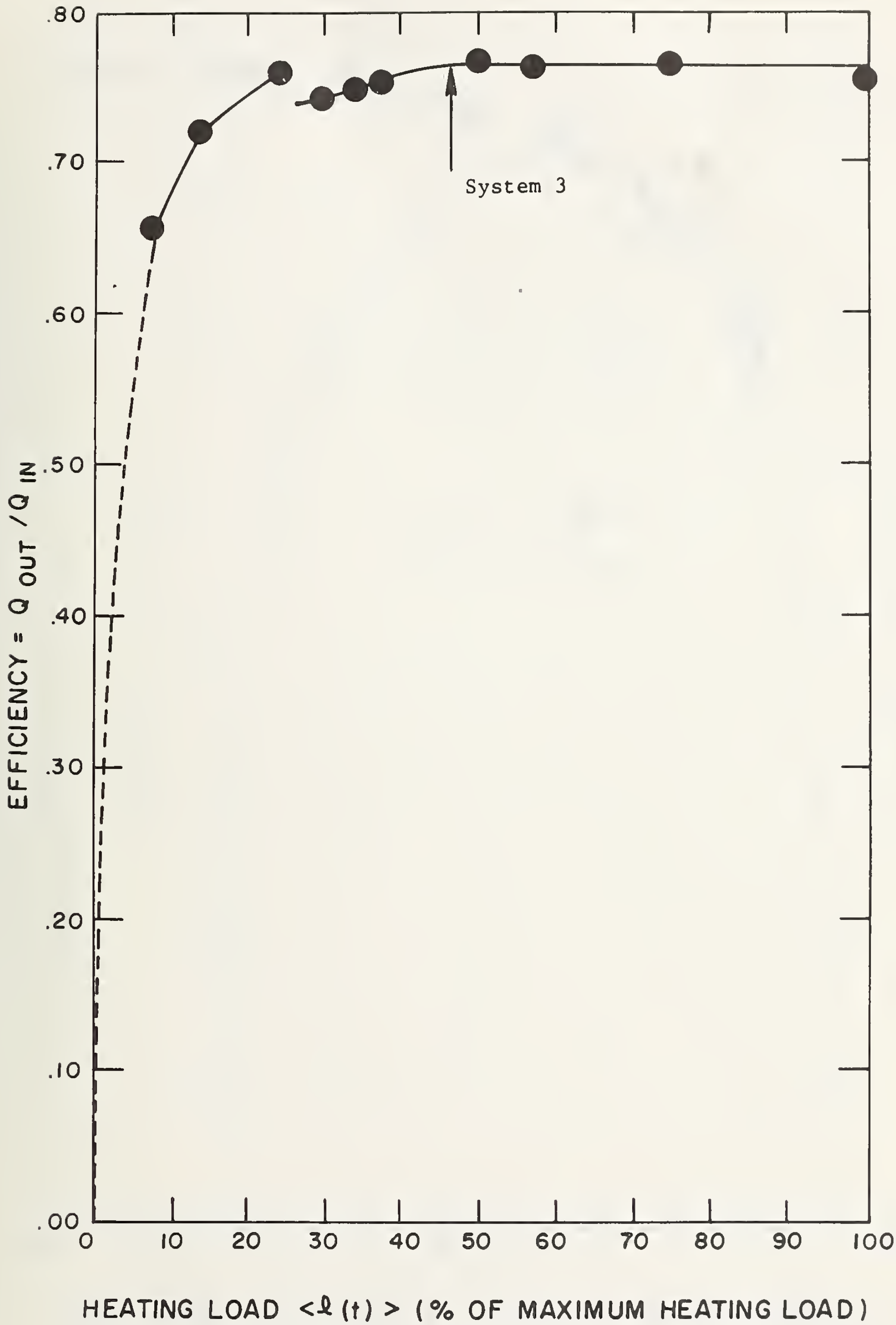

Figure 8. Efficiency versus heating load for system 3.

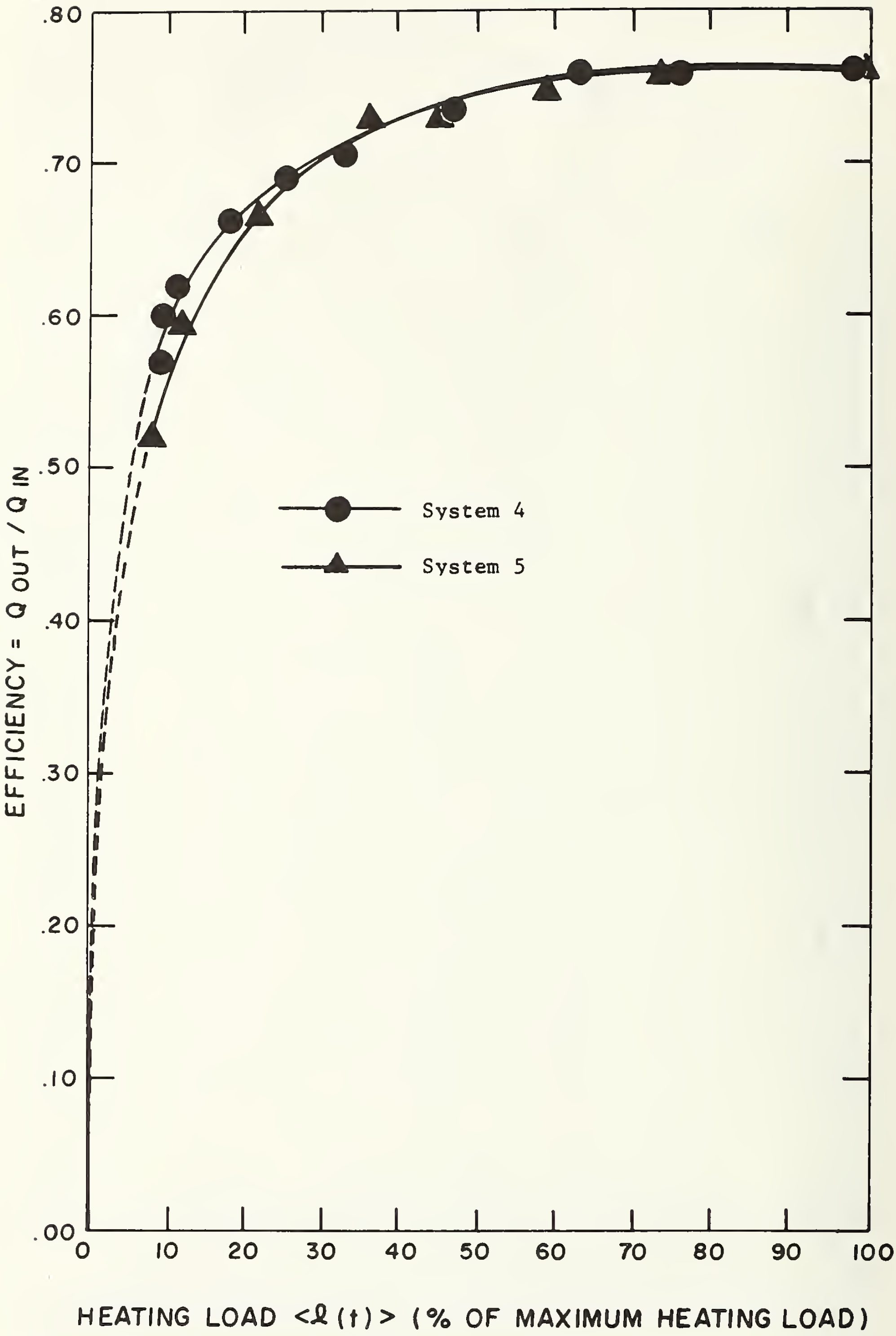

Figure 9. Efficiency versus heating load curves for systems 4 and 5.

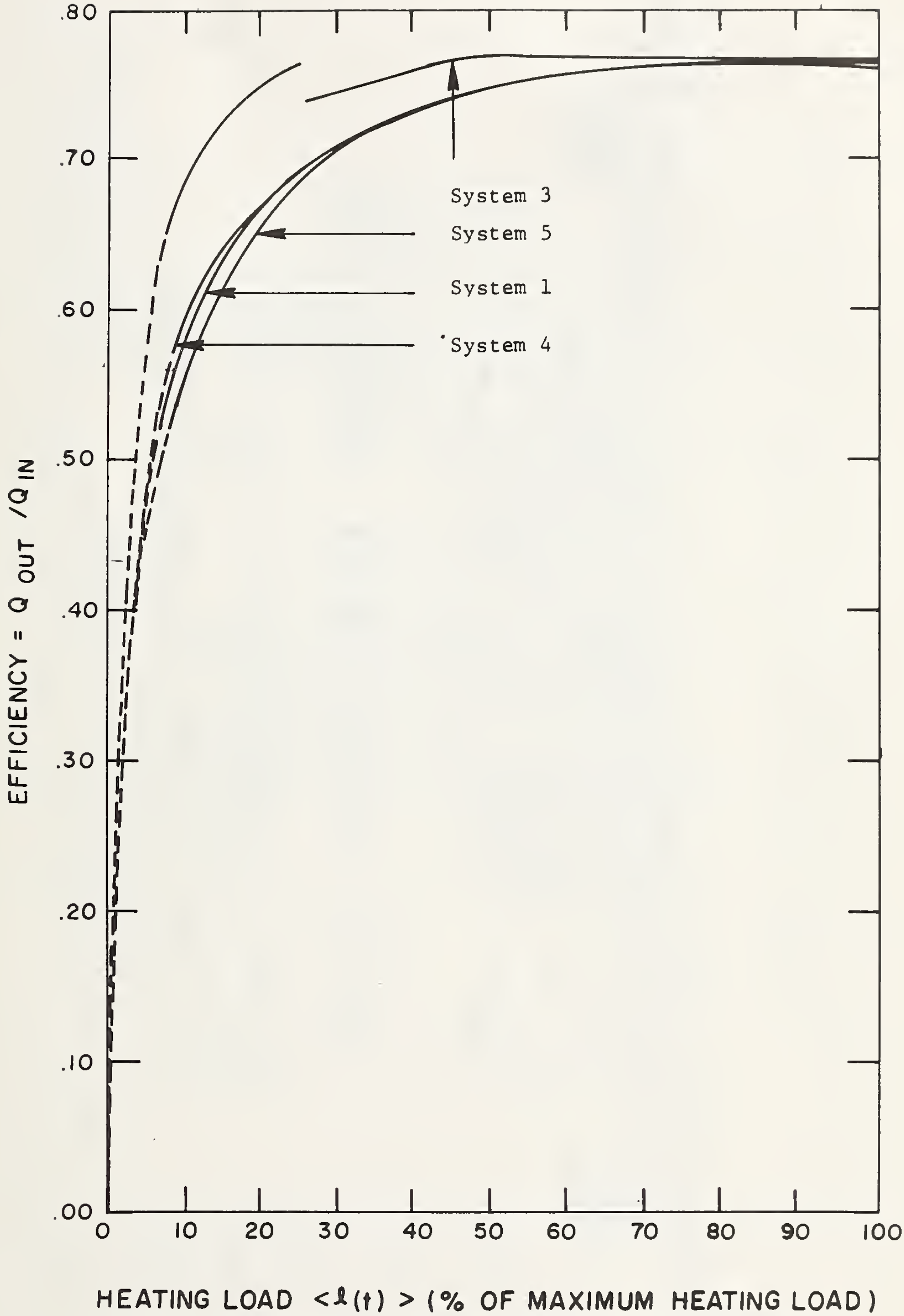

Figure 10. A comparison of efficiency versus heating load curves for systems 1, 3, 4, and 5.

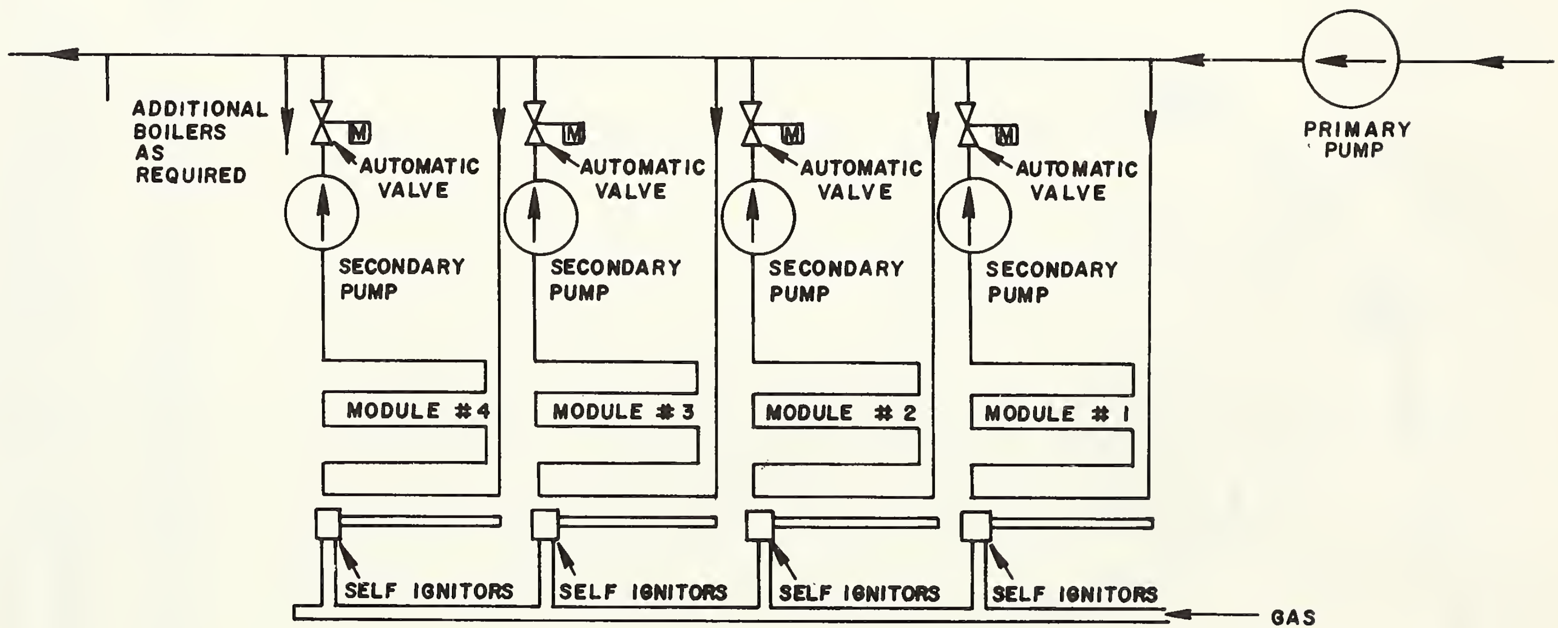

Figure 11. A heating system employing "primary-secondary pumping."

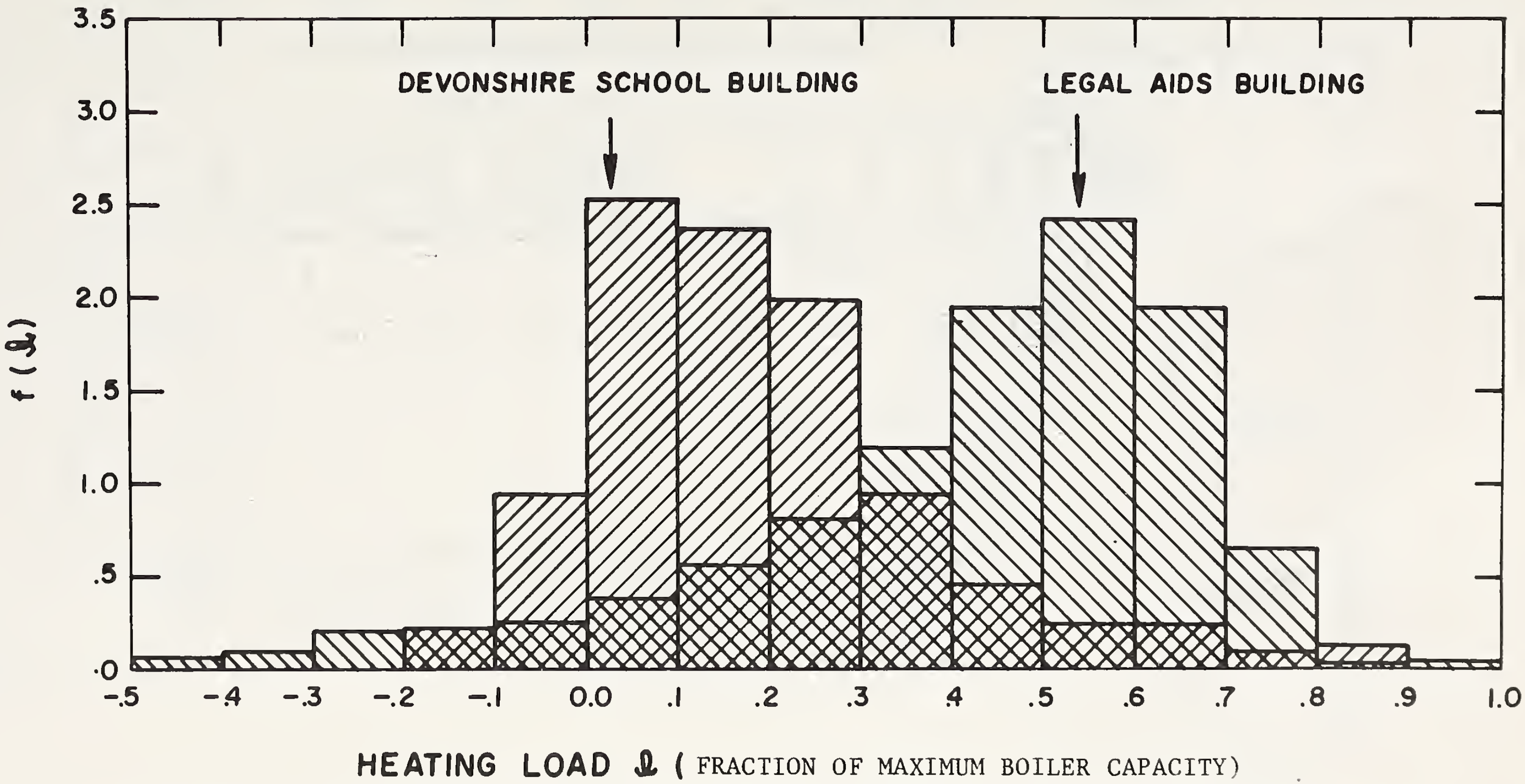

Figure 12. A plot of the heating load distribution function f(e) for the Devonshire School Building and Legal Aids Building in Columbus Ohio (6).

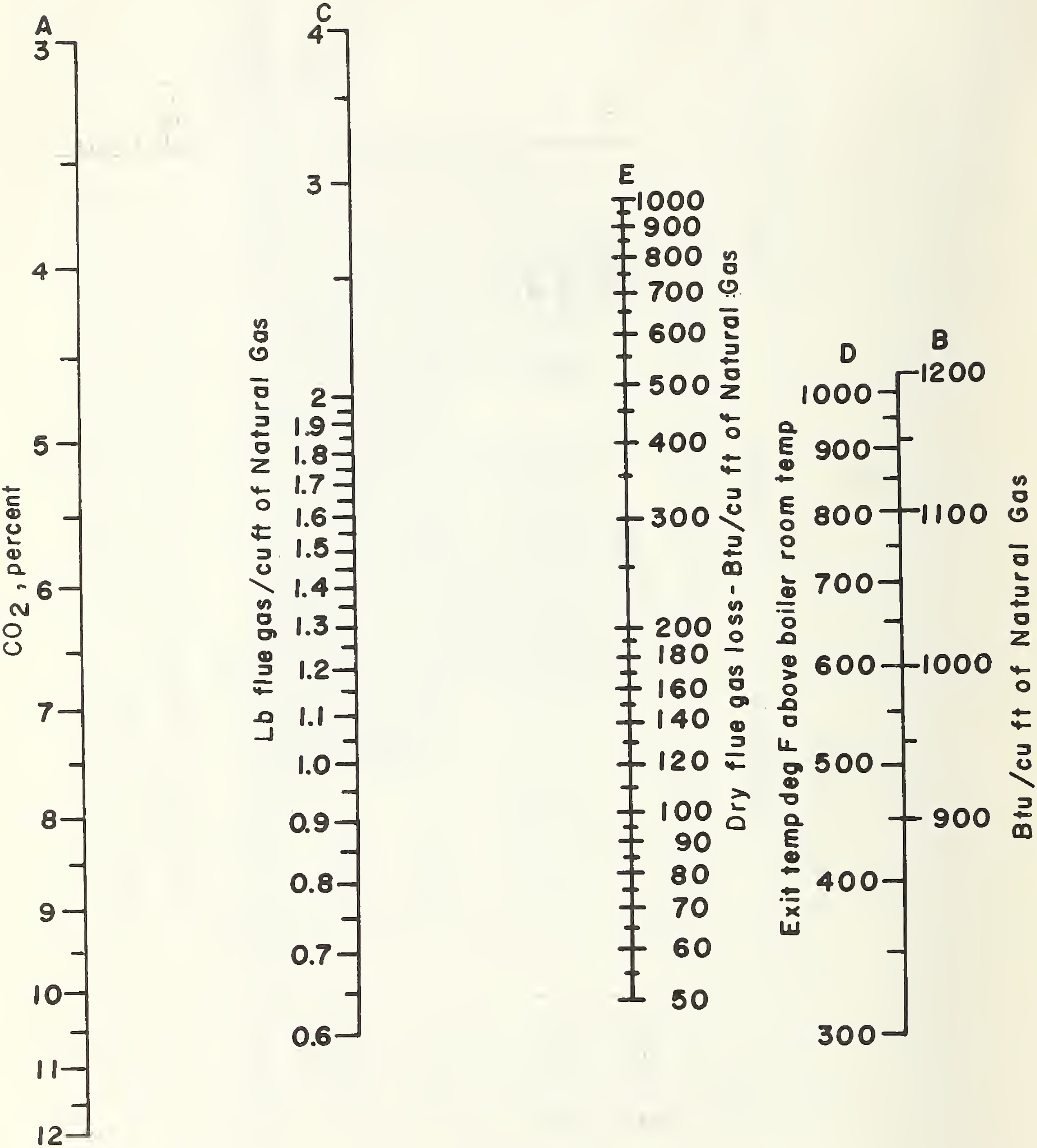

Figure 13. Nomograph giving dry flue gas loss /3/.

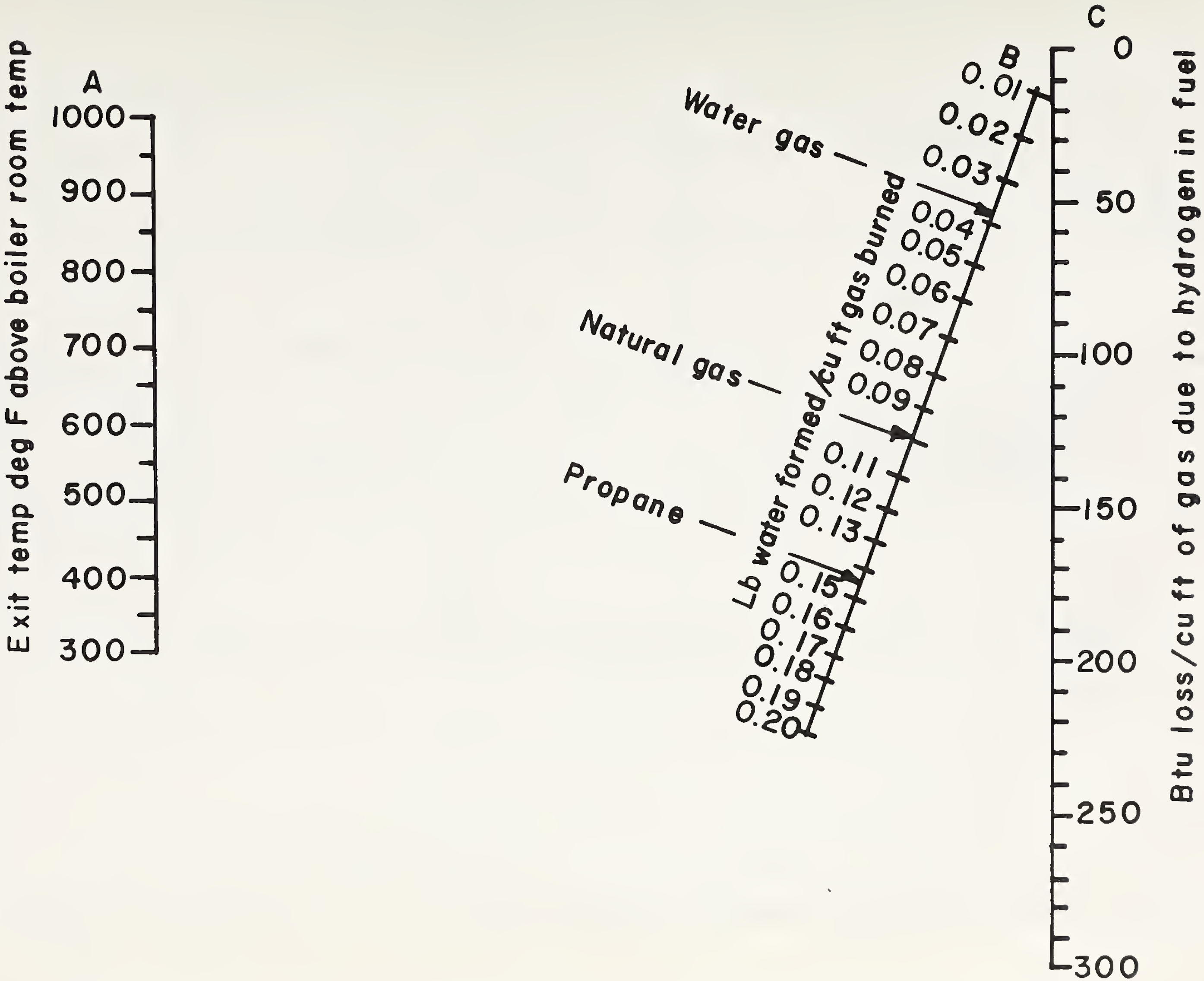

Figure 14. Nomograph giving loss due to hydrogen in the natural gas /3/.

Table 1. Ratios of Actual to Indicated Volumes for Gas Meter

Gas Flow Rate (ft^3/hr) (Specific Gravity = 0.62)	$R = \frac{\text{Actual Volume}}{\text{Indicated Volume}}$
75	1.006
230	1.006
330	1.009

Table 2. Heating Load Distributions Obtained for Two Buildings in Columbus, Ohio [6]

Heating Load*	% of Testing Period at Each Load**	
	Legal Aids Building Testing Period: 12/9/65 to 5/29/69	Devonshire School Building Testing Period: 10/19/66 to 5/31/62
-0.5 to -0.4	0.4	0.0
-0.4 to -0.3	0.8	0.0
-0.3 to -0.2	1.9	0.0
-0.2 to -0.1	1.9	2.2
-0.1 to 0.0	2.5	9.4
0.0 to 0.1	3.5	25.3
0.1 to 0.2	5.2	23.7
0.2 to 0.3	7.6	19.8
0.3 to 0.4	10.8	9.2
0.4 to 0.5	17.9	4.4
0.5 to 0.6	22.1	2.4
0.6 to 0.7	17.9	2.4
0.7 to 0.8	5.9	1.0
0.8 to 0.9	1.2	0.2
0.9 to 1.0	0.4	0.0

* A negative heating load corresponds to a cooling load.

** The data presented in this table has been extracted from the curves in figures (2) and (3) of reference [6].

Table 3. Calculated seasonal Efficiencies Obtained by Using the Efficiency vs. Load Curves for Configurations #3 and #4 and Load Distributions in Table 2.

Cases Examined	Calculated Seasonal Efficiencies (%) — Modular System with Secondary Pumping (Uses data from configuration #3)	Calculated Seasonal Efficiencies (%) — Single Large Boiler (Uses data from configuration #5)
1. Heating systems installed in Legal Aids building and having the same capacity as existing boiler in building.	75.4%	71.8%
2. Heating systems installed in Legal Aids building but oversized by 50%.	73.9%	68.8%
3. Heating systems installed in Legal Aids building but oversized by 100%.	72.5%	65.6%
4. Heating systems installed in Legal Aids building but oversized by 200%.	65.5%	55.9%
5. Heating systems installed in Devonshire School building and having the same capacity as existing boiler in building.	70.2%	61.2%

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

The modular concept of boiler operation was examined in a laboratory test of five gas-fired, cast iron, hydronic boilers. Four of the boilers, each having an input rating of 85,000 Btu per hour, were arranged so that they could either be operated like a single boiler (i.e., all of the boilers either on or off) or as a modular installation in which the boilers are sequentially fired to match the number in operation with the heating load. The fifth boiler had an input rating of 300,000 Btu per hour and was operated as a single boiler installation. Efficiency vs. heating load curves were obtained for the single boiler installation, the four small boilers run like a single boiler and the modular installation operated with and without water flowing through the "idle" modules. These efficiency curves were then used to theoretically predict the effect of the modular concept and boiler oversizing on the seasonal efficiency of gas-fired heating plants. It was found that under certain conditions the use of a gas-fired modular boiler installation instead of a single large boiler could result in considerable energy savings.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Boiler oversizing; efficiency vs. heating load; modular boilers; modular concept; seasonal efficiency.

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