NASA CONTRACTOR REPORT

NASA CR-2495

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PREVAPORIZATION AND PREMIXING TO OBTAIN LOW OXIDES OF NITROGEN IN GAS TURBINE COMBUSTORS

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MARCH 1975

1. Report No. NASA CR-2495	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle PREVAPORIZATION AND PREMIXING TO OBTAIN LOW OXIDES OF NITROGEN IN GAS TURBINE COMBUSTORS		5. Report Date March 1975 6. Performing Organization Code
7. Author(s) Gerald Roffe and Antonio Ferri		8. Performing Organization Report No. ATL TR 203 10. Work Unit No.
9. Performing Organization Name and Address Advanced Technology Laboratories, Inc. Merrick and Stewart Avenues Westbury, New York 11530		11. Contract or Grant No. NAS 3-17865 13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code

15. Supplementary Notes

Final Report. Project Manager, Cecil J. Marek, Airbreathing Engines Division, NASA Lewis Research Center, Cleveland, Ohio

16. Abstract

Tests were conducted to determine the effectiveness of prevaporization and premixing in reducing the formation of oxides of nitrogen in a gas turbine type combustor using liquid JP-5 fuel at the supersonic cruise condition. The combustor inlet temperature was 833 K (1500° R) at a pressure of 4 atmospheres and a reference velocity of 46 m/sec (150 ft/sec). An order of magnitude reduction in nitric oxide emissions was achieved. Nitric oxide emission indices as low as 0.6 gm NO_2/kg fuel were measured at an equivalence ratio of 0.29 with one percent combustion inefficiency without vitiation of the mixer stream.

17. Key Words (Suggested by Author(s)) Premixing; Prevaporization; Combustion; Oxides of nitrogen; Gas turbine combustors	18. Distribution Statement Unclassified - unlimited STAR category 07 (rev.)

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	55	$4.25

* For sale by the National Technical Information Service, Springfield, Virginia 22151

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SECTION I

INTRODUCTION

The problem of reducing emission levels of nitrogen oxides (NO_x) in gas turbine exhaust without sacrificing engine performance is an extremely important one. This is especially true for gas turbines operating at high altitude supersonic flight conditions where the high combustor inlet temperatures increase NO_x production rates and the effects of NO_x on stratospheric ozone may be potentially hazardous. In Reference (1), a method was proposed for the reduction of NO_x in combustion processes based on an appraoch wherein liquid fuel is evaporated and thoroughly premixed with enough air to produce a low overall equivalence ratio prior to ignition. When such a mixture is burned, local temperature peaks which accelerate the production of NO_x are eliminated and NO_x production rates are kept low.

This report describes an experimental program whose aim was the demonstration of the prevaporization-premixing technique at conditions representative of gas turbine operation at high altitude and high Mach number using liquid JP-5 fuel. To do this, air was supplied by a pebble bed heater to a combustion apparatus at a temperature of 833K (1500^{o}R) and a pressure of $4 \times 10^5 N/m^2$ (4 atmospheres). Liquid JP-5 was sprayed into the combustor entrance duct at a station several diameters upstream of a flameholder. The duct section between the fuel injection and flameholder stations provided time for the liquid fuel droplets to evaporate and the gaseous fuel to mix with air prior to combustion. Gas samples were taken downstream of the flameholder station to determine emission indices and combustion efficiencies. The system studied was intended to represent the combustor primary zone and did not include the secondary or dilution zone.

Current engines produce NO_x emission indices of from 10 to 20 gm NO_2/kg fuel at the supersonic cruise condition. It was the goal of this program to demonstrate that a NO_x emission index of 1 gm NO_2/kg fuel could be achieved with CO and unburned hydrocarbon emission indices of 1 gm/kg fuel and 0.5 gm/kg fuel respectively and a combustion inefficiency no greater than 1%. It was a further object of this program to determine whether evaporation and gas phase mixing could be completed without producing either auto-ignition in the mixer tube and/or flashback from the combustor.

SECTION II

DESCRIPTION OF THE PREVAPORIZATION-PREMIXING CONCEPT

The production of NO_x during the combustion of nitrogen-free hydrocarbon fuels is extremely sensitive to temperature. This extreme sensitivity derives from the fact that the principal NO_x production reactions involve either atomic oxygen, atomic nitrogen or the hydroxyl radical, none of which appear in significant concentrations at low temperatures. This sensitivity is compounded by the fact that the rate constants for the reactions in which these species take part are themselves highly sensitive to temperature.

In the standard diffusion flames characteristic of present combustors, mixing and reaction occur simultaneously and produce local regions in which stoichiometric proportions of fuel and air exist. The temperature level in these regions is extremely high, especially when the combustor inlet air has been preheated by the effects of compression, and results in the production of large quantities of NO_x which subsequent dilution causes to freeze chemically.

In Reference (1), a method was proposed to reduce total NO_x production which consisted of prevaporizing and premixing liquid fuel with air prior to combustion, thus separating the mixing and reaction steps entirely. If this can be accomplished in a mixture whose overall equivalence ratio is less than one, the local temperature can never exceed the average sub-stoichiometric level. In the case of a gas turbine, where allowable turbine entrance temperature restrictions require operation at low overall equivalence ratios, this method is particularly promising since local temperature levels can be reduced substantially by premixing.

In Reference (2), a series of computations were carried out to determine the rate of NO production in premixed flames of vaporized JP fuel in air accounting for the effects of finite rate chemistry and turbulent transport processes. The results of these computations are shown in Figure (1). Taking a combustor entrance temperature of 833K (1500°R) as typical of operating conditions for

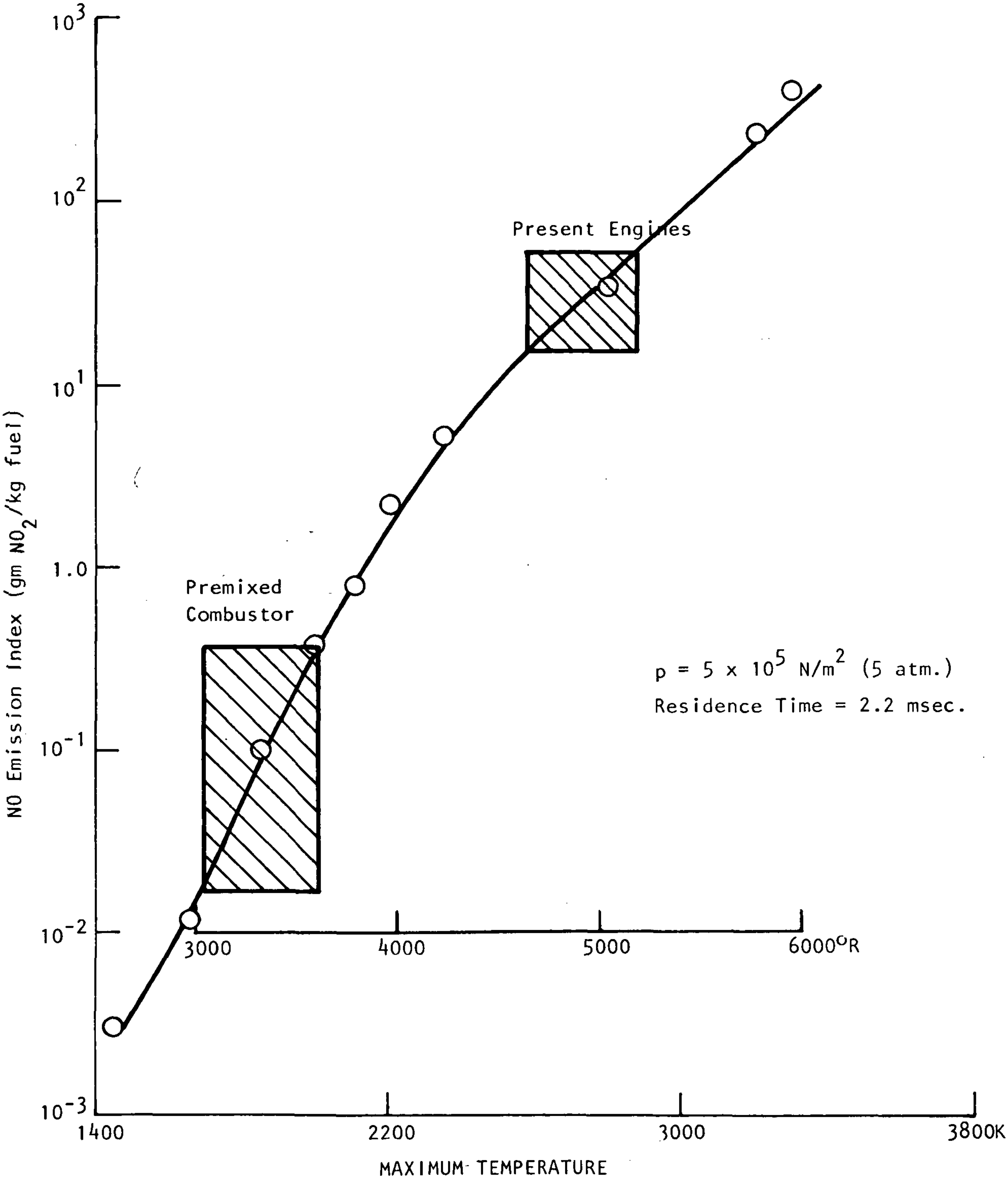


FIGURE 1. CALCULATED NO_x EMISSION FROM PREMIXED FLAMES REFERENCE (2)

an SST-like application, the results of Reference (2) may be replotted as a function of engine equivalence ratio (fuel/air ratio divided by stoichiometric fuel/air ratio) as shown in Figure (2). It should be noted that in converting the results of Reference (2) to the form presented in Figure (2), NO_x emission indices have been multiplied by the ratio of the molecular weight of NO_2 to that of NO in order to put them into the standard form used in reporting engine emission data.

Figure (2) illustrates the extremely sensitive dependence of NO_x level on equivalence ratio for premixed systems. It can be seen from the figure that to achieve an emission index below one, an equivalence ratio below 0.6 is required. However, the sensitivity to equivalence ratio places rather stringent requirements on the degree of mixing which must be achieved. For example, Figure (2) indicates that a completely uniform mixture at an equivalence ratio of 0.6 will produce twenty three times the NO_x that would be produced at an equivalence ratio of 0.4. As a result, if a mixture with an overall equivalence ratio of 0.4 were to contain nonuniformities amounting to only 10% of the flow where the local equivalence ratio was 0.6, then the total NO_x produced would be three times higher than that of a completely uniform mixture. Therefore, it is clear that the application of the prevaporization-premixing concept to gas turbine combustion problems requires the ability to produce a very high degree of premixing if the potential gains of the technique are to be fully realized.

It must be emphasized that vaporization of the fuel prior to combustion is fundamental to the approach demonstrated in this work. The alternate approach of burning in a uniformly lean mixture of fuel droplets and air is not sufficient to reduce NO_x formation. The reason for this difference again lies in the sensitivity of NO_x production rates to temperature. When liquid droplets are present in a combustion zone, evaporation creates a layer of gaseous fuel surrounding their surfaces. As this gas diffuses outward, the local concentration of fuel decreases from the fuel rich condition which exists near the surface to the fuel lean condition which must eventually prevail. The transition from fuel rich to fuel lean produces regions of stoichiometric and near stoi-

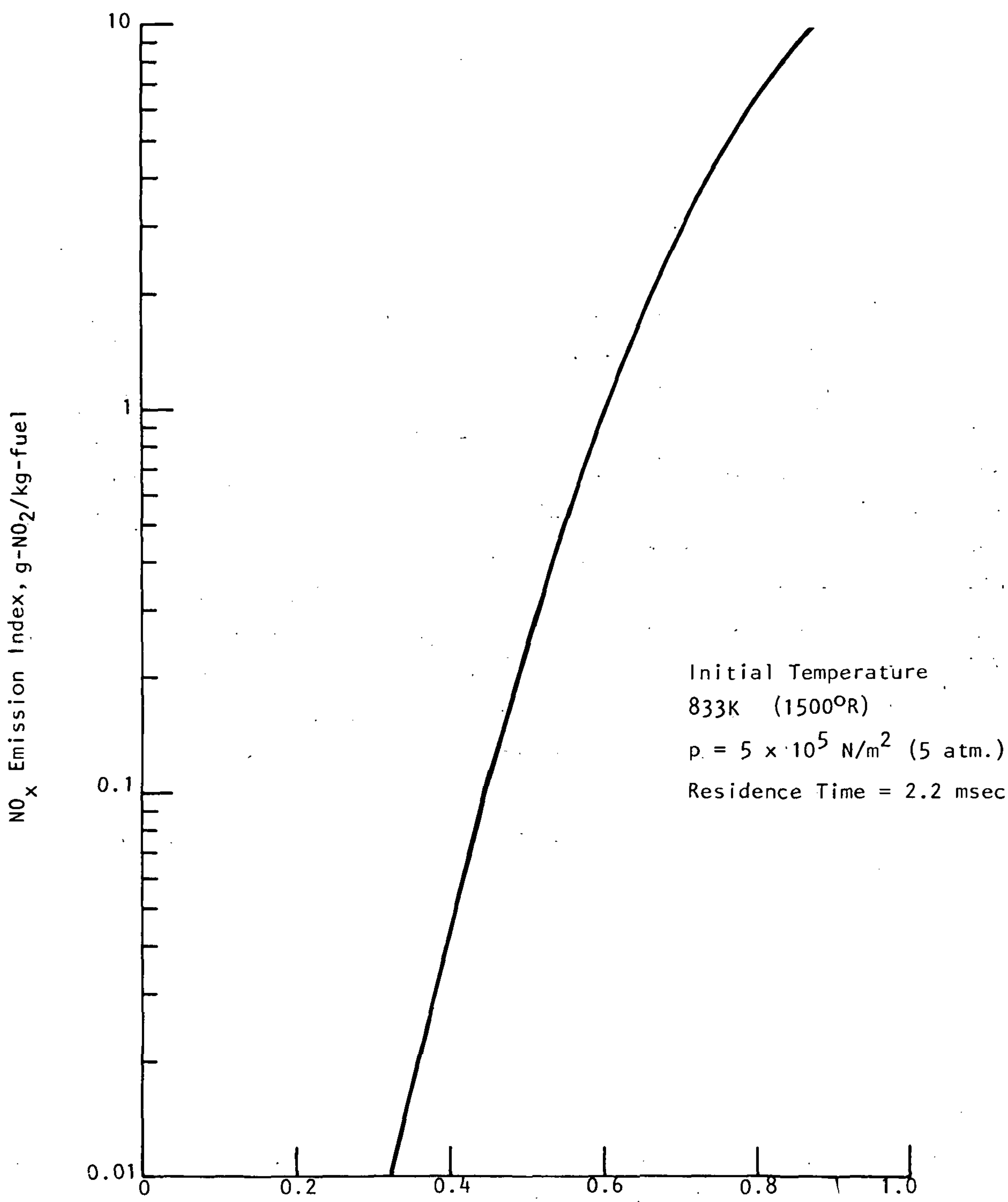


FIGURE 2. NO_x EMISSION INDEX AS A FUNCTION OF EQUIVALENCE RATIO

chiometric proportions and the higher reaction rates which prevail at these conditions result in zones of extremely high temperature. Since the time for NO_x formation at these temperatures is much shorter than any diffusion time, even for droplet sizes of a few microns, NO_x is formed. On the other hand, since the time to reach thermal equilibrium on a molecular scale is much shorter than that required for NO_x formation, little NO_x is formed when the combustion occurs in a premixed gas whose composition is such that a low final temperature is produced. The NO_x levels which result when liquid droplets are present are thus typical of ordinary diffusion flames and little advantage is gained over non-premixed systems.

SECTION III

TEST APPARATUS AND PROCEDURES

The experiments described here were carried out using the pebble bed blow-down facility of General Applied Science Laboratories whose pertinent components are illustrated in Figure (3). Mechanical compressors fill a bank of storage bottles with air at pressures ranging from 1×10^7 to 1.4×10^7 N/m^2. The air is dried prior to storage and contains less than 2×10^{-4} kg of water per kg of air. Prior to a test, the bed of aluminum oxide pebbles is heated by electric glow-bars to a preset temperature. Air from the storage bank is passed through the bed of heated pebbles and into the combustion apparatus. The experiments were conducted at a nominal flow rate of 1.36 kg/sec (3 bl/sec) under which condition the pebble bed pressure was approximately 14×10^5 N/m^2 (200 psia).

The basic layout of the combustion apparatus is shown in Figure (4). Four specific versions of this apparatus were tested and their details will be presented later but the basic functioning of the device is illustrated in Figure (4). Heated air from the choked pebble bed exit orifice enters a plenum chamber from which it flows into two separate supply lines. The principal line, a 10.2 cm (4-inch) pipe equipped with a 3.43 cm (1.35-inch) diameter metering orifice section, carries approximately 92% of the total air flow to the toroidal mixer plenum. The plenum distributes the air uniformly by exhausting through 40 choked 0.64 cm (1/4-inch) diameter orifices into the combustion apparatus. The entering air flows through the fuel injection section which produces a 7.62 cm (3-inch) inside diameter constriction.

Fuel is sprayed through four pressure atomizing nozzles inclined in the downstream direction at 75^o to the flow axis. The four nozzles are each rated at 0.0124 kg/sec (15 gal/hr) with a 7×10^5 N/m^2 (100 psi) pressure differential and produce a 60^o half angle spray in quiescent air. This fuel injection system is rather coarse, and would be expected to produce relatively large initial nonuniformities of fuel/air ratio across the duct. Subsequent mixing is then required to reduce the level of nonuniformity to an acceptable value.

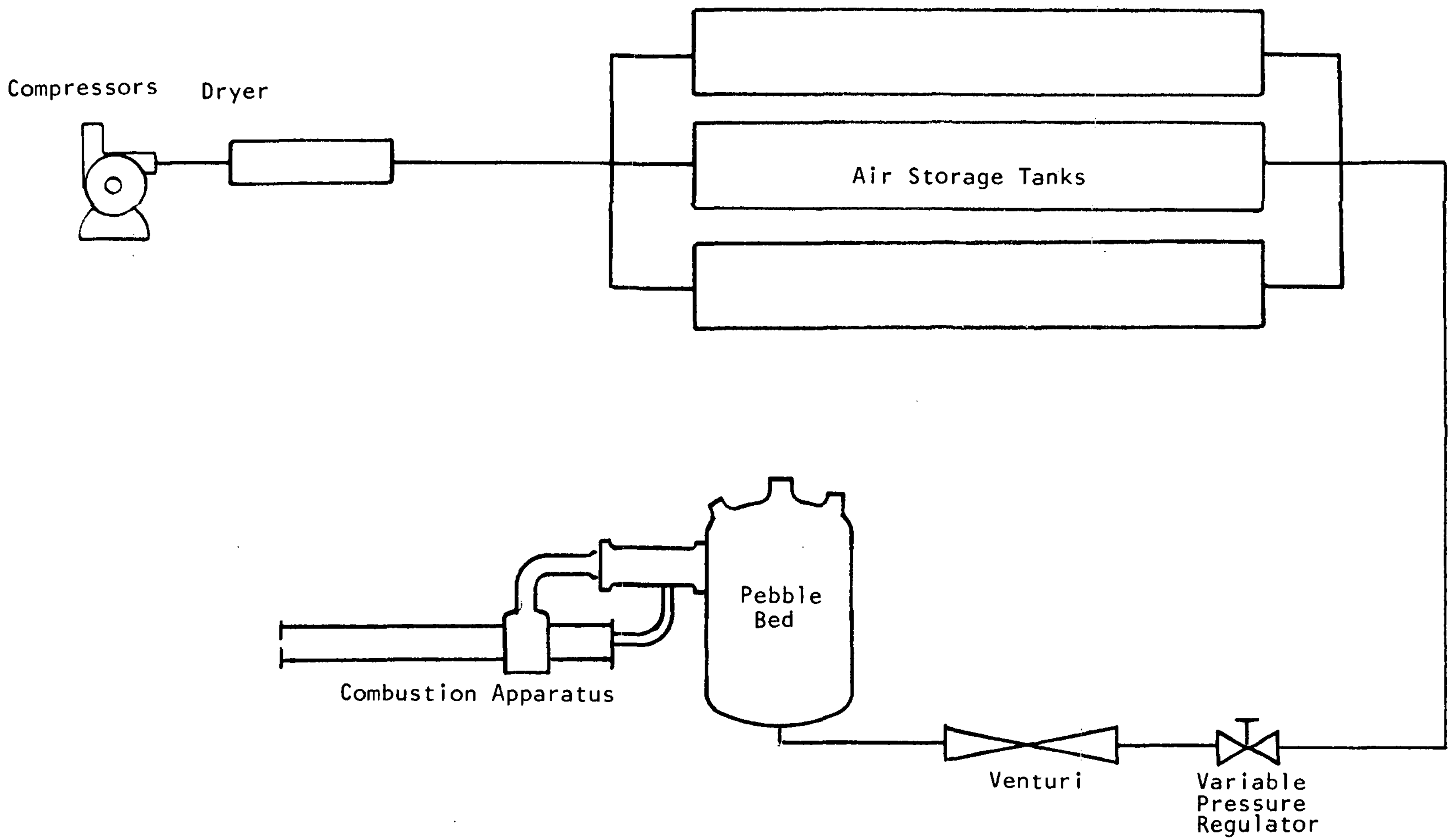


FIGURE 3. SCHEMATIC OF GASL PEBBLE BED BLOW-DOWN FACILITY

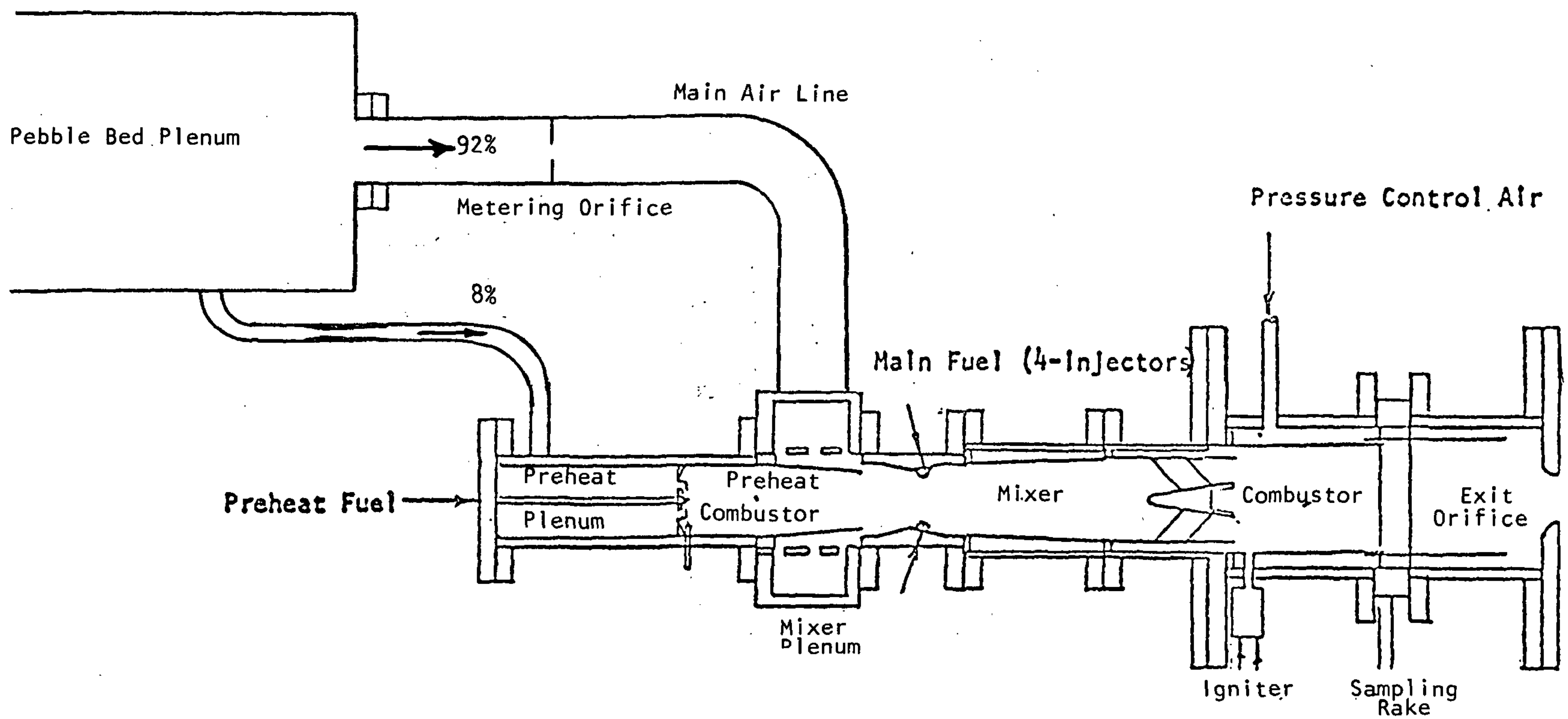


FIGURE 4. BASIC COMBUSTION TEST APPARATUS

The test combustor consists of a vitiating preheater, vaporizer-mixer and reactor. The preheater was installed to improve fuel vaporization and combustion stability at the fuel lean conditions.

Several mixing ducts were used in the experiments. Each was constructed of 0.16 cm (1/16-inch) thick stainless steel sheet and was separated from the outer pressure wall by a dead air gap to minimize heat loss from the apparatus. The liner wall thickness was kept small to reduce the thermal lag of the system to less than 10 seconds.

The basic flameholder consisted of a 10° half angle conical centerbody and a sudden expansion from the mixer exit diameter of 11.4 cm (4.5-inches) to the 15.2 cm (6-inch) diameter combustor section. The centerbody was supported by two 6.3 mm (0.25-inch) struts swept forward from the cone at an angle of 45°. The relative axial position of the cone and outer expansion step was varied as was the constrictive area of the flameholder. The detailed geometry of the configurations tested will be presented later.

The 15.2 cm (6-inch) diameter combustor liner was also fabricated from 0.16 cm (1/16-inch) stainless steel sheet and was separated from the pressure wall by a 2.5 cm (1-inch) gap through which compressed air was passed during the test to prevent burning the liner. This air was exhausted along the rear wall of the combustor where it served as a coolant for the 7.62 cm (3-inch) diameter orifice plate which terminated the apparatus. The exit orifice was sized to produce a combustor pressure of 3×10^5 N/m^2 (3-atmospheres) for the 1.36 kg/sec (3 lb/sec) flow condition at the maximum equivalence ratio tested. The liner cooling air flow rate was adjusted during each test to provide enough additional mass flow to raise the combustor pressure to 4×10^5 N/m^2 (4-atmospheres).

The remaining 8% of the total air flow entered a small preheat section through a 2.54 cm (1-inch) pipe and 0.89 cm (0.35-inch) throat diameter venturi. This flow was straightened in a small settling chamber which was terminated by a perforated plate and a set of straightening screens and supplied air to the preheater. The preheater consisted of a 6.6 g/sec (8-gal/hr) nozzle mounted axially in a set of 60^{o} swirl vanes with a 3.8 cm (1.5-inch) diameter plate surrounding the nozzle to create a flame stabilization zone. The preheat section was operated at a nominal equivalence ratio of 0.6.

The JP-5 fuel was stored in two tanks: one which supplied the main fuel injectors and a second which supplied the preheat fuel nozzle. The tanks were pressurized with nitrogen to force out fuel during the tests. The main fuel flow rate was measured by a cavitating venturi placed upstream of the manifold supplying the four main fuel nozzles; the preheat fuel flow rate was obtained by calibrating the preheat nozzle as a function of driving pressure less combustor pressure and measuring both pressures during each run. A physical analysis of the JP-5 used in the test program is shown in Table I.

TABLE I

JP-5 PHYSICAL ANALYSIS

Specific gravity at 288K (60^{o}F)	0.815 (42.1 deg A.P.I.)
Flash point, PM	329K (134^{o}F)
Pour point	227K (-50^{o}F)
Viscosity at 310K (100^{o}F)	1.5×10^{-6} m^2/sec (31 sec. S.S.U.)
Initial boiling point	452K (355^{o}F)
10% Distillate	469K (385^{o}F)
20% Distillate	477K (400^{o}F)
50% Distillate	499K (440^{o}F)
90% Distillate	544K (520^{o}F)
Final boiling point	555K (540^{o}F)
Residue, % by volume	3

The combustor section was equipped with a cruciform sampling rake which provided a water-cooled support for 16 individual stainless steel tubes (2.1 mm I.D.). The tubes were spaced such that their radial locations were at the centers of equal areas and their manifolded reading was taken as a mass average across the combustor.

The sampling system is shown schematically in Figure (5). The sampling manifold was connected by a 6.4 mm (1/4-inch) stainless steel line to a high flow pressure regulator and dump valve. The sample, collected at a pressure of 4×10^5 N/m^2 (4-atmospheres) was regulated down to a pressure of 2×10^5 N/m^2 (2-atmospheres) before being divided by a set of metering valves into four individual streams. The sample line was heated to a temperature of 450K (350^{o}F) up to a Beckman Model 402 hydrocarbon analyzer which accepted one of these streams; the remaining three sample streams were allowed to cool to 365K (195^{o}F). One of these was passed through a Beckman Model 951 NO/NO_x analyzer (chemiluminescence), another led to a Beckman Model 864 infrared analyzer (CO_2), a Beckman Model 742 oxygen analyzer (polarographic) and a Beckman Model 315B infrared analyzer (CO), connected in series. The last line was used as a dump. Flow rates through the system were kept high by maximizing the amount of sample dumped both up and downstream of the pressure reduction regulator. Calibration gas was introduced through a three way valve located just downstream of the sample manifold. Zero gas for the oxygen, CO_2 and CO analyzers (dry nitrogen) entered through a three way valve located just upstream of the set of metering valves. The NO/NO_x and hydrocarbon analyzers provided internal sources of zero gas. Gas analysis procedures and data reduction equations are in accordance with ARP 1256, Reference (3).

The test apparatus was equipped with temperature and pressure rakes at the inlet to the preheat section, thermocouples and pressure taps in the mixer plenum, two wall thermocouples on the mixer liner and one which protruded just through the liner to measure gas temperature at the wall. The sampling rake was used to measure pitot pressure profiles across the combustor section and a special cruciform rake was provided as a replacement for the flameholder which yielded temperature and total pressure profiles at the mixer exit as well as gas samples for species measurements. Air flow rates were obtained by measuring upstream and downstream pressures at the orifice plate and static and stagnation pressure at the venturi. Stagnation temperature was measured at each flow metering station to account for heat losses in calculating mass flows. All parameters measured were recorded on Honeywell visicorders.

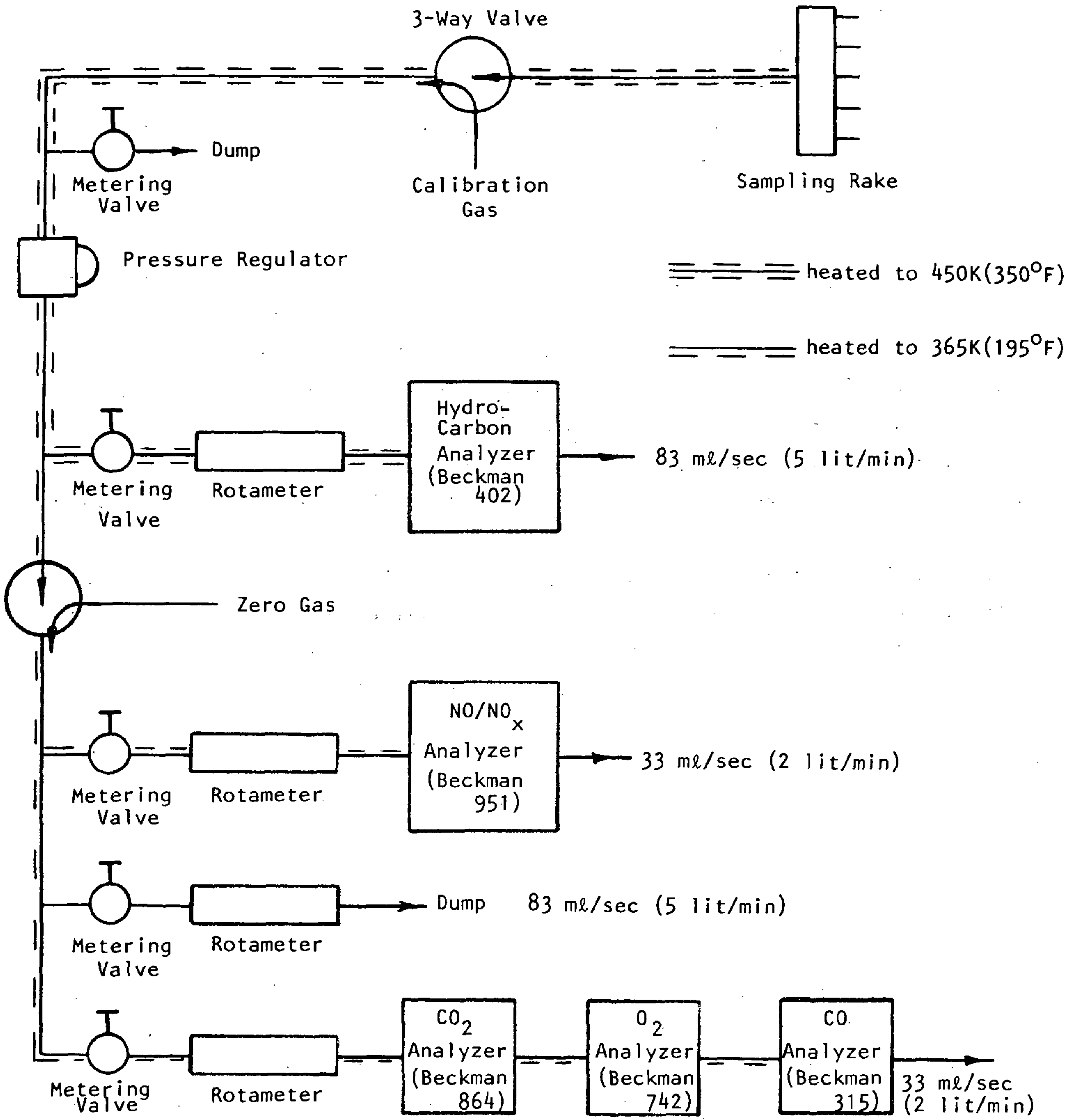


FIGURE 5. SAMPLING SYSTEM SCHEMATIC

As with any blow-down facility, useful test time was limited by the ability to store an adequate amount of high pressure air and to provide sufficient heat. In this particular case, heat storage capacity was the limiting quantity. Figure (6) shows measured temperature histories in the pebble bed and the mixer plenum. At the time that tunnel flow is initiated, the pebble bed is at its maximum temperature but the air delivery lines and combustion apparatus are at ambient temperature and produce the maximum cooling of the air entering the combustor.

As the test progresses, although the bed temperature drops, prior heating of the delivery line reduces subsequent losses and the temperature of the air at the mixer plenum actually rises. The considerable thermal inertia of the system results in a fairly flat peak and slow decline in the temperature curve. By allowing the apparatus to warm up for approximately 45 seconds, a useful test time of over 180 seconds is obtained with a temperature variation of $\pm$ 6K (. 10^{o}R).

Once the warmup period is completed, the gas igniter is lit and fuel injected. No purge is necessary prior to the ignition cycle because of the almost instantaneous ignition produced by the torch type igniter. The igniter is extinguished as soon as the increase in exhaust temperature at the exit orifice raises the combustor pressure to the desired value. Thermal response time of the thin wall liners was always less than 10 seconds and a period of 60 seconds was allowed between the time that gas sample data reached steady state and that at which data was recorded.

The program of experiments was divided into two basic phases. In the first, the preheat section was operated alone and its emission characteristics were determined. This experiment was then repeated while adding the main air to the small fraction (8%) of preheated air to dilute and quench the preheat stream. This series of tests established a base level for the NO_x contribution due to the conventional combustion in the preheat chamber. The second phase of testing, which constituted the bulk of the effort reported on here, consisted of operating the full combustion apparatus and measuring emission

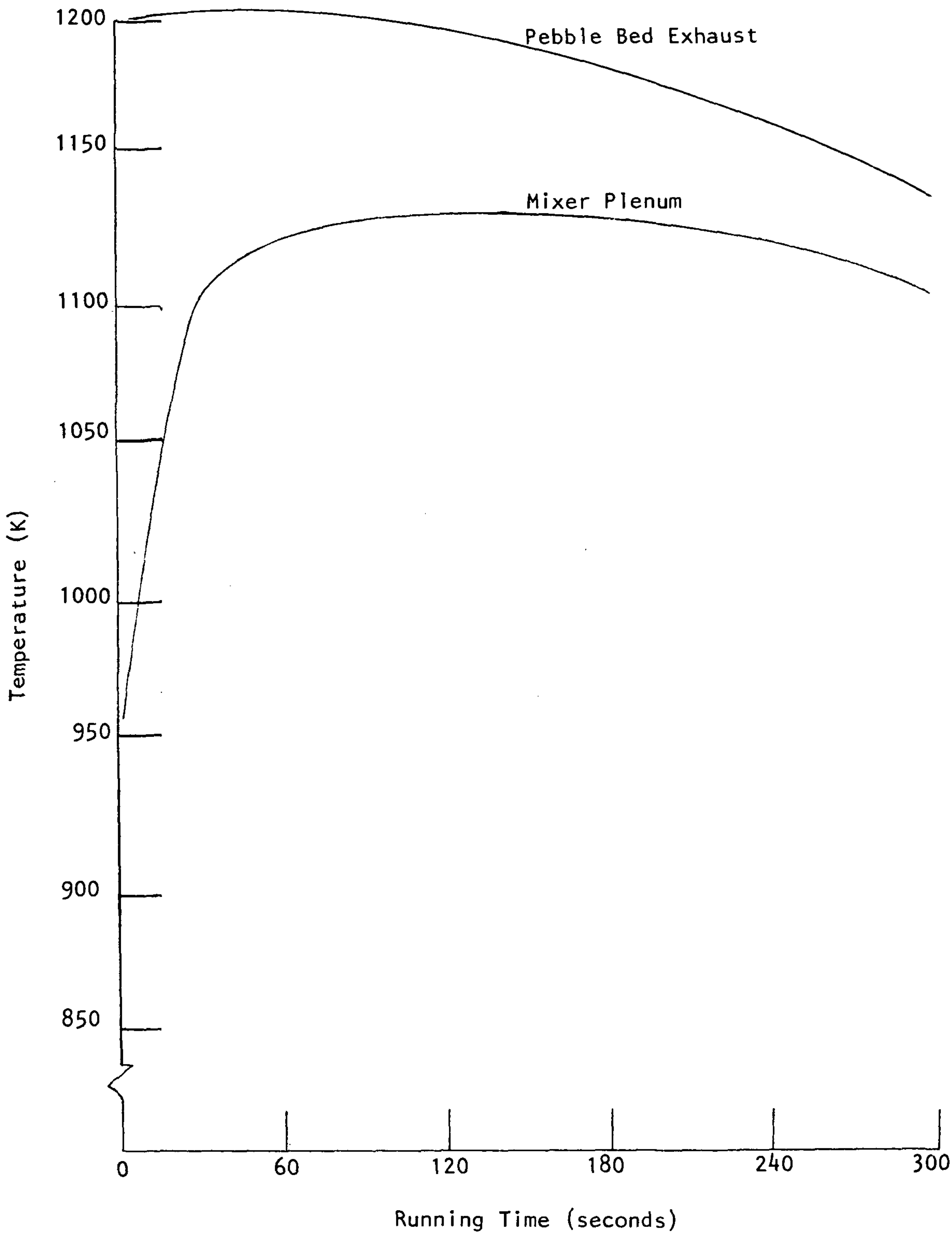


FIGURE 6. TEMPERATURE HISTORIES IN AIR DELIVERY SYSTEM

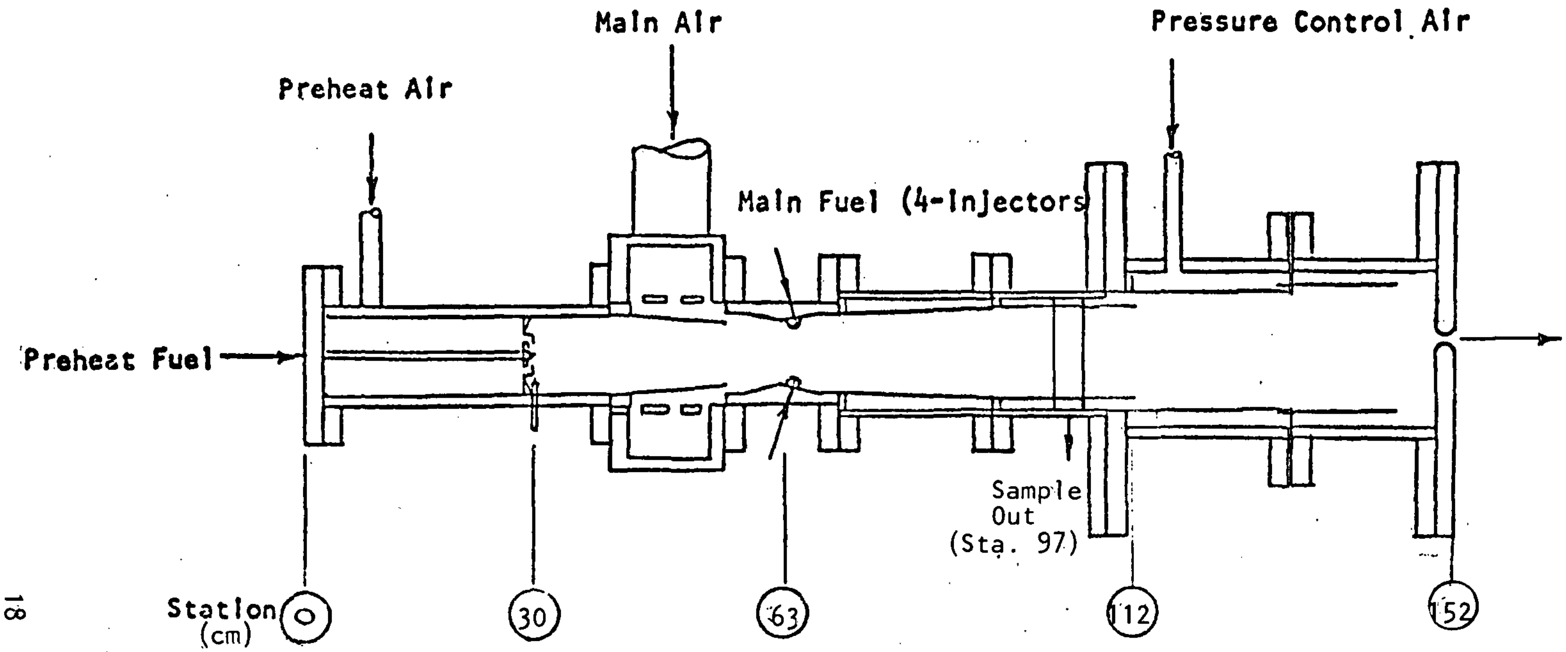
levels and combustion efficiency. Several configurations were tested, differing from one another in mixer diameter, flameholder geometry and combustor length. These configurations were tested both with and without preheat.

The physical arrangement of the apparatus for the preheat emission test phase is shown in Figure (7). It differs from that used for full combustion testing in three respects: the sampling rake normally located in the combustor section was not present, the flameholder normally located at the exit of the mixer tube was replaced by a 10 cm diameter cruciform rake equipped with thermocouples and sampling ports, and finally, the combustor section exit orifice was made smaller to maintain a pressure of 4×10^5 N/m^2 (4-atmospheres) even though no combustion would occur in the main flow.

Four configurations of the full combustion apparatus were tested. These configurations, designated A through D, differed in mixer diameter, flameholder geometry and combustor length. Figures (8) through (15) show the details of these configurations along with tabulations of their essential properties. Configuration A is characterized by an 11.4 cm (4.5-inch) mixer diameter and a 23 cm (9-inch) combustor length. The base of the conical flameholder is aligned with the sudden expansion step and produces a velocity of 10.3 m/sec (340 ft/sec) at the combustor entrance station as compared with the 83 m/sec (273 ft/sec) velocity which prevails in the mixer. The details of the flameholder are shown in Figure (9).

Configuration B, illustrated in Figures (10) and (11), differed from A in that an annular constriction was added to the flameholder nearly doubling the combustor entrance velocity to improve flashback characteristics. This configuration was tested with two combustor lengths, 23 cm (9-inches) as in Configuration A, an 46 cm (18-inches).

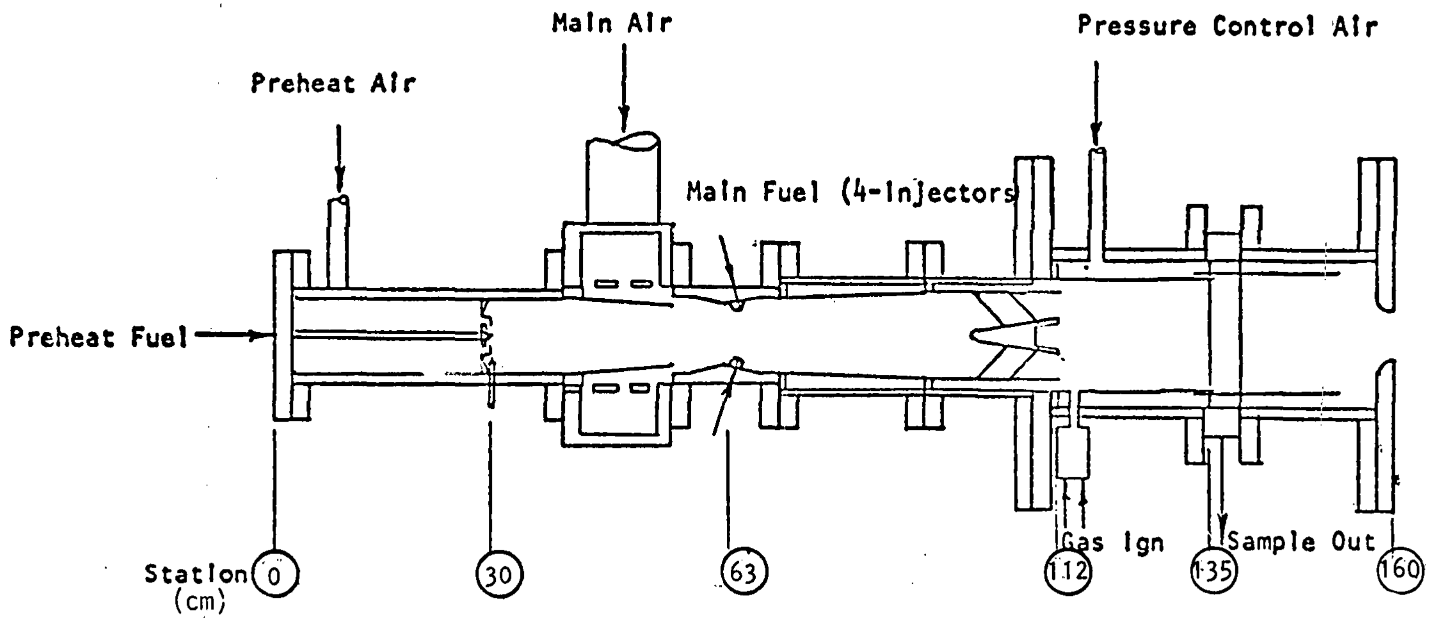
In configuration C, the flameholder geometry was altered by moving the area constriction ahead of the centerbody and using the remainder of the flameholder section as a diffuser to reduce combustor entrance velocity. In addition, the centerbody was displaced downstream by 4.45 cm (1.75-inches) to



Function	Preheat Plenum	Preheat Burner	Fuel Injector	Mixer
Air Flow Rate *	0.11 kg/sec (0.24 lb/sec)	0.11 (0.24)	1.36 (3.0)	1.36 (3.0)
Cross Section	81 cm^2 (12.6 in^2)	81 (12.6)	46 (7.1)	103 (15.9)
Temperature (Typical)	833K (1500°R)	2220 (4000)	945 (1700)	890 (1600)
Velocity * (Typical)	7.9 m/sec (26 ft/sec)	21.3 (70)	198 (651)	83 (273)

*For tests with preheat plus dilution air.

FIGURE 7. APPARATUS CONFIGURATION IN PREHEAT EMISSION TESTING



Function	Preheat Plenum	Preheat Burner	Fuel Injector	Mixer	Flame-holder	Combustion Chamber	Pressure Control
Air Flow Rate	0.11 kg/sec 0.24 lb/sec	0.11 (0.24)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	2.5 (TYPICAL) (5.5)
Cross Section	81 cm^2 (12.6 in^2)	81 (12.6)	46 (7.1)	103 (15.9)	83 (12.8)	183 (28.3)	54 (8.3)
Temperature (Typical)	833K (1500°R)	2220 (4000)	945 (1700)	890 (1600)	890 (1600)	2220 (4000)	1330 (2400)
Velocity (Typical)	7.9 m/sec (26 ft/sec)	21.3 (70)	198 (651)	83 (273)	103 (340)	117 (385)	700 (2300)

FIGURE 8. COMBUSTION APPARATUS CONFIGURATION A

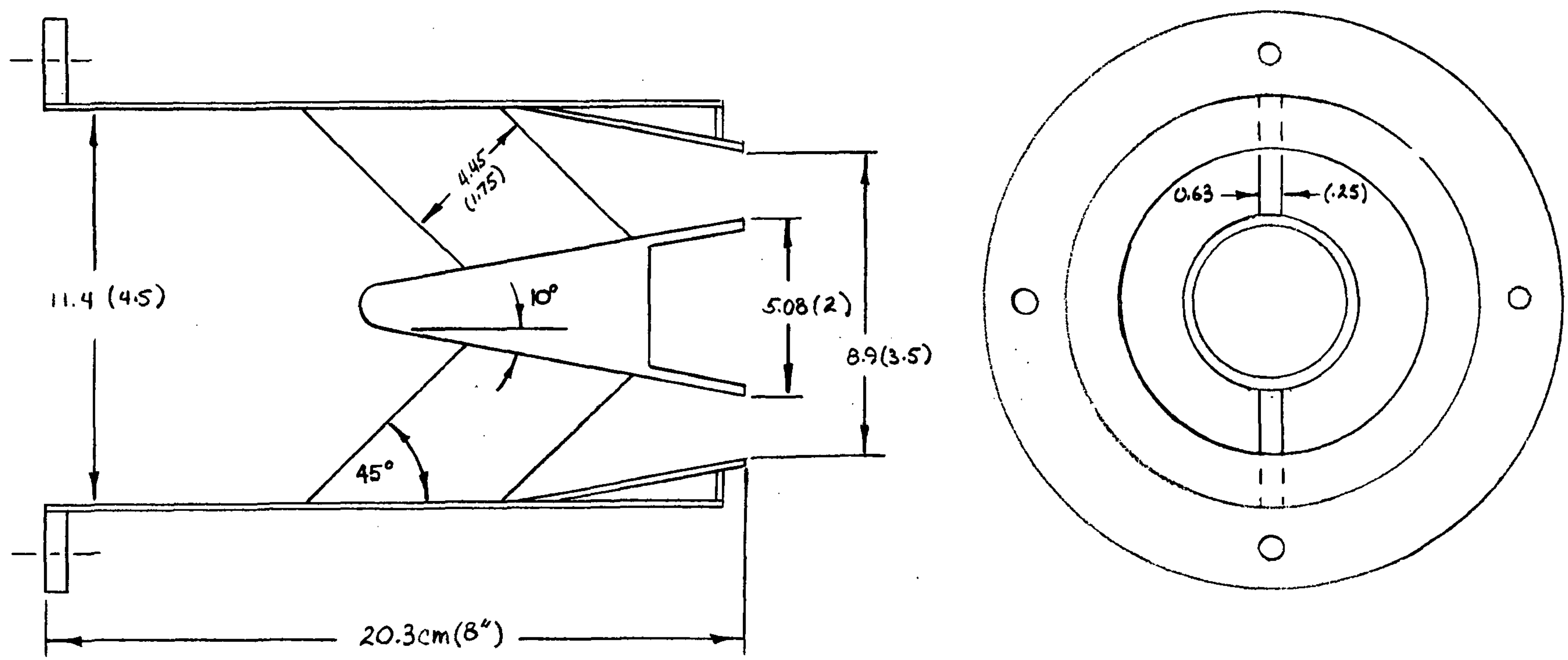
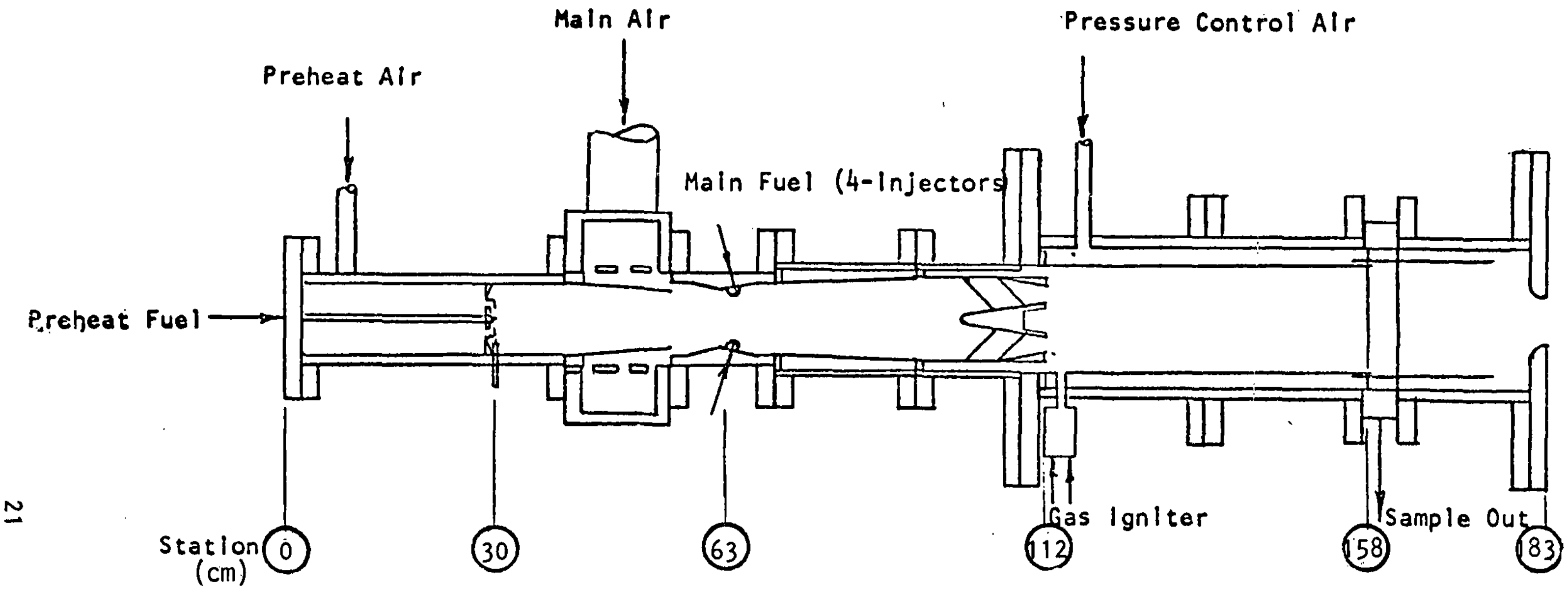


FIGURE 11. FLAMEHOLDER DETAIL - CONFIGURATION B



Air Function	Preheat Plenum	Preheat Burner	Fuel Injector	Mixer	Flame-holder	Combustion Chamber	Pressure Control
Flow Rate	0.11 kg/sec (0.24 lb/sec)	0.11 (0.24)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	2.5 (TYPICAL) (5.5)
Cross Section	81 cm^2 (12.6 in^2)	81 (12.6)	46 (7.1)	103 (15.9)	42 (6.5)	183 (28.3)	54 (8.3)
Temperature (Typical)	833K (1500°R)	2220 (4000)	945 (1700)	890 (1600)	890 (1600)	2220 (4000)	1330 (2400)
Velocity (Typical)	7.9 m/sec (26 ft/sec)	21.3 (70)	198 (651)	83 (273)	204 (670)	117 (385)	700 (2300)

FIGURE 10. COMBUSTION APPARATUS - CONFIGURATION B

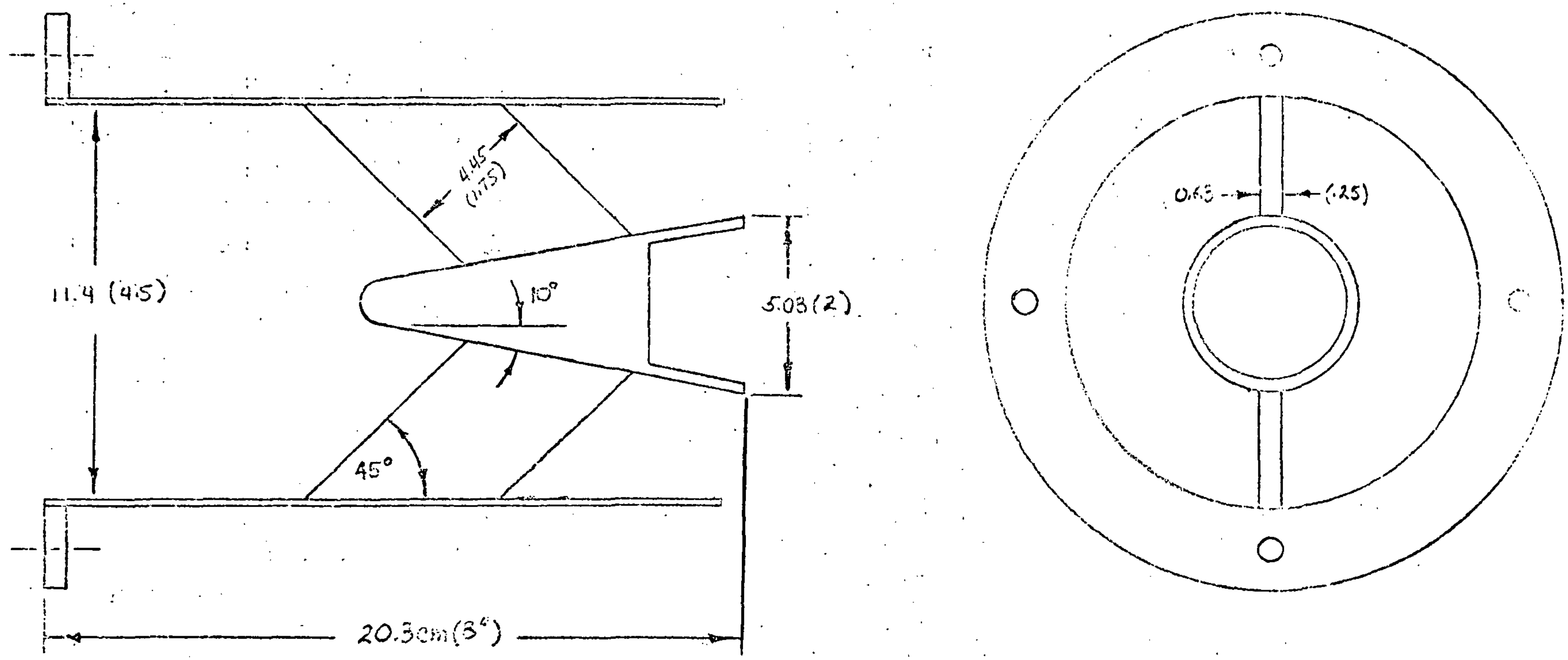
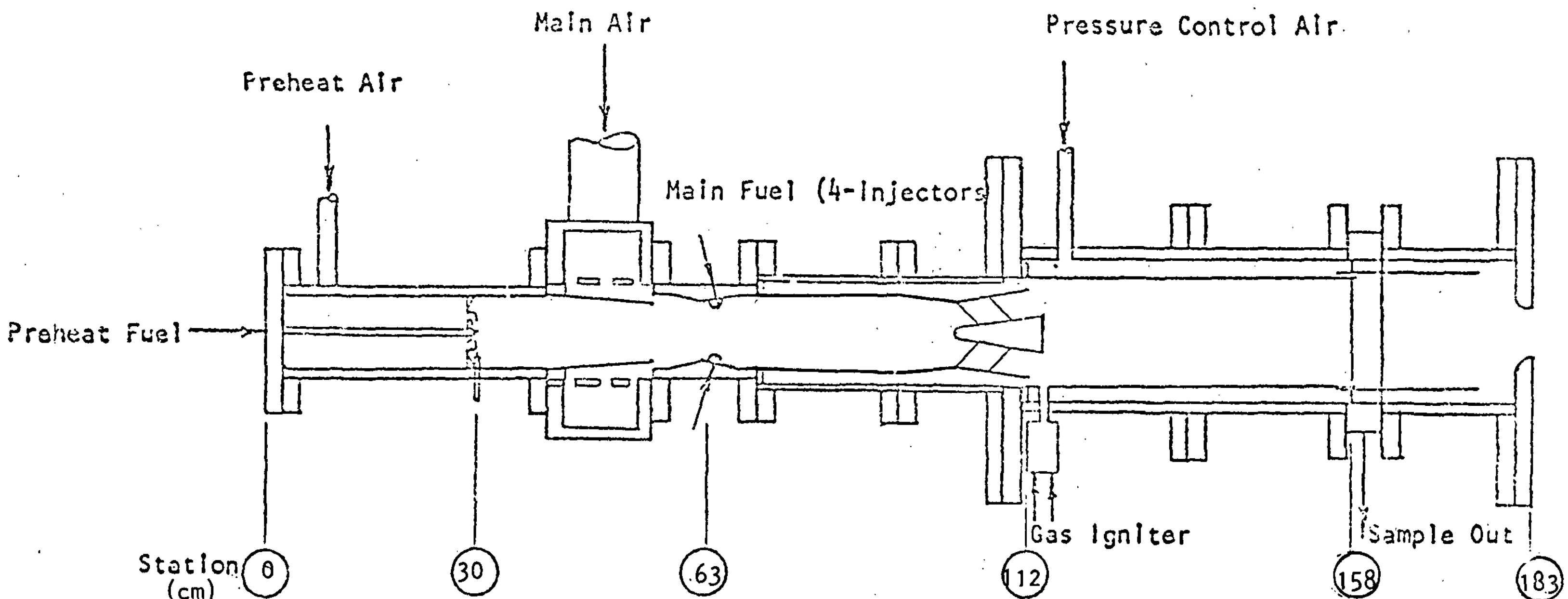


FIGURE 9. FLAMEHOLDER DETAIL - CONFIGURATION A

further reduce velocity and improve combustion efficiency and pressure drop characteristics of the system. The details of configuration C are shown in Figures (12) and (13).

For the last configuration the mixer tube diameter was reduced from 11.4 to 7.6 cm (3-inches) to provide a constant area duct from the fuel injection station to the flameholder constriction. This mixer produced the shortest gas residence time of any configuration for the most favorable auto-ignition characteristics. The essential details of configuration D are presented in Figures (14) and (15).



Function	Preheat Plenum	Preheat Burner	Fuel Injector	Mixer	Flame-holder	Combustion Chamber	Pressure Control
Air Flow Rate	0.11 kg/sec 0.24 lb/sec	0.11 (0.24)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	2.5 (TYPICAL) (5.5)
Cross Section	81 cm^2 (12.6 In^2)	81 (12.6)	46 (7.1)	103 (15.9)	88 (13.67)	182 (28.3)	54 (8.3)
Temperature (Typical)	833K (1500°R)	2220 (4000)	945 (1700)	890 (1600)	890 (1600)	2220 (4000)	1330 (2400)
Velocity (Typical)	7.9 m/sec (26 ft/sec)	21.3 (70)	198 (651)	83 (273)	97 (318)	117 (385)	700 (2300)

FIGURE 12. COMBUSTION APPARATUS - CONFIGURATION C

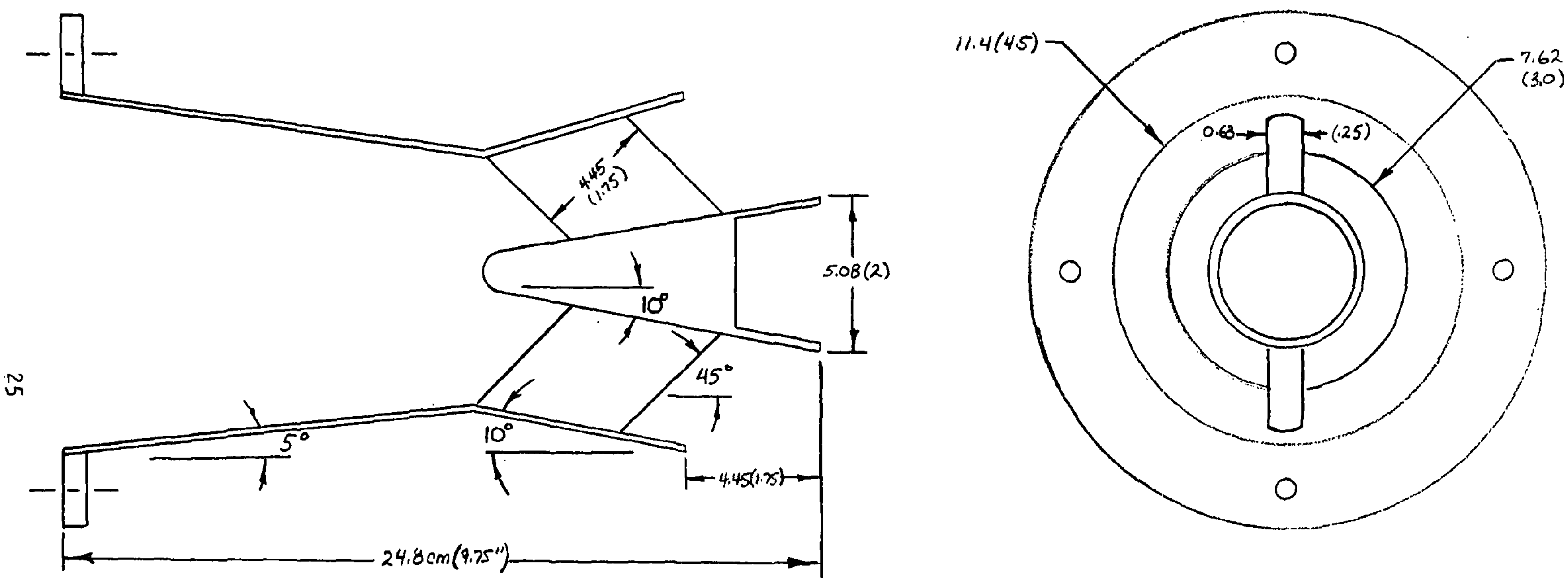
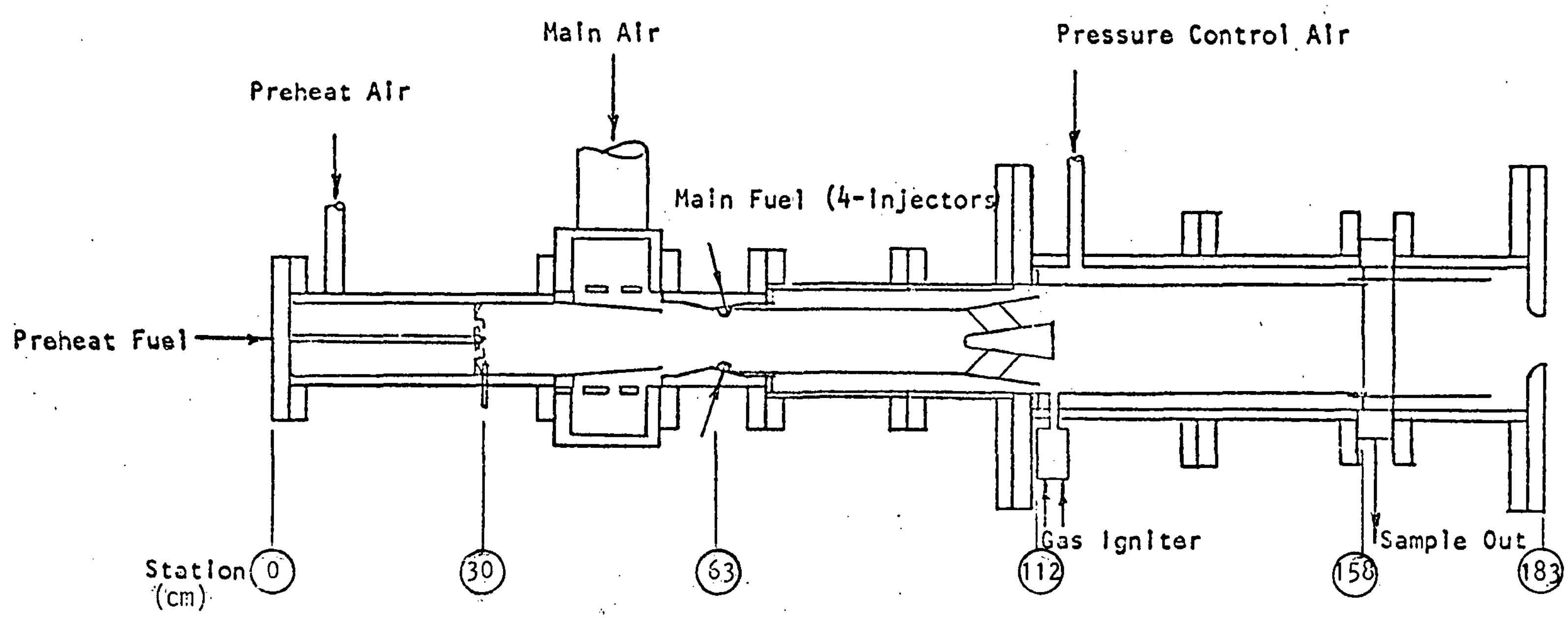


FIGURE 13. FLAMEHOLDER DETAIL - CONFIGURATION C



Function	Preheat Plenum	Preheat Burner	Fuel Injector	Mixer	Flame-holder	Combustion Chamber	Pressure Control
Air Flow Rate	0.11 kg/sec (0.24 lb/sec)	0.11 (0.24)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	1.36 (3.0)	2.5 (TYPICAL) (5.5)
Cross Section	81 cm^2 (12.6 in^2)	81 (12.6)	46 (7.1)	45 (7.07)	88 (13.67)	182 (28.3)	54 (8.3)
Temperature (Typical)	833K (1500°R)	2220 (4000)	945 (1700)	890 (1600)	890 (1600)	2220 (4000)	1330 (2400)
Velocity (Typical)	7.9 m/sec (26 ft/sec)	21.3 (70)	198 (651)	187 (614)	97 (318)	117 (385)	700 (2300)

FIGURE 14. COMBUSTION APPARATUS - CONFIGURATION D

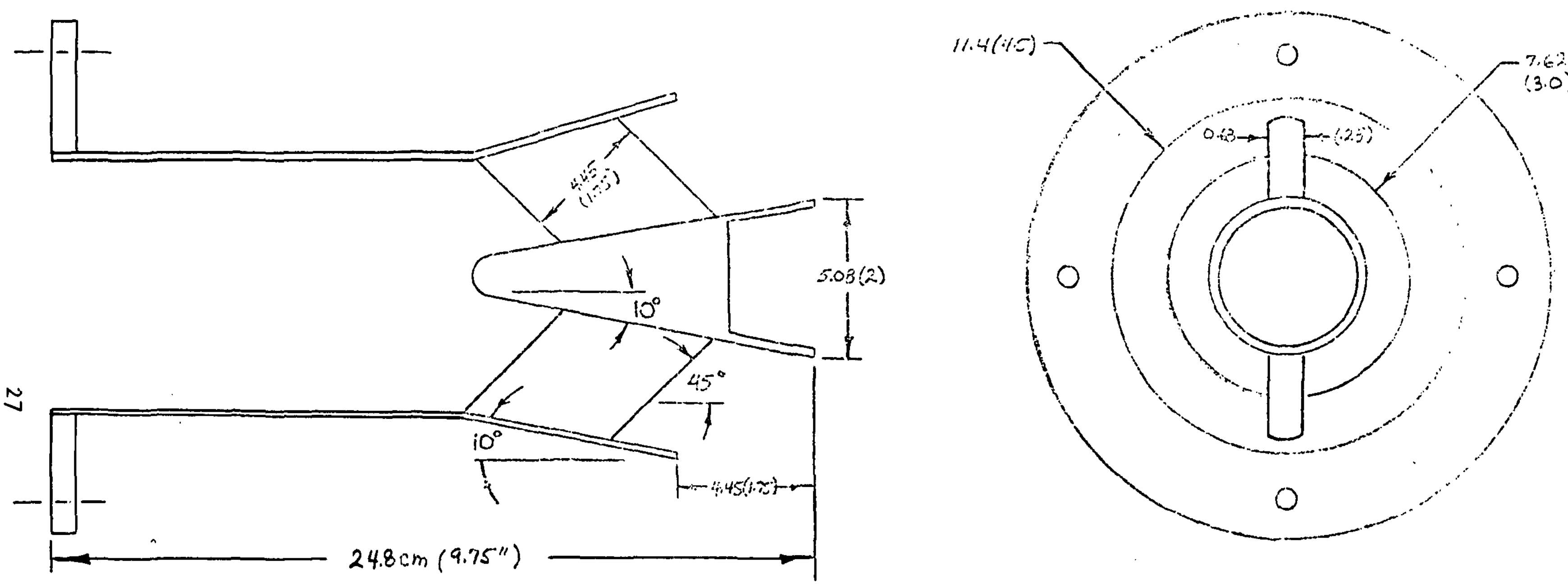


FIGURE 15. FLAMEHOLDER DETAIL - CONFIGURATION D

SECTION IV

TEST RESULTS

Tests were made to determine the emission levels and combustion inefficiency of four configurations of a prevaporizing, premixing combustor. The apparatus was equipped with a preheat device which could be used to boost the mixer entrance temperature by partial vitiation of the inlet airstream. The emission levels of the preheat combustor were also measured along with the resulting combustor entrance temperature profiles. Figure (16) shows the measured temperature distributions at the flameholder station. The circular symbols, denoting the "air only" condition, show a fairly uniform temperature distribution with a slight scatter around the nominal entrance temperature of 833K (1500°R). The use of the preheat section to vitiate the inlet airstream produces a slightly asymmetric temperature profile (triangular symbols) with the upper portion of the duct approximately 65K (120°R) hotter than the lower. The addition of fuel to the vitiated airstream (diamond shaped symbols) lowers the temperature by absorbing heat for heating and vaporizing the fuel and also produces a slightly asymmetric effect, implying that less than complete mixing has occurred by the flameholder station.

The emission characteristics of the vitiated airstream are shown in Figures (17) through (20). For the operational range of preheat equivalence ratio ($0.4 < \phi_p < 0.7$) the NO_x emission index is nearly constant at just over 10 g-NO_2/kg of preheat fuel. Since the fuel injected into the preheat section represents 8% of the total, the NO_x level entering the combustor for tests with vitiation varies from approximately 1 to 2 g-NO2/kg of total fuel as the overall equivalence ratio varies between 0.6 and 0.3. The combustion inefficiency, unburned hydrocarbon and carbon monoxide emission indices for the diluted preheat gas stream are presented in Figures (18) through (20). These measurements clearly indicate the quenching effect of the rapid addition of primary air which results in a high level of CO and unburned hydrocarbons entering the combustion chamber. However, since the downstream combustion zone will affect these species, high concentrations in the mixer tube do not pose a great problem.

One of the most sensitive problems associated with the design of a premixing-prevaporizing burner is the avoidance of mixer tube combustion. This problem may result from either flashback or autoignition. The term flashback refers to the condition of flame propagation upstream from the combustor into the mixer tube. This may result from a mixer tube velocity which is less than the flame propagation speed, from feedback through the boundary layer or from a transient related phenomenon such as combustion generated pressure pulses.

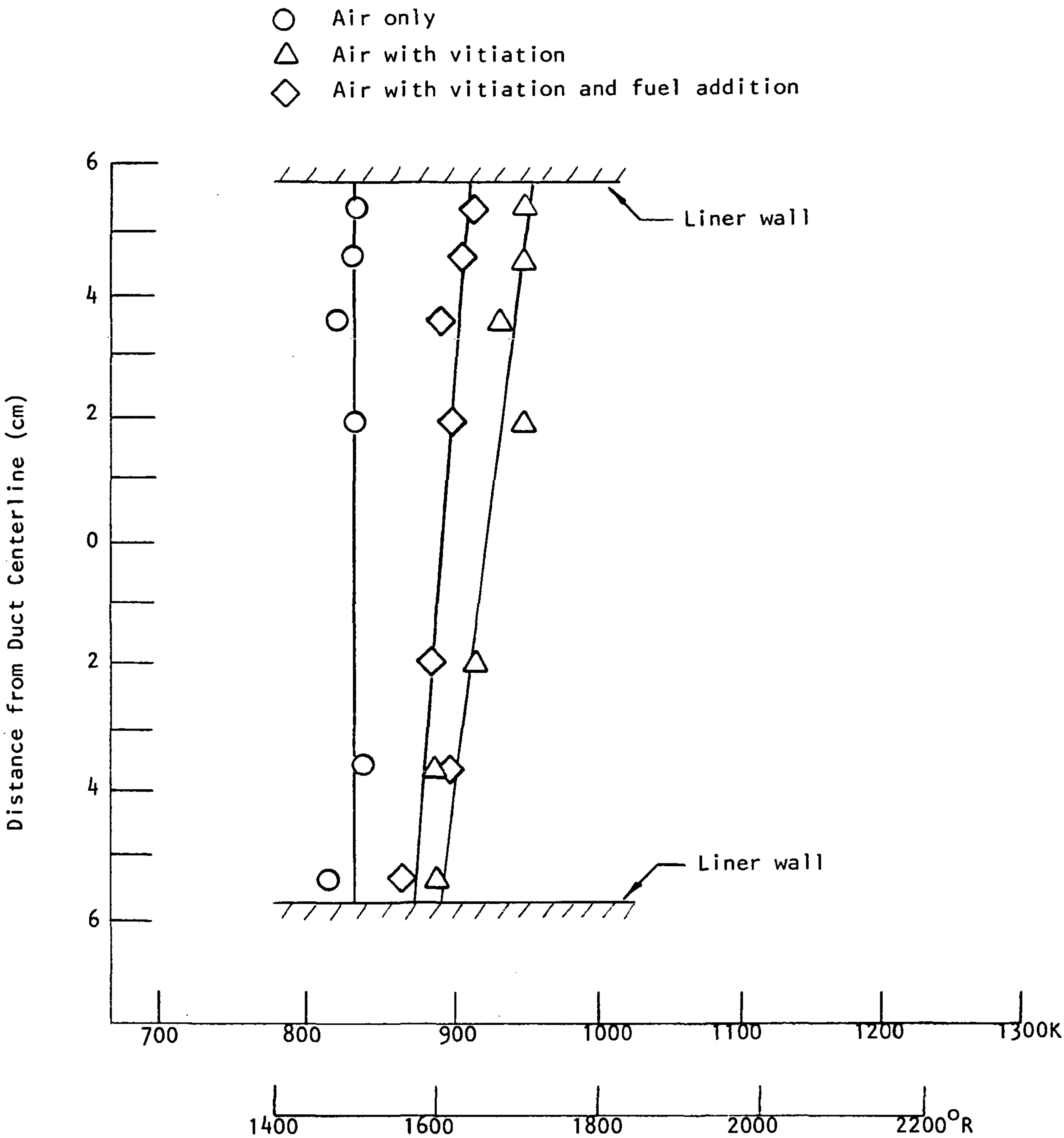


FIGURE 16. TEMPERATURE PROFILES AT FLAMEHOLDER STATION (FULL OPERATION, $\phi_{PREHEAT} = 0.5$)

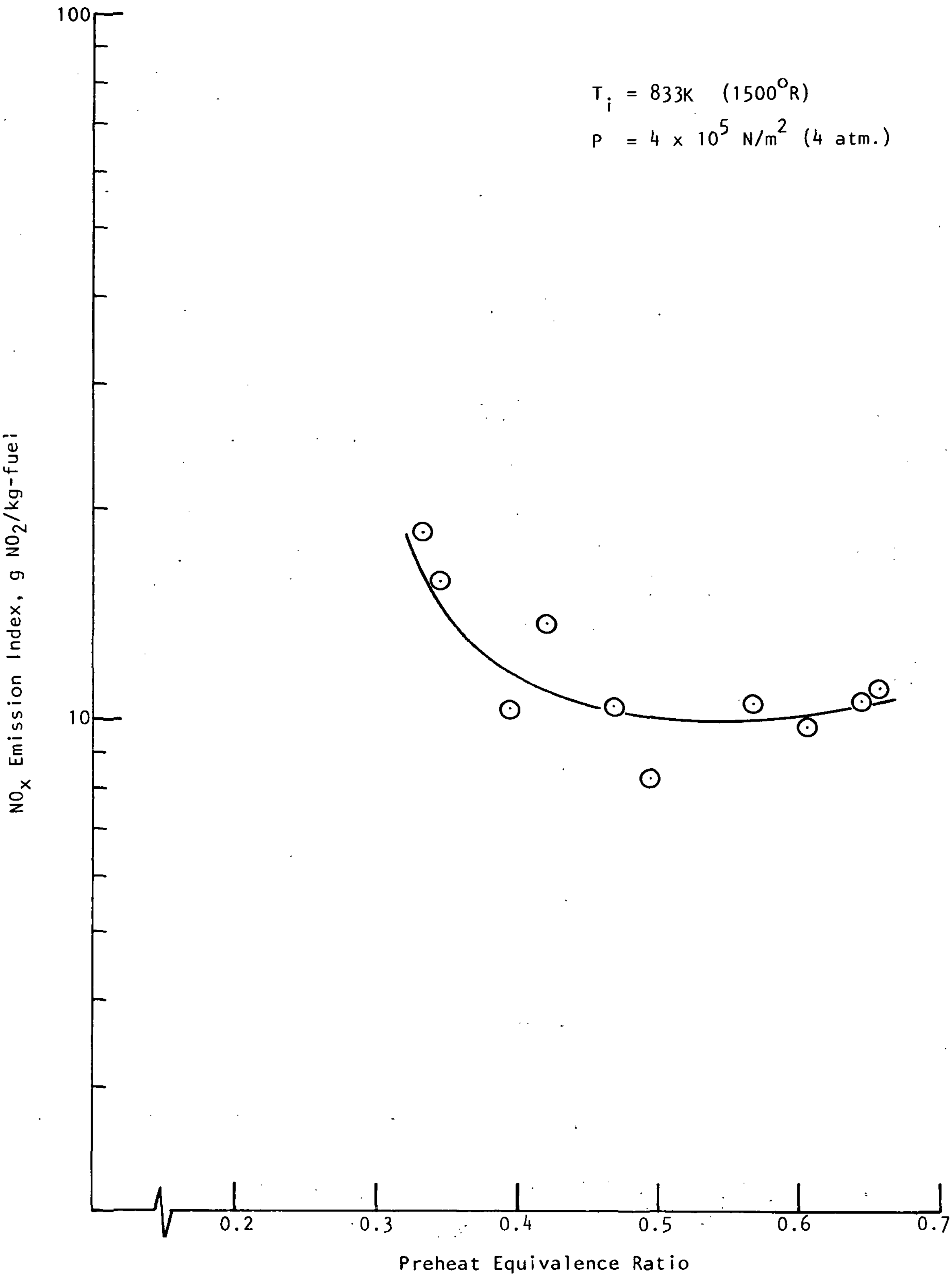


FIGURE 17. NO_x EMISSION INDEX FOR PREHEAT EXHAUST QUENCHED BY MAIN AIR

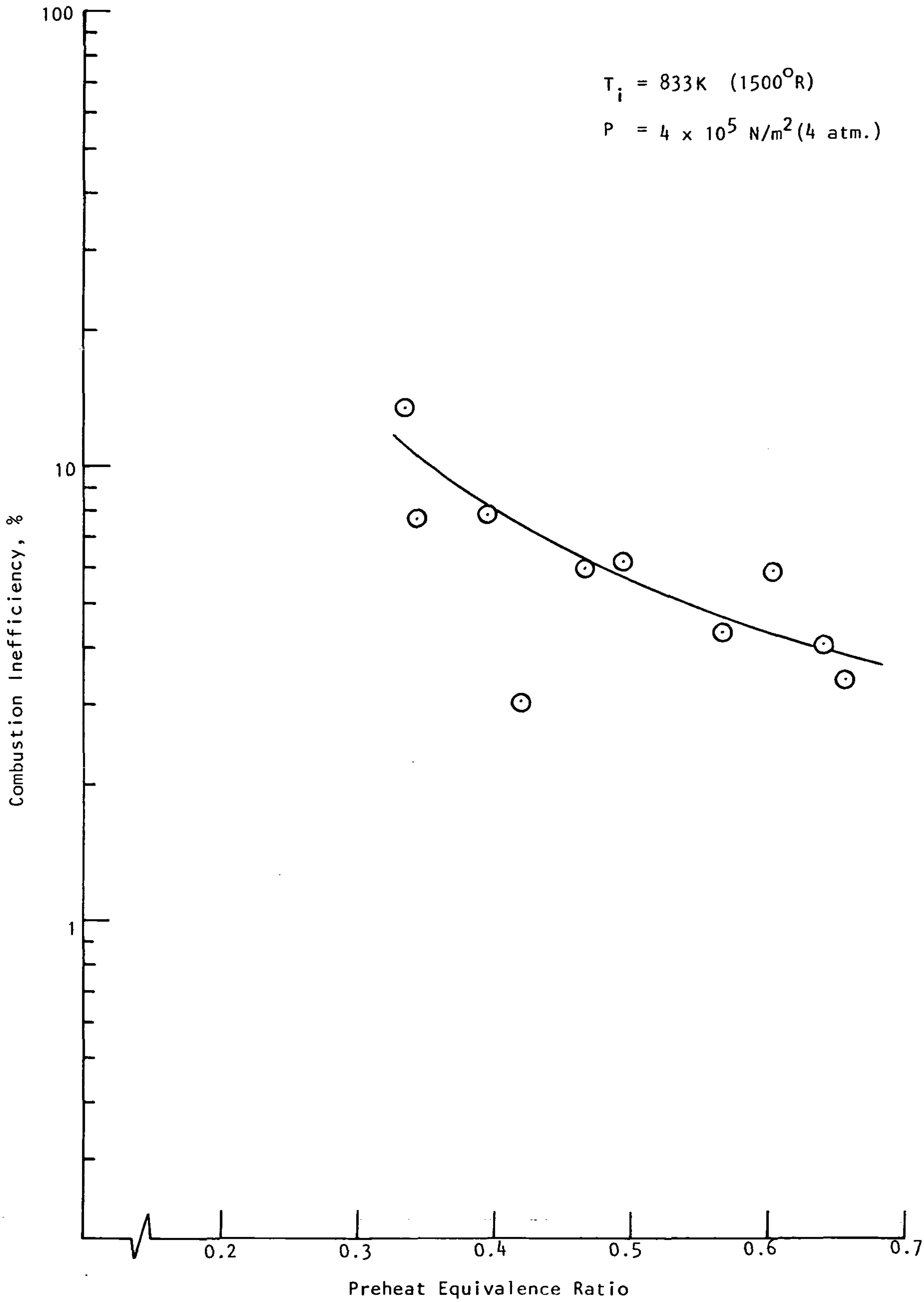


FIGURE 18. COMBUSTION INEFFICIENCY FOR PREHEAT EXHAUST QUENCHED BY MAIN AIR

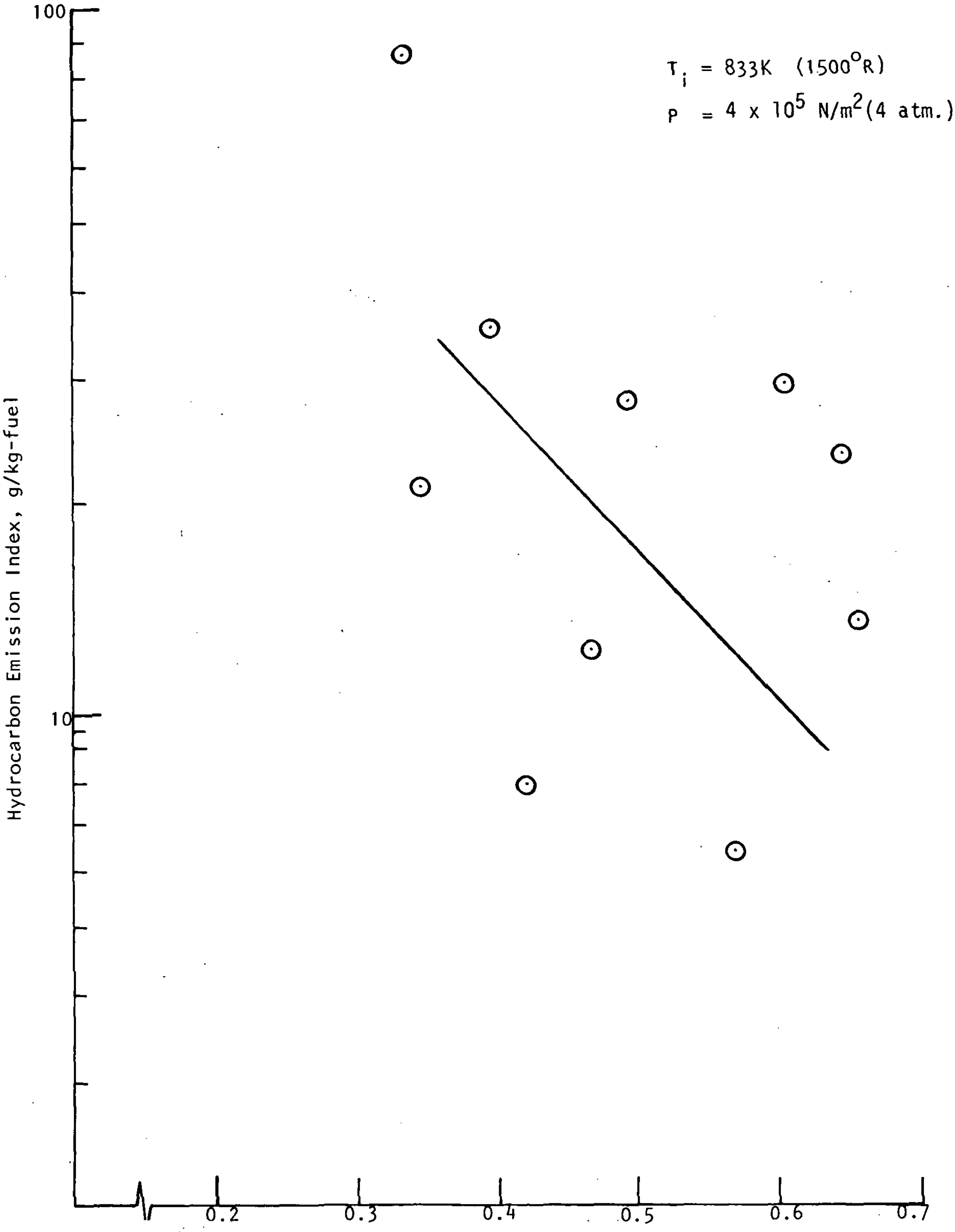


FIGURE 19. HYDROCARBON EMISSION INDEX FOR PREHEAT EXHAUST QUENCHED BY MAIN AIR

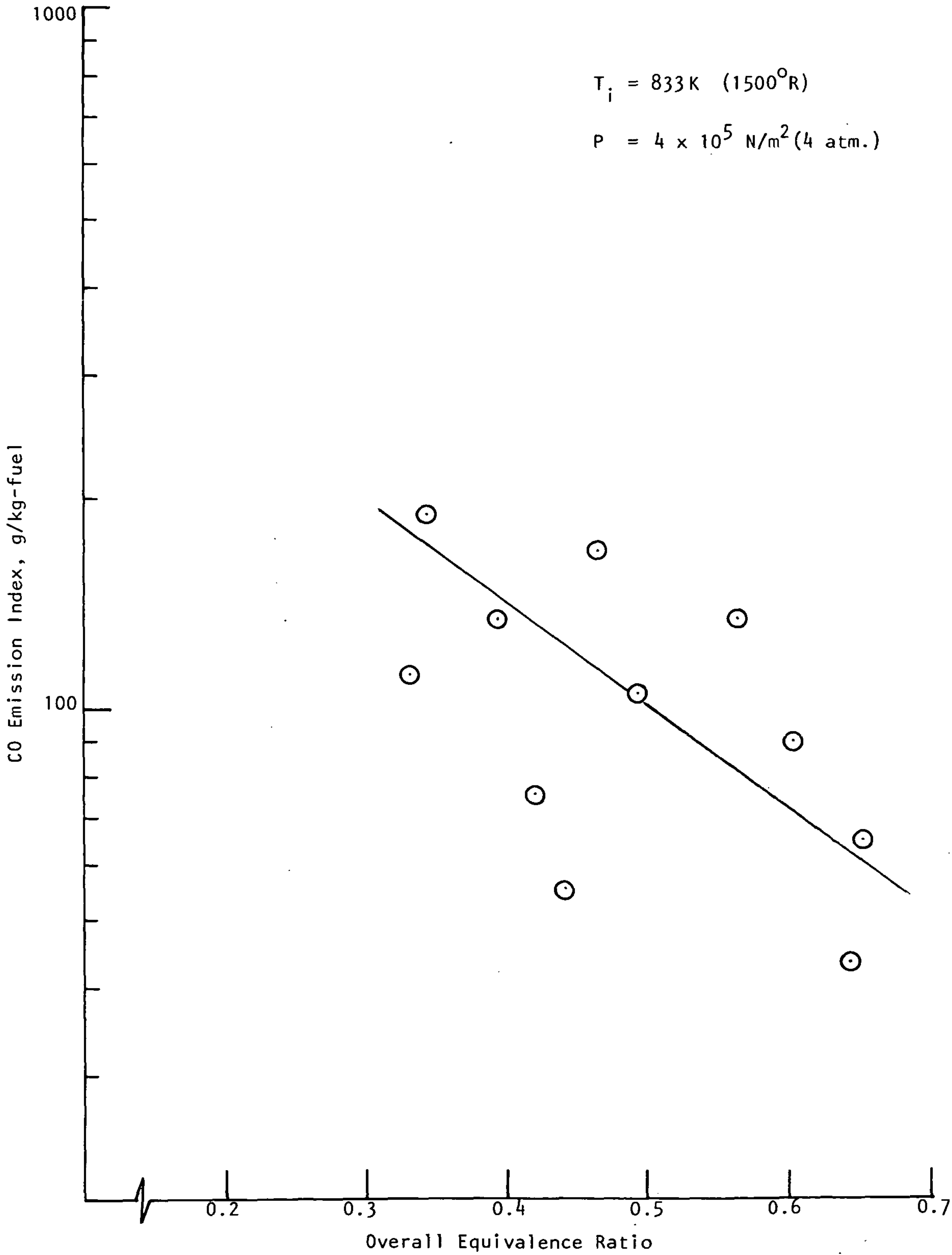


FIGURE 20. CARBON MONOXIDE EMISSION INDEX FOR PREHEAT EXHAUST QUENCHED BY MAIN AIR

The autoignition problem occurs when the residence time of the fuel-air mixture exceeds the ignition delay time for the prevailing temperature, pressure and equivalence ratio.

Configuration A displayed the undesirable tendency to produce combustion in the mixer tube. When tests were run without the use of the igniter or reheater, ignition in the mixer did not occur. However, when the combustion zone gas was ignited, burning in the mixer tube occurred after a short delay indicating a flashback problem. Moreover, when the preheater was used, mixer section combustion occurred even when the combustor section igniter was inoperative, indicating autoignition as well.

Since vitiation was accomplished by heating 8% of the inlet air to a temperature of approximately 2100K (3800°R) and mixing it with the remaining air a short distance upstream of the fuel injectors, it is possible that incomplete mixing produced a local region of high temperature in the center of the inlet airstream. Since the pressure atomizing nozzles produced greater lateral penetration of the fuel as the fuel flow rate (nozzle pressure drop) was increased, it appears that ignition occurred whenever fuel reached the central region of the duct before the hot products of vitiation had diffused sufficiently to reduce the local temperature below the autoignition limit.

The measured emission characteristics of Configuration A (Figure 8) are presented in Figures (21) through (24). The NO_x measurements shown in Figure (21) display considerable scatter but appear to cluster around the calculation of Reference 2. Although the tendency of this design to produce combustion in the mixer section made it impossible to gather enough data to completely define its characteristics, it appears that Configuration A produced a NO_x level on the order of 1 g-NO_2/kg-fuel at an equivalence ratio of 0.6. The combustion inefficiency data presented in Figure (22) indicate that combustion was only marginally stable for this configuration with large variations in combustion inefficiency occurring for small changes in equivalence ratio. Nevertheless, the data indicates a combustion inefficiency on the order of 1% for the unity NO_x condition. The unburned hydrocarbon level, shown in Figure (23) is seen to

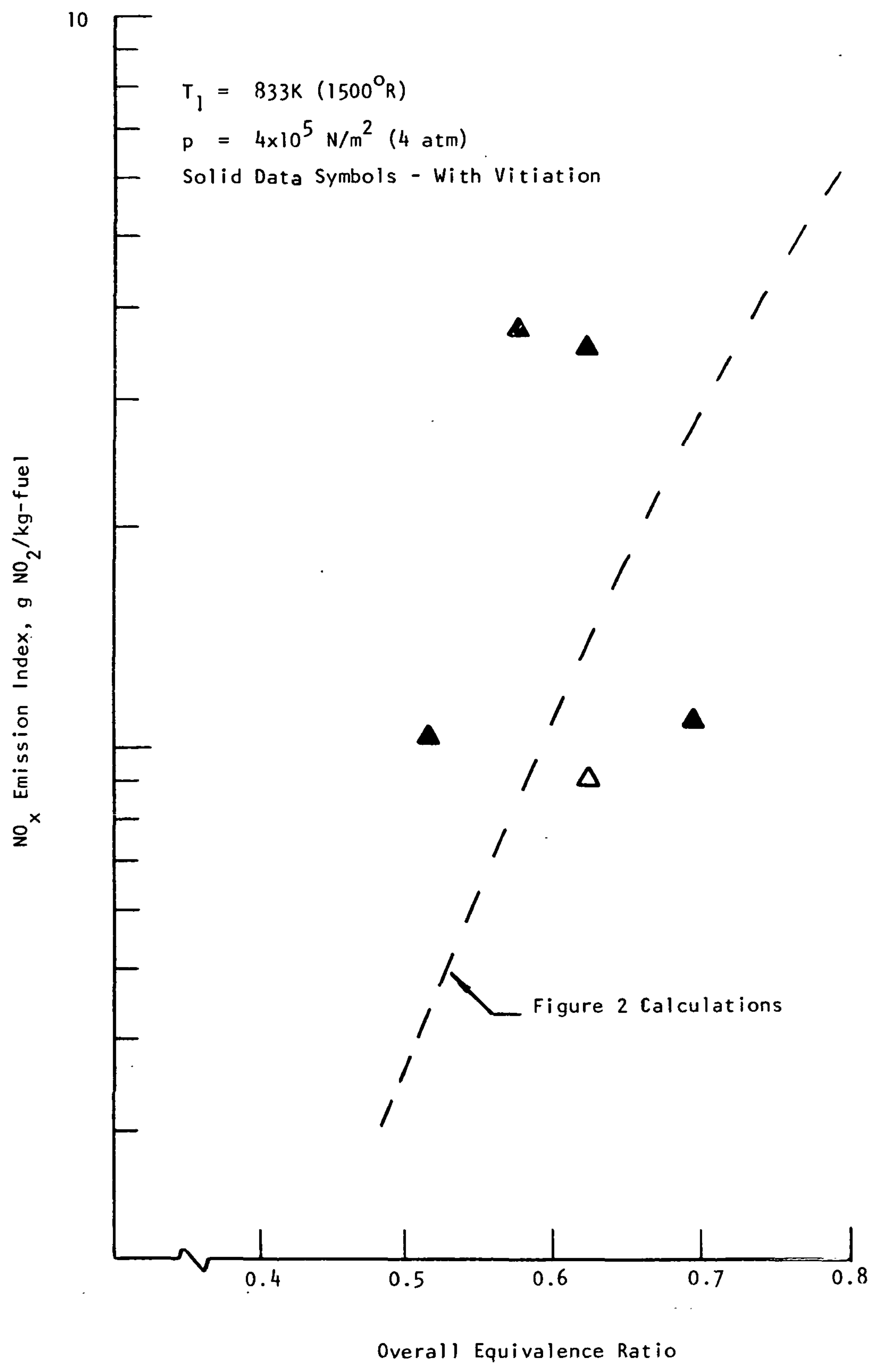


FIGURE 21. NO_x EMISSION INDEX FOR CONFIGURATION A

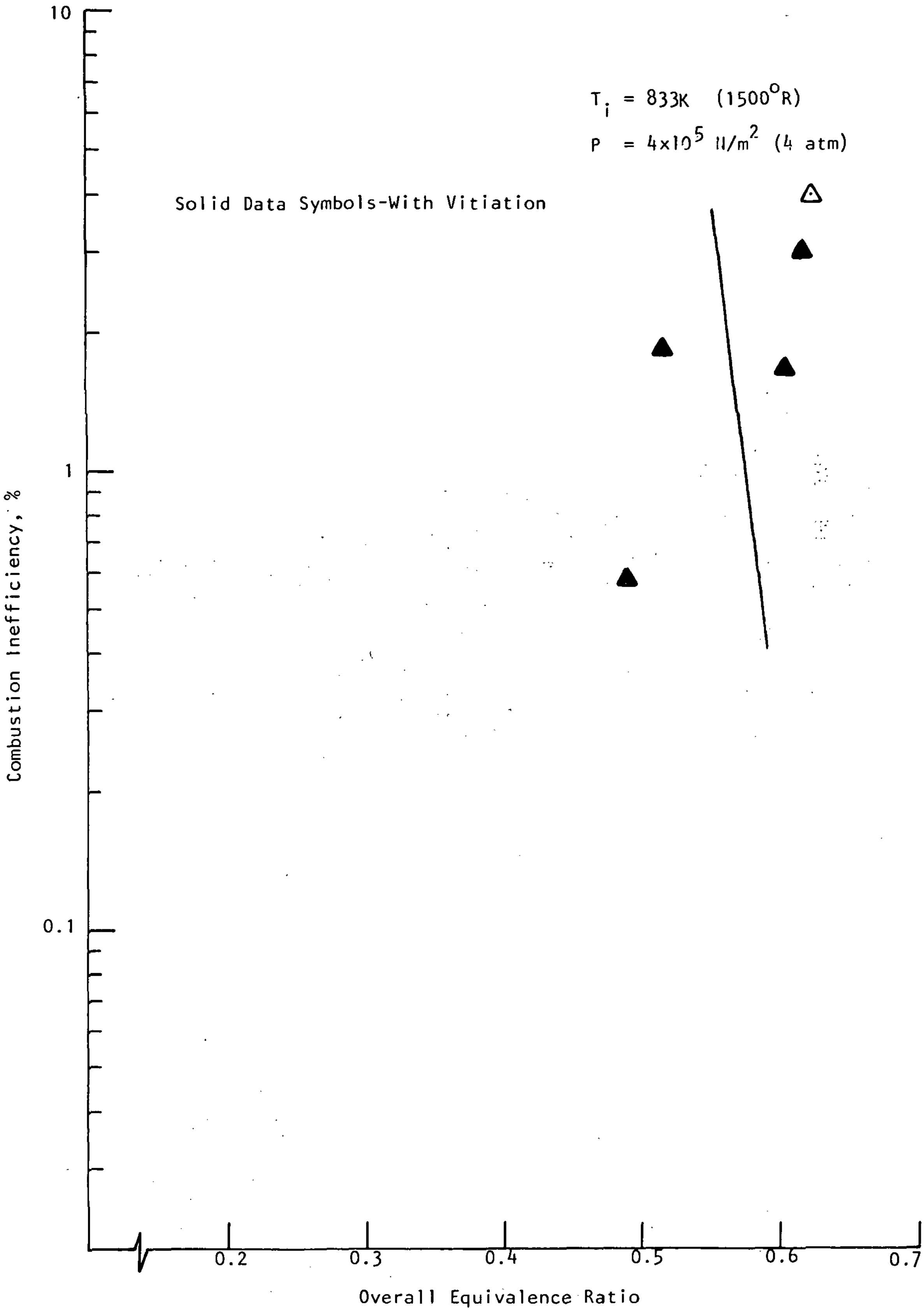


FIGURE 22. COMBUSTION INEFFICIENCY FOR CONFIGURATION A

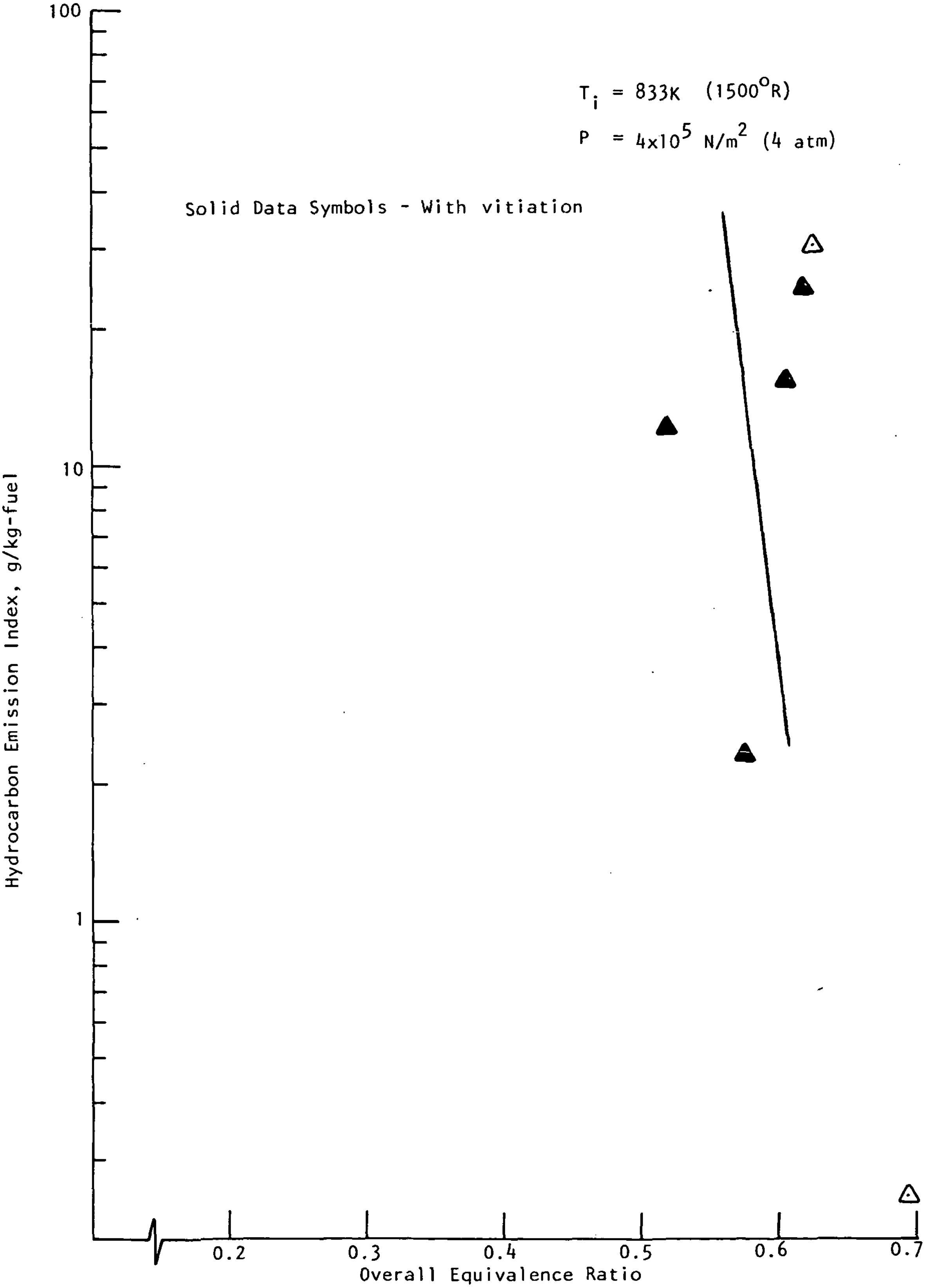


FIGURE 23. HYDROCARBON EMISSION INDEX FOR CONFIGURATION A

vary as the combustion inefficiency while the CO level, shown in Figure (24), is close to equilibrium.

Combustion inefficiency is a measure of the degree of heat release which could be obtained by completing all reactions bringing the combustion products to chemical equilibrium. At the initial temperature of 833K (1500°R) which characterized these experiments, the oxidation reactions will all eventually attain equilibrium since there is no quenching of the primary combustion zone gas with dilution air. Therefore, the combustion inefficiency measurement is an indication of how closely the reactions have approached equilibrium within the given combustor residence time.

The Configuration A data points with high combustion inefficiency display low CO levels combined with high unburned hydrocarbon concentrations and indicate that a significant portion of the flow at the sampling rake station has not started to react. This implies poor ignition and flame propagation characteristics at the flameholder. It would appear that combustion did not spread from the ignition point to all ofthe flameholding zones and that the ignition and start-up procedure used in each test affected this spread.

Configuration B (Figure 10) utilized the same mixer tube as A and therefore had the same residence time characteristics. However, the constriction added at the flameholder exit station increased the velocity from its previous value of 103 m/sec (340 ft/sec) to 204 m/sec (670 ft/sec) and substantially altered its flashback characteristics. It was thus possible to operate Configuration B without vitiation and not encounter combustion in the mixer tube. However, attempts to run with vitiation still produced mixer tube ignition for the higher equivalence ratios.

The emission characteristics of Configuration B are shown in Figures (25) through (28). Figure (25) indicates that the NO_x emission goal of unity was achieved, as before, at an equivalence ratio of approximately 0.5. However, the combustion inefficiency for this configuration (Figure 26) was approximately 2% for the unity NO_x condition and did not drop below 1% until equivalence ratio reached 0.53 at which point the NO_x level was 1.4 g-NO_2/kg-fuel. Hydrocarbon emission is shown in Figure (27) and displays the same

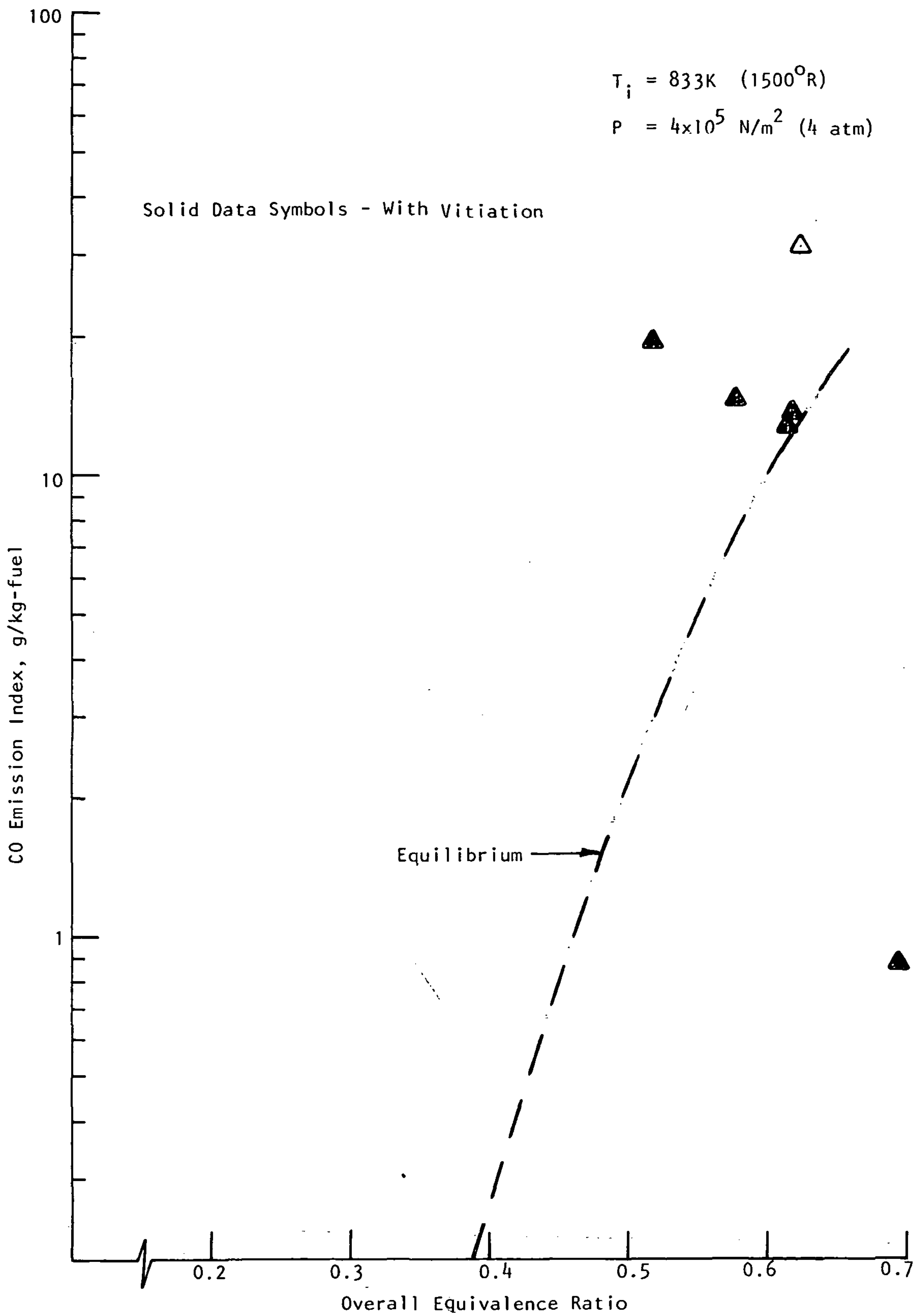


FIGURE 24. CO EMISSION INDEX FOR CONFIGURATION A

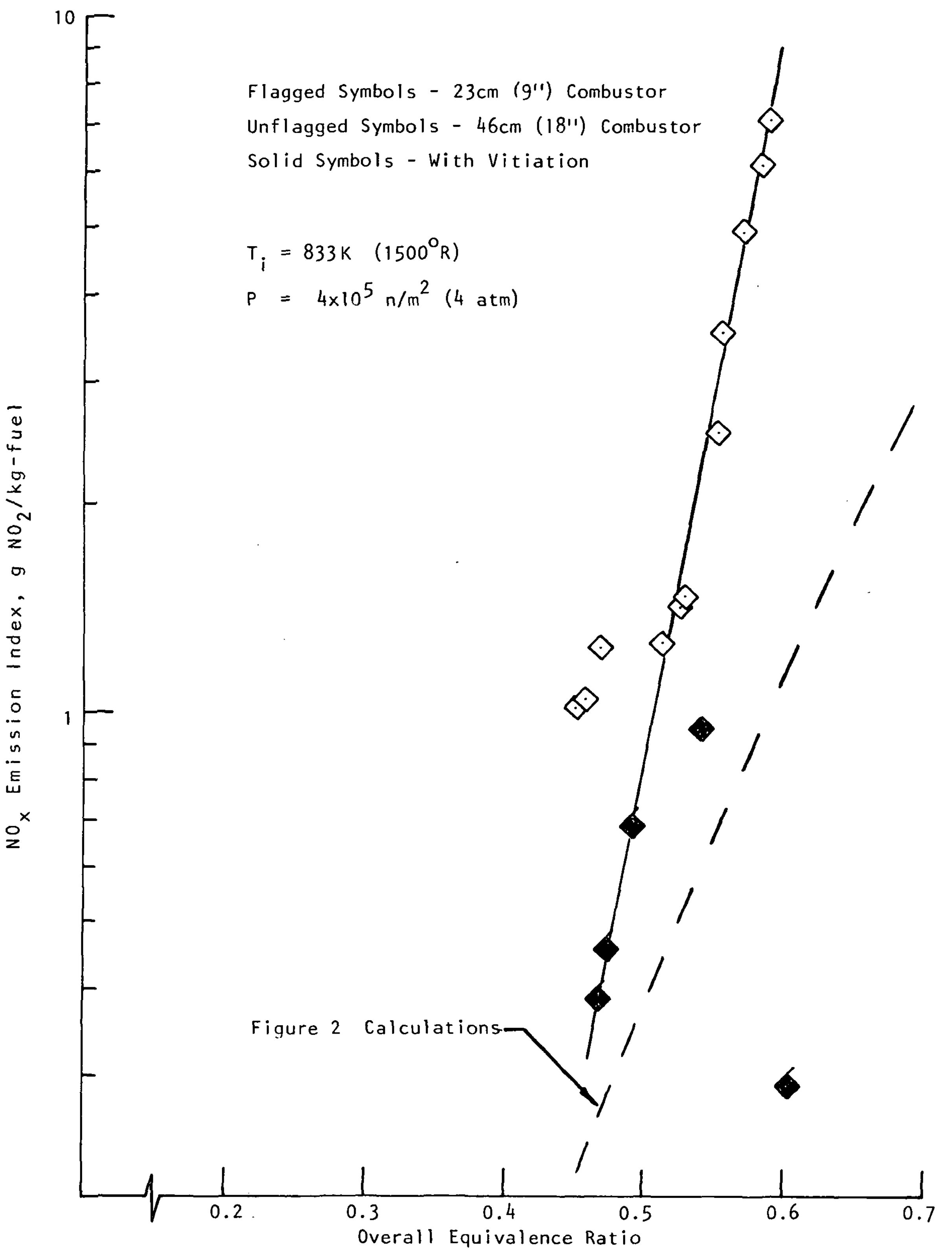


FIGURE 25. NO_X EMISSION INDEX FOR CONFIGURATION B

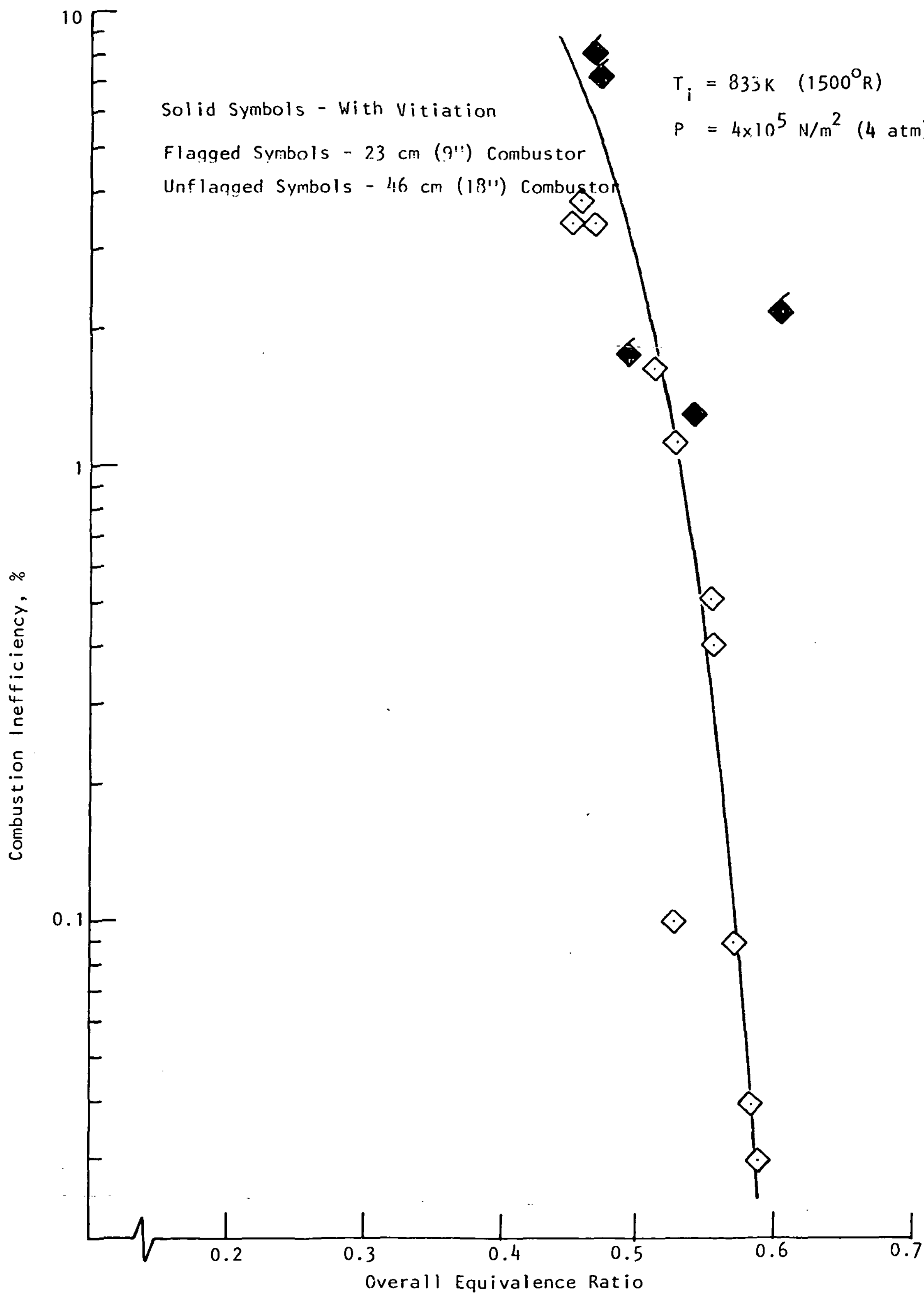


FIGURE 26. COMBUSTION INEFFICIENCY FOR CONFIGURATION B

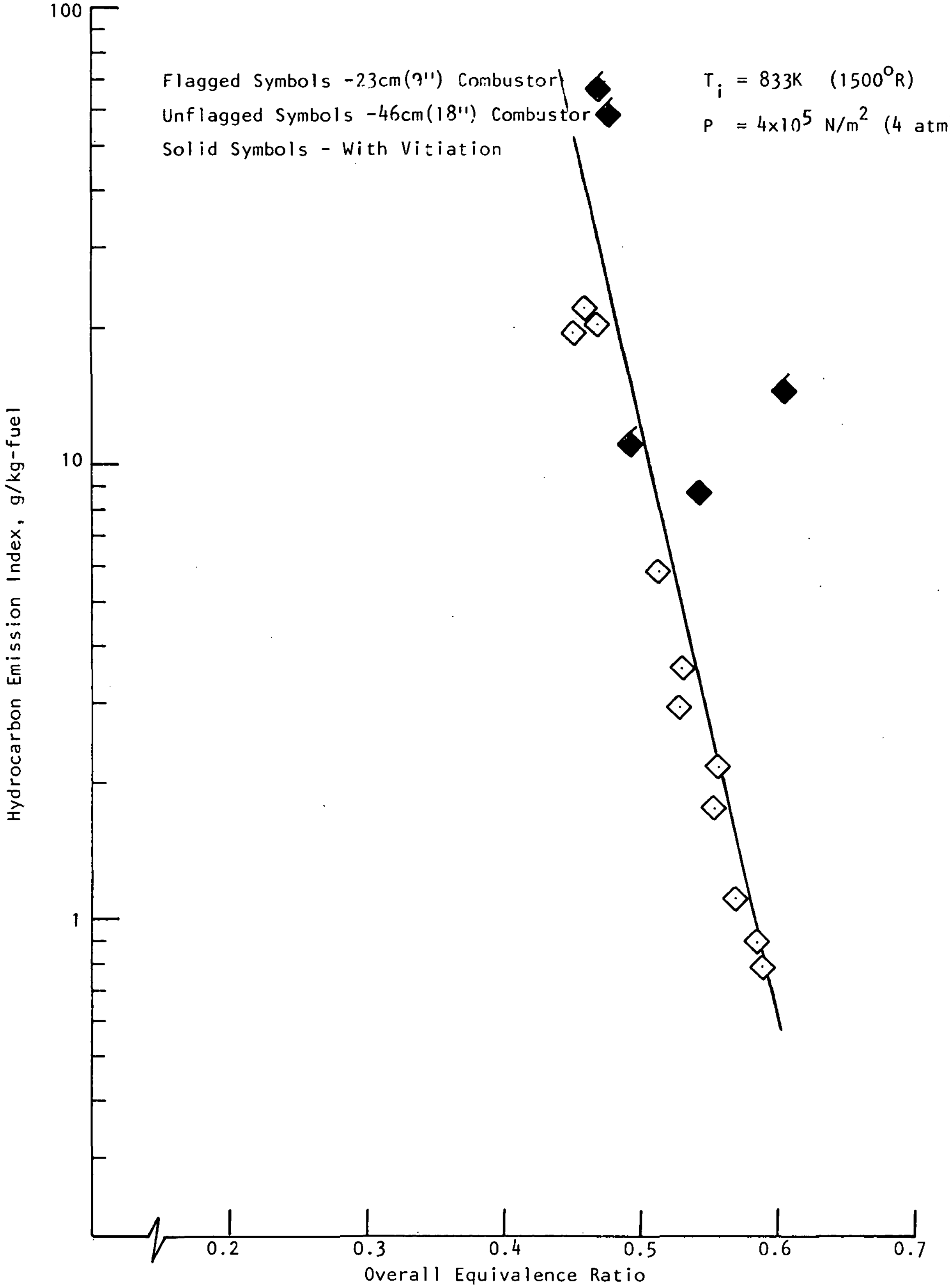


FIGURE 27. HYDROCARBON EMISSION INDEX FOR CONFIGURATION B

behavior as combustion inefficiency. The CO emission levels drop as equivalence ratio increases until chemical equilibrium is reached. Beyond that point they follow the equilibrium curve as shown in Figure (28). The anti-flashback constriction added at the flameholder exit station appears to decrease the flame propagation angle from the sudden expansion step, where combustion initiates, by increasing the axial component of velocity. As a result, the length required for complete combustion is increased by the use of this constriction.

Configuration C (Figure 12) employed the same mixer tube used in Configurations A and B but utilized a modified flameholder geometry which moved the anti-flashback constriction forward to the flameholder nose station and decreased the flameholder exit velocity from 198 m/sec at the constriction to 97 m/sec at the combustor entrance station. This change did not affect the flashback or autoignition characteristics which were the same as those of Configuration B, that is, no flashback but autoignition with preheat at higher equivalence ratios. Configuration D (Figure 14) differed from C in that the mixer tube diameter was kept constant at 7.6 cm (3 inches) from the fuel injection station to the anti-flashback constriction. This modification eliminated autoignition problems, even when the preheat section was used to vitiate the air.

Difficulties were encountered with the cavitating venturi fuel flowmeter during tests of Configurations C and D. These difficulties stemmed from the fact that the venturi mass flow is independent of downstream pressure as long as the venturi pressure drop is sufficient to produce cavitation. Thus, fuel flow rate was obtained by measuring the fuel line pressure upstream of the venturi throat. However, in tests with very low flow rates and in tests where the apparatus was operated for a considerable time prior to fuel injection, the nozzle pressure drop increased considerably above its cold flow value for the same flow rate, probably as the result of partial boiling of the fuel. This increased fuel line back pressure to the point where cavitation ceased and the single pressure measurement was no longer adequate to define flow rate. As a result, the usual cross check for sample validity (i.e., fuel/air ratio obtained from carbon balance must be within 15% of the fuel/air ratio obtained from flow metering) could not be obtained for many of the tests run.

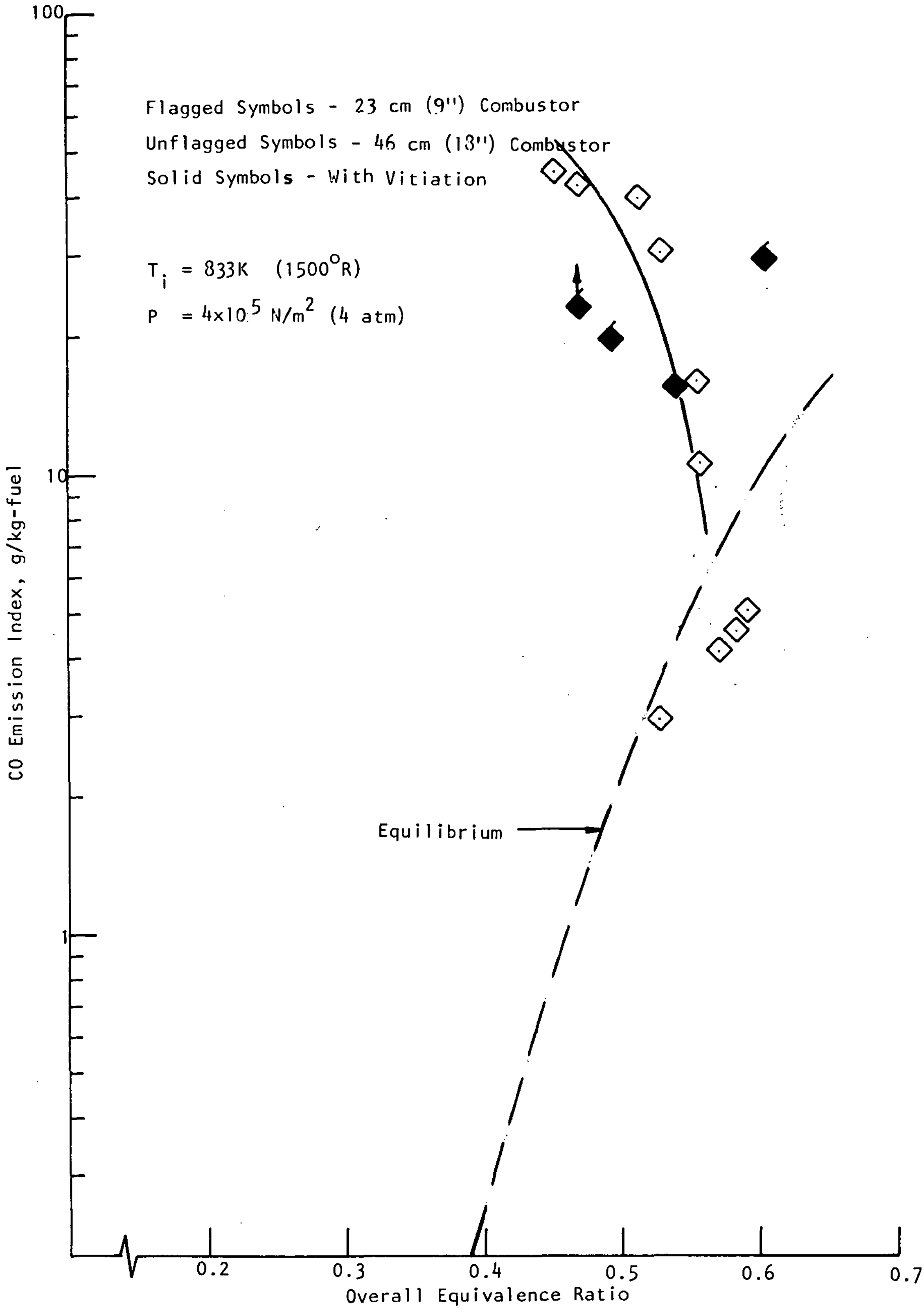


FIGURE 28. CO EMISSION INDEX FOR CONFIGURATION B

The measured NO_x emission characteristics of Configurations C and D are shown in Figure 29. The figure presents the results of all tests run. Data symbols representing those runs for which a sample cross check could not be obtained have been flagged to indicate that fact. The line drawn through the data points represents a fit to the unflagged data only, but appears to be an adequate representation of the flagged data as well.

There was no appreciable difference between the emission characteristics of the two configurations implying an insensitivity to mixer diameter. The NO_x levels shown in Figure 29 display the type of temperature dependance predicted in Reference 2, but are displaced toward lower equivalence ratios. As before, vitiation appears to have no significant effect at the conditions tested here.

It is interesting to note that the NO_x level after secondary combustion can actually be lower than that entering the combustor. Tests of the vitiated air alone (Figures 17-20) indicated a NO_x level of 2 gm-NO_2/kg of total fuel at the combustor entrance when vitiation was employed, with an overall equivalence ratio of 0.3 However, Figure 29 indicates a NO_x level of 1 gm-NO_2/kg fuel at that equivalence ratio with no apparent change using unvitiated air. Therefore, it would seem that the NO_x entering the combustor is chemically reduced during the early stages of the combustion reaction and that subsequent formation is independent of this preliminary reaction.

The combustion inefficiency for Configurations C and D is shown in Figure 30, where the beneficial effect of lowering flameholder exit velocity is clearly evident. The combustion inefficiency at the unity NO_x condition is just under 1%, and decreases logarithmically with increasing equivalence ratio. Hydrocarbon emission levels, shown in Figure 31, vary as the combustion inefficiency and carbon monoxide levels, shown in Figure 32, approach equilibrium as the equivalence ratio increases.

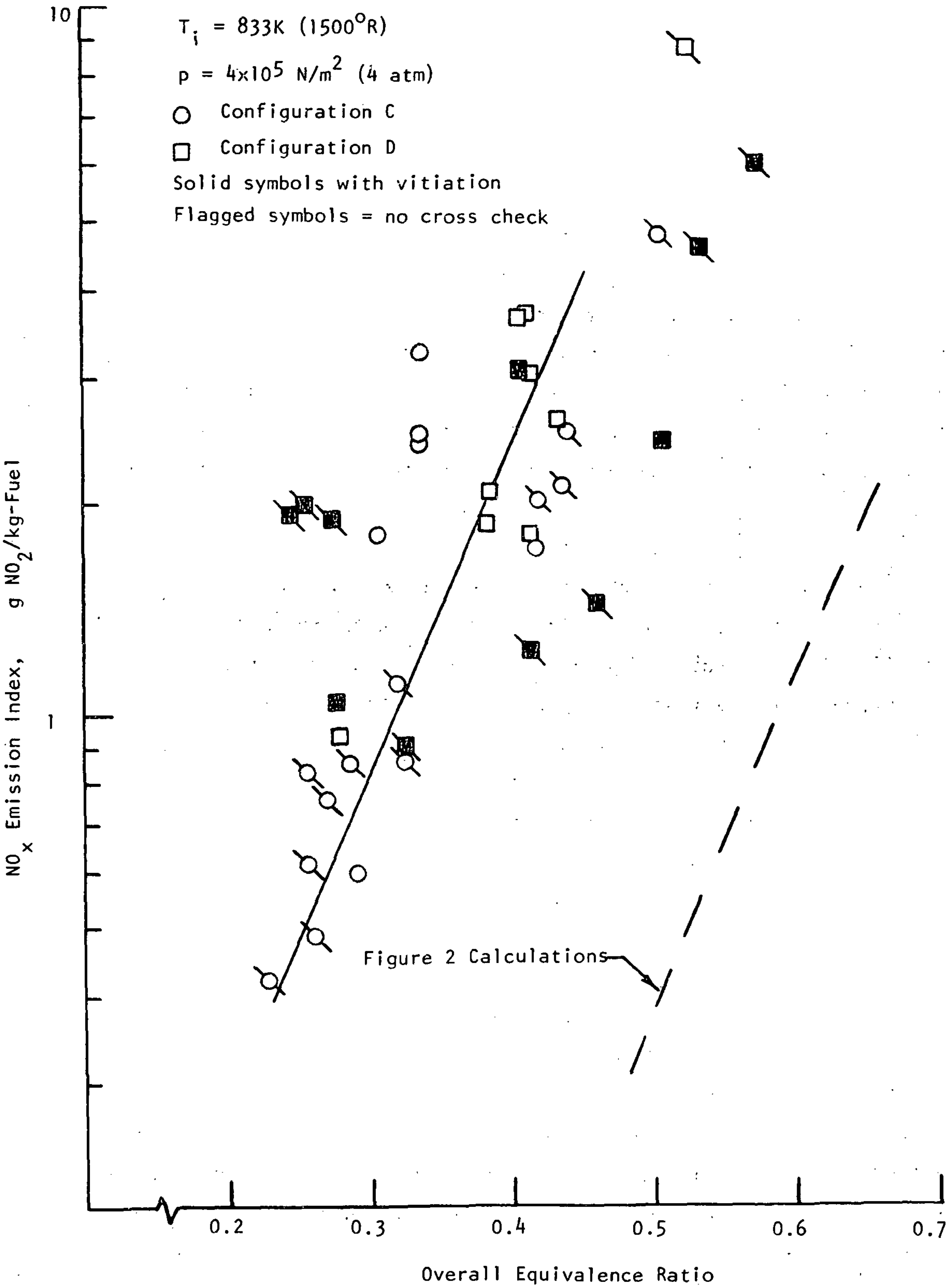


FIGURE 29. NO_x EMISSION INDEX FOR CONFIGURATIONS C AND D

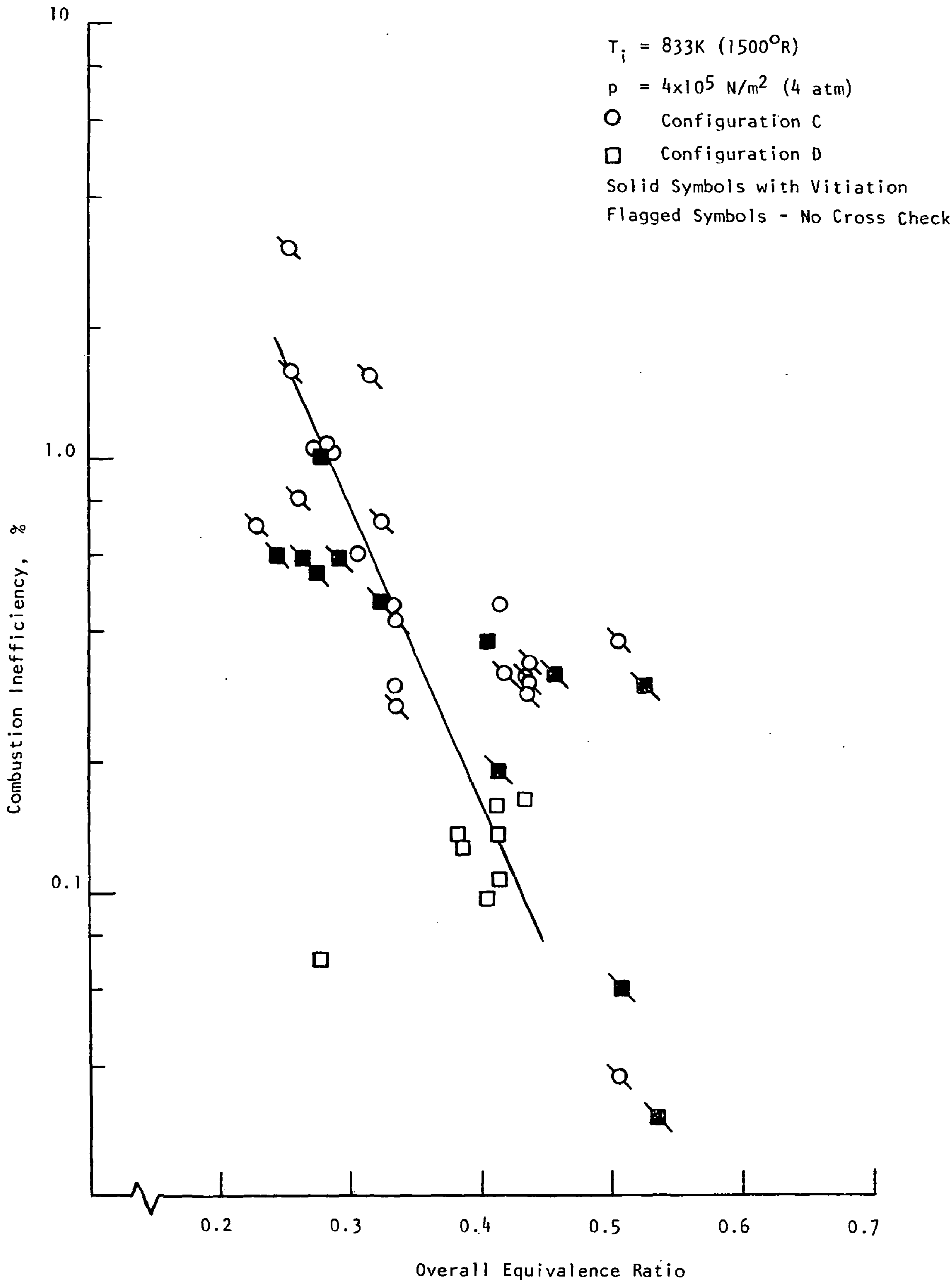


FIGURE 30. COMBUSTION INEFFICIENCY FOR CONFIGURATIONS C AND D

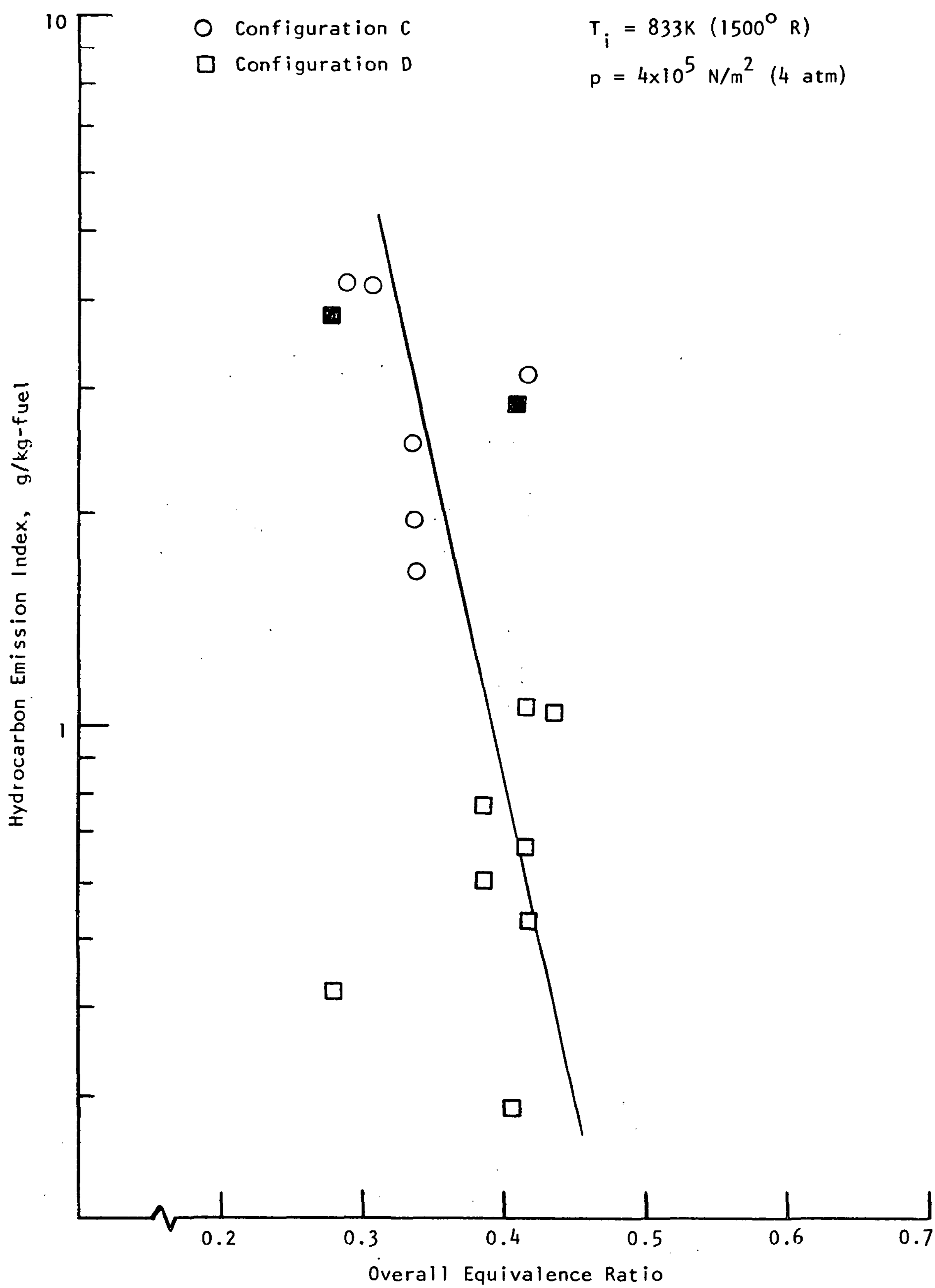


FIGURE 31. HYDROCARBON EMISSION INDEX FOR CONFIGURATIONS C AND D

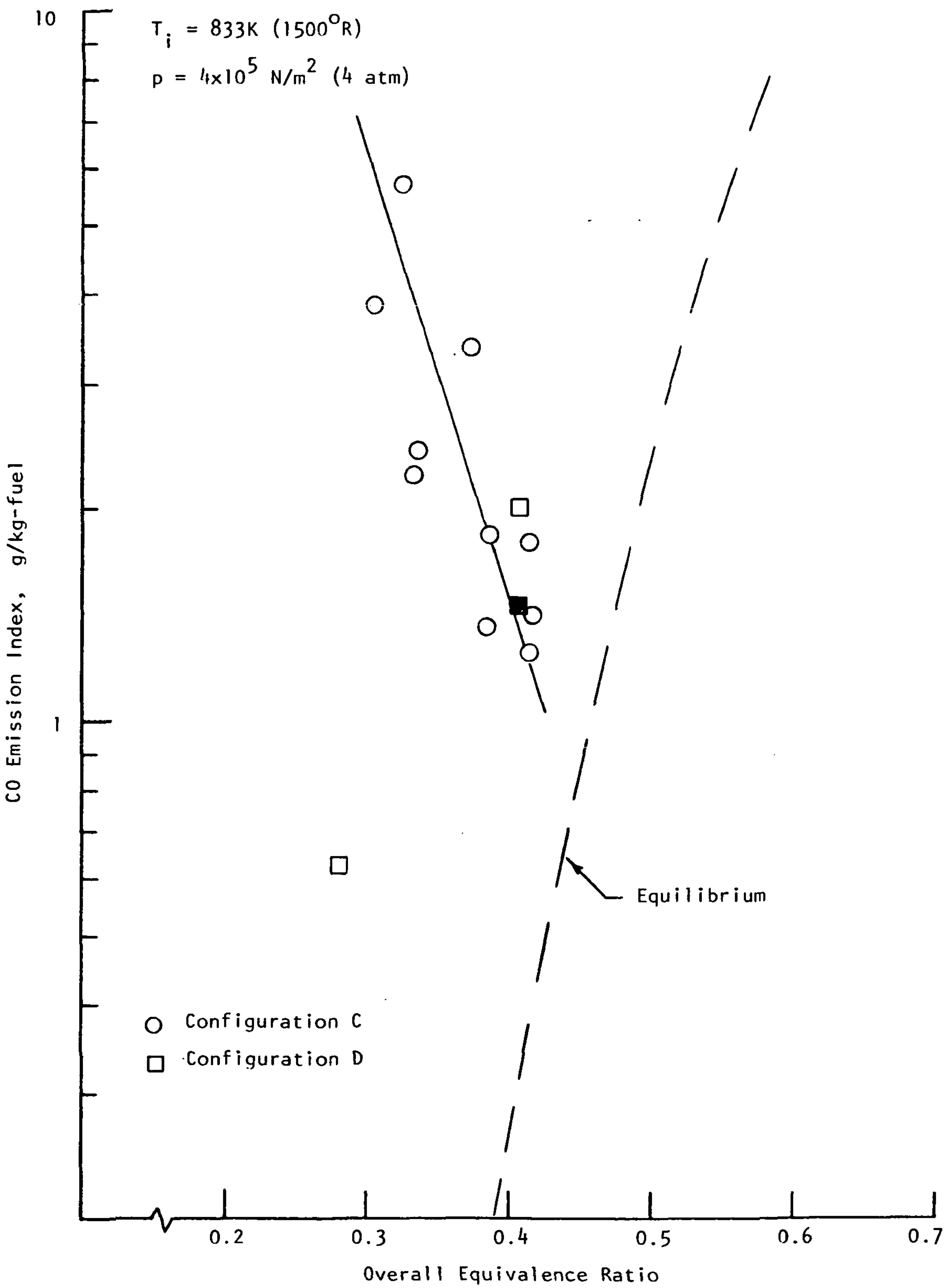


FIGURE 32. CO EMISSION INDEX FOR CONFIGURATIONS C AND D

On the general subject of pressure drop, it should be noted that the plenum chamber used to introduce air uniformly into the combustion apparatus produced a pressure drop of approximately 50% through the use of an array of choked orifices. Although clearly not a pressure drop attributable to the burner itself, there is the possibility that this drop at the very entrance to the mixer may have artificially increased turbulence levels and thus mixing rates. This possibility must certainly be investigated further. For the last configuration tested, the flameholder diffused mixer velocity down to 97 m/sec (318 ft/sec) of Mach 0.15. Assuming a total loss in dynamic pressure at the flameholder, this implies a pressure drop of 1.5%. Adding the Rayleigh line total pressure loss due to combustion brings the frictionless loss to 2.5%. Had the mixer gas not been diffused prior to the sudden expansion at the flameholder, the 187 m/sec (614 ft/sec) mixer velocity would have resulted in a combined total pressure drop of nearly 7%. Therefore, it is clear that premixing burners cannot employ excessive mixer velocity and should utilize isentropic diffusers whereever possible.

The fact that preheating the combustion air produced no apparent change in either NO_x level or combustion inefficiency is particularly interesting. It implies first that vaporization of pressure atomized droplets of JP-5 can be accomplished within the 2.7 milliseconds residence time provided by the 46 cm (18-inch) mixer tube since the presence of liquid droplets should substantially increase NO_x levels. It also implies that the flame in the combustor is primarily diffusion controlled since preheating the combustion air, which considerably speeds chemical reaction rates, had no effect on combustion inefficiency should be possible by providing a suitably designed flameholder. Although preheat proved unnecessary at the 833K (1500^{o}R) combustor entrance conditions tested here, its value at lower combustor entrance temperatures remains to be determined.

SECTION V

SUMMARY OF RESULTS

Four configurations of a prevaporizing, premixing gas turbine combustion apparatus were tested, using liquid JP-5 fuel, at the supersonic cruise condition ($T=833K$, $p=4 \times 10^5$ N/m^2). The apparatus consisted of an array of four pressure atomizing nozzles, a mixing and vaporization duct, a flameholder and combustion chamber. The first configuration employed a mixer velocity of 83 m/sec (273 ft/sec) and was incapable of preventing flashback. The second configuration was equipped with a constriction at the mixer exit which increased the combustor entrance velocity to 187 m/sec (614 ft/sec). This constriction prevented flashback from the combustor into the mixer. However, attempts to increase combustor inlet temperature by vitiating a portion of the inlet air resulted in autoignition. The high combustor entrance velocity produced by the mixer exit constriction increased combustion inefficiency by increasing flame length.

In the third configuration, the antiflashback constriction was moved upstream and a conical diffuser added to reduce combustor entrance velocity. This change substantially deceased combustion inefficiency and did not alter the antiflashback qualities of the design. Autoignition still occurred when the inlet air was heated beyond the 833K level by partial vitiation.

In the last configuration tested, the mixer duct diameter was reduced to produce a constant velocity from the fuel injection station to the flameholder entrance. This design displayed the same low combustion inefficiency as its predecessor and was capable of operation without flashback or autoignition in the mixer.

Although temperature measurements at the combustor entrance station revealed that a completely uniform mixture of fuel and air had not been attained, NO_x levels in the premixing designs displayed the logarithmic dependence on equivalence ratio predicted by finite rate chemical calculations. The NO_x levels measured were higher than those predicted by calculations which assume complete mixing. Combining this with the fact that a combustion inefficiency of less than 1% was obtained at an overall equivalence ratio on the order of 0.3, it appears that incomplete mixing produced combustion in regions of locally elevated equivalence ratio. Even with this apparent deficiency in mixing, a NO_x

level of 0.6 g-NO_2/kg-fuel was obtained with 1% combustion inefficiency at an overall equivalence ratio of 0.29.

Preheating the combustor inlet air above the 833K level by partial vitiation produced no apparent changes in NO_x level or combustion inefficiency, although this did exacerbate the autoignition problem. Measurements of the NO_x level produced by vitiation showed 2 g-NO_2/kg-fuel entering the combustor and only 1 g-NO_2/kg-fuel at the combustor exit. It would appear that NO_x entering the combustor undergoes some degree of reduction in the early stages of reaction since NO_x levels downstream of the main combustion zone are the same with or without vitiation.

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