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AN EXPERIMENTAL INVESTIGATION OF  
FUEL VAPORIZATION ON COMBUSTION EFFICIENCY

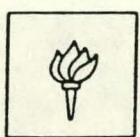
by

V: Zakkay  
A. Agnone  
H. Clisset

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NEW YORK UNIVERSITY  
FACULTY OF ARTS AND SCIENCE  
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## ABSTRACT

The development of a residential-size vaporizing oil burner is presented along with its operational and performance characteristics. The vaporization scheme consists of spraying No. 2 oil onto a regeneratively heated surface at a temperature above the oil vaporization temperature ( $650^{\circ}\text{F}$ ). The vaporized oil mixes with a preheated air ( $T = 250^{\circ}\text{F}$ ) stream to produce a uniform fuel air mixture. The premixed vaporized fuel/air stream produces short blue flamelets anchored to a steel screen flame holder. The operational and performance characteristics of this burner are presented for a range of the air and oil flow parameters around the stoichiometric condition, and for a nominal firing rate of  $\approx 1.2$  gph. Operation with less than 3% excess air is demonstrated with very little soot formation. The combustion quality of the vaporizing oil burner is substantially improved compared to conventional spray combustion and recirculation type blue flame burners.

The vaporizing oil burner was adapted to a conventional boiler and the thermal efficiency was determined by a calorimeter technique and compared to the stack method. The thermal efficiency with the vaporized combustion mode is about 4% greater than conventional spray combustion burners. The increase is realized through the reduced excess air requirements. The increased efficiency can result in reduced oil consumption from 12% to 20% depending on the location and usage of the burner unit. Items for further research are enumerated.

## I. INTRODUCTION

In the combustion process of industrial and residential oil burners, spray combustion of a liquid fuel is utilized. This process results in a large variation of local fuel/air ratios from lean to rich values. The average combustion is presently carried out at a global equivalence ratio,  $\phi$ , of 70% of stoichiometric. This implies that about 40% excess air is needed for soot-free combustion. Therefore, the heat rejected with this excess air is lost to the environment, resulting in a lower efficiency for the process.

Since chemical reactions occur on a molecular level, optimum efficiency requires that the fuel and air be thoroughly mixed at the molecular level. In spray combustion, this is not the case, because local regions of high air/fuel concentration exist around each fuel droplet or fog of liquid fuel droplets even if uniformly distributed in the air. Hence, the combustion occurring at a global  $\phi = 0.7$  will actually have widely varying local fuel to air ratios within the combustion zone. Moreover, locally the flame is always at an equivalence ratio of unity. These conditions result in excessively high local temperatures with corresponding inefficient combustion and large levels of pollutants, such as  $\text{NO}_x$  and CO production.

Vaporization of the fuel and mixing it with the air prior to combustion to achieve a uniform mixture offers significant combustion improvements and energy savings compared to current combustion practices. With vaporized fuel, the mixture can be made uniform throughout the combustion region on a molecular scale, and result in significantly lower losses since combustion processes with prevaporized fuel can be conducted at close to the theoretical stoichiometric condition to attain maximum efficiency by eliminating excess air.

A review of thermal-vaporizing burner developments up to the early 1970's is presented in Ref. 1. Notably, the work performed by the American Petroleum Institute on a prototype vaporizing burner (rated at 0.5 gph), is described in detail in Ref. 2. In this unit, a vaporizing chamber separate from the combustion chamber, and heated to about 800<sup>0</sup>F, is used to vaporize the oil prior to combustion. Air is used to purge deposits from the vaporizing chamber. However, vaporization deposits, sometimes as much as 750 grams of deposit per 1000 gallons of oil burnt, were found in this unit. While conditions for reducing deposits were generally established, no satisfactory equipment for a residential size vaporizing burner for #2 oil has as yet been devised. A vaporizing burner for #2 fuel oil remains of considerable interest because of its potential for energy saving. Recent progress on commercial size (firing rates greater than 10 gph) units was summarized in Ref. 3. The adaptation of the vaporization schemes used in commercial size units to residential sized ones was found to be impractical.

The present research on vaporized and premixed oil combustion involves the development of a burner head that could possibly be retrofitted to residential oil burners to improve their efficiency and emissions. The developmental vaporizing burner unit is described below, along with its operational and performance characteristics for a range of fuel and air flow parameters. Clean soot-free operation is demonstrated with less than 3% excess air. This vaporizing oil burner was evaluated by adaptation to a conventional boiler and the thermal efficiency was measured by a calorimeter technique. Significant improvement with the vaporizing burner is found. Further research to determine long-term, transient operation and adaptation to other fuels is suggested.

## II. VAPORIZING BURNER HEAD DESIGN AND DEVELOPMENT

### A. Conceptual Design

In the first phase of the current research, four vaporization schemes were evaluated. This research is summarized in Ref. 3. Injection in heated air ( $T_{\text{air}} \sim 500^{\circ}\text{F}$ ), and spraying fuel onto a heated surface ( $T_{\text{wall}} \sim 700^{\circ}\text{F}$ ), were found to give good vaporization, combustion, and operational characteristics. Using these experimental results as a guide, the research in the second phase focussed on adapting either one, or a combination, of these two vaporization schemes to an actual burner. The ultimate objective is to develop a prototype vaporizing oil burner for possible retrofit to residential oil burners.

Conceptually, the burner head had to be designed so that it could be fired in a conventional spray combustion when initiating from a cold start. Once the vaporization surface temperature reached about  $700^{\circ}\text{F}$ , the preheat oil flow would be shut off, and spraying onto the vaporization surface initiated. The vaporized fuel would then mix with the theoretically correct amount of air in a cylindrical duct. The mixture would auto-ignite at the flame holder, and a blue flame would be produced. The combustion products would then regeneratively heat the vaporization surface and sustain the vaporization and combustion process.

Although the nominal operating flow (fuel and air) conditions were well established from the first phase, the configuration and a number of parameters were uncertain and could not be ascertained with sufficient accuracy from theoretical considerations. In particular, the following determinations had to be made in a developmental phase:



- (1) The best preheat and vaporization nozzle arrangement in the burner head so that:
  - (a) the preheat time of the vaporization surface would be less than 30 seconds,
  - (b) the mixing length for the air and oil vapor would be sufficient to provide a uniform mixture,
  - (c) the minimum amount of air preheat necessary to avoid recondensation of the oil vapors would be provided,
  - (d) there would be compatibility of the preheat flame air flow requirements with those of the vaporization condition.

To quantitatively address these problems, it was necessary to vary the preheat and vaporization nozzle(s) size, type, location, and fuel injection pressure systematically around nominal values. Also, to insure good mixing, the mixer length and amount of air swirl also had to be varied.

- (2) For proper vaporized and premixed oil combustion, the values of the parameters which control auto-ignition, flashback, and flame blowoff, also had to be determined. The transition from the preheat to the vaporized combustion mode had to be dependable and repeatable, i.e., not subject to spurious, unstable behavior.
- (3) Lastly, the geometric configurations and flow parameters had to be defined so that a satisfactory equilibrium steady-state operating mode could be attained.

## B. Developmental Vaporizing Burner Head Design

To obtain quantitative answers to the above questions, an experimental configuration was designed so that most of the aforementioned parameters could be easily varied around a nominal configuration. A photograph of this burner design is shown in Fig. 1.

In this design, the preheat spray combustion nozzle (0.75 gph) and vaporization nozzles (1.20 gph) were selected so that  $\phi \approx 0.7$  for the preheat and  $\phi \approx 1.0$  in the vaporization mode for a constant air flow rate. The 1.2 gph firing rate with vaporized oil was selected to match that of the conventional burner supplied with the boiler so that we would have direct comparison of results between the spray combustion burner and the vaporized oil burner with the same boiler.

The vaporizing surface and spray nozzle configuration were designed to meet the following requirements:

1. evaporation temperature of 700<sup>0</sup>F with oil spray,
2. vaporization rate  $\approx 40$  gph per ft<sup>2</sup> of vaporization surface,
3. heat flux rates  $\geq 100,000$  BTU/hr/ft<sup>2</sup> to minimize deposit formation and droplet life time,
4. good atomization and spray performance of the nozzle.

The spray angle and distance from the vaporization surface were selected to satisfy these criteria.

Experiments were conducted with ambient temperature and preheated air at a temperature of about 250<sup>0</sup>F. This is below the lower distillation limit of 365<sup>0</sup>F for No. 2 fuel oil, and produces a mixture temperature greater than the dew point of the fuel vapors. The combustion quality, i.e. sooting, appeared greatly improved with preheated air.

The exploratory experiments, coupled with an analysis of the heat flux requirements of the vaporization surface, showed that a number of improvements of the experimental design could be made. These consisted of:

1. Increasing the heat flux to the external surface by circulating the combustion products over this surface, and by contracting the flow area, (Fig. 2), in the vicinity of the vaporization surface to increase the local flow velocity and hence Reynolds and Nusselt numbers. A 9.5" diameter by a 15" long chamber was used. This provided sufficient combustion volume.
2. Retracting the preheat flame into the mixing and vaporizing chamber so that radiation from the preheat flame could be advantageously utilized to reduce the heat-up time, and thus also provide a step-change reduction in heat flux as required for vaporization alone.
3. Reducing the vaporization surface thickness from 0.067" to 0.028" to decrease the thermal inertia of the vaporizing surface.
4. Distributing the fuel more uniformly over the vaporization surface to promote quicker mixing.
5. Installing a helical swirler to increase the turbulence and residence time of the vapors in the mixing-vaporizing chamber, in order to attain a more uniform fuel air mixture.
6. Installing a flow acceleration device and flame holder at the mouth of the firing tube, to avoid flashback and to improve the combustion efficiency.
7. Providing regenerative preheating of the air to above the mixture dew point ( $T_{dp} \approx 250^{\circ}\text{F}$ ) to avoid recondensation of the oil vapor.

The modified burner configuration is shown in Fig. 3. Figure 4 shows a photocopy of the improved oil burner head.

Experiments with this configuration were performed in the Axeman Anderson 87CPO Boiler.

### III. ADAPTATION TO THE BOILER AND THERMAL EFFICIENCY SETUP

#### A. Burner Head Adaptation

The vaporizing oil burner was adapted to an Axeman-Anderson Mark III conventional spray combustion burner. This is rated for a 1.25 gph firing rate. The boiler is a wet base, double pass type. No modification to the boiler unit itself was made during this investigation. The burner head design, used for evaluation in the present experimental program, is shown in Fig. 3. This consists of a 0.75 gph mechanical atomization nozzle (70° S.S.). The nozzle is located nine inches upstream of the flameholder screen. During the preheat phase, the nozzle is supplied with oil at 126 psig. Hence, the actual firing rate is nominally  $\sqrt{1.26} \times 0.75 = 0.84$  gph. When burning in the vaporized oil mode, the nozzle is supplied with oil at 300 psig. Hence, the oil flow rate corresponds to nominally 1.20 gph during this mode, which is the rated value of the boiler. The fuel pump was modified to be able to supply the two oil pressure levels. Also, three-way and oil dump solenoid valves were installed in the oil feed supply system to prevent oil dripping during the transition from the preheat to the vaporized combustion mode. The air flow rate is held at the same value for both the preheat and vaporized combustion mode. The swirlers provide a swirl number of about 1.5. These are needed mainly to improve the preheat flame characteristics.

The mixing chamber tube wall is 0.028" thick. This was selected to minimize thermal inertia and thereby provide short preheat times. Also, axial conduction losses are minimized with this tube thickness. The tube was instrumented with a dozen Chromel-Alumel thermocouples distributed axially and peripherally around the tube. These were attached to the tube

wall and were used to monitor the axial temperature gradients, to determine the heat fluxes during the preheat and vaporized combustion modes, and to determine when to transition from preheat to vaporization. The vaporizing oil burner head was installed in the conventional burner installation mounting as shown in Fig. 5. The combustion products are recirculated over the vaporization surface to provide the necessary heat fluxes for preheat and vaporization. The installation of the recirculating can increase the flow velocity over the burner surfaces.

The experiments showed some air preheat ( $T_{\text{air}} \sim 250^{\circ}\text{F}$ ) was necessary to avoid recondensation of the vaporized oil during the mixing process with ambient temperature. For these tests, the air preheat was performed electrically external to the boiler. Several regenerative air preheat schemes were considered. The use of a stack recuperator was considered to be inadequate because of insufficient heat flux at this point in the system to provide short preheat times, especially when initiating from a cold start since the boiler and resident water mass provide a high thermal inertia in the system. Circulation of the combustion air over the firing tube surfaces was also found to be inadequate due to flow area constraints and interference with the heating scheme of the vaporization surface.

The technique chosen for preheating the air was the installation of an air heat exchanger on the firing wall (back plate) of the burner chamber as depicted in Fig. 5. This was the best alternative since the back plate is simply insulated and is the largest source of jacket losses in this burner. To provide a steady state inlet air temperature rise of approximately  $200^{\circ}\text{F}$ , the surface area of the heat exchanger was estimated to be of the order of ten square feet.

This area is approximated by the hollow wall baffle can designed and built but as of this writing not yet evaluated.

The inlet air part of this hollow wall can be equipped with a butterfly-type valve to regulate the air flow in order to insure that the proper stoichiometric fuel-to-air ratio is obtained.

#### B. Instrumentation

Besides monitoring the oil and air flow rates, the oil injection pressure and temperature, the air inlet temperature, and the combustion products were samples inside the combustion chamber (point A on Fig. 5), just prior to (point B), and immediately at the outlet of the boiler (point C). The samples were analyzed for carbon dioxide, carbon monoxide, and oxygen content. The first two were measured with both an Infrared Industries' dual gas analyzer, and with a Dwyer absorption kit. The oxygen content of the flue gases was measured with a Bacharach Fyrite oxygen analyzer. Comparison of the oxygen content data for points A, B, and C indicates substantial air leakage into the flue gases along their path through the boiler. This necessitates correction of the % CO<sub>2</sub> and stack temperature data at points B and C. Smoke samples were also taken with the spot technique primarily after the boiler (point C). The combustion gas temperature was also monitored with a Pt-Pt-10% Rh thermocouple inserted near point A. This, however, did not show an accurate indication of the adiabatic flame temperature due to large radiative and convective losses. The temperature of the flue gases immediately after the boiler (point C) was also measured with a Chromel-Alumel thermocouple. This, in conjunction

with the corrected CO<sub>2</sub>, CO, and O<sub>2</sub> measurements were used to evaluate the thermal efficiency of the boiler by the stack technique.

### C. Thermal Efficiency (Calorimeter) Setup

The Axeman-Anderson Mark III oil burner was instrumented to be able to measure the thermal efficiency of the vaporized premixed combustion mode by a calorimeter technique.

The calorimeter setup is shown in Figs. 6a and 6b. Water from a tank 20 ft above the experimental setup was used to provide a steady water flow rate to the boiler. The flow rate was measured with a water flow rotameter. A gravimetric technique for measuring the water flow rate by weighing nominally 40 gallons of water collected over a 15 minute has been implemented for greater accuracy measurements in the future. The inlet and exit temperature of the water were measured with Iron-Constantan thermocouples. A  $\Delta-T^*$  temperature transducer was used to measure the water temperature rise. The oil mass flow rate was monitored with an oil flow rotameter. To insure accurate and repeatable results, the oil consumed over a 70 minute period was accurately weighed with a beam balance. The air flow rate was also verified through the oxygen content in the flue gases. The measurements were taken after an essentially steady state condition was attained. The steady-state thermal efficiency was determined from the following thermal efficiency equation using the same technique as in Ref. 4.

$$\eta_{Th} = \left( \dot{m}_w \Delta T_w / \dot{m}_{oil} \right) \left[ 60 C_{p_w} / SPG \Delta H_{fuel} \right] \times 100\%$$

\*  $\Delta-T$  Tradename for the Delta-T differential temperature transducer.



$\dot{m}_w$	-	Volume flow rate of H <sub>2</sub> O, gallons/min.
$\dot{m}_{oil}$	-	Volume flow rate of oil, gallons/hour
$\Delta T_w$	-	Steady rise in temperature of water, °F
$C_{pw}$	-	Specific heat of water, 1 BTU/lbm °F
$\Delta H_f$	-	Gross heating value of #2 fuel = 19560 BTU/lbm
SPG	-	Specific gravity of #2 oil = 0.855

The accuracy of the thermal efficiency determination depends upon the accuracy of the individual measurements of the water and oil flow rate, the water temperature rise, the specific gravity, and the higher heating value of the oil. The oil flow rate determined from the gravimetric technique was more accurate than that determined by either the rotameter or the nozzle, hence this was used in the efficiency calculation. The oil flow rate was estimated to be measured accurately to within  $\pm 0.125\%$ . The accuracy of the water temperature rise was determined from the  $\Delta$ -T transducer to be about  $\pm 0.25\%$ . The error in the water flow rate was estimated to be less than 1.0%. The accuracy of the higher heating value of the oil was estimated from a specific gravity determination of the oil and the data given in Ref. 5. The overall accuracy of the thermal efficiency determination is estimated to be within 2% of the actual value.

The boiler and combustion chamber were thoroughly scrubbed of any soot deposited on the walls prior to the efficiency measurements determination. Also, the fuel rich tests were performed last since at these conditions sooting was prevalent as indicated by the smoke numbers presented below.

#### D. Vaporizing Oil Burner Performance and Evaluation

The vaporizing oil burner steady-state performance was evaluated through the measurements indicated above and compared to the conventional spray combustion mode. The evaluation was made for two nominal firing rates of 0.84 and 1.20 gph, and for a range of fuel/air equivalence ratios, two combustion air temperatures, and atomization pressures of 125 and 300 psig. A number of other parameters such as amount of swirl and swirl type, nozzle type, vaporizing chamber penetration into the burner, mixer length, etc. were also experimentally investigated. The data gathered with these configurations were sparse and served only as a guide to establish an operating configuration; thus, these data will not be presented here. A complete set of data is presented here for the operating configuration shown in Fig. 3.

A qualitative picture of the quality of the combustion is indicated from the flame photographs of both the preheat spray combustion flame and the vaporized oil combustion flame shown in Figs. 7a to 7d, for both ambient temperature air and preheated air. The improved combustion characteristics of the vaporized oil combustion and the improvements of the air preheat are clearly evident from these pictures. The effects of the air swirl on the vaporized oil combustion flame are also evident.

To quantitatively evaluate the combustion, the flue gases were analyzed for carbon dioxide, oxygen, and carbon monoxide content for ambient temperature ( $T \approx 50^{\circ}\text{F}$ ) and preheated air ( $T = 250^{\circ}\text{F}$ ) for both the preheat spray combustion and vaporized oil combustion. The  $\text{CO}_2$  content of the flue gases is shown in Figs. 8a and 8b for the ambient temperature air and the preheated air case respectively. The theoretical  $\text{CO}_2$  curve is shown for comparison. The oil vaporization effects on the combustion are seen to be a significant

improvement even with ambient temperature air. A comparison of the effects of the air preheat on the vaporized oil combustion mode is shown in Fig. 8c. This shows the carbon dioxide content in the combustion product does not reach the stoichiometric level when ambient temperature air is used.

To determine how close the vaporized oil combustion process actually approached the stoichiometric value, the oxygen content of the flue gases was also measured. The data taken with preheated air is shown in Fig. 9 for the V.O.C. as a function of the equivalence ratio  $\phi$ . The theoretical curve is shown for comparison. The data shows the actual combustion approached the ideal process to within an excess oxygen content of the order of 1%.

A further indication of the improved combustion characteristics of the V.O.C. is evident from the smoke numbers shown in Figs. 10a and 10b for the ambient temperature and preheated air respectively. With preheated air and V.O.C., the smoke number is nearly zero up to an equivalence ratio of about 1.0. The smoke number of the preheat flame is about 1.0 and increases rapidly beyond an equivalence ratio of about 0.7. The smoke numbers with ambient temperature air, Fig 10a, of the preheat flame are nominally 3 for equivalence ratios less than 0.7. While with the V.O.C. the smoke number is about 2 for equivalence ratios less than unity. Hence, this also shows the need for preheated air.

The carbon monoxide content of the combustion products with V.O.C. and preheated air are shown in Fig. 11 for a range of equivalence ratios. The theoretical equilibrium curves of the CO content in the flue gases are shown in the figure for comparison for several assumed stack temperatures. The data shows the CO is nominally less than 50 ppm for equivalence ratios

less than unity, and rapidly increases to about 100 to 200 ppm when the stoichiometric ratio is increased to 1.1. The data appears to follow a 440°F stack temperature curve.

Preliminary thermal efficiency measurements were made for the vaporization mode of operation. These measurements were limited to the preheated air case for the calorimetric method whereas for the stack method some ambient air data is included as well. The results are presented in Figs. 12 and 13. In these figures, an attempt has been made to correct the test data for two factors which became manifest during the tests: 1) evidence of air leakage in the boiler for which stack temperature and %CO<sub>2</sub> at the stack requires correction; 2) not quite steady state conditions for the calorimeter technique requiring a heat source/sink correction in thermal efficiency calculations. Although the results indicate the need for further refinement of the thermal efficiency set-up some preliminary qualitative conclusions about the pre-vaporized pre-mixed combustion technique can be drawn: 1) ambient air combustion gives thermal efficiencies equal to preheated air combustion so that air pre-heating has its most conspicuous benefit in reducing smoke numbers; 2) pre-vaporized-premixed combustion appears to produce an increase in thermal efficiency over conventional spray combustion of the order of 4%, the theoretical improvement value. The thermal efficiency measured at Brookhaven National Laboratories with the Axeman-Anderson boiler, using the conventional spray combustion mode, is also shown for comparison. In

Ref. 3, the present authors reported the potential fuel savings obtainable with improved thermal efficiency. The curve is reproduced here in Fig. 14. For a 5% improvement in thermal efficiency, this curve shows fuel savings close to twenty percent.

Typical temperature histories of the vaporization surface are shown in Figs. 15a and 15b for the ambient temperature and preheated air combustion. These curves show the rapid initial heating due to the preheat flame. Nominal net heat flux rates of  $6 \text{ BTU/sec/ft}^2$  have been determined from these using the thin-skin technique, i.e.  $\dot{q} = \rho C_p \tau \frac{dT_w}{dt}$ . This compares well with a heat flux rate of about  $3.5 \text{ BTU/sec/ft}^2$  needed to vaporize one gallon per hour of oil on a surface of  $A = \pi D \ell = \pi(4) 2/144 = 0.17 \text{ ft}^2$ . With preheated air, the initial wall temperature is higher and the heat fluxes are nominally the same as those with ambient temperature air. The time needed to heat up the vaporization surface to about  $700^\circ \text{F}$  is nominally 15 to 20 seconds.

At switchover from the preheat to the vaporized and premixed combustion mode, the surface temperature continues to increase for about six seconds, then a cooling period of about a half-a-minute is followed by a gradual heating of the surface. The initial cooling following transition is due to both the oil vaporization from the heated surface and the air flow inside the mixing tube. The reheat is obviously due to the external heating by the combustion products. The vaporization surface attains nearly an equilibrium temperature well in excess of the desired vaporization surface temperature. The slow temperature rise in the vaporization surface after several minutes is due to heat conduction along the surface to the colder parts of the burner and boiler.

### E. Comments on Achieving Perfect Combustion in Practice

An examination of the oxygen, carbon monoxide, and smoke data, points out the difficulty of attaining and maintaining stoichiometric operation in practice. Besides errors in the determination of, and setting the air and fuel flow rates, diurnal and local variations in air humidity, barometric pressure, temperature, and air impurities, as well as the No. 2 fuel oil temperature and source supply, etc., influence the exact determination of stoichiometric condition. These spurious effects can very easily influence the instantaneous stoichiometry of the combustion by a few percentage points. Automatic monitoring and flow adjustments to compensate for these effects appears impractical and economically prohibitive even under controlled conditions.

Also, flow fluctuations caused by the air and fuel delivery equipment preclude stoichiometric operation at every instant. An elementary fluctuation analysis of the air and fuel supply system, where the instantaneous air and fuel mass flow rates are the sums of a mean (time averaged) flow rate and a random fluctuation whose root mean square values are known, shows the deviation from the mean value is proportional to the square root of the sum of the rms values of the air and fuel flow rate less the cross correlations of the two, that is

$$\frac{\sqrt{(f/a)'^2}}{(f/a)} = \sqrt{\left(\frac{f'}{\bar{f}}\right)^2 + \left(\frac{a'}{\bar{a}}\right)^2 - \left(\frac{f' a'}{\bar{f} \bar{a}}\right)}$$

This shows that a 10% rms value of either the fuel or air flow rate produces a 3% change in the stoichiometry about a mean value. If both the fuel and air flows fluctuate about a mean value, then the cross-correlation term

subtracts from the sum of the rms values of the two flow rates. If the fuel and air fluctuations are uncorrelated, this last term vanishes. If the fuel and flow fluctuations are in phase, the cross-correlation term decreases the deviation from the mean. It would be desirable to experimentally measure these supply fluctuations to assess the actual departure from the mean stoichiometry. The interactions of the supply fluctuations with the natural turbulence produced by the swirler, injection, mixing, and combustion phenomena under vaporization conditions is also highly desirable.

D

#### IV. SUMMARY AND CONCLUSIONS

An experimental residential oil burner which utilizes vaporization by spraying onto a preheated plate has been designed, built, and tested. Experiments have been conducted to: (1) determine the operation as a function of the flow parameters; (2) establish dependable and repeatable transition from a short preheat phase with a conventional spray combustion flame to the vaporized combustion mode; (3) the equilibrium vaporization surface temperature attained with the steady vaporized combustion mode; and (4) determine the performance of the vaporizing oil burner. The developmental experiments have shown:

1. To obtain a reasonable preheat time ( $< 30$  secs) it was necessary to: (a) flow the combustion products over the exterior of the vaporization surface; (b) establish the preheat flame inside the mixing and vaporization chamber, and (c) reduce the mixing chamber wall thickness from 0.067" to 0.028". A preheat time from ambient to the desired vaporization temperature ( $T_w \approx 700^{\circ}\text{F}$ ) of less than 20 seconds has been obtained.
2. Transition to the vaporization mode could be established spontaneously by momentarily diverting the oil flow from the injection nozzle back to the reservoir for sharp fuel cutoff during transition. The optimum transition time was found to be approximately 300 msec. of fuel flow interruption. This time was sufficient to clear the mixing chamber of the preheat flame. A longer transition time resulted in transition failure because of rapid cooling of the flame holder and vaporization surface which inhibited spontaneous ignition of the vaporization flame.



The experiments were performed with: (1) a number of air flows corresponding to stoichiometric levels between  $\phi = 0.7$  and 1.3; (2) several air preheat temperatures from ambient to 250°F; (3) two nominal levels of firing rates of 0.85 and 1.20 gph; (4) different fuel nozzle types; (5) nozzle locations relative to the flame holder screen at the end of the vaporizing mixing chamber; (6) air swirl; (7) injection oil pressures of 125 and 300 psig to obtain different firing rates with the same nozzle; and (8) different mixing turbulence baffle configurations.

The measurements included the vaporization surface, stack and oil temperature histories, carbon dioxide, carbon monoxide, oxygen content of the flue gases, smoke numbers, and thermal efficiency. Flame photographs were also taken. These measurements have shown that air preheat is necessary to prevent the vaporized fuel oil from recondensing and forming yellow tipped blue flames. Satisfactory preheat times and transition from a preheat flame to a vaporizing premixed flame could be obtained. However, the vaporization mixing chamber wall temperature showed axial variations of -150°F per inch equilibrium wall temperatures decreasing from about 1400°F close to the flame holder. The carbon dioxide and oxygen levels were found to agree with the theoretical trends. Smoke numbers less than one were obtained with the vaporized oil combustion mode even with less than 3% excess air.

For this work, the experimental burner was adapted to the Axeman-Anderson Mark III, 87CPO boiler. The adaptation involved designing a regenerative air heater exchanger integrated with the combustion chamber and the burner head. The burner operation consists of a preheat phase of about 20 seconds to increase the vaporization surface temperature from a cold start to the desired temperature. The air heat exchanger shell is

also heated during this period. The preheat flame is contained inside the vaporization and mixing chamber during this phase in which a stoichiometric ratio of about 0.7 is used. Transition to the vaporization and premixed combustion mode is initiated when the vaporization surface temperature is 800°F. The oil flow rate was increased by spraying at a higher pressure (300 psig) to adjust the stoichiometry to unity while retaining the same air flow rate as in the preheat phase. Auto-ignition of the vaporized fuel is automatically obtained at the hot flame holder screen. The combustion occurs in a nearly flat flame composed of many blue flamelets.

Experimental data with the boiler adapted vaporized premixed combustion have been presented. In particular, the flue gas carbon dioxide, oxygen, and carbon monoxide contents have been correlated with theoretical trends for a range of equivalence ratios for both the preheat and vaporized premixed oil combustion mode using both ambient temperature and preheated air. Generally, better agreement was found with the preheated air and the vaporized-premixed combustion than with ambient temperature air and the preheat spray combustion. The measured values of the smoke numbers under these conditions agree with these trends and indicate the relative merits of the vaporized-premixed oil combustion.

The thermal efficiency measurements also indicate the improved combustion when vaporizing and premixing.

Further research is being planned to:

1. Optimize the burner configuration.
2. Examine transient phenomena more closely and cyclic operation over 15 minute intervals.
3. Perform  $\text{NO}_x$  and UHC measurements at the same conditions reported herein.

4. Evaluate the air preheater hallow-wall baffle can.
5. Measure the thermal efficiency of the boiler optimized burner combination.
6. Redesign controls so that this unit can be self-contained free-standing prototype for delivery to Brookhaven National Laboratories for confirmation of the thermal efficiency measurements obtained at New York University using the standard technique and equipment used for other burner units. Also install a blue flame combustion verification control.
7. Examine the adaptability of vaporized and premixed combustion to other liquid fuels such as No. 6 oil and synthetic fuels.

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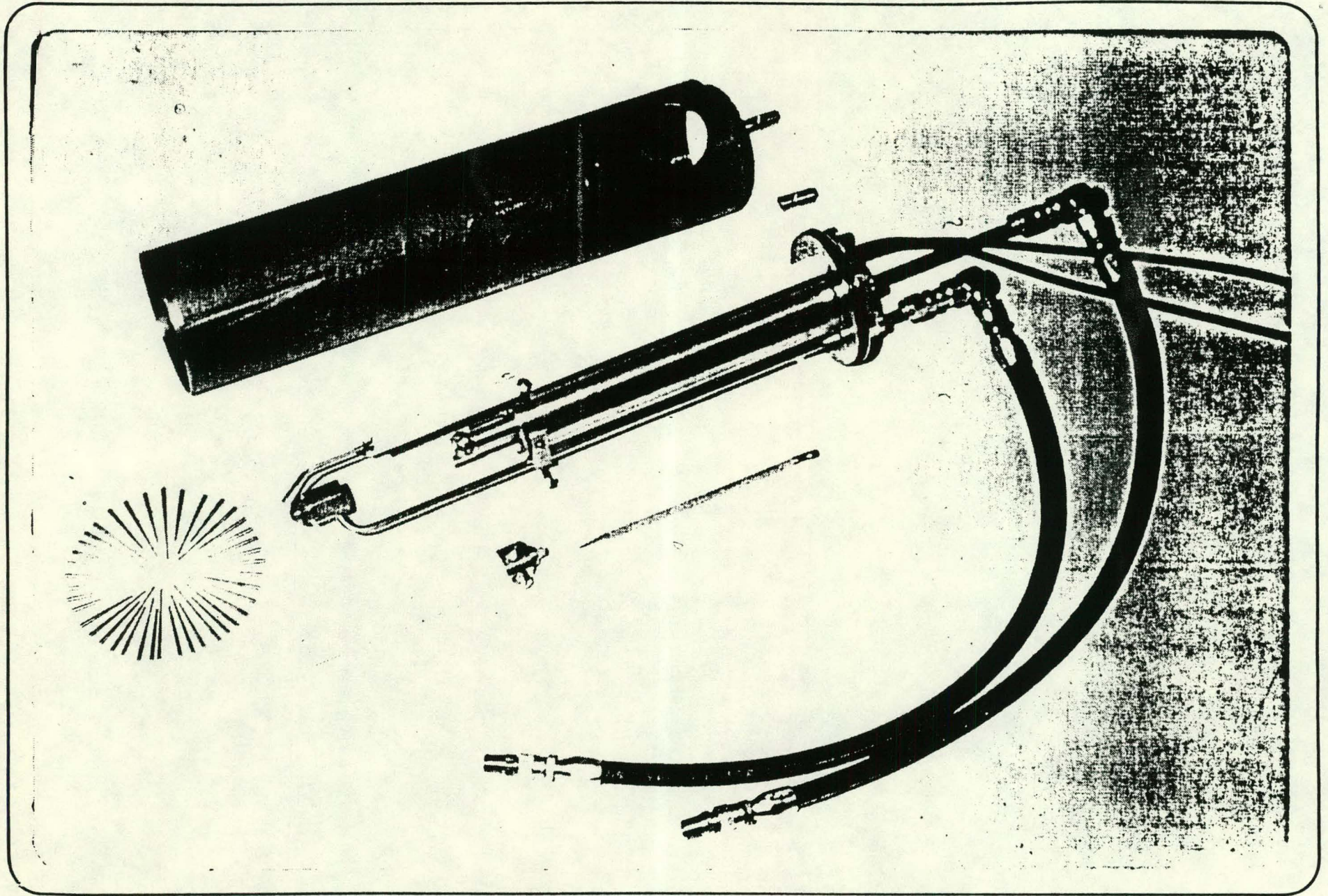


FIGURE 1 : VAPORIZING OIL BURNER HEAD ASSEMBLY - DEVELOPMENTAL CONFIGURATION

25

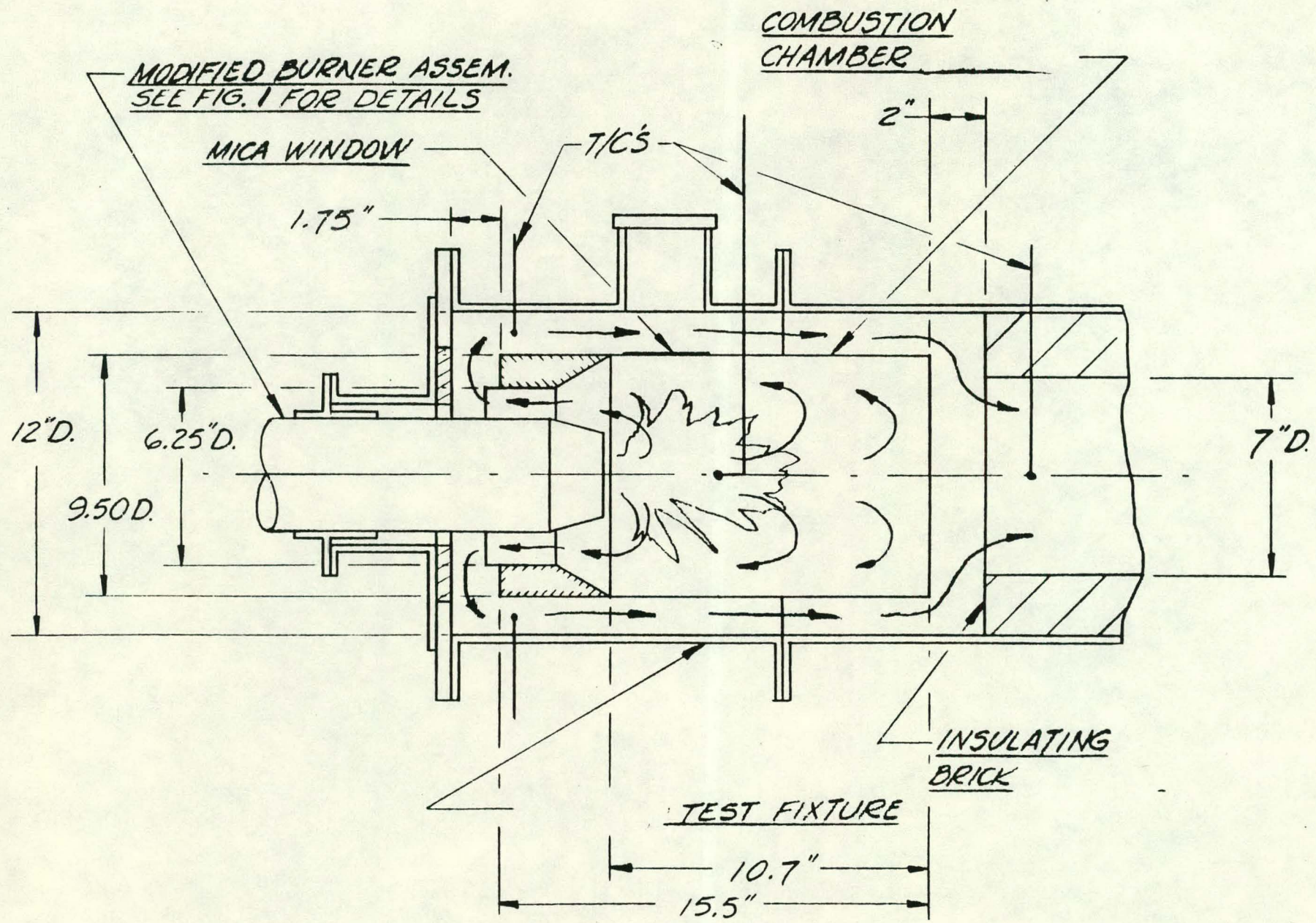


FIGURE 2 : MODIFIED TEST STAND

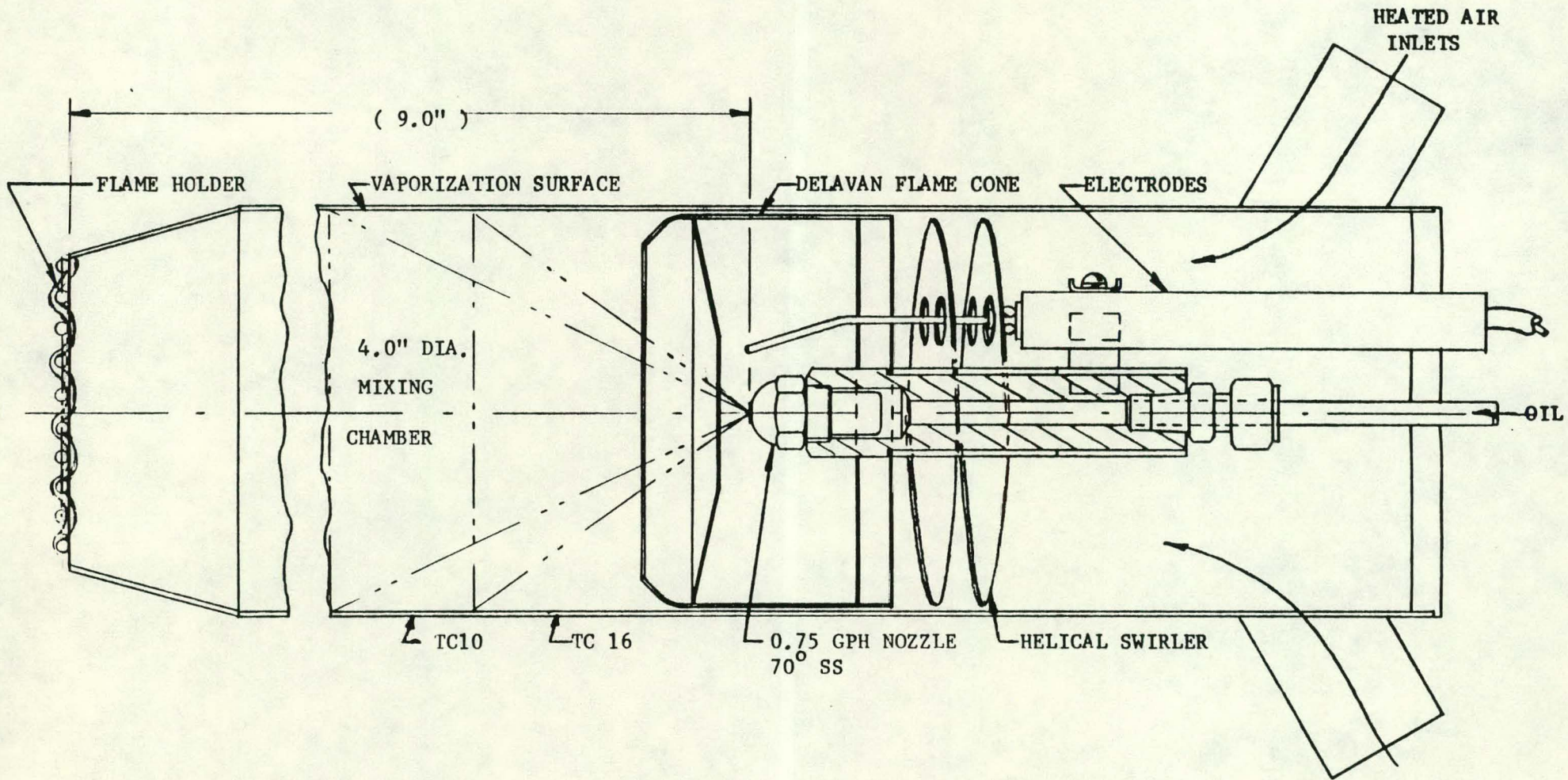


FIG. 3 VAPORIZED AND PREMIXED OIL BURNER HEAD

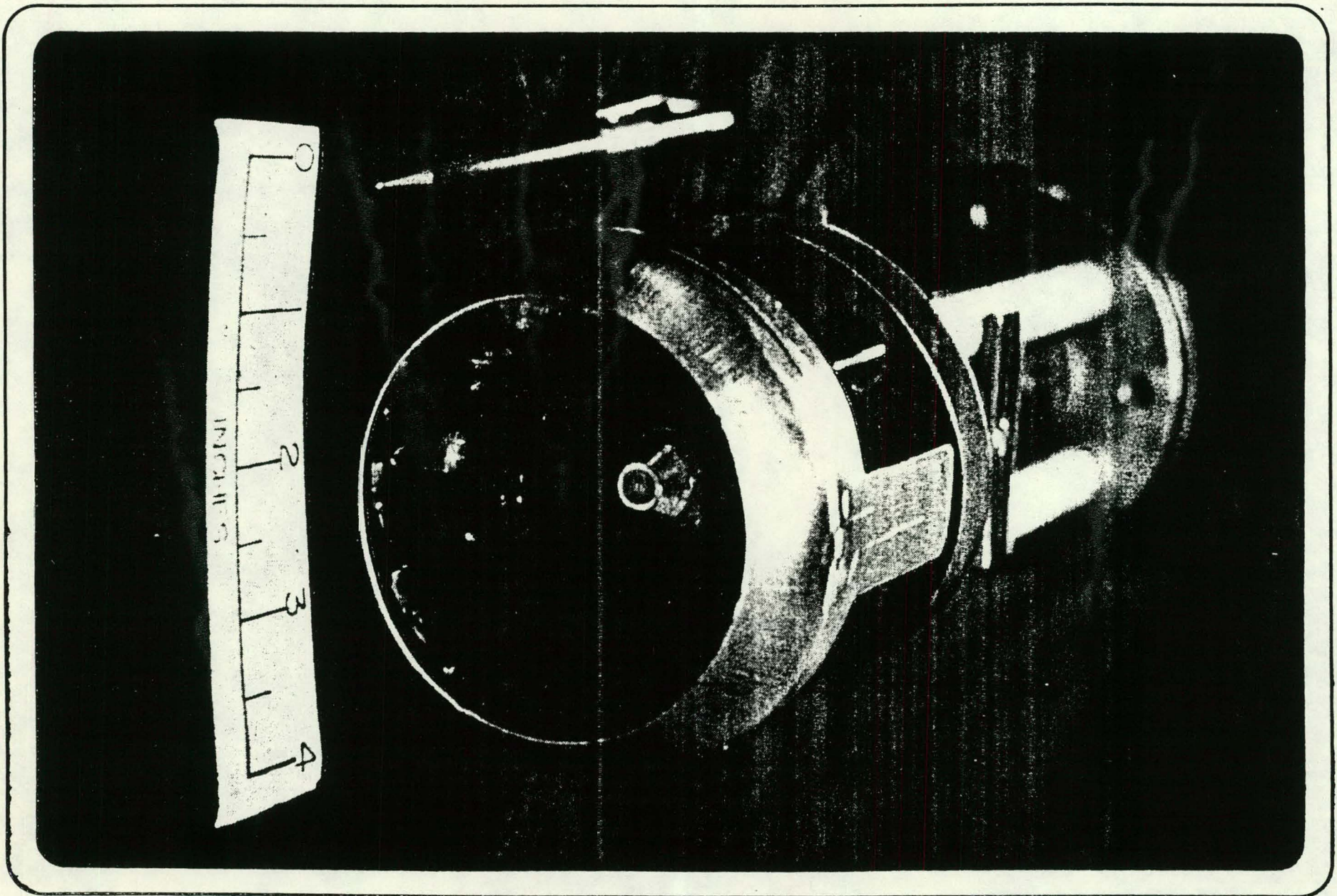


FIGURE 4: PHOTOGRAPH OF THE VAPORIZING BURNER HEAD



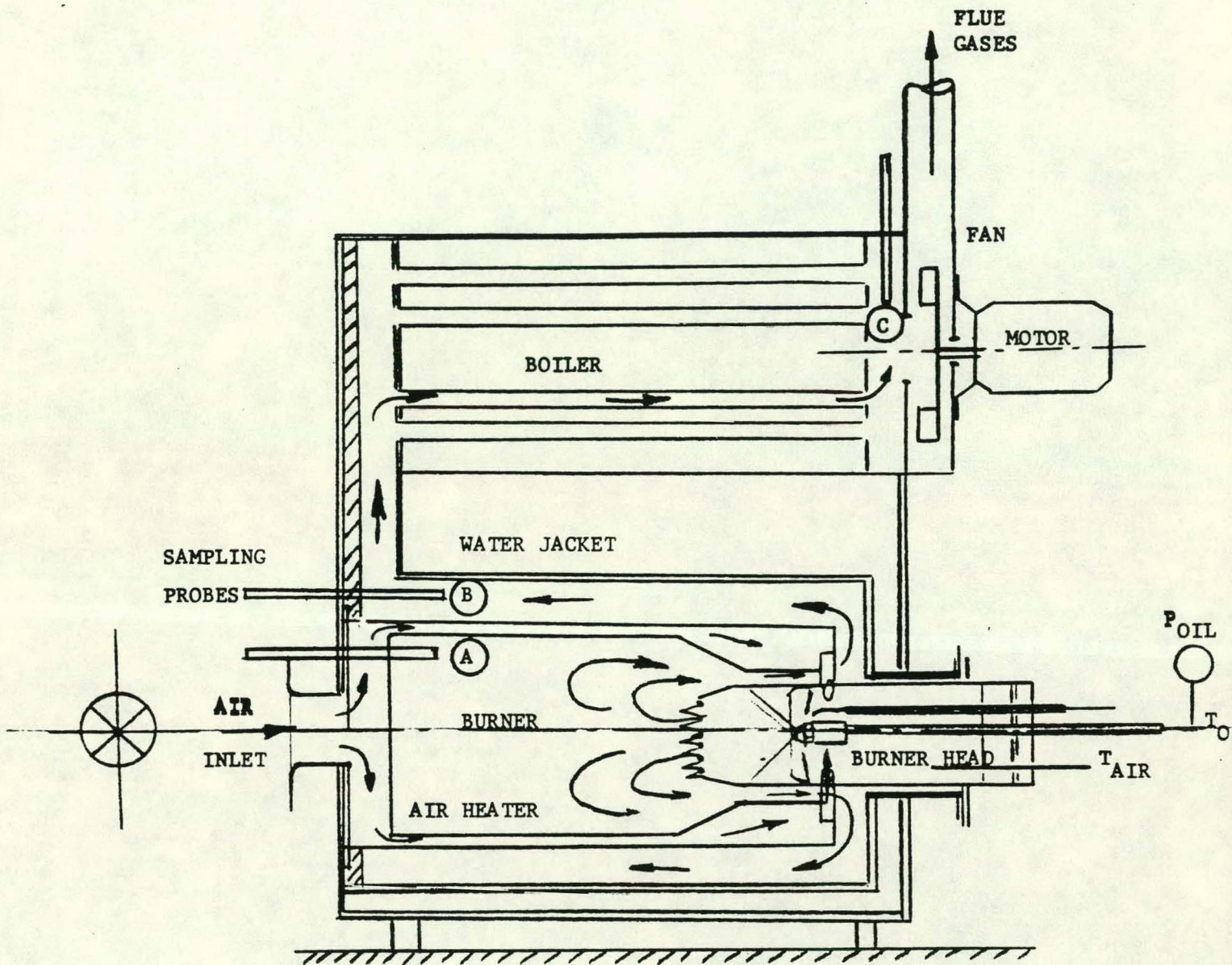
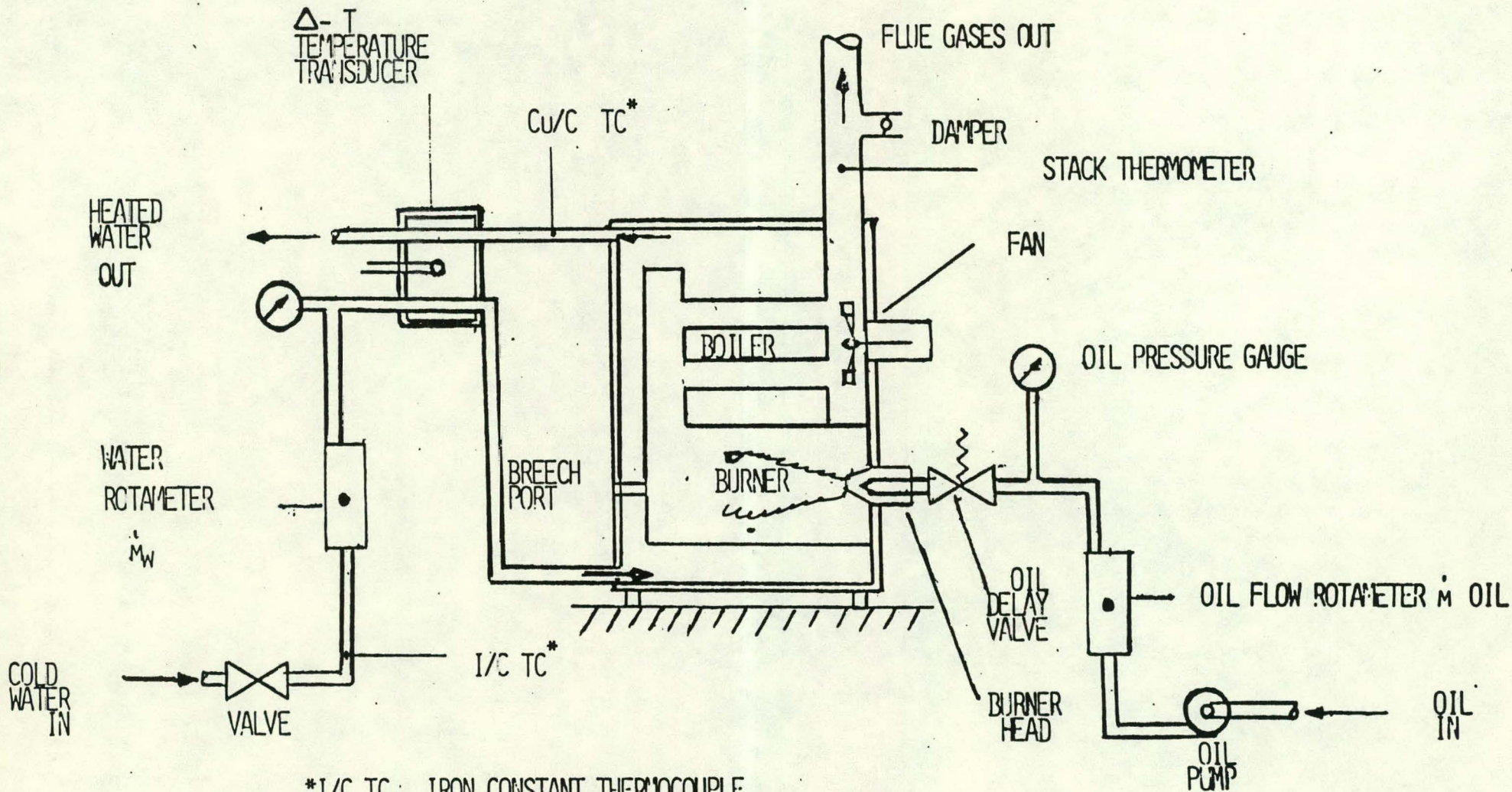


FIG. 5 BOILER ADAPTATION OF VAPORIZING OIL BURNER



\*I/C TC IRON CONSTANT THERMOCOUPLE  
 Cu/C TC COPPER CONSTANT THERMOCOUPLE

FIGURE 6A: THERMAL EFFICIENCY APPARATUS

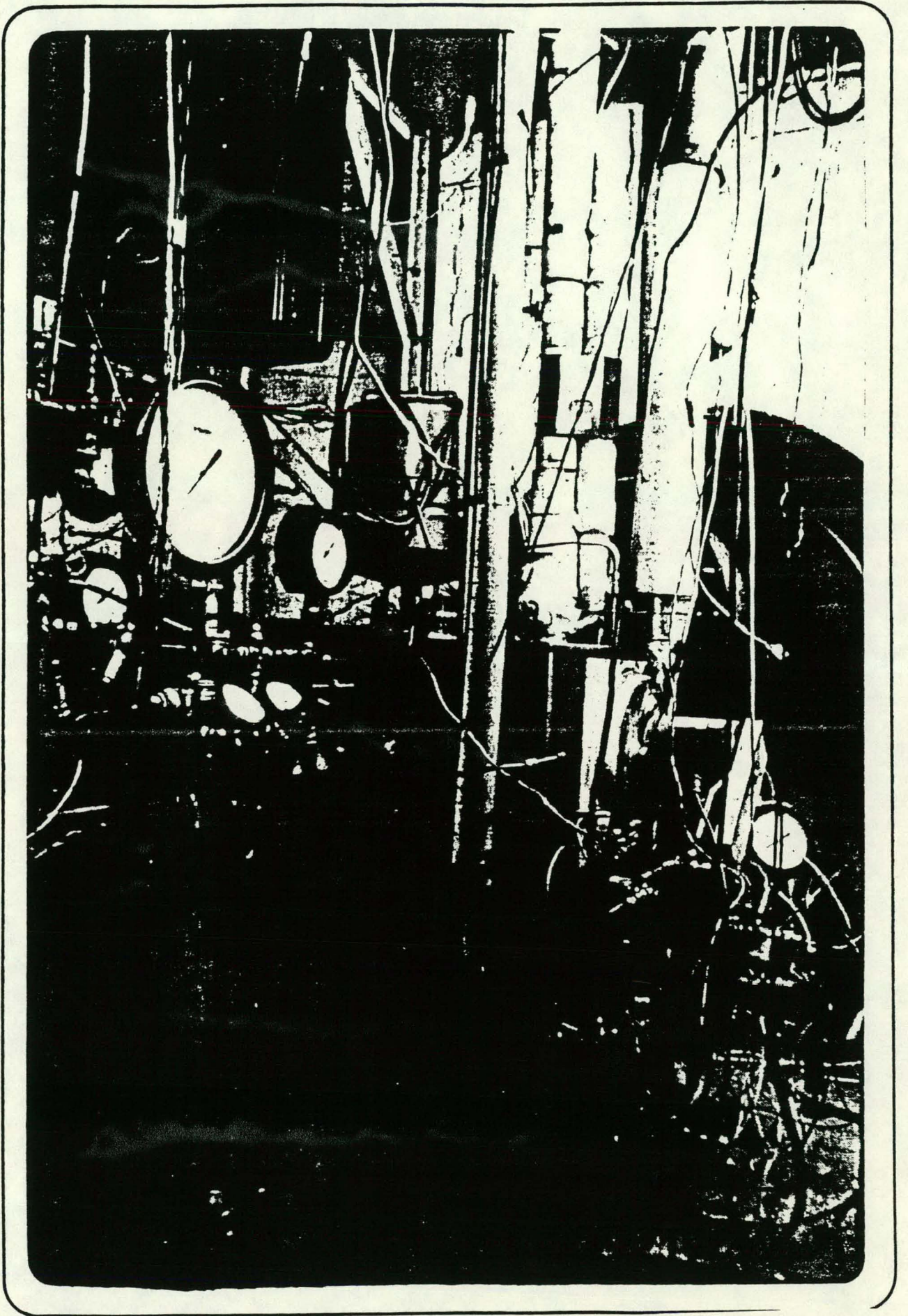


FIGURE 6B: PHOTOGRAPH OF CALORIMETER SET-UP

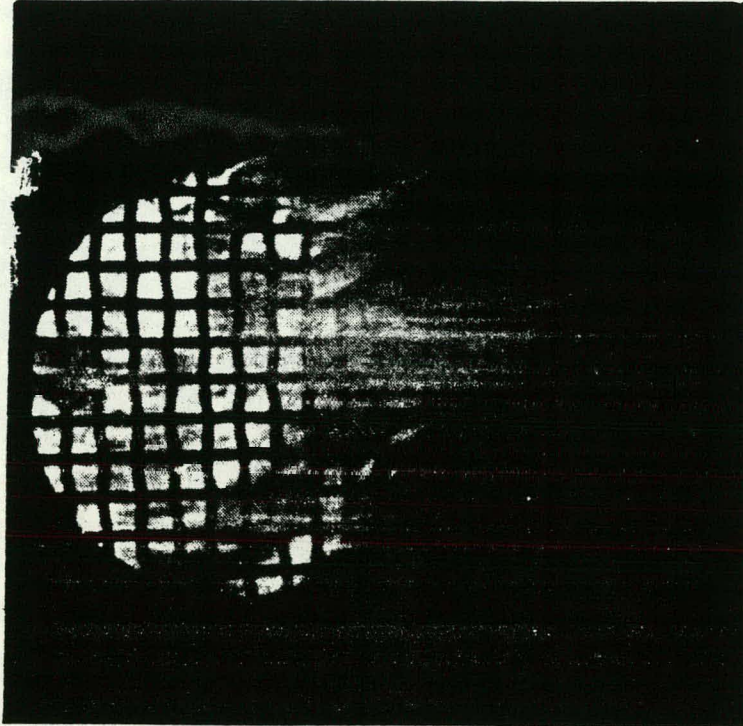


FIGURE 7A: AMBIENT TEMPERATURE AIR

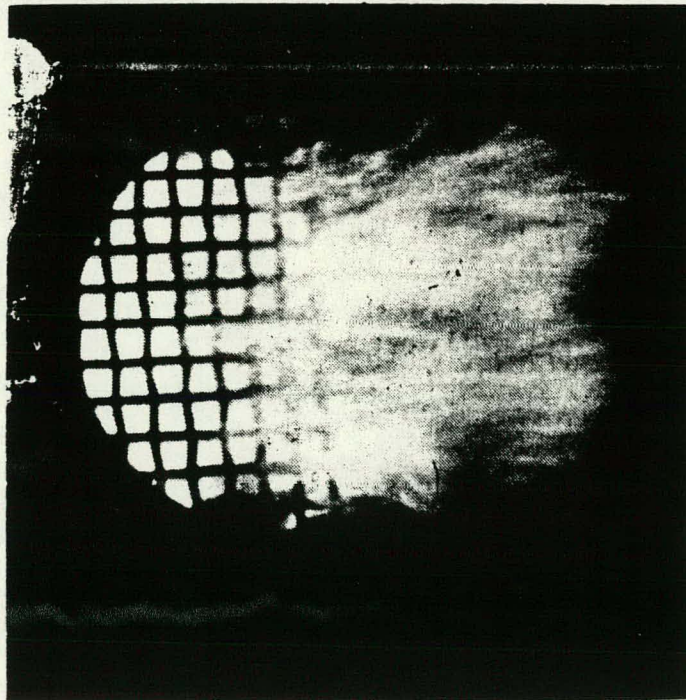


FIGURE 7B: PREHEATED AIR

FIGURE 7: PHOTOGRAPHS OF THE PREHEAT SPRAY COMBUSTION FLAMES

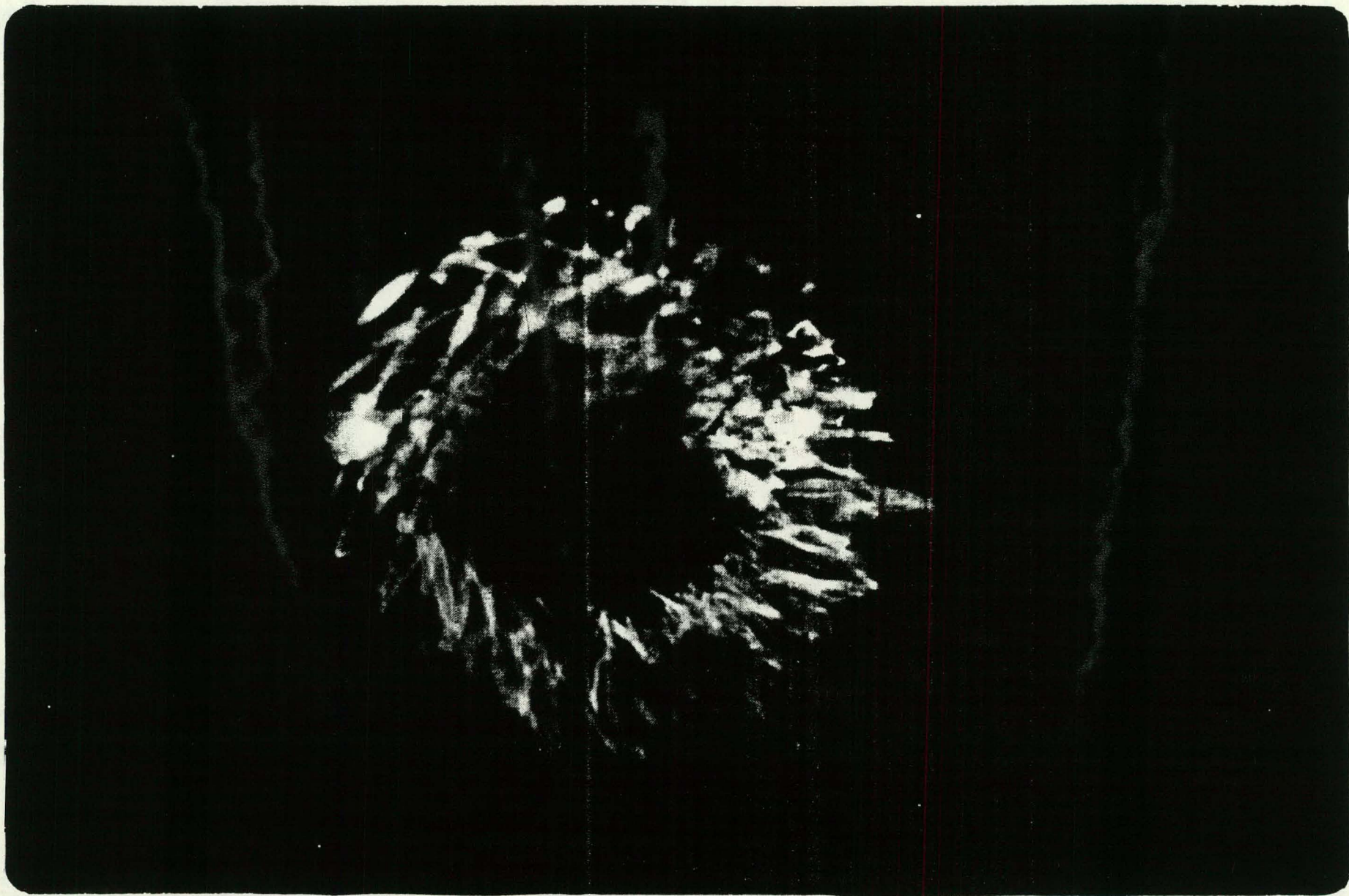


FIGURE 7C PHOTOGRAPH OF BLUE FLAME

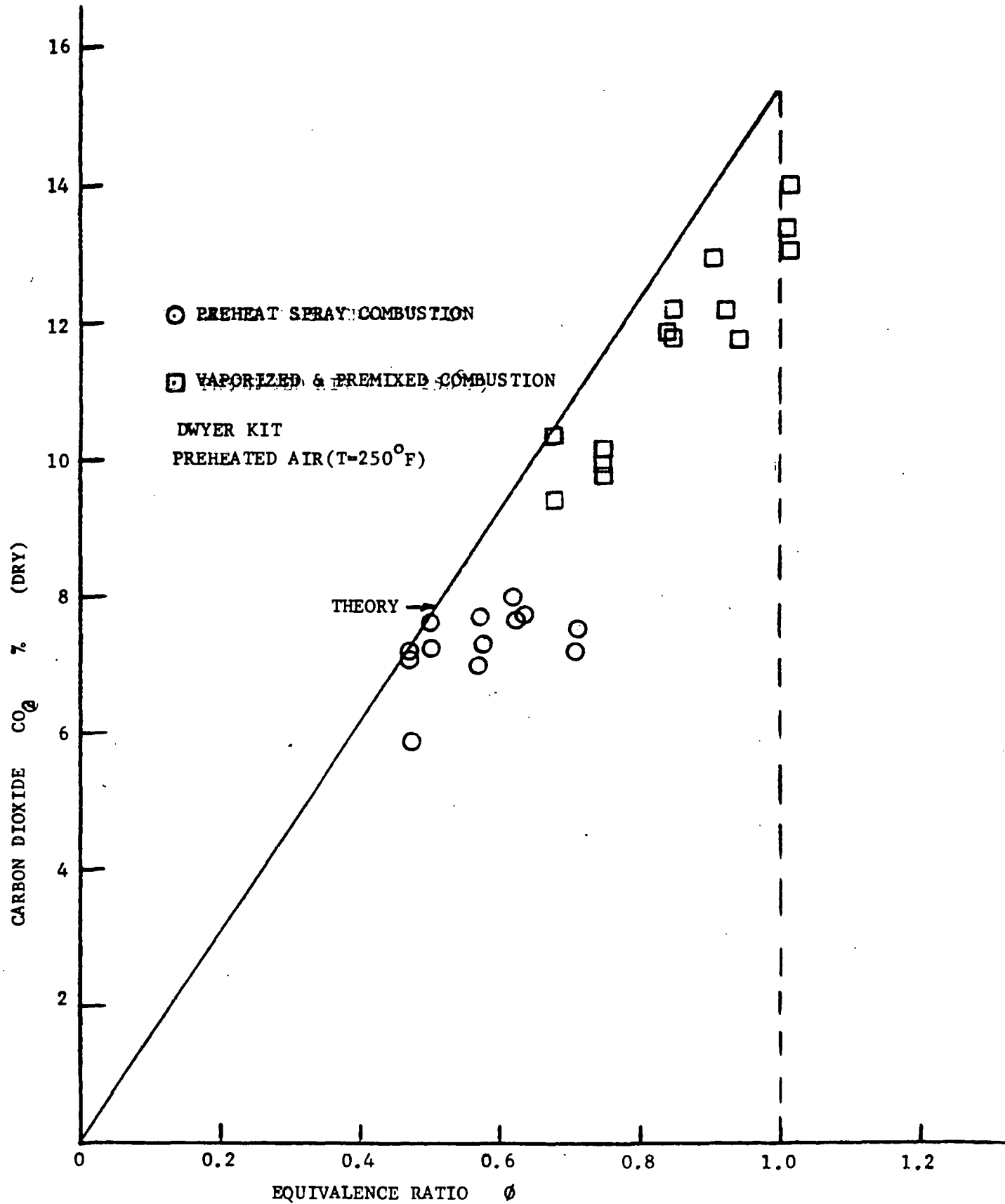


FIGURE 8B: CARBON DIOXIDE PRESENT IN THE COMBUSTION PRODUCTS OF NO. 2 OIL

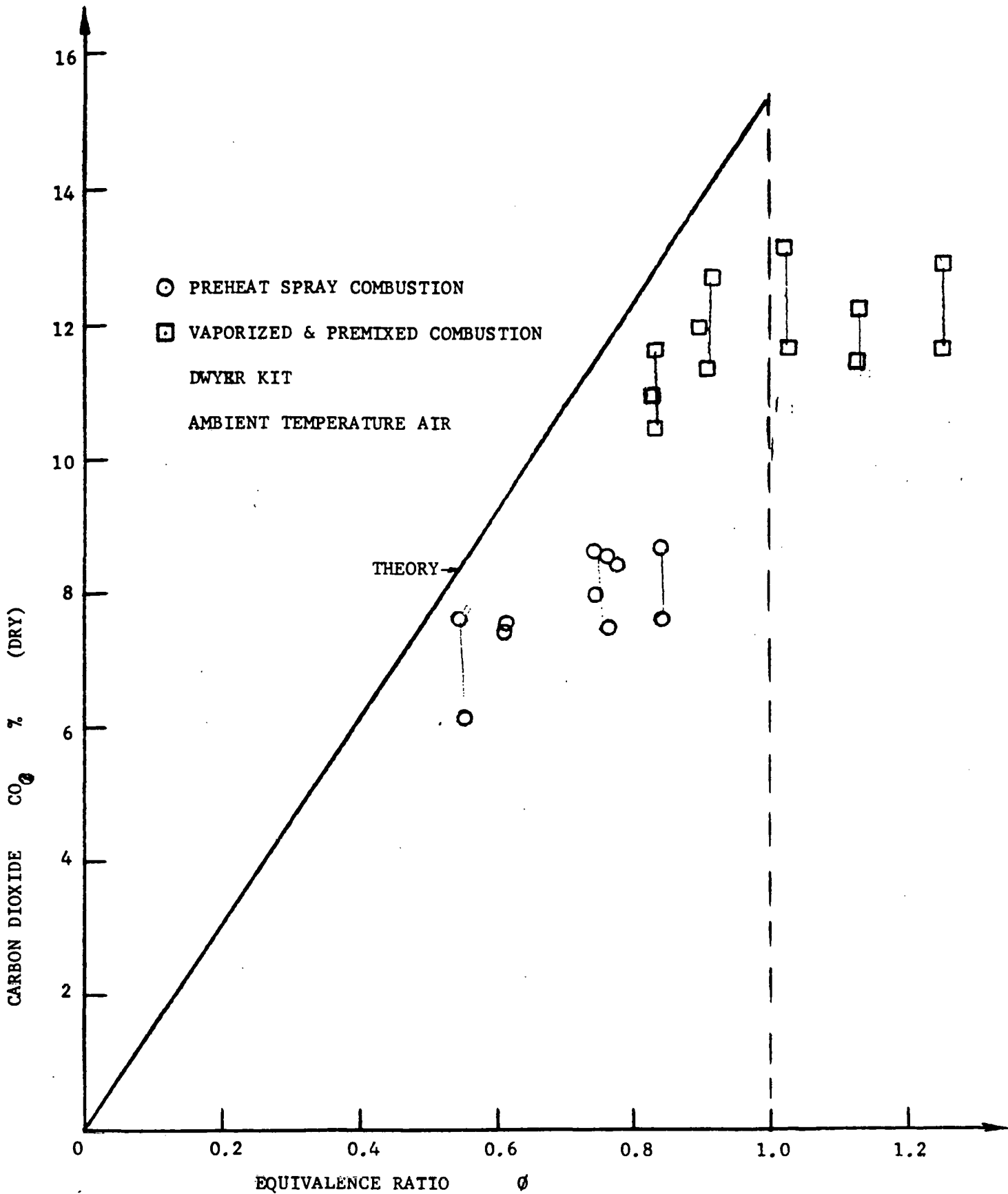


FIGURE 8A: CARBON DIOXIDE PRESENT IN COMBUSTION PRODUCTS OF THE COMBUSTION OF NO. 2 OIL.

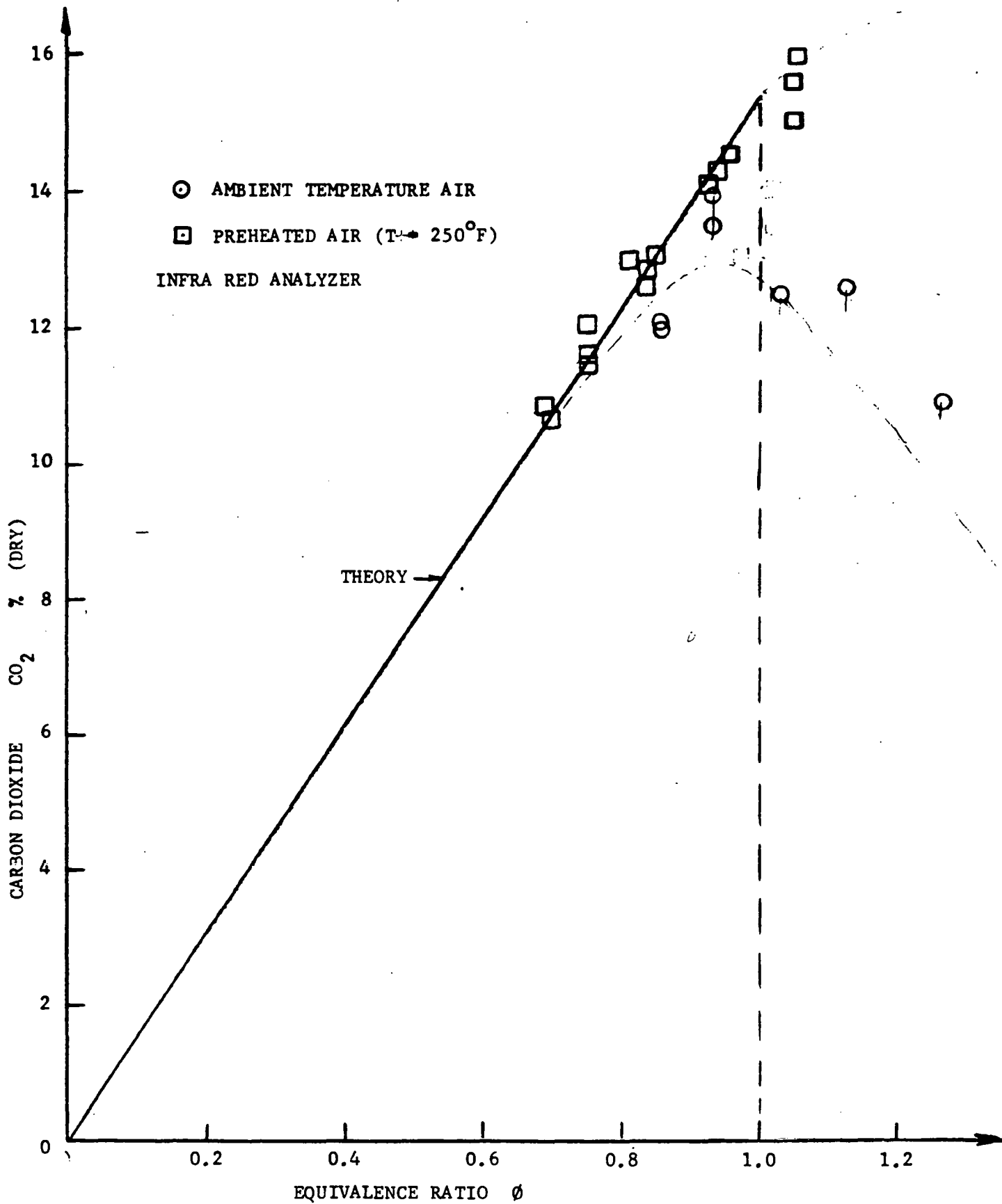


FIGURE 8C: CARBON DIOXIDE PRESENT IN COMBUSTION PRODUCTS OF VAPORIZED AND PREMIXED COMBUSTION OF NO.2 OIL.



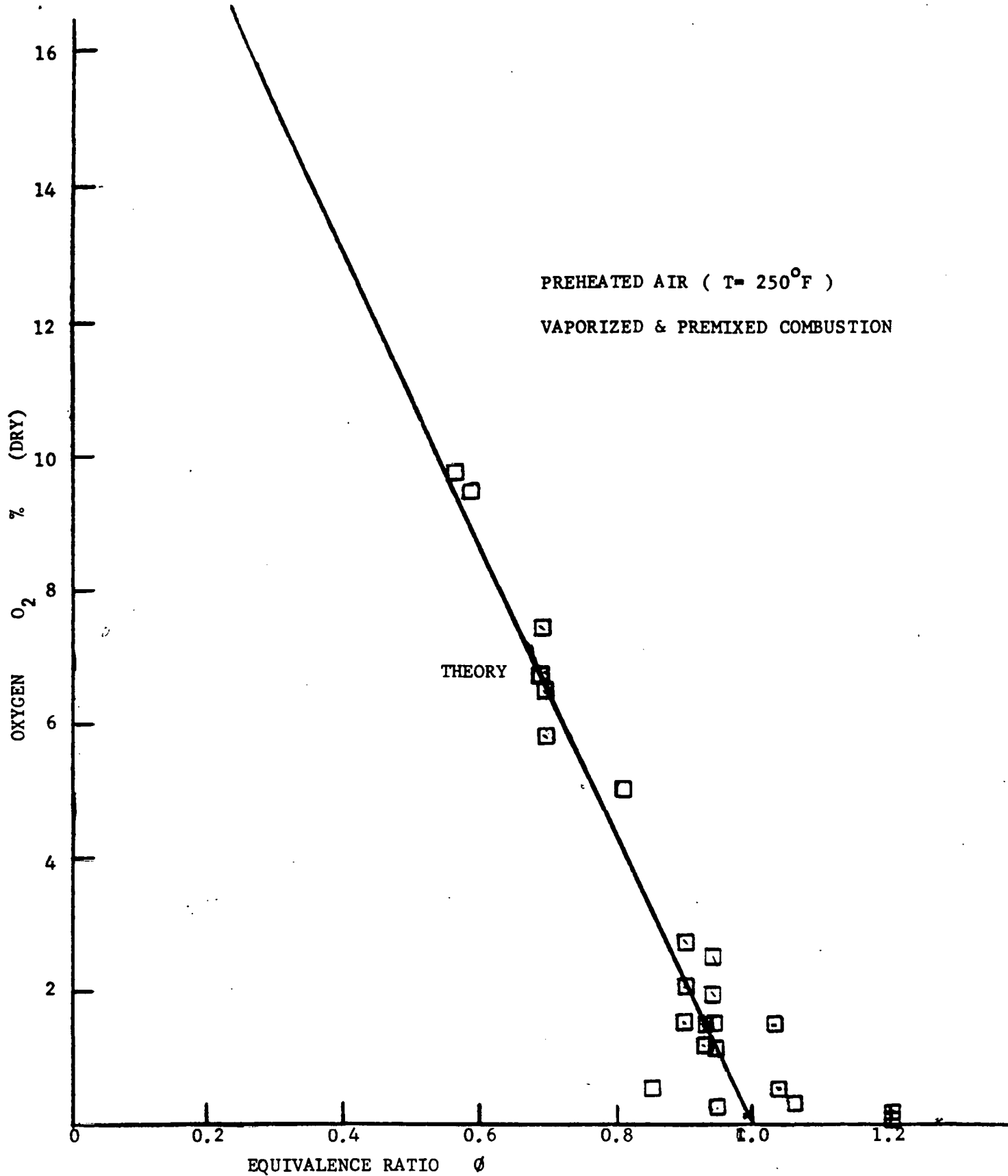


FIGURE 9: OXYGEN PRESENT IN THE COMBUSTION PRODUCTS OF VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL.

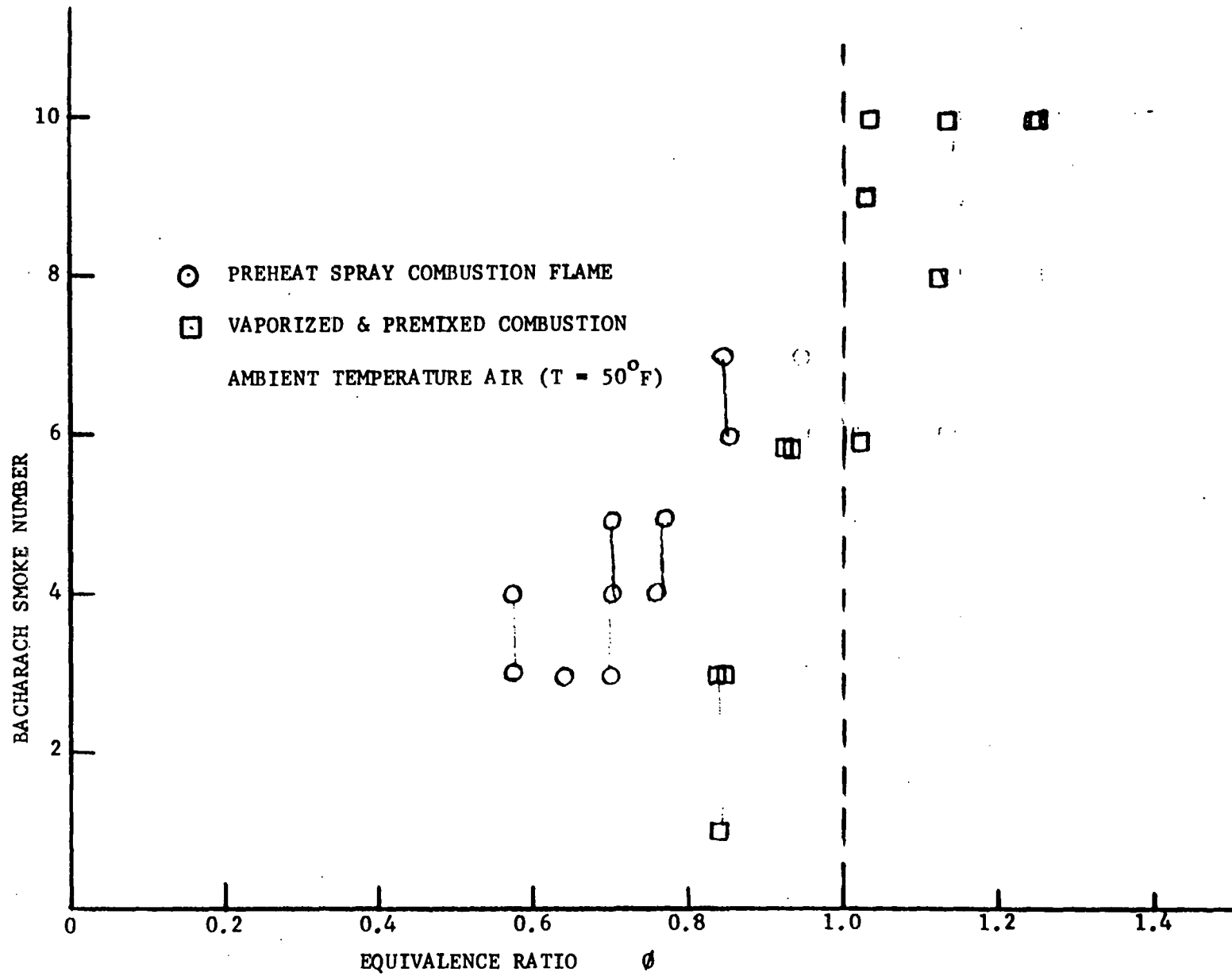


FIGURE 10A: MEASURED BACHARACH SMOKE NUMBERS OF THE PREHEAT SPRAY COMBUSTION FLAME AND OF THE VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL - AMBIENT TEMPERATURE AIR.

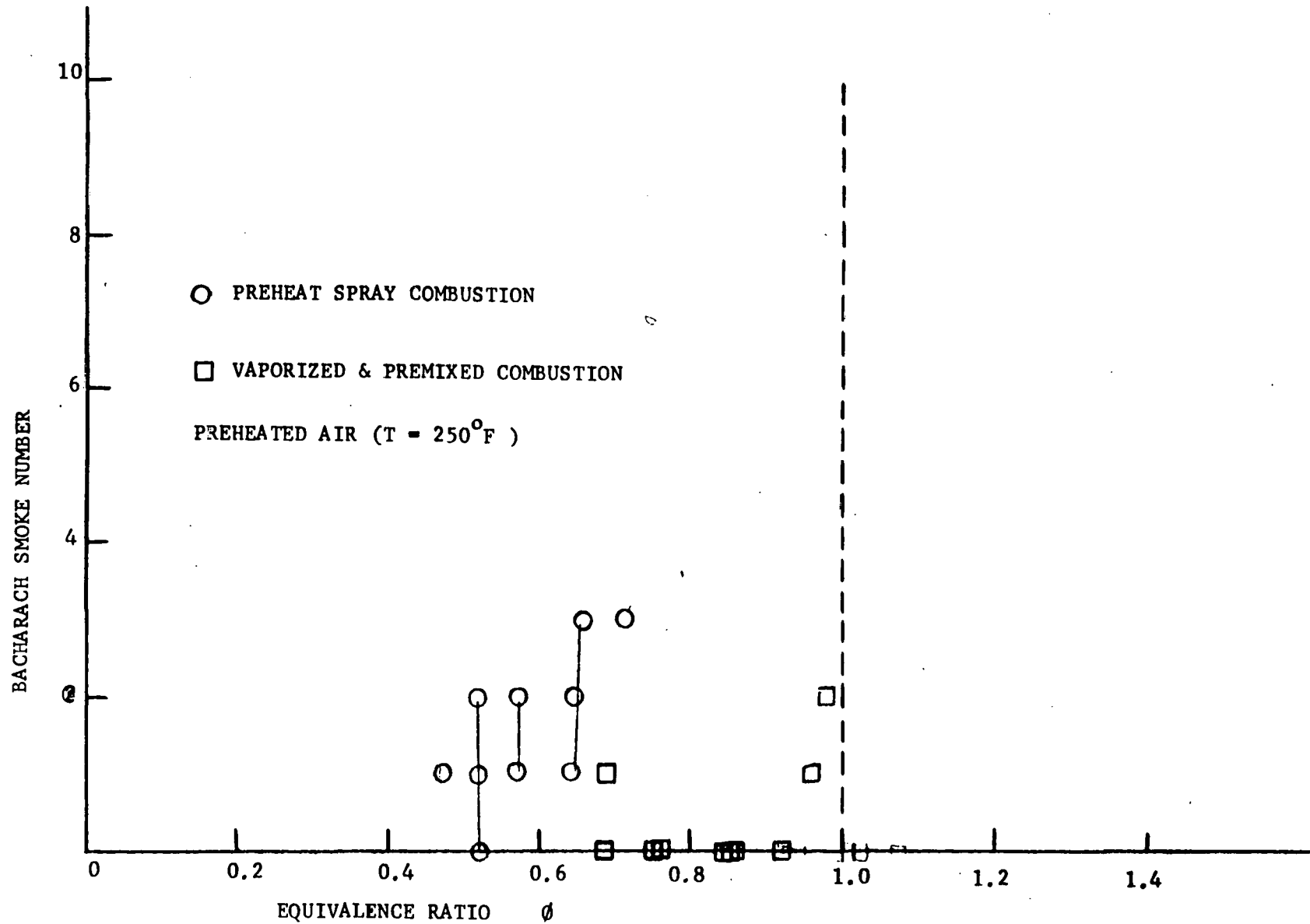


FIGURE 10B: MEASURED BACHARACH SMOKE NUMBERS OF THE PREHEAT SPRAY COMBUSTION AND OF THE VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL - PREHEATED AIR

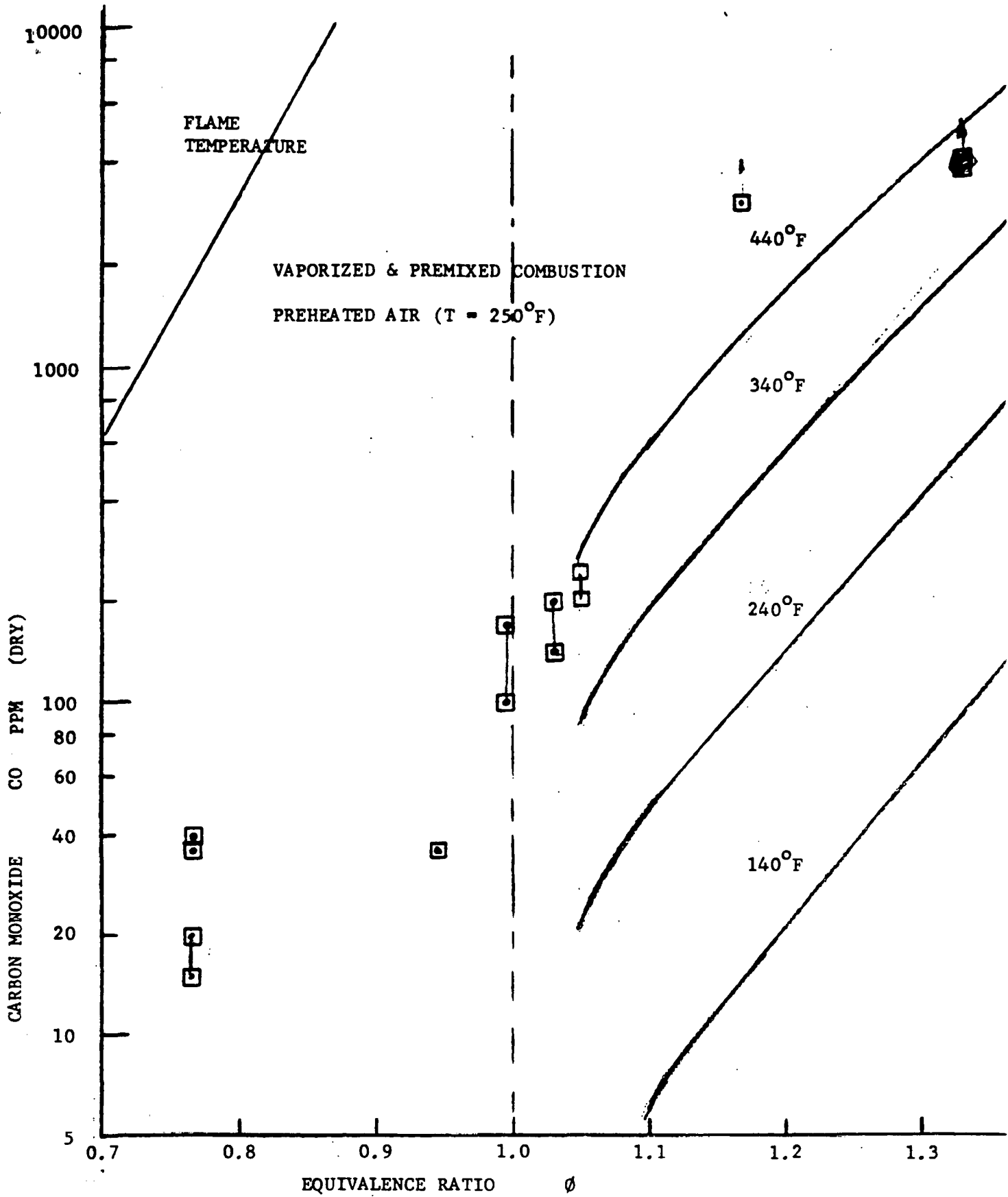


FIGURE 11: CARBON MONOXIDE PRESENT IN FLUE GASES OF VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL.

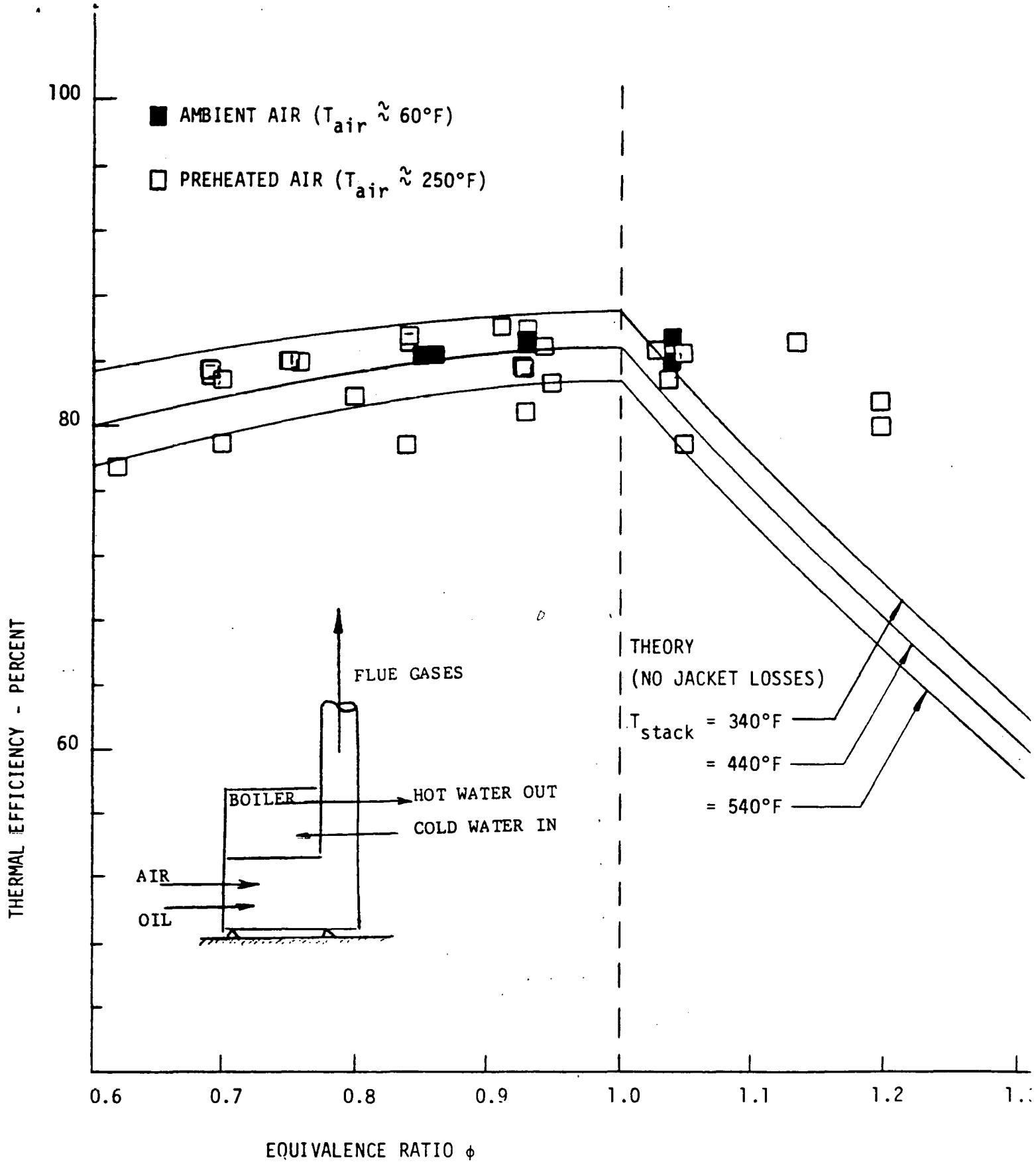


FIGURE 12: THERMAL EFFICIENCY OF VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL MEASURED WITH STACK TECHNIQUE.

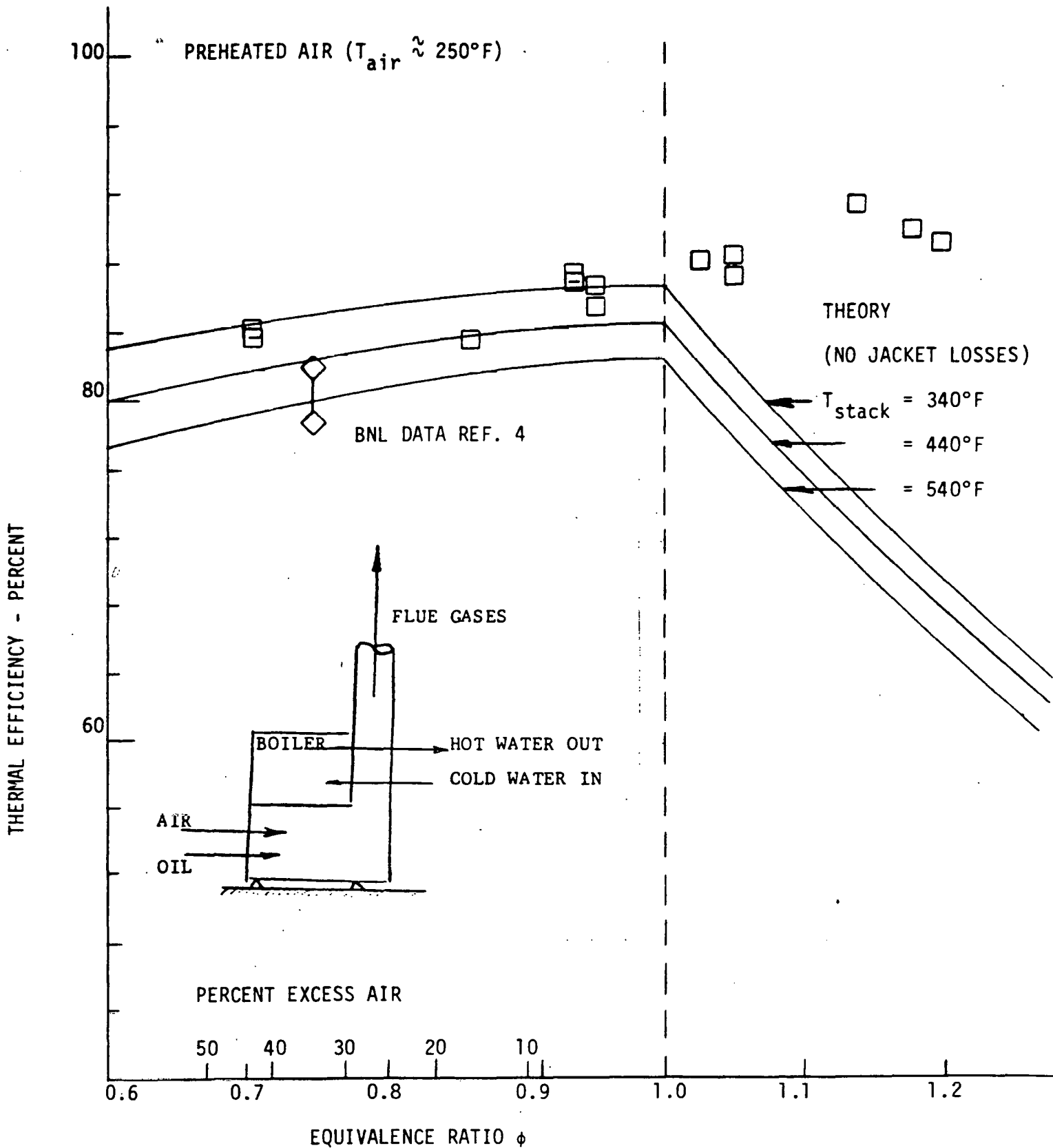


FIGURE 13: THERMAL EFFICIENCY OF VAPORIZED AND PREMIXED COMBUSTION OF NO. 2 OIL MEASURED WITH CALORIMETER TECHNIQUE.

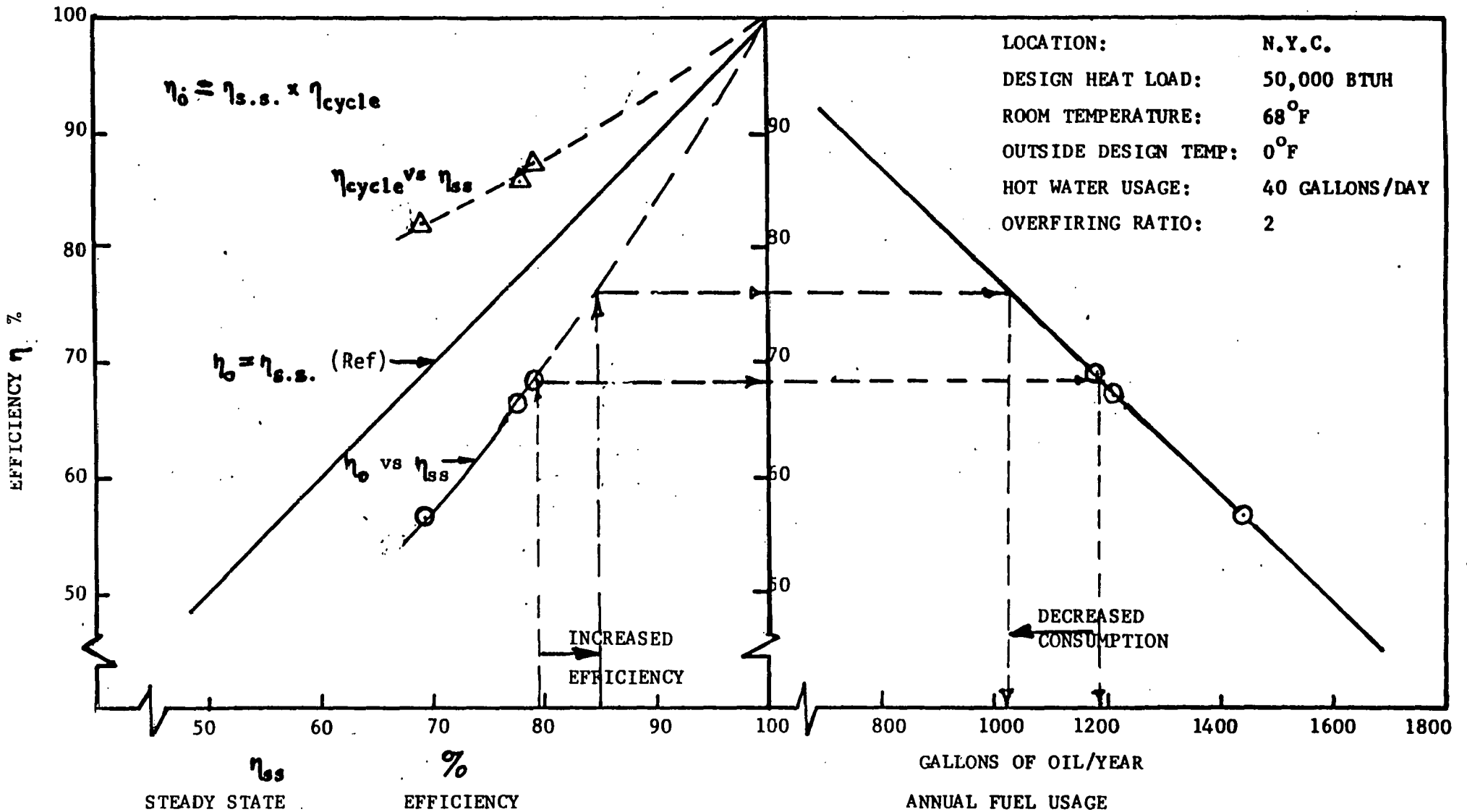


FIGURE 14: VARIATION OF SEASONAL EFFICIENCY WITH STEADY-STATE EFFICIENCY AND ANNUAL FUEL CONSUMPTION.

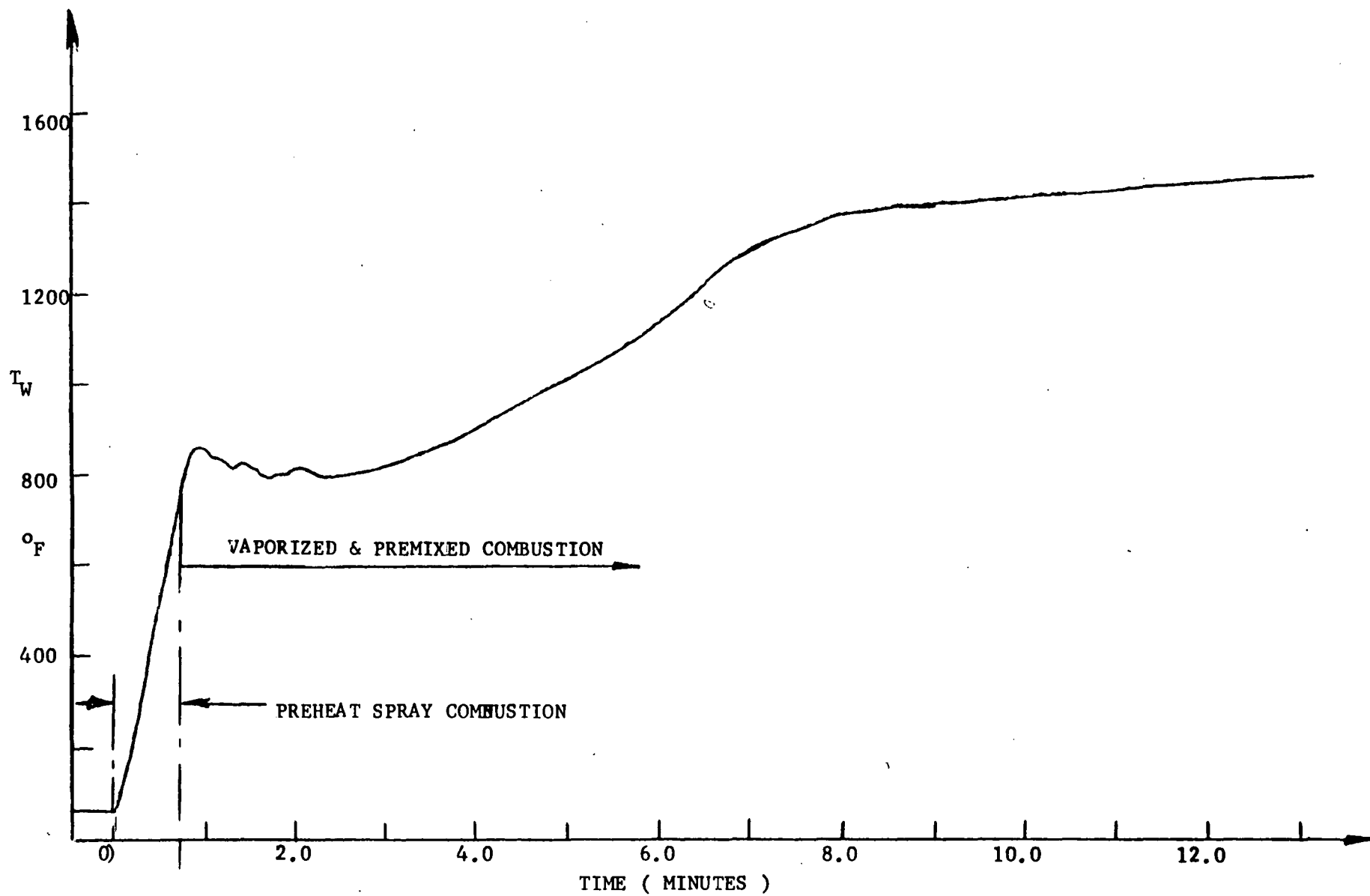


FIG. 15a TYPICAL VAPORIZATION SURFACE HISTORY - AMBIENT TEMPERATURE AIR



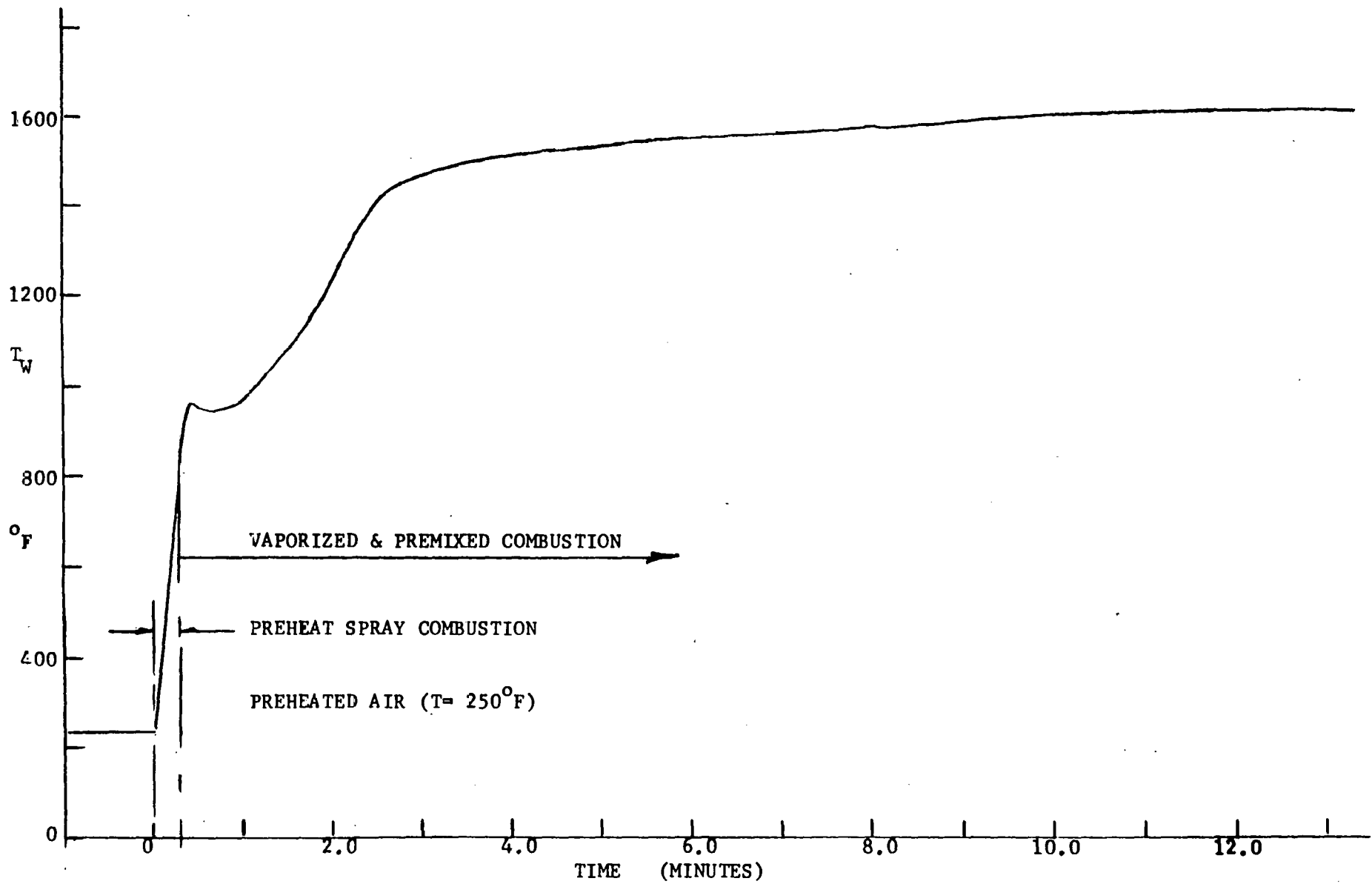


FIG. 15b TYPICAL VAPORIZATION SURFACE TEMPERATURE HISTORY - PREHEATED AIR