DEVELOPMENT OF A HIGH INTENSITY, LOW EMISSIONS BURNER

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Abstract

In the past decade increasing environmental awareness has brought attention to the emissions levels of gas turbine combustors, especially in the case of potential automotive applications. In order to meet present and projected pollution requirements, researchers have been investigating the performance of premixed, lean primary zone fuel-air ratio combustors to replace the present diffusion flame, stoichiometric burners. However, the premixed lean combustor has a very narrow acceptable operating range as a function of fuel-air ratio. The burner development described here was undertaken to demonstrate that the emission index (g pollutant/kg fuel) levels of CO and NO_x can be lowered by an order of magnitude from previously attained levels, while maintaining very low unburned hydrocarbons levels. Flameholder geometry was found to have a very strong effect on the emissions levels through its influence on the lean flammability limit, and hence was carefully studied. It was also noted that the lean operational limit imposed by the rise in CO levels there is primarily a residence time phenomenon.

Table of Contents

List of Tables

List of Figures

List of Symbols

a	distance from the dead end of a can-type flameholder to heat ports
A	bluff body area
A_j	jet port area
D	flameholder diameter
d	jet port diameter
h	distance downstream from the flameholder
p	pressure
S_u	laminar flame propagation velocity
t^*	ignition delay time
UHC	unburned hydrocarbons
v_a	bluff body flameholder approach velocity
v_{bo}	blow-off velocity
v_j	jet velocity
v_r	reference velocity (based on properties of the cold gases)
α	thermal diffusivity
φ	equivalence ratio
η	efficiency

I. Introduction

Pollution standards are becoming increasingly stricter for all types of power plants, both stationary and mobile. One important source of pollutants are the burners used in continuous flow engines. Such engines include the gas turbine Brayton cycle engines which, besides being widely used in planes, are finding wider application in marine, power production, pumping, and automotive uses.

(Although Brayton cycle engines are commonly touted for their low pollution potential, especially compared to internal combustion engines, they are still "dirtier" than could be desired.) The burners used in such engines are presently incapable of meeting 1976 Federal emissions standards for automobiles, and standards for flight emissions will be adopted soon.

Gas turbine combustor design practice has largely been influenced by the performance requirements of flight propulsion applications, even when units have been used on the ground or on the seas. Design philosophies date back to World War II when pollution wasn't a concern, and a narrower range of technologies were available. In recent years, it has been proposed and demonstrated that by changing the way in which fuel and air are delivered to the combustor, drastic decreases in the emissions levels are possible. This is at the cost of increased complexity and is made possible by the introduction of new materials and control systems.

Premixing of fuel and air before combustion, and lean primary zone fuel-air ratios offer a way of drastically reducing unburned hydrocarbons, carbon monoxide, and oxides of nitrogen. The basic validity of this approach has been demonstrated in the laboratory and has reached developmental status in some cases.

It was the goal of this project to demonstrate that NO_x emission indexes of between 0.01 to 0.1 g NO_x/kg fuel could be obtained while simultaneously keeping the CO emission index between 0.1 to 1.0 g CO/kg fuel and hydrocarbon levels at the negligible levels routinely obtained by most researchers. It was also desired to investigate the rise of CO near the lean limit reported by most researchers.

II. Conventional Combustors

Most gas turbine combustors are of the type shown in fig. 1. Because of the temperature limitations of the expanders, overall fuel-air ratios are of the order of 1:50, with an extreme lean limit of about 1:200. With modern compressors, pressure ratios of 20:1 and higher are used, although regenerative cycles are on the order of 5:1. Because most gas turbine combustors have been developed for flight applications, or have been derived from flight versions, they have been designed for high combustion intensities and stability over a wide range of pressures, fuel-air ratios, and reference velocities. Standard liner materials are alloys such as Inconel and Nimonic, so that primary zone wall temperatures have to be controlled by such means as film cooling.

In order to meet the stability requirements, most combustors use diffusion flames in which a stream of atomized liquid fuel is sprayed directly into the combustion area, so that mixing and burning occur almost simultaneously. Since the overall fuel-air ratios are far outside of the normal flammability limits, part of the air is allowed to bypass the primary combustion region, and is added back to the combustion products after reaction has taken place. The primary zone combustion occurs near stoichiometric and is stabilized by recirculating flows created by impinging jets or swirl action.

Mixing of fuel and air in the primary zone is usually poor, and locally the fuel-air ratio may vary from unburnably lean to unburnably rich. Under such conditions, even if the overall primary zone fuel-air ratio is not optimum, there are bound to be enough local zones where the fuel-air ratio is able to maintain combustion.

Typical emissions levels of conventional gas turbines are presented in Table I.

There exists at least one exception in this diffusion burner world. About 30 years ago Armstrong-Siddely, now part of Rolls-Royce, developed a combustor in which liquid fuel and air were premixed and sent through a "candy-cane" vaporizing tube positioned directly in the annular combustion region. The upstream jet serves to stabilize the burning. Although it is generally agreed that the mixing in the candy-cane tube isn't complete, combustors of this type emit no smoke. Vaporizing combustors of this type were first used in the "Mamba" jet engine and are best known in this country in the Curtiss-Wright J65 (1).

III. Pollutants

There are three types of pollutants which are of interest here. They include unburned hydrocarbons (UHC), oxides of nitrogen (NO_x), and carbon monoxide, and are produced by the combustion of air with hydrocarbon fuels. The global reaction is represented by the equation:

$$C_xH_y + (x + \tfrac{1}{4}y)\,O_2 \rightarrow xCO_2 + \tfrac{1}{2}H_2O \tag{1}$$

The concentration by volume of O_2, CO_2, and H_2O in the special case of propane reacting with air as a function of equivalence ratio φ (fuel-air ratio divided by the fuel-air ratio at stoichiometric) is plotted in fig. 2a and b for lean ($\varphi < 1.0$) fuel-air mixtures. Also included are plots of NO, CO, and H_2 (2).

UHC and CO are the result of incomplete reaction of the fuel with the oxygen in the air. If reaction (1) went to completion, no UHC or CO would be produced for $\varphi < 1$. Because of equilibrium considerations, this is of course impossible. The sequence of events which occurs during the combustion of a hydrocarbon fuel is not understood in detail. To simplify greatly, fuel first undergoes pyrolysis and partial oxidation to H_2 and CO, respectively. These products then react with oxygen to form water and carbon dioxide:

$$CO + \tfrac{1}{2}O_2 \rightarrow CO_2 \tag{2}$$

$$H_2 + \tfrac{1}{2}O_2 \rightarrow H_2O \tag{3}$$

The equilibrium constants for these reactions are strong exponential functions of temperature, being highest for low temperatures and low for high temperatures. The combustion efficiency, η_c, the fraction of the maximum possible energy which has been released during a combustion reaction, and the most important combustion performance parameter, can be calculated from the concentrations of UHC, CO, and H_2 in the combustion products since they are the principal species representing unreleased enthalpy. One equation which expresses this relationship is: (from ref. 3)

$$1 - \eta_c = 10^{-3}(.232\ EI_{CO} + EI_{UHC} + 2.76\ EI_{H_2}) \qquad (12)$$

where levels of UHC, CO, and H_2 have been given in terms of "EI" or emissions index, which is grams of species per kilogram of fuel consumed.

Inefficient combustion is the result of inadequate flame stabilization, premature quenching of the post flame reactions, and/or poor premixing of fuel and air before combustion. According to previous combustor models, (4) (5) poor mixing is the major culprit in conventional combustors.

Although most hydrocarbon reactions occur immediately in the flame front or not far from it, and so the levels of UHC are fixed immediately after the flame front, CO levels are determined by reactions in the post flame region. In the flame itself the hydrocarbons are reduced to CO, H_2, and H_2O along with various radicals, such as H, O, and OH. After the flame front the CO and H_2 are oxidized, as shown in reactions (2) and (3). The CO reaction is catalyzed by the presence of water, and is represented by the sequence of steps:

$$CO + O_2 \rightleftarrows CO_2 + O \qquad (13)$$

$$O + H_2O \rightleftarrows 2\ OH \qquad (14)$$

$$OH + CO \rightleftarrows CO_2 + H \qquad (15)$$

$$H + O_2 \rightleftarrows OH + O \qquad (16)$$

with equation (15) being the important one. If these reactions are quenched by the premature addition of cool diluent, such as wall-film cooling or bypass air, the CO will be prevented from reacting to CO_2 and will be present in the exhaust in superequilibrium quantities dependent on when the quenching occurred (how far downstream from the flame front.)

The production of NO_x is governed by reactions separate from the actual combustion process, except for activation energy. The primary source of NO_x is generally agreed to be the pair of reactions referred to as the Zeldovich mechanism: (6)

$$O + N_2 \rightleftarrows NO + N, \quad k = 1.4 \times 10^{14} \exp(-78500/RT) \qquad (9)$$

$$N + O_2 \rightleftarrows NO + O, \quad k = 6.4 \times 10^{9} \exp(-6280/RT) \qquad (10)$$

where the k's are the equilibrium constants, T temperature, and R the universal gas constant. Because of its high activation energy, reaction (9) controls the system. The activation energy is provided by the combustion process, and hence the NO_x emissions levels are highly dependent on flame temperature. From equation (9) it is also evident that NO_x levels are dependent on O radical concentration, which is also dependent on temperature. A small additional source of NO_x under rich conditions is the reaction:

$$N + OH \rightleftarrows NO + H \ , \quad k = 2.8 \times 10^{13} \qquad (11)$$

Super equilibrium concentrations of O atoms may make contributions, but their importance decreases with pressure increase. Reactions in the flame front involving HCN and CN may make small contributions under rich conditions. Generally the Zeldovich mechanism is the main source of NO_x in lean flames, with the reactions occurring in the post flame area because of the high activation energy and the requirement for radicals which appear only at high temperatures. Because of this, NO_x formation is also sensitive to premature quenching of the post flame reactions.

A generalized plot of UHC, CO and NO_x concentrations as a function of φ is given in fig. 3. While UHC levels of less than 1 ppm on a volume basis for equivalence ratios from the lean limit to 1.25 are possible, CO levels usually tend to gradually increase from the lean limit to stoichiometric where they rapidly climb. NO_x is proportional to flame temperature, with its peak on the lean side of stoichiometric because of nonequilibrium effects. A plot of the theoretical adiabatic flame temperature of propane is shown in fig. 4 as a function of φ. In general the most important factor affecting levels of pollutants from a burner is φ. NO_x is affected by such factors as pressure and burner inlet temperature primarily insofar as they affect flame temperature.

IV. The Importance of Being Lean and Premixed

As fig. 3 indicates, UHC, CO, and NO_x become simultaneously low as the fuel-air ratio becomes lean. Since UHC are low from the lean limit to slightly past stoichiometric, the critical quantities to control are CO and NO_x. Unfortunately, the reactions which govern their levels are relatively slow and occur in the postflame region. It is possible even in conventional diffusion flame combustors (see Table I) to keep either NO_x or CO at federally acceptible levels, but it is not possible to keep them simultaneously low. Conditions which are conducive to complete combustion and low levels of CO and UHC also lead to high levels of NO_x.

Most methods for lowering the NO_x level without sacrificing CO performance involve lowering the flame temperature, since that is the factor which most strongly governs the NO_x formation rate. One method is water injection into the primary zone. However to be effective, water flow rates must be of the same order as the fuel flow rates, making water injection impractical in propulsion applications for steady-state operation. It would be practical only for certain transient conditions such as large acceleration. There is also some question of effects on components (6) (7).

Another approach is to reduce the combustion temperature adding an inert diluent gas to the combustion zone, such as recirculated exhaust gas. Conceptually, the most straightforward way of reducing the flame temperature is to make the equivalence ratio in the primary zone as small as possible. However, merely reducing the overall primary zone fuel-air ratio in the case of a diffusion flame is inadequate. As the liquid droplet in a diffusion flame begins to vaporize and the resulting vapor diffuses outward from the droplet and begins to mix with the air, there exists locally a wide range of fuel-air ratios, including ones near or at stoichiometric. Because of the sharpness of the NO_x curve near stoichiometric, these local hot spots can create an inordinate amount of NO_x, far above acceptable standards. To effectively control the flame temperature by controlling the fuel-air ratio, the fuel-air composition must be as uniform as possible throughout the mixture. Efforts are being made to make diffusion flames more nearly homogenous by reducing the droplet size, but this is of limited effectiveness since the diffusion phenomenon is the inherently limiting factor. However, measurements by Burgoyne and Cohen (9) using tetraline as a fuel implied that fuel droplets with a diameter below 10 microns burn as a vapor while those above 40 microns burn individually in the diffusion mode. Droplets of intermediate size behaved in a transitional manner.

Since NO_x reactions are relatively slow and a majority of the NO_x production occurs downstream of the flame front when the equivalence ratio is near one or greater, another approach is to quench the NO_x reactions immediately after the flame front. Under ideal conditions it might be possible to remove enough heat from the primary zone to halt the NO_x formation but not enough to halt the CO to CO_2 reactions. Such an approach might be possible in a stationary application where steady-state conditions can be maintained, but probably would not be suitable for a propulsion application.

In general, the most straightforward approach to simultaneously reducing emissions of NO_x, CO, and UHC is to burn homogenous and lean in the primary zone and allow enough residence time for the CO to drop to an acceptable level. Although burning lean and premixed has stability and operating range problems associated with it, it also offers some advantages. Additional benefits include longer combustor line life because of the lower temperatures and more even outlet temperature distribution because of the homogeneity.

V. Premixed Flames and Stability

In the recent past premixed flames have been of limited usefulness in practical applications, being chiefly of use in after-burners and reheat stages. However, premixed flames have been of great interest in the laboratory. They are one of the simplest types of flame phenomena and so the most amenable to analytic and experimental study, although understanding of diffusion flames has been of more applied importance in the past.

There are many fundamental differences between a premixed and diffusion flame, but the most important, from a combustor design viewpoint besides that of emissions levels, is that of stability. In general a diffusion flame is stable over a much wider range of fuel-air ratios, while a premixed flame is stable to greater velocities, as illustrated in fig. 5 (10). While the narrow flammability limits are a drawback, the emissions requirements of a low pollution combustor limit the usable range of fuel-air ratios anyway. On the positive side, the increased velocity stability gives the possibility of high combustion intensities.

The stability curves for a typical flame are shown in fig. 6 as a function of equivalence ratio and approach velocity. Approach velocity is here defined as the volume flow rate upstream of the flame front divided by a characteristic cross sectional area of the burner mixture passages. The stability curve is usually closed and may be arbitrarily divided into four sides, called the strikeback, or flashback, rich, lean, and maximum blow-off limits. The position of these boundaries is governed by the chemical composition of the fuel, pressure, inlet temperature, and the burner.

Pressure has the effect of increasing the rich limit, but only a slight effect on the lean limit as shown in fig. 7. Increased inlet temperature has a very strong effect on the lean limit. Correlations of the form:

$$\varphi\ (T_2) - \varphi\ (T_1) = C \times 10^{-4}\ (T_1 - T_2) \qquad \text{(ref. 3)} \qquad (17)$$

have been made, where T_1 and T_2 are the initial mixture temperatures before combustion and C is a constant of the order of 5. This formula is probably valid over a limited range only.

Extending the lean limit by increasing the inlet temperature has a limited effect on the production of pollutants. Although for a fixed value of φ increased inlet temperature does raise the flame temperature, the flame temperature at the new lean limit remains the same as at the old lean limit (11). The combination of reduced temperature increase due to combustion and increased inlet temperature cancel each other out, and hence the NO_x level remains unaffected if φ is reduced to take advantage of the new lean limit. The effect on CO and UHC is not so obvious, although measurements indicate that their levels are reduced as inlet temperature is raised (12).

The strikeback region's extent is a strong function of the physical geometry of the apparatus involved. Up to a certain point, the smaller the dimension of the port through which the flame must strikeback, the lower the velocity at which it occurs. There is a critical inlet port dimension for which strikeback will not occur, called the quenching distance, d_T. In general:

$$d_T \propto 1/p$$

while increasing temperature also causing d_T to decrease. Strike-back, and blow-off, can be correlated by combustor loading parameters, an example of which, used in ref. 10, is the $1/\theta$ parameter: (13)

$$1/\theta = \dot{m}/(p^{1.75} A_r D_r^{0.75} e^{T/300}) \times (kg/(atm^{1.75} meters^{2.75} sec)) \quad (18)$$

With the exception of some experimental catalytic burners, most combustors operate in a region far above the strikeback boundary, but in the premixed case are prone to another type of strikeback which is not as strongly dependent on velocity as it is on heat balances. This other sort of strikeback occurs when the upstream flow reaches the spontaneous ignition temperature before it reaches the intended combustion zone.

Plee and Mellor (13) list three varieties of this "nonclassical" strikeback:

1) autoignition due to high upstream temperatures or hot walls;

2) flame propagation through reversed flow fields; and

3) preignition of a separated flow field.

The latter two are caused by upstream flow disturbances such as upstream fuel injection and physical obstacles which can cause flow separation and so recirculate flow. Almost any kind of obstruction or irregularity, such as a poor match where two sections of ducts meet, can act as undesired flameholders under the wrong conditions. Reversed flow fields and separation can be avoided by careful design and assembly of the ducting between the mixer and flameholder. Autoignition is the greatest problem since inlet temperatures are not under the control of the burner designer. However, since there is a delay time associated with autoignition, the problem can be approached by controlling the residence time, as is discussed in a following section.

VI. Premixed Lean Burners

The principal components of a premixed lean burner include:

1) flameholder

2) fuel and air mixer

3) vaporizer (for liquid fuels)

4) bypass air control valve

5) control system

6) starting circuits, such as atomizer (for liquid fuels)

The relationships of these components to one another is illustrated in the block diagram of fig. 8. This component breakdown is mainly for conceptual purposes, since in an actual design, some functions might be combined in a single device. These components are not necessarily unique to premixed lean (PML) burners although diffusion burners do not need vaporizers or separate starting circuits, even if they use liquid fuels. Variable geometry and control systems may also be applied to diffusion burners (14).

In PML burner design there are at least five major areas which deserve attention: residence time; nonclassical strikeback; liner cooling; geometry control; and mixing. The subject of flameholders will be covered in a separate section.

Residence Time. Most conventional combustors have been designed to be as compact as possible with emphasis on short length. The exceptions are some which have been designed for stationary gas turbine applications, and their size approaches that of the rest of the engine. Compact size favors NO_x control since the primary zone combustion is quenched as soon as possible, but penalizes the oxidation of CO to CO_2. Increasing the combustor size would appear to have no advantage, except that at lean fuel-air ratios the rate of NO_x production decreases rapidly, while the rate of CO oxidation becomes greater. Hence, the benefits from allowing the CO to react to CO_2 are greater than the penalty paid in NO_x production. This situation is illustrated in fig. 9.

Residence time, the time it takes exhaust gases to travel from the flame front to the plane where quenching occurs, is defined as the combustor volume divided by the volume flow rate of combustion products. Combustor volume is that volume bounded by the beginning of the flame front, the liner walls, and the cross-sectional plane at which the bypass air rejoins the primary zone flow. The desired CO level should be the criterion for choosing the residence time at the maximum flow rate, and hence the combustor volume, since the NO_x criterion can be ignored. However, physical constraints may put a limit on the residence time.

Nonclassical Flashback or Autoignition. If a mixture of fuel and air is elevated to a certain pressure and temperature and the fuel-air mixture is of the right value, ignition will occur. The time the mixture may remain at this temperature before a reaction occurs is called the ignition delay time, t*. It should be noted that as the temperature of a mixture is raised, a continuous series of chemical reactions occurs with "ignition" being defined when a violent reaction occurs which is usually accompanied by an audible report. In a poorly designed burner it is possible for t* to be less than the upstream residence time, which is the volume between the point where fuel is introduced into the air flow and the flameholder divided by the volume flow rate of the air plus fuel. This can be due to high inlet temperatures, thermal soak-back from the flameholder, or wall hot spots. This situation is illustrated in fig. 10. Within the area bounded by the combustor "length" axis, the mixture limit line, and the danger point curve, the temperature must be kept below the ignition temperature. This ignition temperature is a function of the upstream residence time, pressure, duct design, insulation, and inlet temperature. As noted earlier, anything which locally increases the residence time, such as recirculation zones which convert the mixer and the duct into a flameholder, must be avoided through careful design of these areas. Soakback and hot spots caused by the flameholder must be avoided by thermally isolating the flameholder from the incoming mixture.

The problem of autoignition caused by inlet temperature is directly related to the problem of mixing of the fuel and air, and is discussed in reference 11. There are basically two cases of interest. Fuel can be introduced into the mixer either as liquid droplets which are vaporized by the inlet temperature or as a vapor. Experimentally measured t*'s for the former case for isooctane fuel are plotted in fig. 11, taken from Mullins, ref. 15. As noted in ref. 11, these t*'s may be in the range encountered in automotive turbine inlet tracts, which is on the order of 100 msecs. In this droplet diffusion case, t* is claimed to be only a weak function of such factors as droplet size, fuel temperature, and coarse air turbulence. Air fuel ratios in the lean range and air velocity were found to have a negligible effect. It was found that t* is inversely proportional to static pressure. Similar measurements were made by Spadaccini (16).

For mixing systems in which the fuel is prevaporized, the t* represents a different sort of phenomenon. In ref. 17 Burwell and Olson measured t*'s for a homogeneous mixture of isooctane rather than the vaporizing droplet mixture of Mullins. The results were correlated by the equation:

$$t^* = (398.7\ W_a + .916) \times 10^{-15}\ \varphi^{-1.04}\ p^{-2.57} \exp(58320/T) \quad (19)$$

where W_a is the air flow rate in lbs/sec. Note that φ has a strong effect. Another major difference is the activation energy. In general, for both the prevaporized and droplet diffusion cases t* is given by an Arrhenius expression:

$$t^* \ \alpha\ \exp(E/RT) \quad (20)$$

where E is the global activation energy. Burwell and Olson measured E= 64.2 kcal/mole, while Mullins measured E= 32.4 kcal/mole. For purposes of comparison, equation (19) with W_a = 0.0132 lbs/sec, P= 1 atmosphere, and φ = 0.5 is plotted in fig. 11 next to the Mullins curve. A partial explanation of this difference is that in the vicinity of a vaporizing droplet there may exist locally high values of φ as the vaporized gas diffuses outwards from the droplet, and high φ implies shorter t* according to Burwell and Olson. Measurements made by Brokaw and Jackson (18) with gaseous propane indicate a strong dependence on fuel concentration. Measurements made by Spadaccini (16) with liquid JP-4 and No. 2 fuel oil droplets showed almost no dependence on equivalence ratio. The situation is analogous to that of the wider stability range of the diffusion burner compared to the premixed burner.

If the reported data are correct, prevaporized mixing, especially for lean equivalence ratios, may allow higher inlet temperatures and/or longer upstream residence times. Although during the mixing process even a prevaporized fuel will temporarily give high equivalence ratios until the mixing is completed, the time these "dangerous" equivalence ratios exist may be shorter than in the diffusion droplet case. Longer residence times may allow the possibility of better mixing. A drawback of vaporizing the fuel before mixing is the problem of fuel breakdown and the resulting deposits build-up on fuel line walls.

Because of the lack of data on t*'s for different fuels and conditions, experimental testing is necessary to determine t* accurately for a particular application.

Liner Cooling. Conventional combustors depend on the injection of bypass air along the walls of the primary zone to provide film or transpiration cooling. In a PML burner the use of such cooling air should have a beneficial effect on NO_x levels by halting the Zeldovich mechanism, but at the same time would quench the post flame reactions which lower the CO level. However, it is not known if any research has actually been undertaken to quantitatively measure these two effects. Ceramic liner walls or convection cooling are sometimes recommended. Since premixed lean flames burn at much lower flame temperatures than the stoichiometric diffusion flames in a conventional combustor, the thermal wall loads should be much lower in a PML burner.

Variable Geometry. Conventional burners operate over a large portion of their stability curves. A PML burner is restricted to a very small variation in φ because of the emissions requirements, forcing the operating region into becoming almost an operating line. Ideally this operating line should be as close to the lean blow-off limit as possible, although some experimenters (19) (20) (21) claim that CO levels rise near the lean limit, restricting the operating range even more.

Since present expander maximum temperature limitations require an overall equivalence ratio which is well below the flammability limit, the possibility exists of keeping the primary zone fuel-air ratio at a fixed point by varying the bypass air ratio. This requires variable geometry air ducting and a control system. It is sometimes claimed that variable geometry flameholders are also required to keep the gas velocity in the primary zone constant. This is because the lean limit becomes "richer" as velocity increases, which compromises the emissions performance.

An alternative approach is to avoid variable geometry by extending the operating limits. One way of achieving this is by refining the design of the mixer and flameholder as much as possible. Another is to design a burner which has two stages of burning. The first stage is usually a stoichiometric diffusion flame which acts as a pilot light for the premixed, lean second stage. Unless the first stage represents a very small portion of the total primary zone flow, this approach appears to be intrinsically less attractive than the pure premixed lean approach.

Mixing. Good microscopic mixing is vital for good emissions performance. Locally rich regions have the effect of raising the level of CO and NO_x. Mixers in general are a subject which is beyond the scope of this discussion, although a few generalizations may be made. There are basically two approaches to mixing which are of interest here. The first type involves the spraying of atomized droplets of fuel directly in the hot inlet air tract where the droplets are flash vaporized and turbulence created downstream of the injection point completes the mixing. In the other case, prevaporized fuel is injected into the inlet tract. As mentioned above, autoignition restrictions may limit the time available for mixing. Also, parts of the mixer, such as turbulence generating devices, may also unintentionally work as flameholders if classical flashback occurs and hence exacerbate the nonclassical strikeback problem as noted in ref. 10.

Another important point is the need for multiple fuel injection points, regardless of whether the fuel is injected in the liquid or gaseous form. This is necessary to insure good mixing distributions and reduce the mixing time, as noted in refs. 22 and 23.

VII. Flameholders

Designing a burner so that it can operate as close to the flammability limits as possible under all conditions is of vital importance if low emissions are to be obtained. It is important to design the flameholder, that part of the combustor where combustion actually occurs, to have good stability properties, where stability is taken to mean a good lean limit.

In high speed gas turbine flows where the velocity is greater than the turbulent flame speed of the mixture by at least an order of magnitude, a flame must be stabilized by the violent recirculation and thorough mixing of the combustion products with the incoming mixture. Devices which accomplish this feat, and hence anchor the flame at a fixed point in the combustor, are called flameholders. There are two basic types of flameholders. The more basic type depends on the flow separation caused by a bluff body. Bluff body flameholders are most commonly used in afterburners and ramjets. The stability of such devices is related to their wake width, which in turn can be related to their shape and size. For a duct of fixed size, as the flameholder scale is increased, a maximum blow-off velocity is reached when the flame wake width is about half the height of the duct in a two-dimensional case and $\frac{1}{3}$ the duct diameter in an axisymmetric case (24). A correlation of experimental results is given in fig. 12.

The other type of flameholder depends on the action of jets pointed at angles ranging from perpendicular to the major flow to directly upstream, or the inducement of swirl flows to cause recirculation. Such devices will be referred to as "can" type flameholders hereafter in this paper, although they may assume other geometries. No simple correlations or analytical methods exist to aid in the design of can-type flameholders. Most development work is based on empiricism, thorough testing, and art. Ref. (11) contains a quick overview of the basic types of can flameholders, though probably many more types exist than are discussed there. Fig. 13 contains simple schematic representations of the way in which bluff body and can-type flameholders work.

Although recirculating flows are the most common way of stabilizing flames, other methods exist which depend on recirculating heat from the combustion products to the incoming flow by conduction or convection instead of by mass transfer. A thin copper strip held parallel to the flow has been reported to be able to function as a flameholder (25). Ramjet experimenters demonstrated that a smooth alumina tube will work (26), while others have used electrically heated low drag flameholders. (27).

While the above systems depend on conduction, a rather ingenious one depends on convection heat transfer (28) (29). In this system the inlet passages are designed to receive heat from exhaust passages. With thermal "positive feedback" ratios of 80%, it is claimed that it is possible to abolish the conventional flammability limits. The system is even claimed to have automatic strikeback control. Since the device burns mixtures which won't usually burn without the high level of preheating, if strikeback does occur, and the heat transfer area is reduced, the preheat is also reduced so that the flame "strikes forward". One inherent problem with such a system is that excessive pressure drops may be created by the large heat transfer surface required.

VIII. Review of Previous Research and Development

Although the study of premixed flames in the laboratory dates back to the earliest days of combustion research, the development of lean premixed flames for the express purpose of lowering emissions levels has occurred mainly within the present decade. Research may be divided into two categories, the study of fundamental properties in the laboratory and applications development. The latter may be further divided into jet engine and automotive gas turbine combustor research. Much of the flight applications research has been funded by the Federal government through NASA, the EPA, and the military. Although the government has sponsored work in the automotive field, some of the most important research has been performed by Ford, GM, and Chrysler. Laboratory research has verified that premixed lean flames can produce lower levels of pollutants, while applications research as focused on the problems of lean blow-off and flashback.

Since 1977 most flight applications research has been managed under the NASA Stratospheric Cruise Emission Program (SCERP) through Lewis Research Center (30) (31). SCERP coordinates the fundamental research being conducted at Lewis, in universities, and in industry, into what is termed lean, premixed, prevaporized (LPP) combustion. The predecessor of SCERP was the Experimental Clean Combustor Program (ECCP) which concentrated on upgrading conventional technology. The pollution levels attained in this program did not meet the order of magnitude NO_x

reduction recommended for cruise conditions (32). The SCERP effort is divided into four phases:

I. Fundamental studies and concept assessment.

II. Concept screening.

III. Experimental combustor development.

IV. Engine verification.

As of 1979 phase I had been completed with the next three phases expected to take until 1983 to complete. Laboratory research has been oriented toward simulating conditions found in actual combustors, including residence time, reference velocity, inlet temperature, fuel type, and pressure. Although conventional liquid fuels have been used, propane has been found to have the same characteristics as Jet A (31) except for ignition delay times. Propane tends to have longer delay times than other hydrocarbons, and hence would give misleading results in autoignition studies. Fuel-air mixture preparation has received a great deal of attention while flameholder design has been ignored until recently. The influence of inlet temperature and pressure have also been studied. Results hitherto reflect the general trends and behavior outlined in section III. Most work has been restricted to steady state performance, since the primary focus of SCERP is on reducing NO_x during cruise conditions.

Although ground level pollution near airports has also become a cause for concern, some of the earliest jet engine pollution research concerned the effects of Advanced Supersonic Transports (AST's) on the upper atmosphere. Ferri was one of the first to propose premixed lean combustion as a solution to this problem. With Roffe he examined the proof of principal aspects (12), effects of temperature and pressure (33), and effects of premixing quality (22). The flameholder used in these experiments consisted of a 10° half-angle cone supported by struts. Reference velocity was 46 m/sec under a pressure of 4 atmospheres. The gas sampling rake could be located at either of two stations, 9" and 13" downstream of the flameholder. Liquid JP-5 fuel was injected 18" upstream of the flameholder and was vaporized by the inlet temperature of 700 to 1000°K. Both the upstream position and number of injection points were varied to control the extent of fuel-air mixing and vaporization. According to work done by Cooper (34), the emissions level sensitivity to thoroughness of vaporization is greatest at leaner equivalence ratios as far as NO_x was concerned. Degree of vaporization also had an influence on CO levels near the flameholder, but the influence was negligible far downstream of the flameholder. Cooper also verified that liquid droplets of a size less than 50 microns Sauter Mean Diameter burn as a "psuedo-vapor".

Roffe and Ferri also showed that elevated inlet temperatures decrease the levels of CO and UHC while increasing NO_x for a fixed value of φ because of the increased flame temperature. Pressures of 4 to 24 atmospheres were also examined, with Roffe and Venkataramani (35) using pressures up to 30 atmospheres. This had the effect of lowering CO and UHC levels, while having no effect on NO_x. It was concluded in general that flame temperature is the only factor which effects NO_x levels. A comparison of temperature effects on diffusion and premixed lean burners is shown in fig. 14.

Roffe and Venkataramani also examined the effect of flameholder geometry on emissions levels (36). Geometries tested included wire screens, perforated plates, multiple cones, single cones, vee gutters, and swirl burners. It was found that geometry had only a weak influence on the lean blow-out limit and on flashback. Emissions levels of CO, UHC, and NO_x were found to be strongly influenced by the blockage ratio. It was proposed that the blockage effect was due to the increased turbulence levels created. All of these flameholders were uncooled. It was noted that the water-cooled perforated plate flameholder used by Anderson (37) and Marek and Papathakos (38) had a slightly lower lean stability limit.

A study was also made using a combustor with an upstream jet for flame stabilization (39). Measurements were made at reference velocities of 7.74 m/sec., inlet temperatures of 300 and 600°K, and at various stations downstream of the flame front. Propane was the fuel. Comparison was made between the experimental results and predictions made with a simplified kinetic model of propane combustion.

The Solar Division of International Harvester (40) has developed a can-type flameholder for both flight and automotive applications. Called the JIC for "jet induced combustion", combustion is stabilized by the 180° change in direction gases must make between inlet and exhaust.

Roberts, et al (41) developed and tested an experimental two stage combustor. The primary stage which acted as a pilot light, used a diffusion flame while the secondary combustor was premixed. Although the fuel was liquid, it was vaporized before being mixed with air.

In the SCERP reports it is emphasized that stability and flashback are the two biggest problems to overcome before the LPP (PML) concept is a viable alternative to present diffusion combustor technology. To clarify the flashback terminology, the term "autoignition" was agreed to apply to ignition occurring upstream of the flameholder while "flashback" applied to the region downstream of the flameholder. Due to a lack of reliable data on the ignition delay times of common jet fuels, a program is under way to measure them.

In the automotive turbine field it is taken for granted that premixed lean combustion is necessary to meet EPA regulations. Ford Motor Company has published some of the most detailed information (11) (19). Their flameholder is of the single-stage can-type using "jet impingement" for stabilization. To cope with an allowable flame temperature range of 1560 to 2000°K electronically controlled variable geometry jet ports and bypass ratio valving were used. Special attention was given to the problems of autoignition and flashback. The upstream part of the combustor was designed to minimize mixture residence times and the can flameholder itself was insulated to prevent the soakback of heat from the combustion zone. Although the details of the fuel-air mixing weren't discussed, it appeared from the diagrams that some mixing quality may have been sacrificed to control the residence time. It was also noted that high inlet temperatures present no obstacle to NO_x control because the lean limit is extended at the same time the flame temperature is increased, allowing the operating range to be shifted to smaller values of φ for an increase in inlet temperature.

The Chrysler (20) and GM (21) combustor designs are both of the two-stage type with pilot light "torches" used to extend the operating ranges. The Chrysler design emphasizes fixed geometry without the need for external control systems while the GM design has electronically controlled variable geometry. The GM burner could operate in a flame temperature window of 1290 to 1400°K while that of the Chrysler burner could operate from 1110 to 1720°K. The smaller temperature does not necessarily reflect a lean limit effect but a CO limit. For most of the burners described in this section, the CO emissions curve as a function of flame temperature or equivalence ratio has a negative slope. This is probably an effect of lean fuel-air ratios and short (1-5 m/sec) residence times before the gas samples are taken, and will be discussed later.

To give an idea of the environment an automotive gas turbine combustor must face, Tables II and III present the operating conditions for the GM and Chrysler combustors as a function of gas generator speed. All present automotive gas turbines undergoing development are of the regenerative type. Chrysler also published some details of the transient conditions a combustor must withstand. Fuel surges of 100 to 1 ratio are claimed to be possible during acceleration to overcome gas generator inertia. Their combustor must have good relight characteristics since fuel flow is shut off during deceleration.

A comparison of the emissions results of some experimental combustors is presented in fig. 15. It is given in terms of EI_{CO} versus EI_{NO_x}. UHC are not included in the comparison since UHC tends to follow the same trends as CO, and UHC levels were in general low, on the order of 1 ppm in the primary zone.

IX. Present Experimental Work

Several important points can be drawn from the research reviewed in the previous section. The most important variables which affect emissions levels are flameholder geometry and fuel-air mixture preparation. Problems which need to be overcome to allow the practical implementation of premixed burners include flashback and autoignition, and limited equivalence ratio operating range.

From an experimental viewpoint, it has been accepted that propane simulates well the emissions characteristics of commercial fuels, although not the autoignition characteristics. It has also been demonstrated that inlet temperature and pressure have a limited effect on emissions. The effect of turbulence still requires further clarification.

In the present research it was decided to look further into the effect of flameholder geometry on emissions. It was also intended to examine the effect of laminarity. The main emphasis was on developing a burner with substantially lower emissions levels than had been previously reached. It was also hoped to investigate the rise of CO levels near the lean limit, an effect which narrows the operating range of the burner. It was speculated that this was mainly a residence time effect, and so the residence times for a particular burner were examined.

It was decided to use room temperature inlet air to the burner to avoid the complication of preheating the air and most of the measurements were made at one atmosphere pressure.

The present research is described in the following part of this paper.

Experimental Procedure. Principal measurements consisted of gas sampling from the flow downstream of a flame. The samples were analyzed in the instrument train shown in fig. 16. The Beckman instruments used included two 3158 infrared analyzers for measuring CO_2 and CO, an OM-11 oxygen detector, a 951 NO/NO_x meter, and a model 400 hydrocarbon analyzer. The hydrocarbon meter had been modified to be accurate to within 1% of full-scale reading.

Two different probes were used to collect samples, one for use at atmospheric pressure, and the other for measurements in a pressure cell. The atmospheric pressure probe, shown in fig. 17a, consisted of a 1/8" OD copper tube to which a U-shaped $\frac{1}{4}$" copper tube had been silver-soldered. Samples were collected through the smaller tube while water was passed through the larger tube to protect the probe and quench the sample gases. This probe was mounted on a mill head to allow accurate positioning in three dimensions.

The stainless steel probe, shown in fig. 17b, was made from a 1/16" inconel tube in a stainless steel water jacket. The design is basically two tubes, the sample and water feed lines, within a larger tube which can be sealed where it enters the pressure cell with a Swage-lok fitting. Again, the water cooling serves to both protect the probe and quench the samples.

Copper and high-nickel alloys are known to have catalytic effects on NO, which constitutes the major portion of the NO_x

produced by a flame. Within a probe, samples from a rich flame can undergo the reactions: (42)

$$CO + H_2O \rightarrow CO_2 + H_2 \tag{21}$$

$$2H_2 + 2NO \rightarrow 2H_2O + N_2 \tag{22}$$

so that NO readings are depressed. However, using quartz or water-cooled metallic probes reduces this problem so that accurate readings may be made up to an equivalence ratio of about 0.7. However, it has been pointed out that quenching may also cause the reaction: (43)

$$NO + O \rightarrow NO_2 \tag{23}$$

so that an investigator may be led into believing that NO_2 forms an appreciable amount of the NO_x produced by the flame (44).

In the case of the atmospheric probe, a sealed bellows type pump was used to draw samples through the probe and supply feed pressure to the instrument train. Chamber pressure obviated the need for a pump to drive the pressure probe. A calcium silicate water trap was used to dry the sample in both cases.

Burners. Two basic burners were used in the measurements, one for use with bluff body flameholders in the laminar region, and the other for use with can-type flameholders.

The bluff body burner is shown in fig. 18. On three sides the flameholder was surrounded by water-cooled aluminum walls and on the fourth by a quartz plate for observing the flame. The flow approaching the flameholder must pass through a series of screens to reduce the turbulence level. A 2" x 2" stainless steel duct could be placed above the flameholder to allow gas sampling far downstream (up to 14") of the flame front. The flameholder itself consisted of 1/8" wide stainless steel strips spaced 1/8" apart.

The second burner, which could be used both under atmospheric and at higher pressures, is shown in fig. 19. It consisted of a steel cylinder with an aluminum bottom and a transite top capped by a stainless steel duct. The cylinder contained the stainless steel (300 series) flameholder itself and the passages for supplying fuel-air mixture to it. The duct and flameholder inside diameters were both approximately 1-7/16". The flameholder itself in its final configuration was surrounded by the transite insulation shown. The cylinder also contained a mixer for combining fuel and air, but was not used because of problems encountered while attempting to sample directly from it, and to maintain mixture consistency between the two burners. Two stainless steel fine mesh screens* were positioned upstream of the flameholder to act as flame traps. A 2" long quartz tube was positioned between the flameholder and its duct to allow direct observation of the flames. Under atmospheric conditions a mirror could be held over the duct to observe the flames from above.

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* For a discussion of the dimensions required to prevent strike-back, and the influences of temperature and pressure, see R. Friedman and W. C. Johnston, "The Wall-quenching of Laminar Propane Flames as Function of Pressure, Temperature, and Air-Fuel Ratio." Journal of Applied Physics, 21 (August, 1950), pp. 791-5.

The pressure cell used to hold the above burner is shown in fig. 20. It was of all steel construction and consisted basically of a 2-1/4" OD, 1/8" wall tube joined to a 4-1/2", 1/4" wall tube via a 1/2" thick plate. House air was used to convectively cool the burner and its duct. This air was combined with the exhaust gases, and three water jets were injected into this mixture before it finally reached the outlet valve, which was kept in a water bath. The sampling probe was located immediately at the end of the burner duct in a fixed position, but could be moved over the width of the duct. A 1-1/4" diameter by 1/2" thick quartz window above the quartz section of the duct allowed direct observation of the flame under pressure conditions. The maximum working pressure to which the cell was subjected was 60 psig.

Premixed fuel and air were supplied to the burners from system shown in fig. 21. The fuel in all cases was commercial grade propane supplied by the Matheson Co. in liquid form. House supply air was used with it. Fuel and air were regulated to pressures of 80 inches of mercury before being metered through sharp-edged orifices which had been calibrated to an accuracy of better than 1% by the water displacement method. The two flows were then passed through control valves to be combined in the mixer. The mixer consisted of 24, 1/32" stainless steel tubes surrounded by a larger tube. Propane passed through the 1/32" tubes while air passed around them, and the two were combined at the outlets of the propane tubes. This mixer was developed and built by Dr. Francis Clauser and is based on the observation that the single fuel injection point usually used in experiments of this sort is inadequate to ensure good microscopic mixing, even if it is very far upstream of the flameholder. The mixer built into the can burner was based on the same idea. A tap in the supply line between the mixer and burner allows a Beckman 402 hydrocarbon detector to monitor accurately the fuel-air ratio.

It should be noted that there are three characteristic velocities associated with the burners. In the case of the bluff body burner, approach velocity, v_a, is the upstream mixture volume flow rate divided by the open area of the flameholder. For the can-type burner, the jet velocity, v_j, was used and is defined as the upstream mixture volume flow rate divided by the total jet port area. Also used was the reference velocity, v_r, the upstream mixture volume flow rate divided by the maximum duct area.

X. Measurements and Observations

The first group of data relates to the bluff body burner. This burner was intended to operate in the laminar region, and by varying the slot area, allowed approach velocities over 750 cm/sec. However, at the very high velocities, the flow became turbulent. The laminar premixed flames produced by this burner served as a baseline for later measurements. The stability curve for this burner is shown in fig. 22. In the central region of the area enclosed by the curve, the flames formed triangular tent shapes over the slots. Their height increased with velocity until they became turbulent and their tips became brush shaped. Before that point, the flames began to oscillate up and down, with an amplitude of oscillation of approximately one third their maximum height and at sonic frequencies. With regard to fuel-air ratio, their height was a minimum at stoichiometric, and was greatest at the rich and lean limits. It should be noted that the boundaries of the stable region weren't very well defined, and there was noticeable hysteresis between blow-off and reattachment (where the flame descends back down the duct to the flameholder). Lean blow-off was much more sharply defined than rich blow-off. Absolute stoichiometric blow-off was never attained, as might be expected from some of the bluff body correlations. Lean blow-off occurred in two stages. As the mixture became leaner, the flames would become longer until they broke-up into a chaotic wall of flame which finally completely detached itself from the flameholder. The leanest ratios reached with this burner were about 2.25% (equivalence ratio of 0.55)*. On the rich side the flames became longer, but eventually "opened-up", without the chaotic structure produced by the lean flames. The boundary of the rich blow-off boundary was extremely vague.

Combustion product samples were taken from this burner as a function of fuel-air ratio for the fixed approach velocities of 160, 350, 500 and 750 cm/sec. The CO, NO_x, O_2 and CO_2 curves for 160 cm/sec are shown in fig. 23a,b, while the CO and NO_x curves for all velocities are compared in figs. 24 and 25. Levels of UHC were almost constant at less than 1 ppm between lean blow-off and a fuel-air ratio to a little over 5%, at which point they immediately climbed up. In general the curves are all very similar in spite of the different approach velocities. The NO_x curve peaks vary somewhat at stoichiometric, probably due to the difficulty encountered there because of their sharpness. The measurements were taken in the free stream over the center of the flameholder to avoid wall effects. Probe height above the burner varied from 5" at 160 cm/sec up to 10" at 750 cm/sec to ensure that all post flame front reactions had gone to completion.

Data collection with the can burner was similar. However, part of this work included the development of different flameholders, a topic which is discussed in the next section. Initial testing of the flameholders was carried out at atmospheric pressure, and one

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* In this section, the term fuel-air ratio is taken to represent the fuel volumetric fraction of the mixture, f/f+a, where f represents volume flow rate of fuel and a represents volume flow rate of air.

from the original collection was chosen for testing under pressure. These flameholders could be operated only over very narrow ranges of fuel-air ratios. At the higher combustion intensities at which these flameholders were tested, enough heat was transmitted to the flameholder itself so that its surface temperature became great enough to ignite the oncoming upstream flow. This was especially a problem in the original version of the burner where a large plenum fed mixture to the flameholder. The final version was redesigned with passages to reduce the contact time of the mixture with the upstream walls of the flameholder, which were insulated with a $\frac{1}{4}$" of transite. However this design was only a stopgap measure since under pressure, the burner was still limited to a very narrow range of less than $\frac{1}{2}$% fuel-air ratio between lean blow-off and nonclassical strikeback.

Behavior of the can burner flames at the lean and rich blow-off limits were similar to those of the bluff body burner. On the lean side the can jets merged together to form a cone-shaped plume which became longer with decreasing fuel-air ratio. Eventually the plume broke-up into a chaotic plug or cylinder of flame, and finally blew-off as the fuel-air ratio became leaner. It should be noted that the break-up of the plume did not necessarily correspond to inefficient burning and a rise in CO and UHC. With the better flameholders, these pollutants were low right to the point of blow-off. Extremely lean, high jet velocity flames tended to produce higher levels of UHC than flames near stoichiometric and at lower velocities, but the difference was on the order of $\frac{1}{2}$ ppm.

XI. Flameholder Development

The design of can type burners is not very amenable to analytical methods, although can type burners are some of the most common types in actual use. The design is usually based on empiricism, misconceptions, and a lengthy development program. Although these combustors can and do perform very well.

The burner project was initially based on earlier research (45) which had implied that the condition of laminarity might be of benefit, in addition to the previously stated ones of thorough premixing and leanness, in order to minimize pollutant generation. This additional condition set an upper limit on the maximum allowable flameholder approach velocity, limiting the combustion intensity greatly if the flameholder area was also limited. To circumvent this problem, it was decided to increase the flame front area by "folding-up" the flat bluff body flameholder geometry into a cylinder. The resulting "folded-laminar" flameholder is shown in fig. 26a. It consisted of a 1-1/2" OD stainless steel cylinder into which 22, 1/16" evenly spaced, lengthwise slots were cut.

Fig. 27a and b shows the emissions profile for v_j = 500 cm/sec while NO_x, CO, and UHC curves for v_j = 500, 1000, 1500, and 2500 cm/sec are compared in figs. 28, 29, and 30.

Note that this flameholder was able to maintain stable operation down to a fuel-air ratio of only 2.8% (equivalence ratio of 0.68). Compared to the flat burner, the NO_x is severely suppressed at stoichiometric and on the rich side. This was probably due to non-adiabatic conditions caused by excessive heat loss to the duct walls. Another possibility is the catalytic action of the stainless steel duct on the NO.

As the velocity varied, the flameholder seemed to experience different modes of operation. At extremely low velocities, the flame shapes were similar to the triangular tent-shaped flames. At higher velocities, modes appeared in which the flames merged together in the center of the flameholder. Transition between the modes was sharp, and operation near the transition point usually resulted in oscillations. However, the pollution characteristics were not significantly degraded at the higher velocities, although above 1000 cm/sec, levels of UHC climb drastically at the lean limit.

From observations of the flame, the flameholder was obviously not acting as a rolled-up bluff body, but as a can-type flameholder. It was also apparent, from observations of this burner and the flat one, that nonlaminar operation did not degrade the pollution characteristics. Since the can burner was working better than expected, at least as far as combustion intensity potential was concerned, it was decided to develop it further.

Many different types and configurations of can-type burners are possible, but since little hard and fast facts are known about their design and relative advantages, the existing design was taken as a starting point. Can-type burners and their descendants were all of the generic class shown in fig. 31. The flameholder consists of a cylinder with one closed end and a series of jet openings evenly spaced in a row a distance a from the dead end. It has been noted (46) that there are two important recirculation zones in such a can. One is downstream of the holes and near the wall, the other upstream against the dead end. Hence, it is suggested that there are three parameters which influence the stability characteristics of such a flameholder: the ratio of a/D; the ratio d/D; and the number of jet holes, n.

Some investigators have attempted to optimize n (47), but their conclusion that $n = 6$ is best is not very convincing. The downstream recirculation zone has been studied to a more successful degree. The parameter d/D is similar to the "slot ratio" used by Broman and Zukoski (46). This parameter has to be small enough so that adequate penetration to the center of the can is ensured. The ratio a/D doesn't appear to have been studied before. Setting $a/D = 0$ was observed to decrease the stability of the can, making the reattachment phenomenon, used to light the burner, almost impossible in some cases. However, no effort was made to optimize a/D. In most cases it was set equal to $1/6$.

It is also possible to have multiple rows of jets. Some research (48) has indicated that having more than one row of jets of equal magnitude has a negative effect on stability. A comparison was made with two flameholders both with $n = 6$, $d/D = 1/12$, and $a/D = 1/6$. However, one had one row of jets while the other had three spaced $\frac{1}{2}$" apart (for a total of $3 \times n = 12$ jets). (See fig. 26b, c). Their lean blow-off limits are compared in fig. 32a on the basis of jet velocity and by reference velocity in fig. 32b and their emissions characteristics in figs. 33, 34 at a jet velocity of 2500 cm/sec. The emissions curves are truncated because of nonclassical strikeback.

A variation on the single row burner had n = 9, but the jet holes were all drilled at an angle so that a swirl flow was set up within the can. (See fig. 26d). At lean fuel-air ratios, the flames began to "unwind" and leave the can.

The final version of the single row burner had 22 evenly spaced 1/16" jet holes and is shown in fig. 26e. Its emission profile is shown in fig. 35a,b for a jet velocity of 1000 cm/sec. Although no better in performance than the n = 6 flameholder, the smaller diameter of its jet holes lowers the velocity at which classical strikeback occurs. No attempt was made to further optimize the number and arrangement of jet holes. It is suspected that as long as d/D ensures adequate penetration of the jets to the center of the can, the performance will not be affected by n.

The n = 22 flameholder was used to measure emissions levels as a function of distance from a fixed reference point, in this case the level of the jet holes. These measurements were made at a jet velocity of 1000 cm/sec under atmospheric pressure, for fuel-air ratios from 2.5% to 3.5%. Results for NO_x, CO, and UHC are presented in figs. 36, 37, and 38. The data from fig. 37 were replotted as a function of f/f+a in fig. 39. From this figure it can be seen that for different stations downstream from the flameholder, the CO concentration as a function of fuel-air ratio curves change their shapes drastically. Far downstream of the flameholder the curve is monotonic and of positive slope while near the flameholder there is a peak near the lean limit.

In fig. 36 the NO_x curves at lean fuel-air ratios are almost flat while the curves at higher fuel-air ratios increase with distance downstream from the flameholder, h, and hence residence time. The UHC curves are fairly flat at all fuel-air ratios except immediately at the flame front where they climb sharply. The point at which the curve for the lowest fuel-air ratio climbs is the farthest to the right because of the elongation of the flames near the lean limit. The CO curves display the highest dependence on h at all fuel-air ratios. The lean curves fall off more quickly than the richer ones, but it is still clear that too small a combustor volume (short a residence time) will increase the CO levels.

As noted before, at high burner intensities, near stoichiometric operation was impossible, due to nonclassical strikeback. This problem was only exacerbated under high pressure conditions, so much so that operation was impossible with the original configuration of the burner. The final insulated, low residence time design was only a makeshift solution, and further work is needed.

The final flameholder was tested at 30 psig with jet velocities of 2000 and 2800 cm/sec, and at 60 psig with jet velocities of 1200 and 1700 cm/sec. These values obtained under pressure are the lowest obtained hitherto, and are lower than those obtained under atmospheric pressure with the same flameholder. Measurements were taken under conditions of stable operation and with burner inlet mixture at room temperature in all cases. Results are presented in Table III.

As a final summary of the above results, selected NO_x, CO, and UHC curves of the various flameholders are plotted together in figs. 40, 41, and 42, respectively. In addition, the lean limit EI values of CO and NO are plotted in fig. 43. Although the bluff body results are slightly worse than those of the can-type burners, this is primarily due to the fact that no effort was made to optimize the bluff body dimensions and blockage ratio. The results do demonstrate the strong effect of flameholder design on the lean limit and hence on the lowest level of pollutants possible.

XII. Summary and Conclusions

Although premixed lean combustors present the possibility of greatly reducing pollutant emissions from Brayton cycle engines or any other engine in which continuous flow combustion occurs, there are still a number of problems to overcome. The two main ones are the inherently narrow operating range as a function of equivalence ratio and the danger of strikeback. Although the former may be remedied by variable geometry and elevated inlet temperatures, it is still important to pay attention to flameholder design, since that is the primary factor in obtaining good stability performance. The latter problem is of importance because it has an indirect effect on mixing. Adequate microscopic mixing is of crucial importance in preparing the fuel-air mixture. Although the performance of developmental premixed lean burners is vastly improved over that of conventional diffusion burners, even greater improvements might be possible with better mixing. Some indication of this is that the UHC and CO levels began to rise near the lean limits of the burners developed by the automotive companies. Although a similar trend was noted with the original can-type flameholder described in this paper, the better burners had CO levels which continued to decrease or leveled off near their lean limits, while the UHC levels were uniformly low until the lean limit.

The lean limit rise in CO is apparently produced by a combination of operating near the lean limit and using residence times on the order of a few milliseconds. Residence times on the order of 100 m/secs produce almost monotonic CO curves as a function of equivalence ratio for $\varphi = 1$. This may allow wider operating ranges and cleaner emissions through operating closer to the lean limit.

Research oriented towards developing combustors for AST's may give misleading results when applied to stationary or automotive combustors because of the above described effect. Although long residence times may be impractical for flight applications, there may be some uses where frontal area, bulk, and weight are not at so great a premium.

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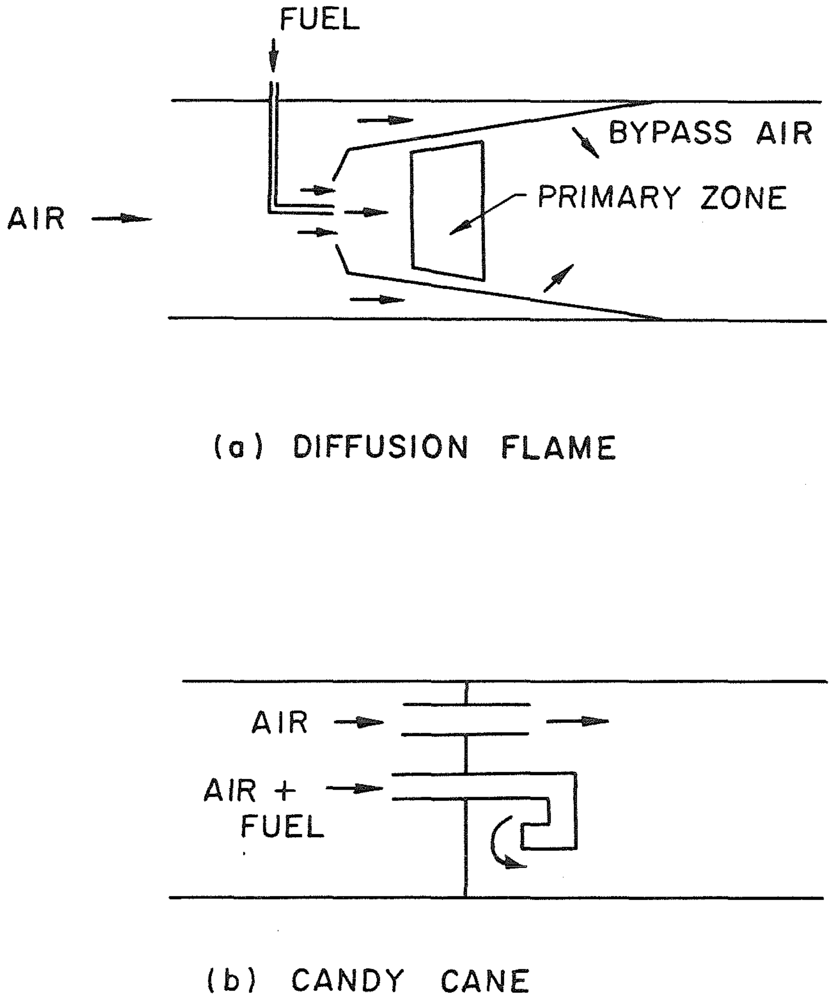

Figure 1. Conventional gas turbine combustor configuration.

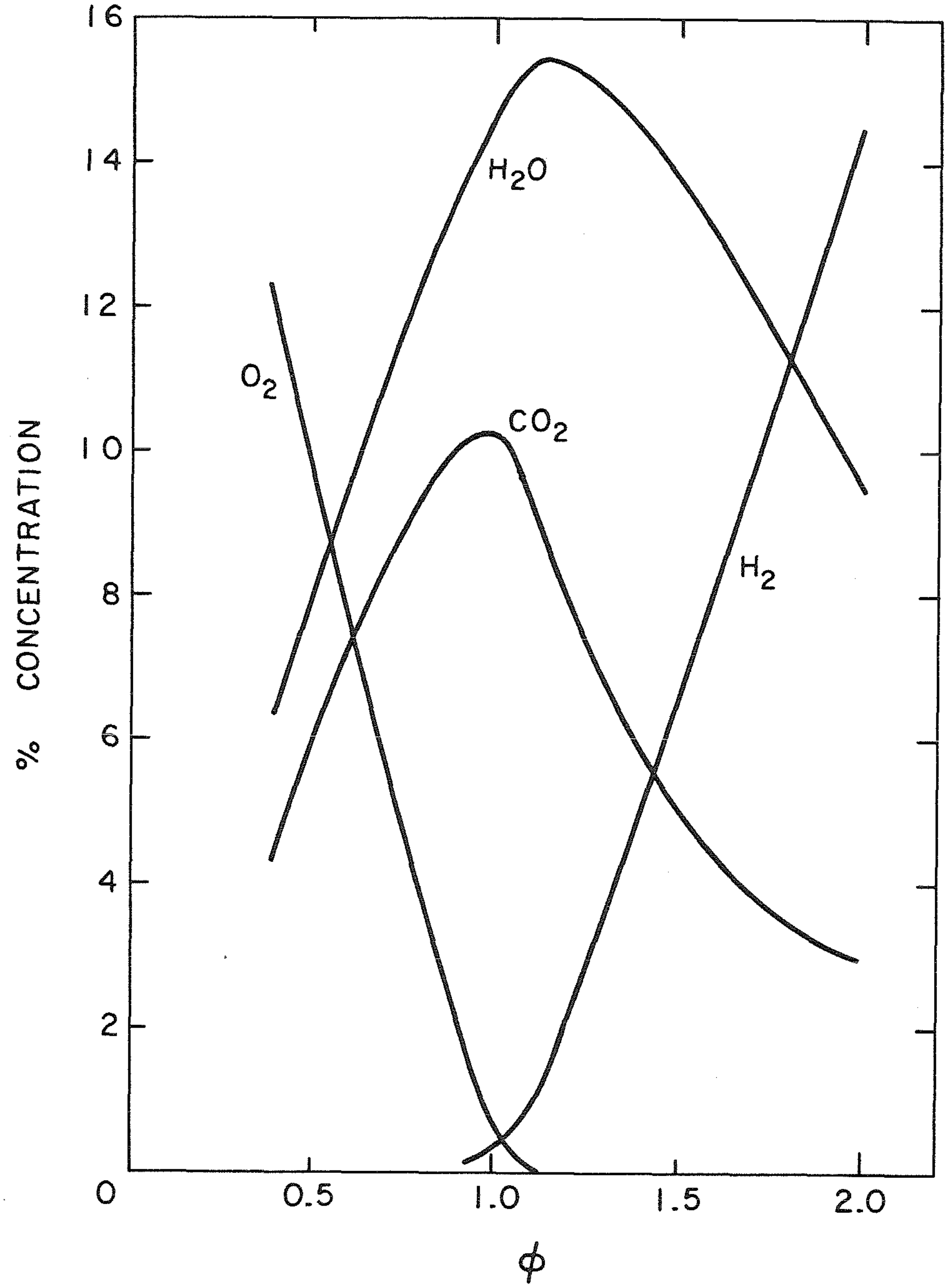

Figure 2a. Equilibrium concentrations of O_2, CO_2, H_2O, and H_2 for propane at room temperature and atmospheric pressure. (ref. 2)

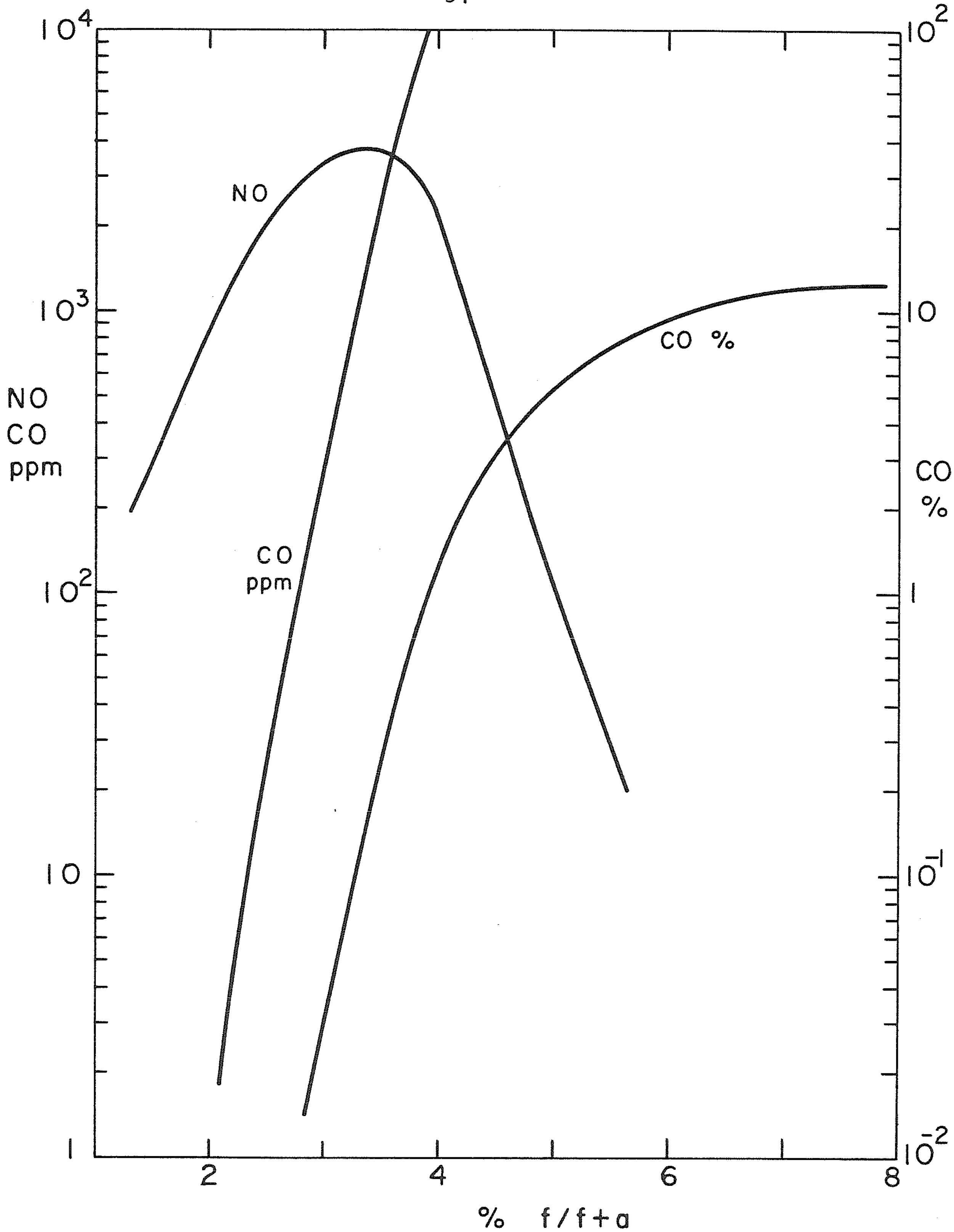

Figure 2b. Equilibrium concentrations of CO and NO_x for propane at room temperature and atmospheric pressure. (ref. 2)

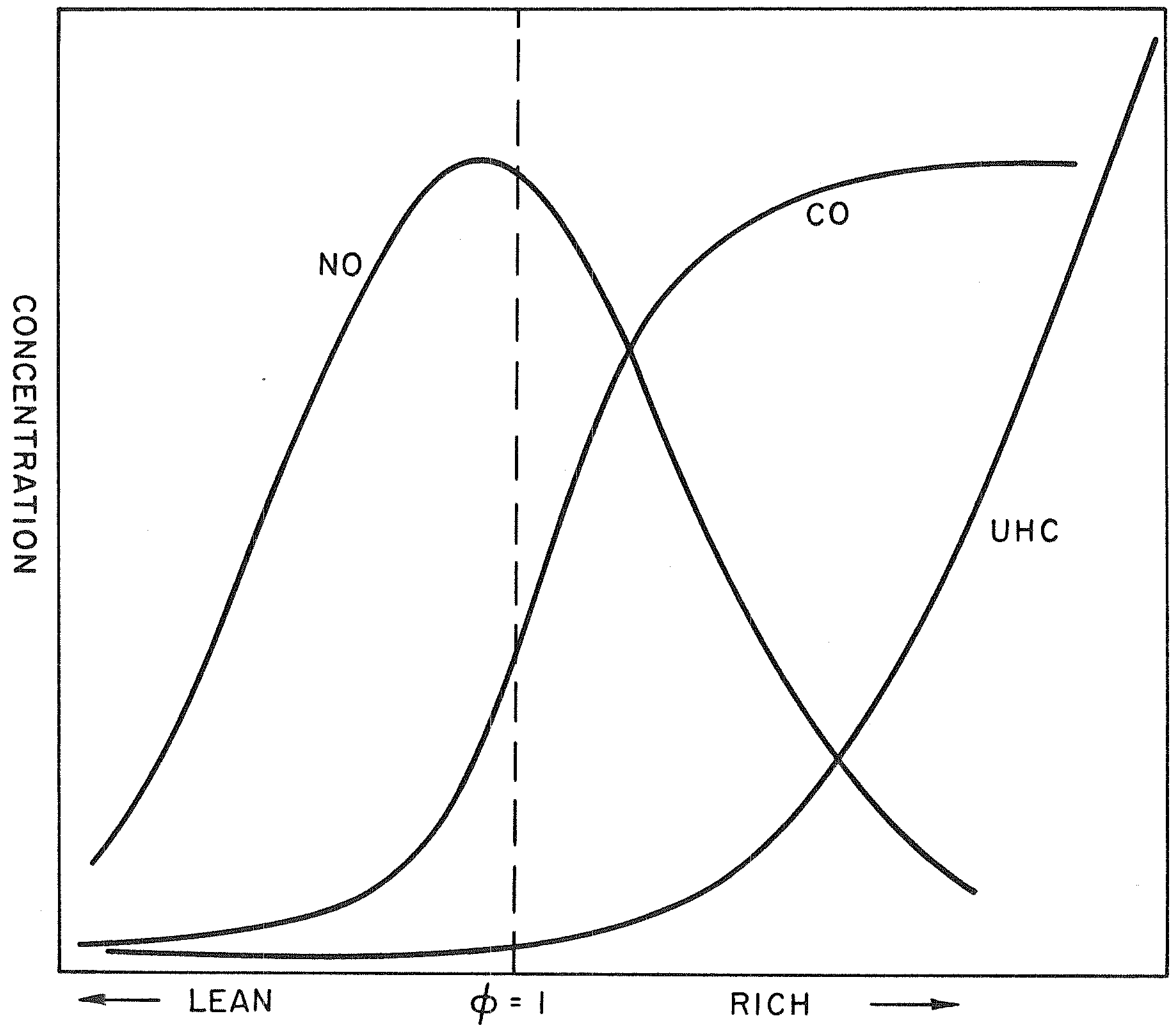

Figure 3. Basic pollutant curves as functions of equivalence ratio.

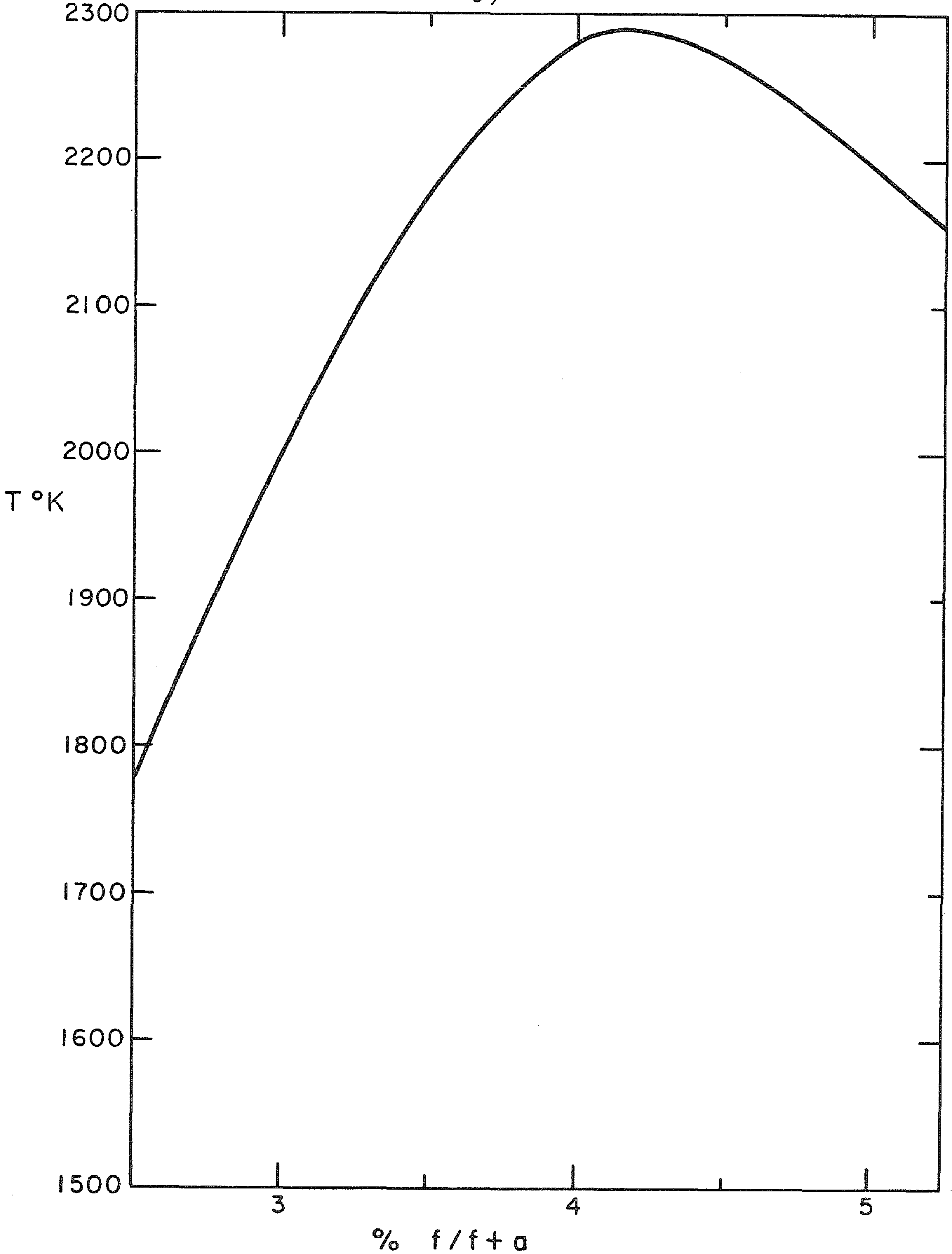

Figure 4. Theoretical adiabatic flame temperature of propane at room temperature and atmospheric pressure. (ref. 26)

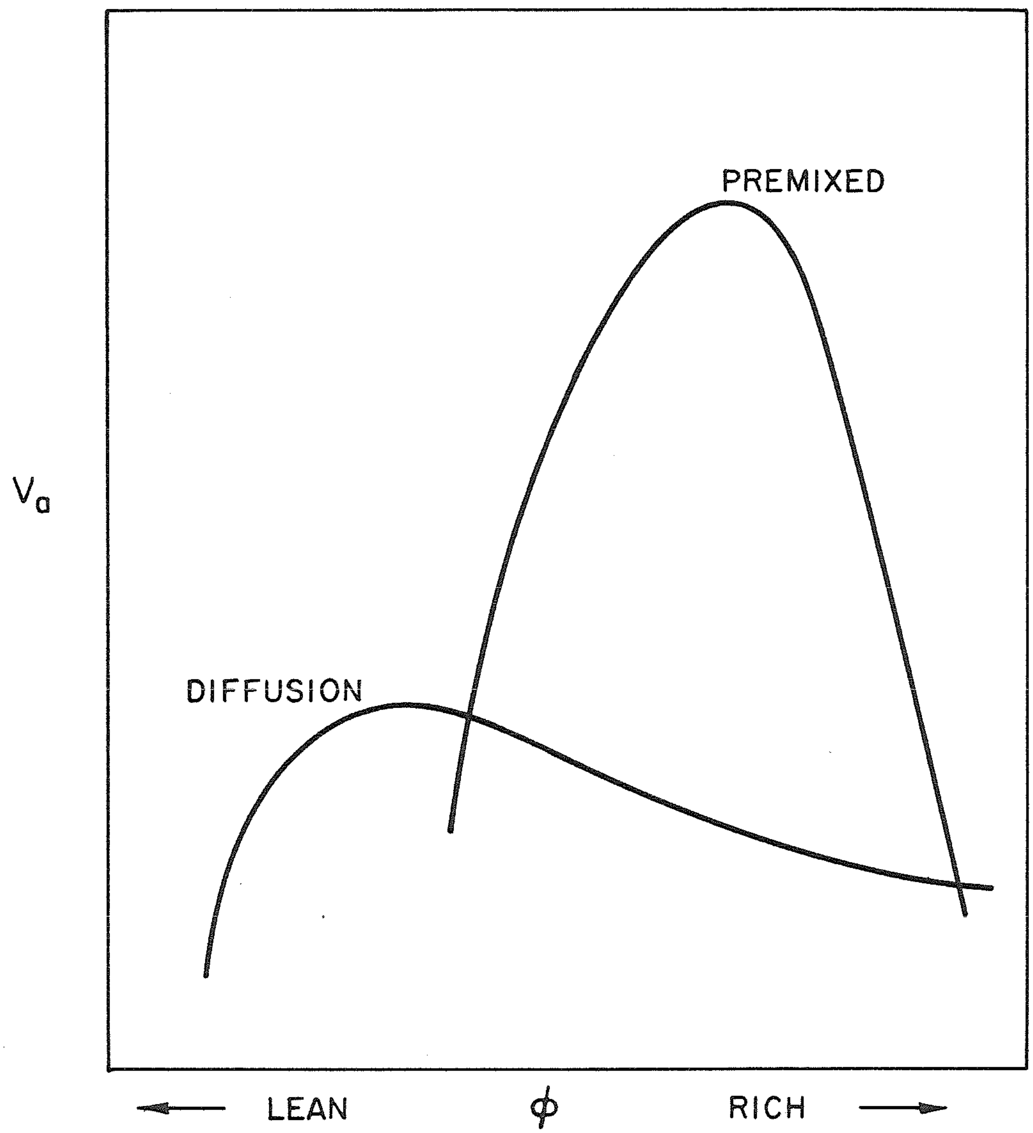

Figure 5. Comparison of v_{bo} curves of premixed and diffusion flames. (ref. 10)

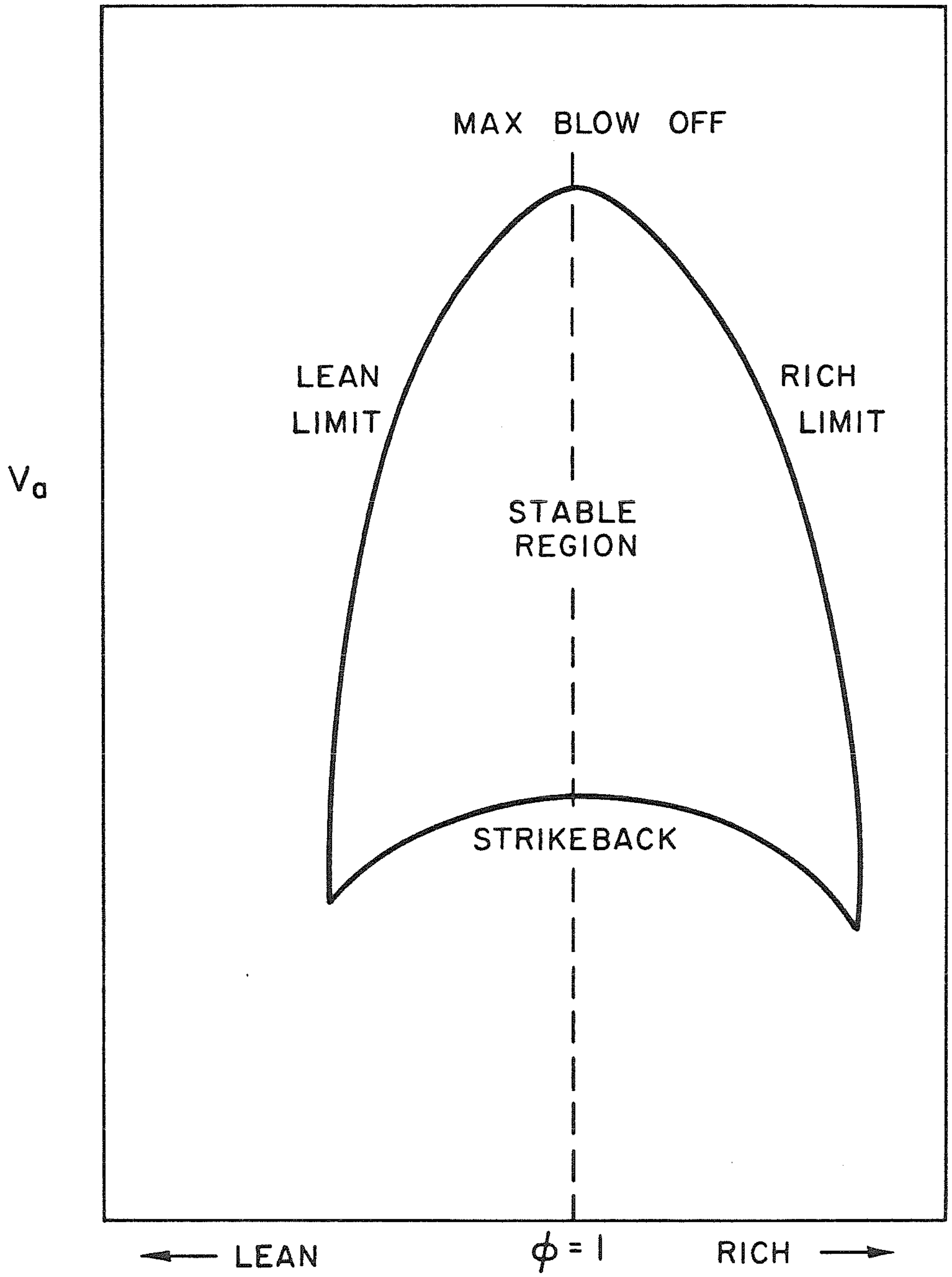

Figure 6. Generalized flame stability curve.

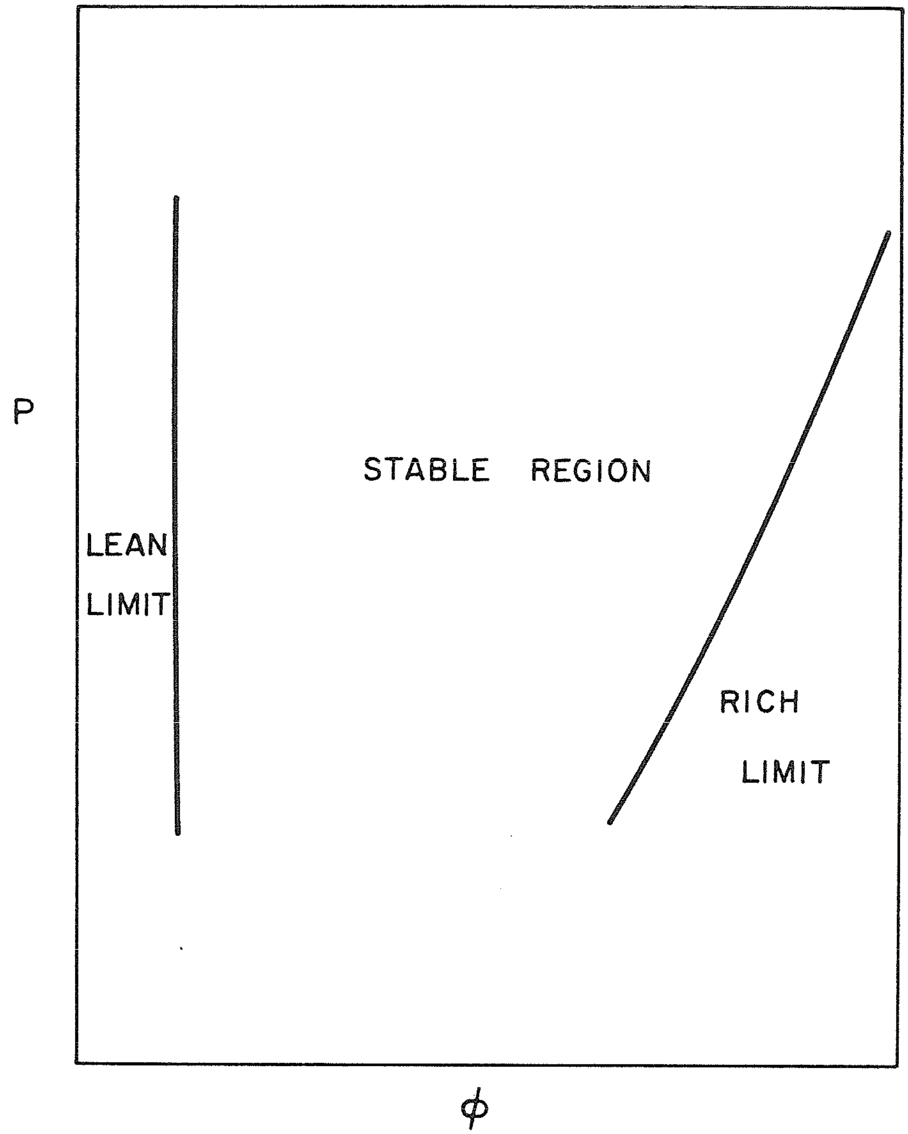

Figure 7. Effect of pressure on flame stability.

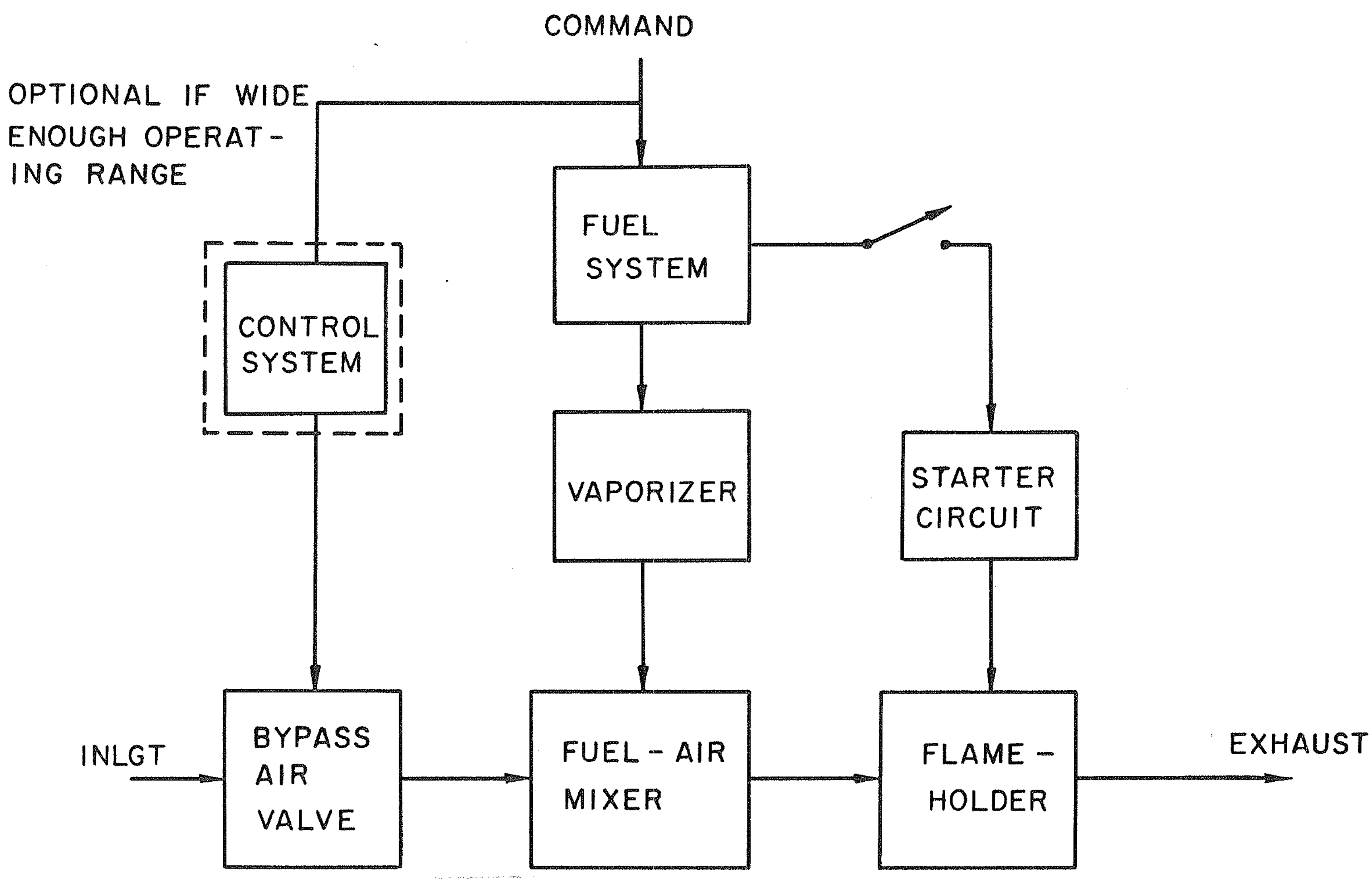

Figure 8. Block diagram of premixed lean combustor components.

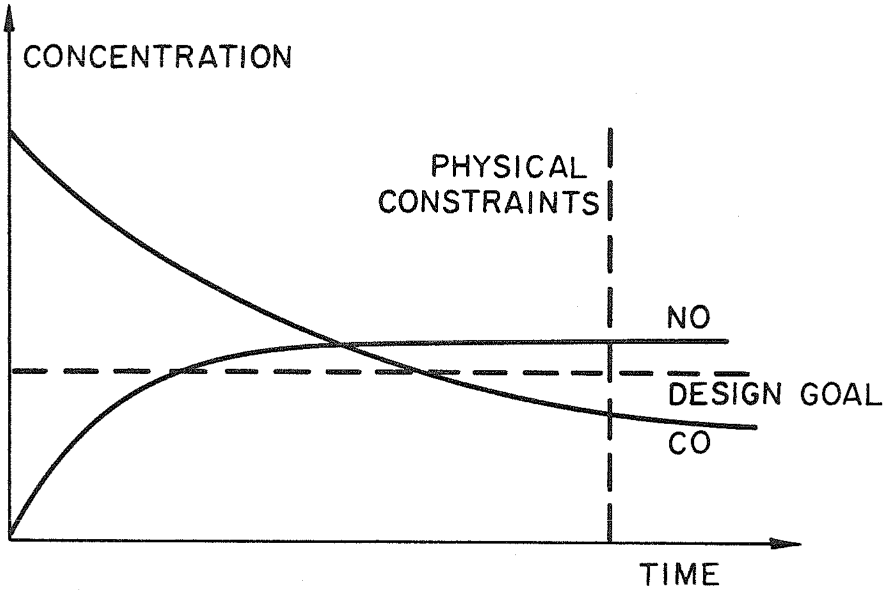

Figure 9. Residence time and combustor size.

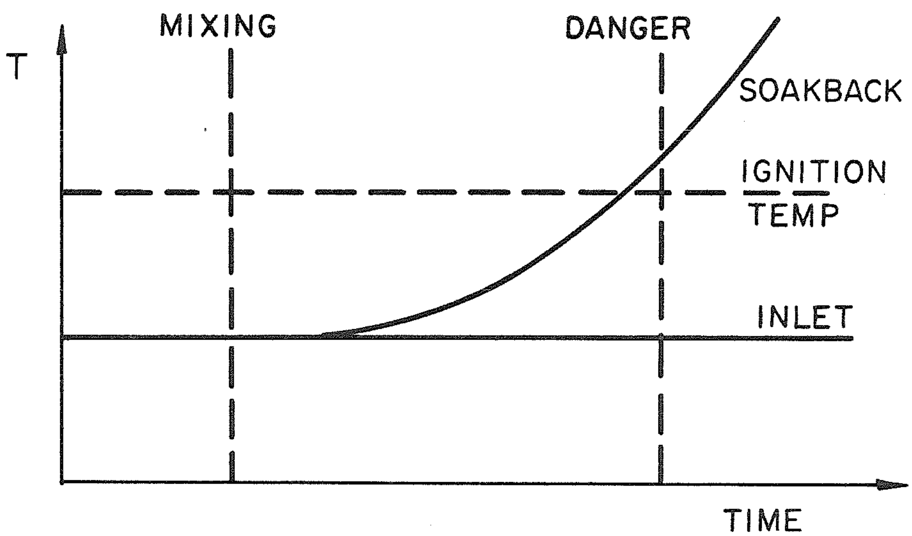

Figure 10. Combustor temperature relationship.

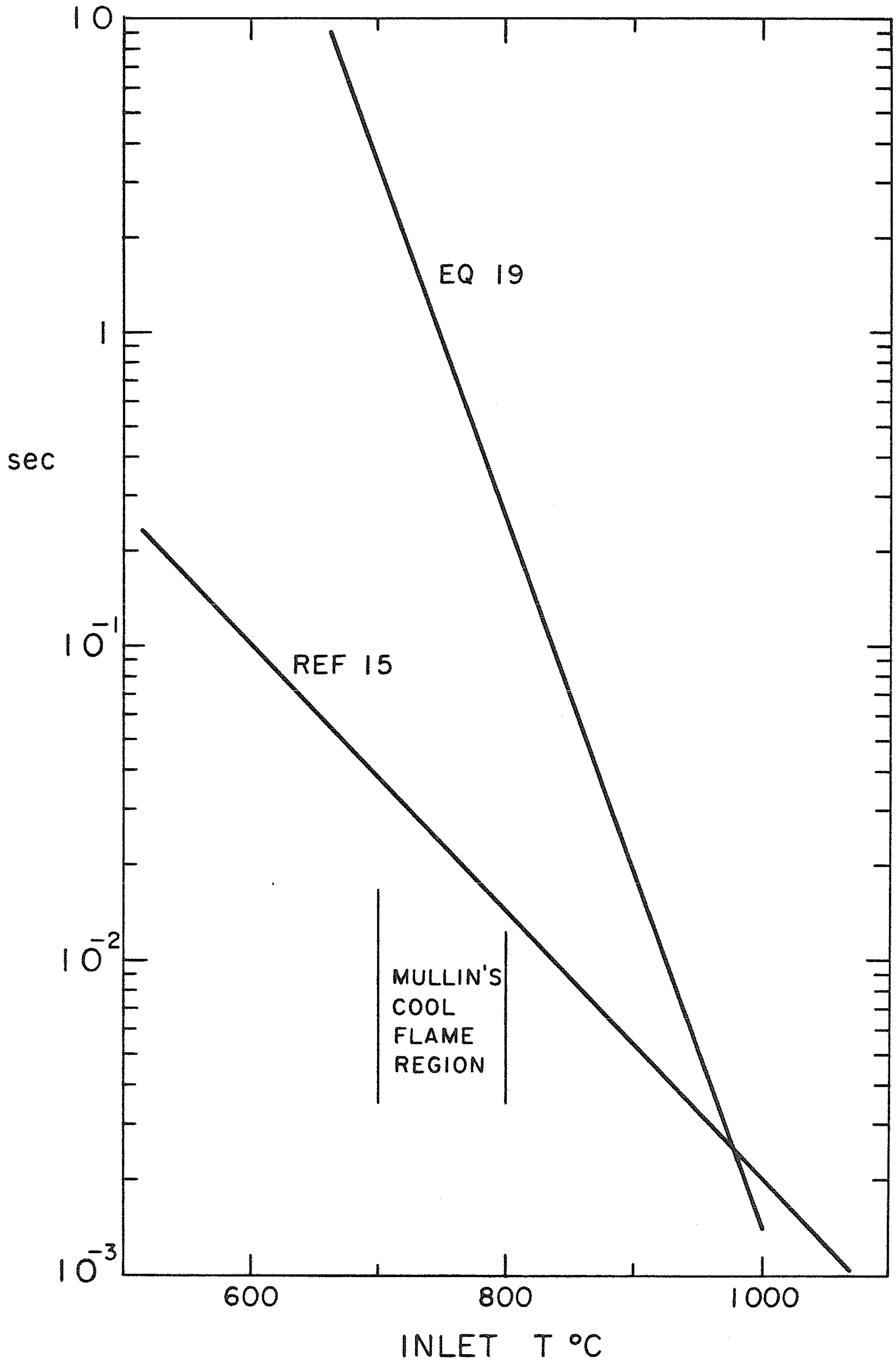

Figure 11. Comparison of t* for prevaporized and droplet diffusion mixing.

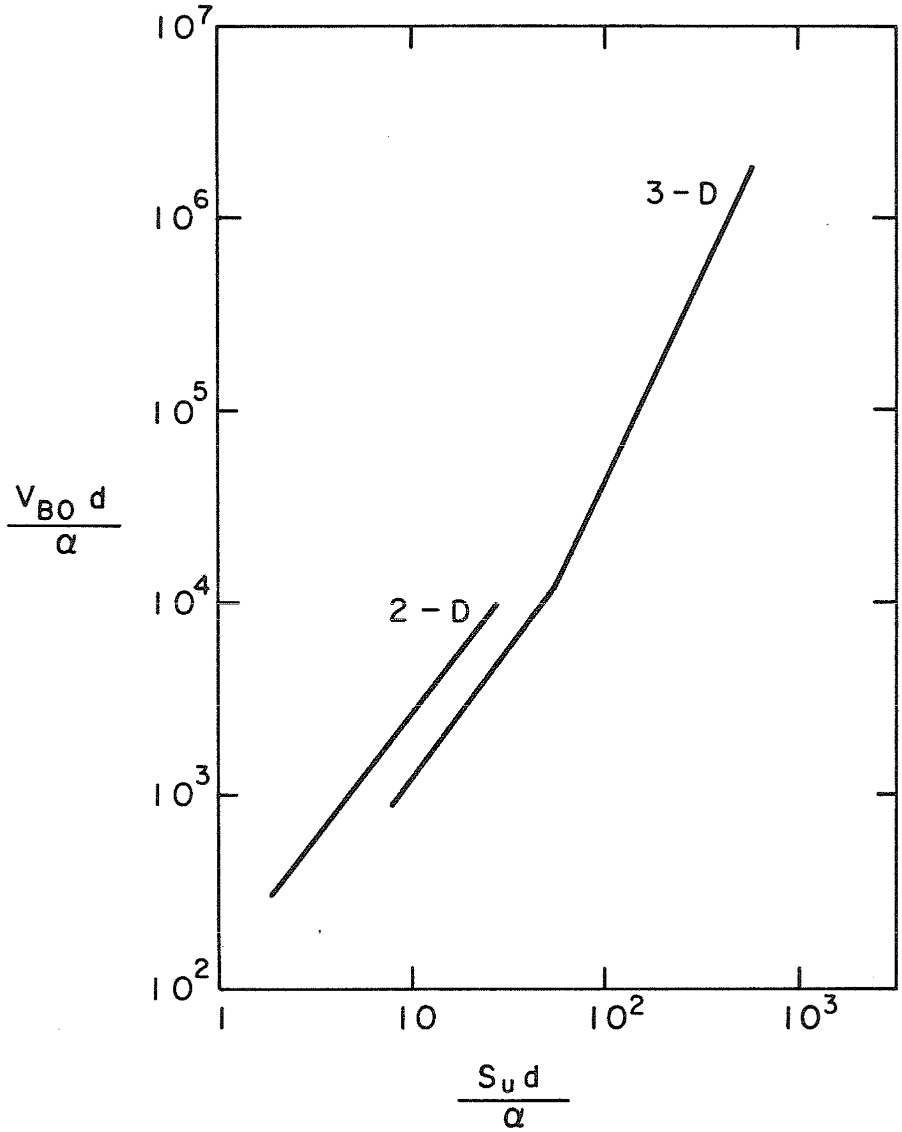

Figure 12. Bluff body v_{bo} correlation.

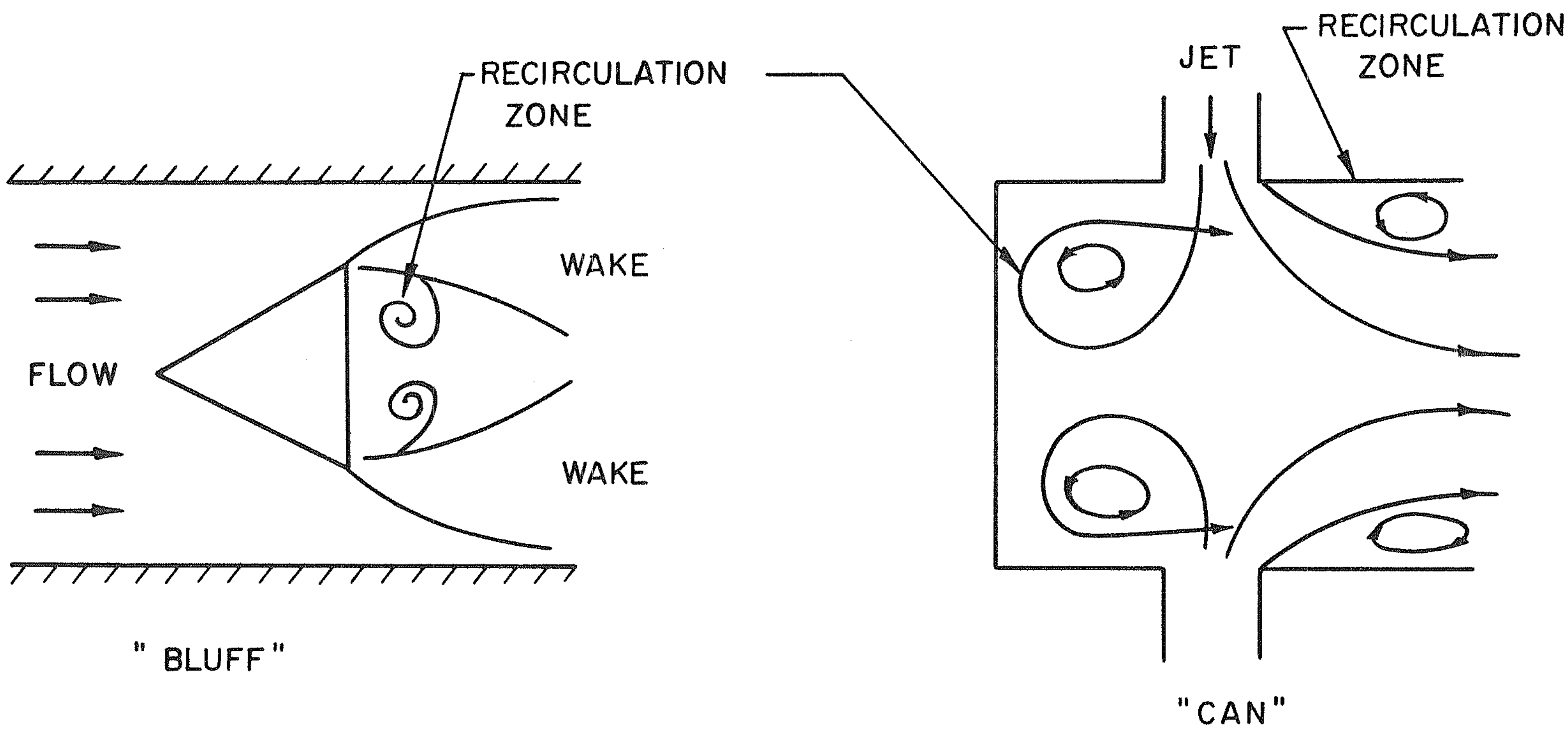

Figure 13. Bluff body and can type flameholders.

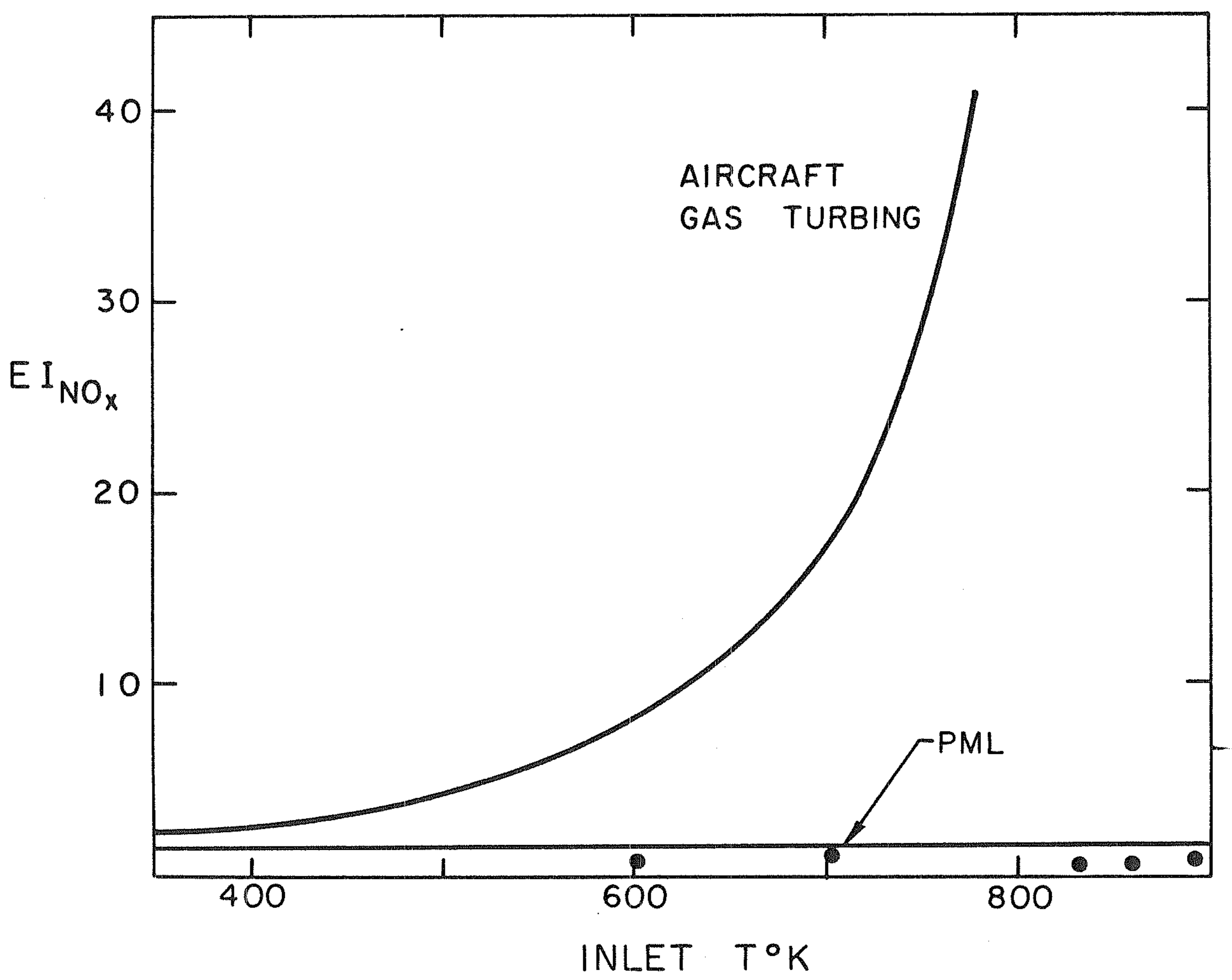

Figure 14. Effect of combustor inlet temperature on EI_{NO_x}.

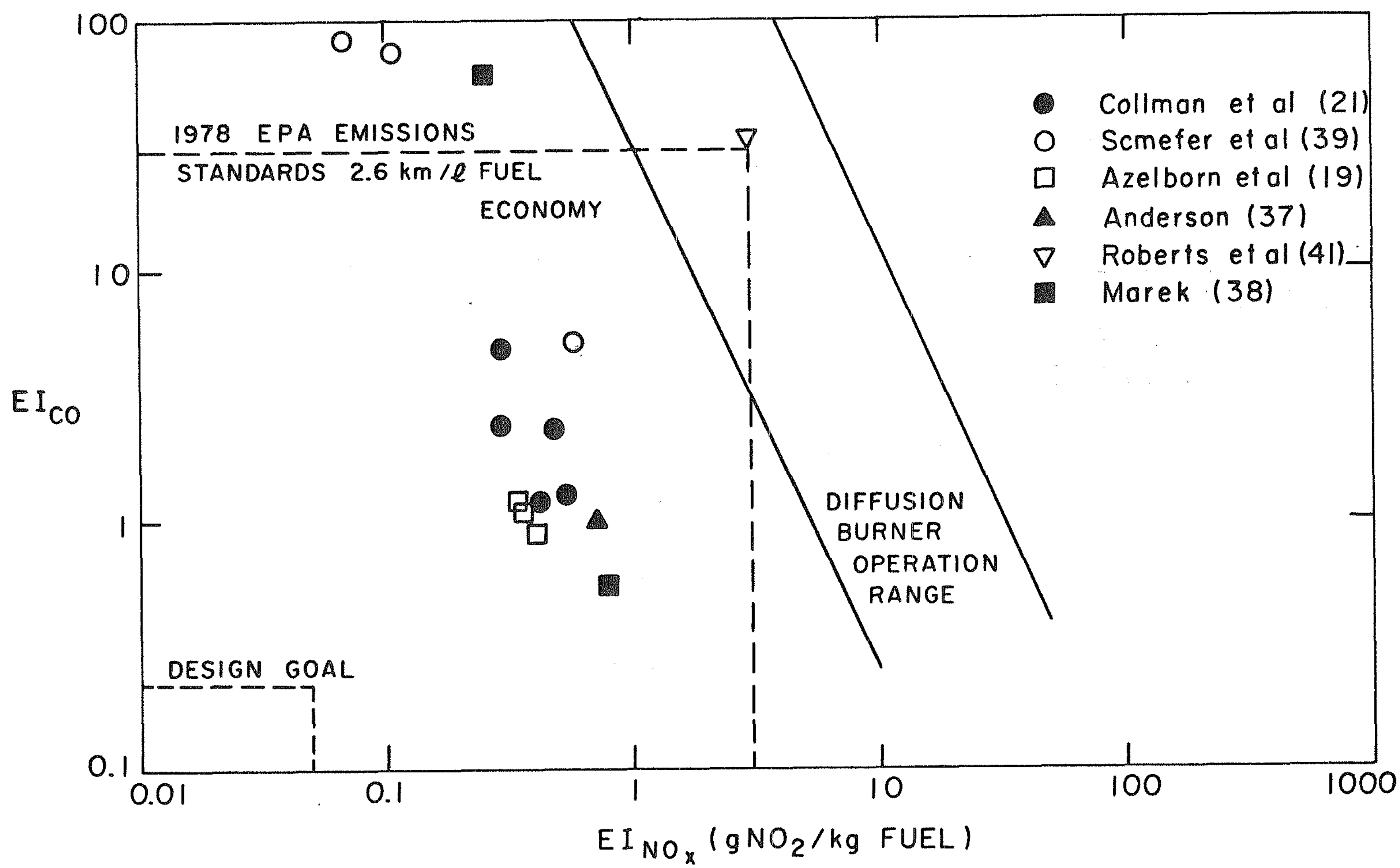

Figure 15. Emissions performance of burners used in previous research.

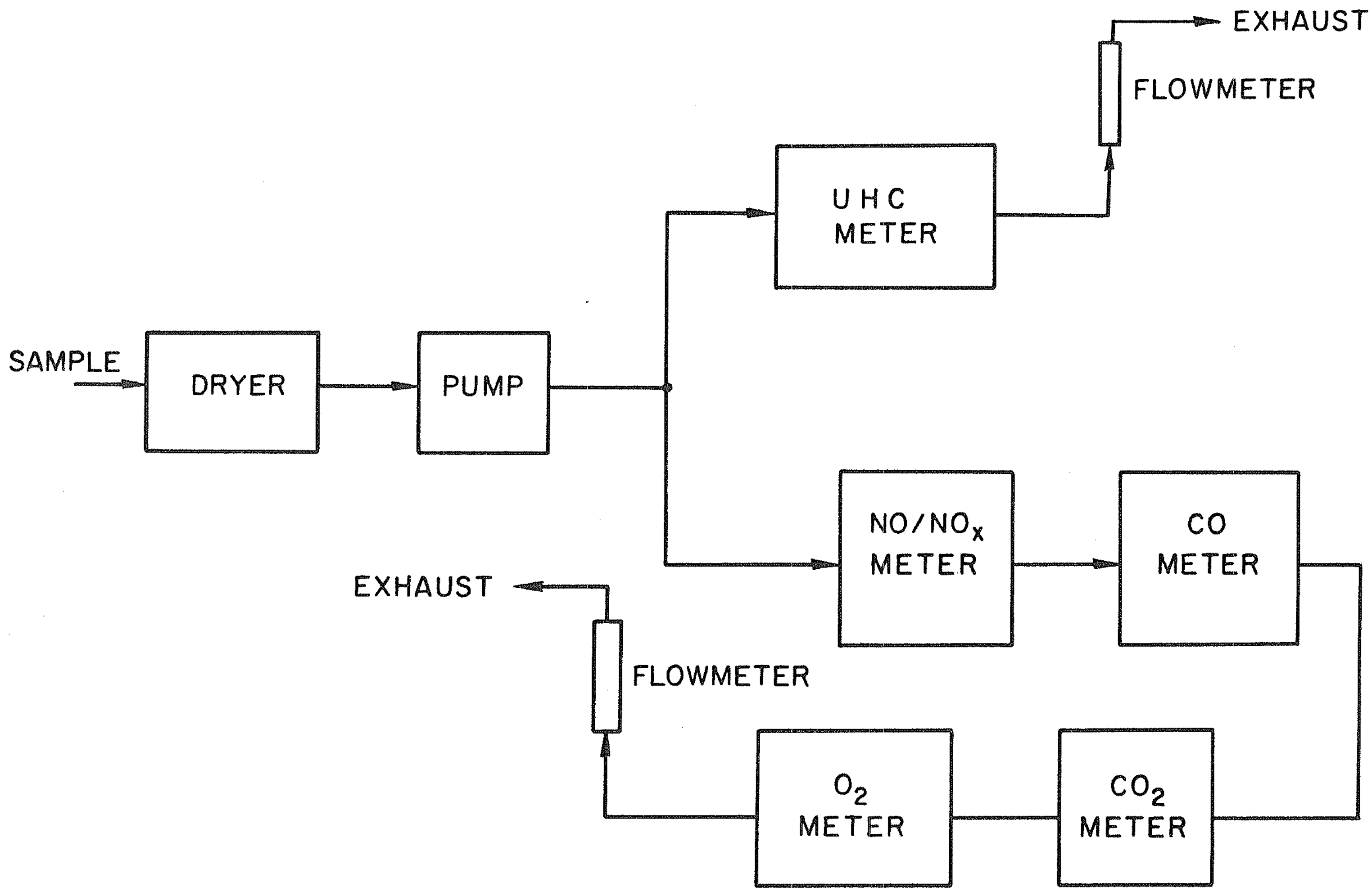

Figure 16. Instrument train.

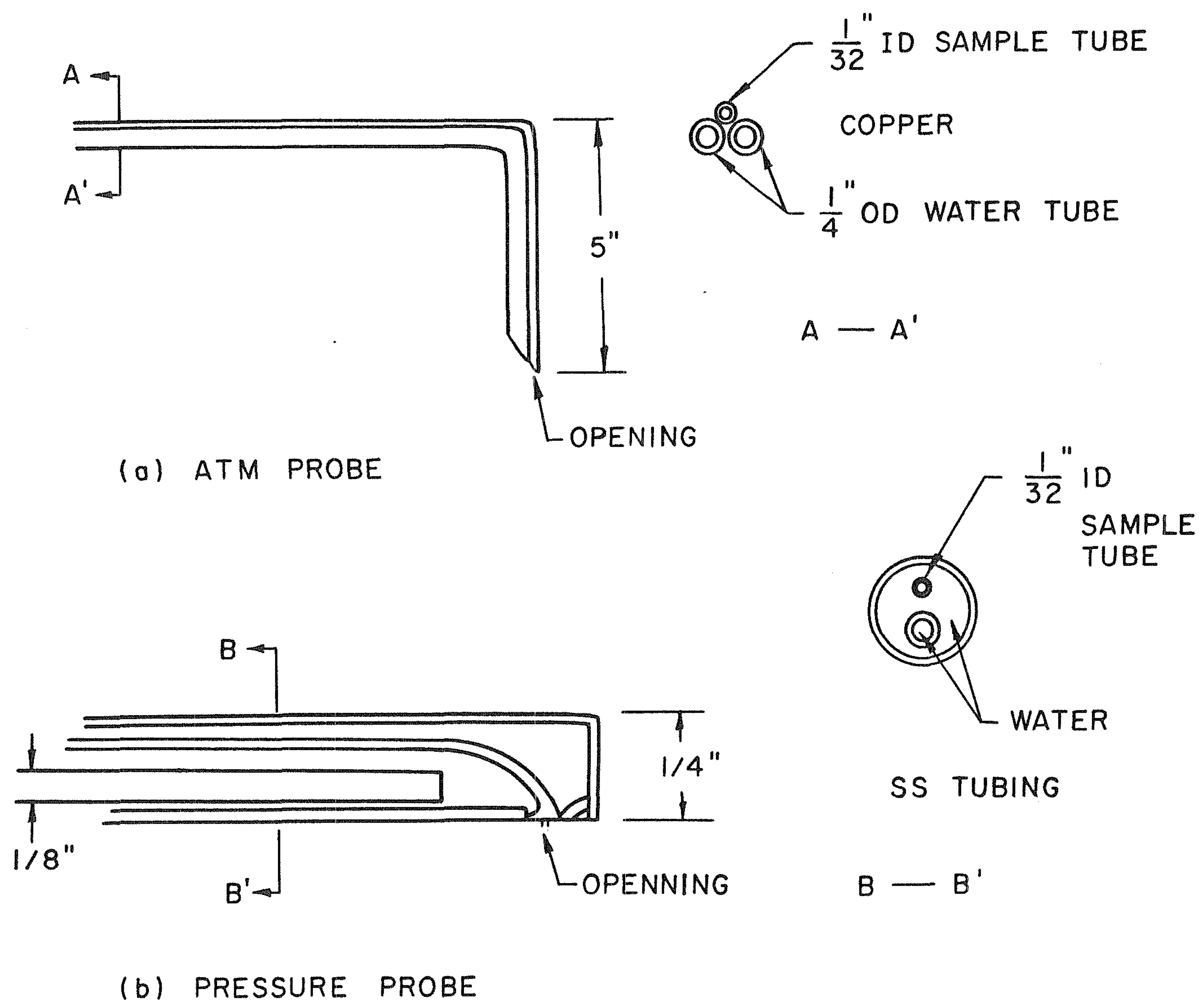

Figure 17. Gas sampling probes.

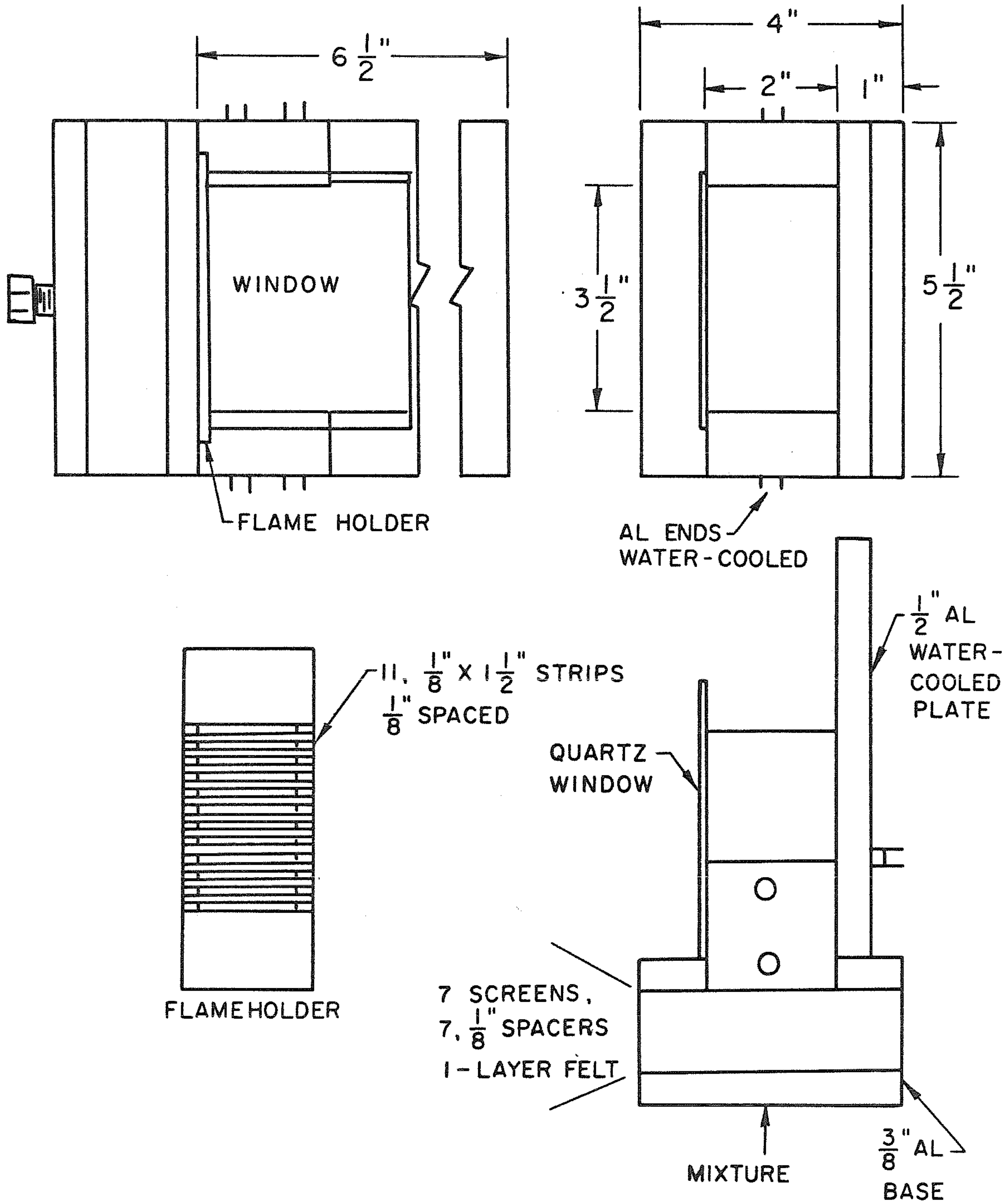

Figure 18. Bluff body flameholder burner.

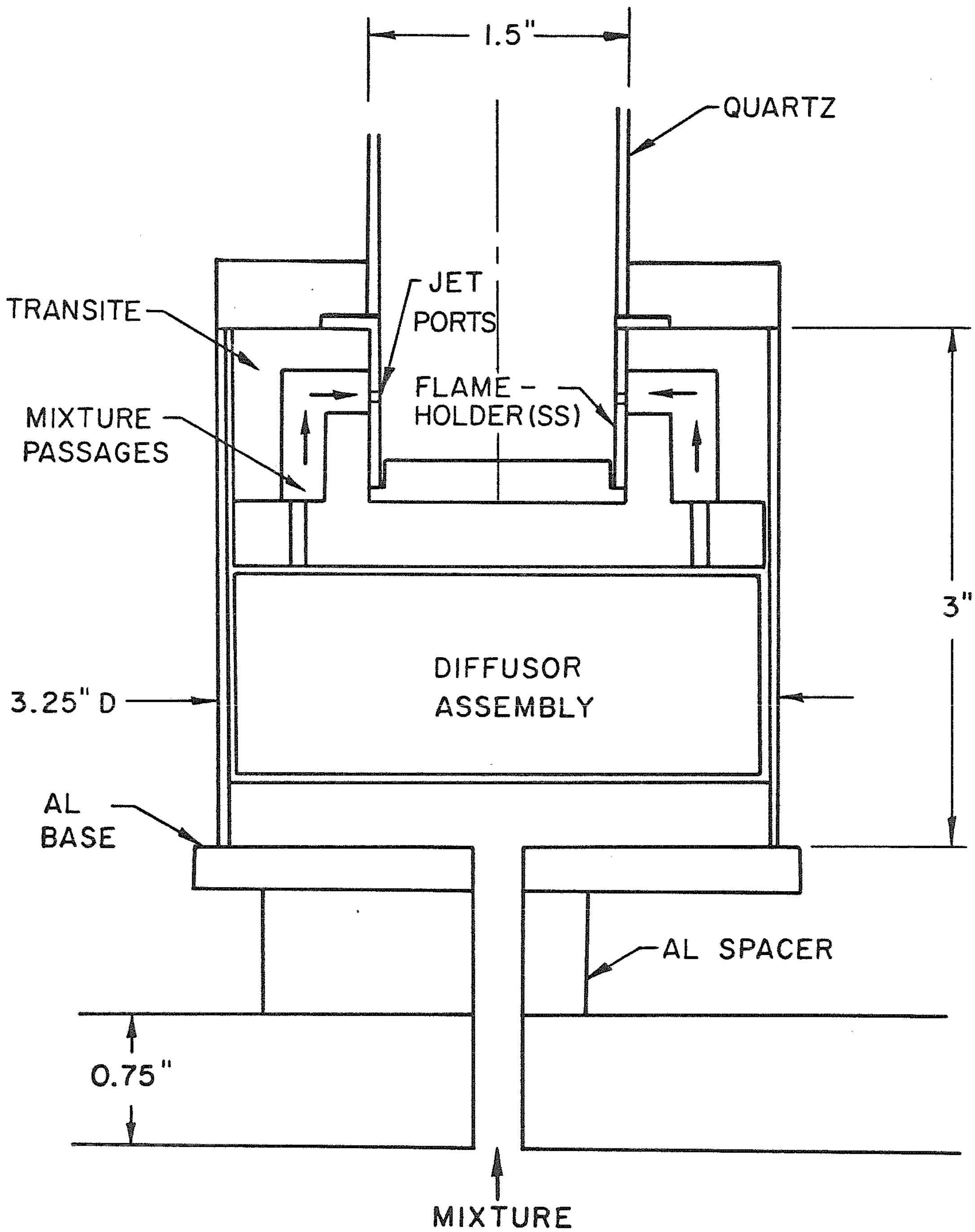

Figure 19. Can type burner. (cross-sectional view)

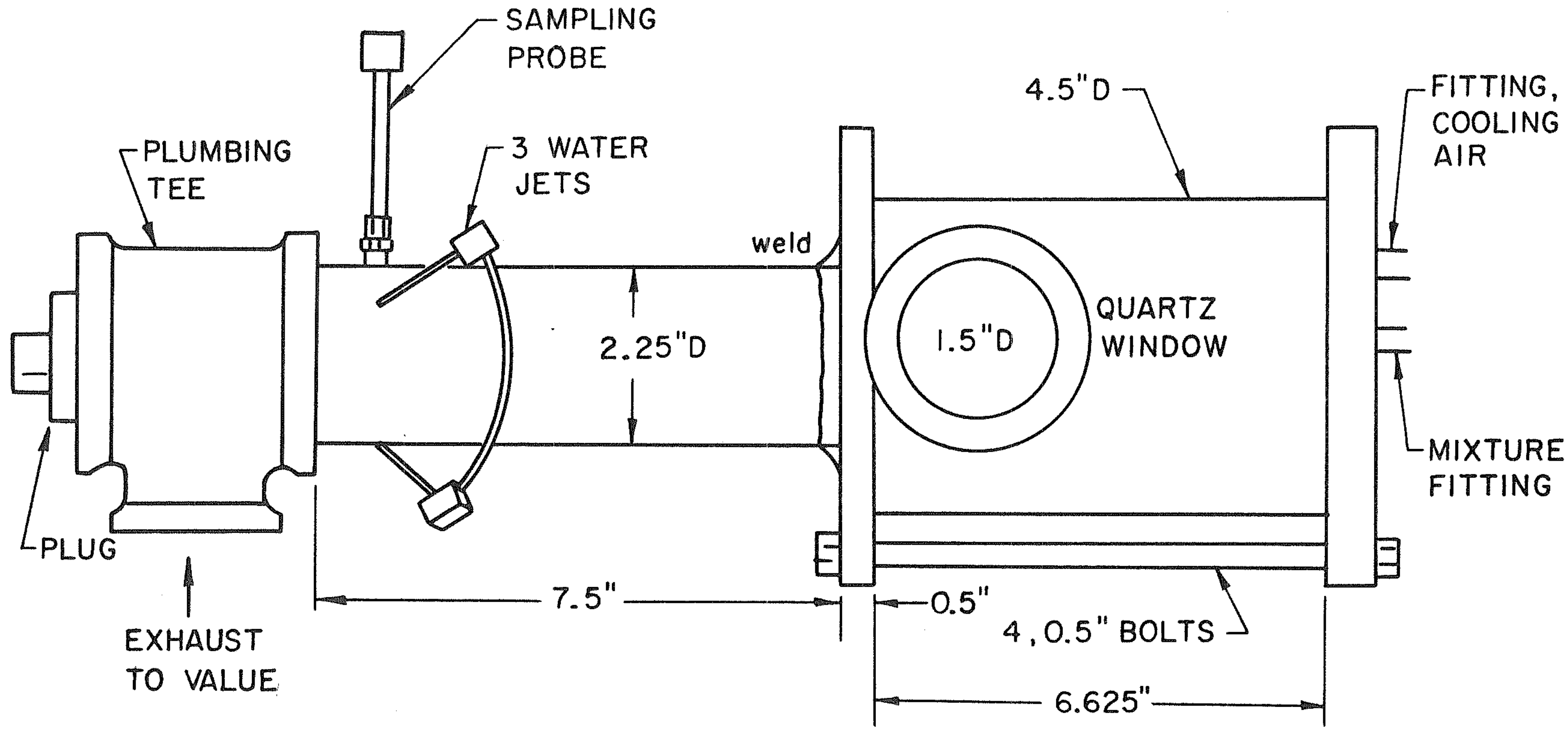

Figure 20. Pressure cell.

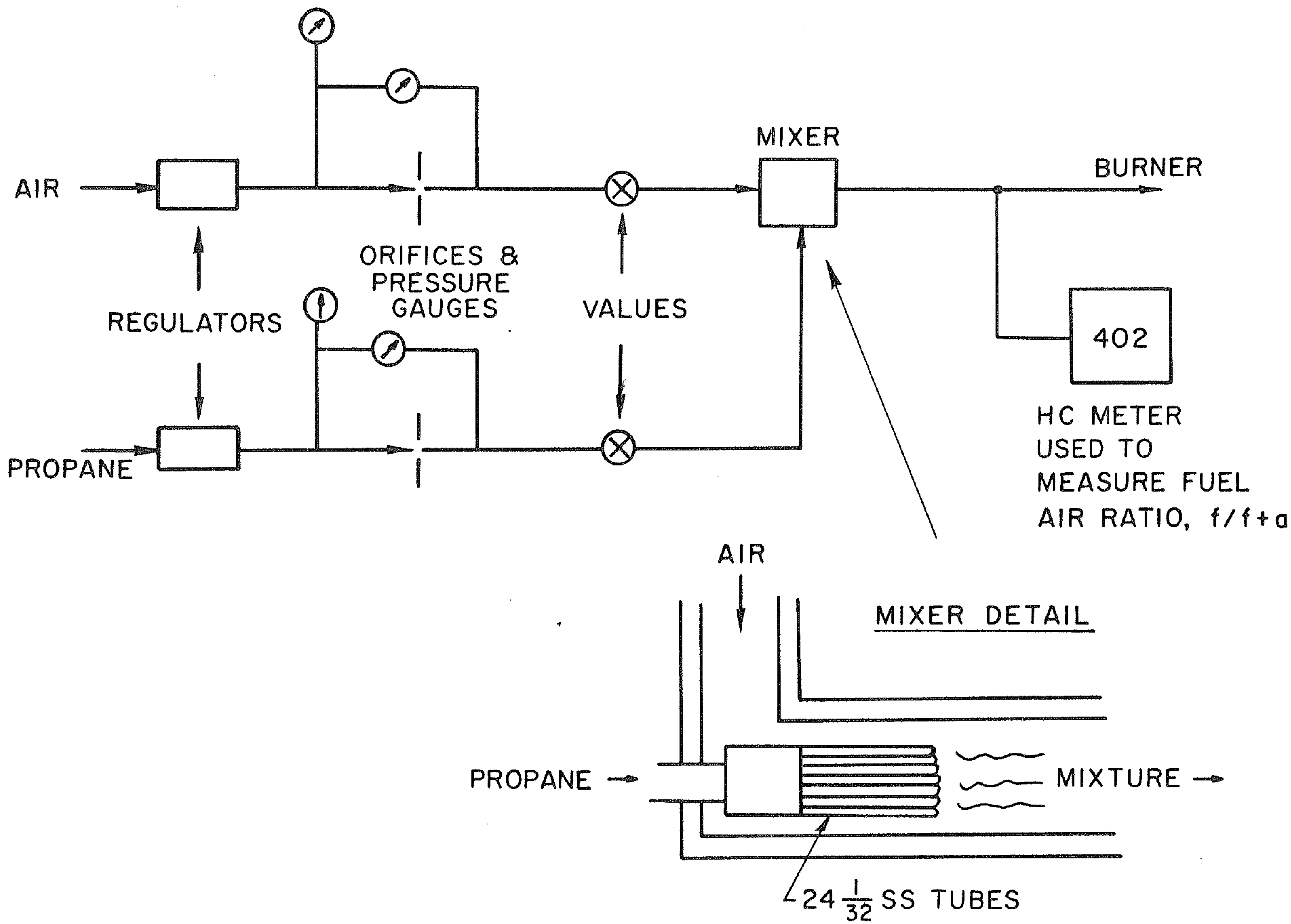

Figure 21. Fuel-air mixture supply system.

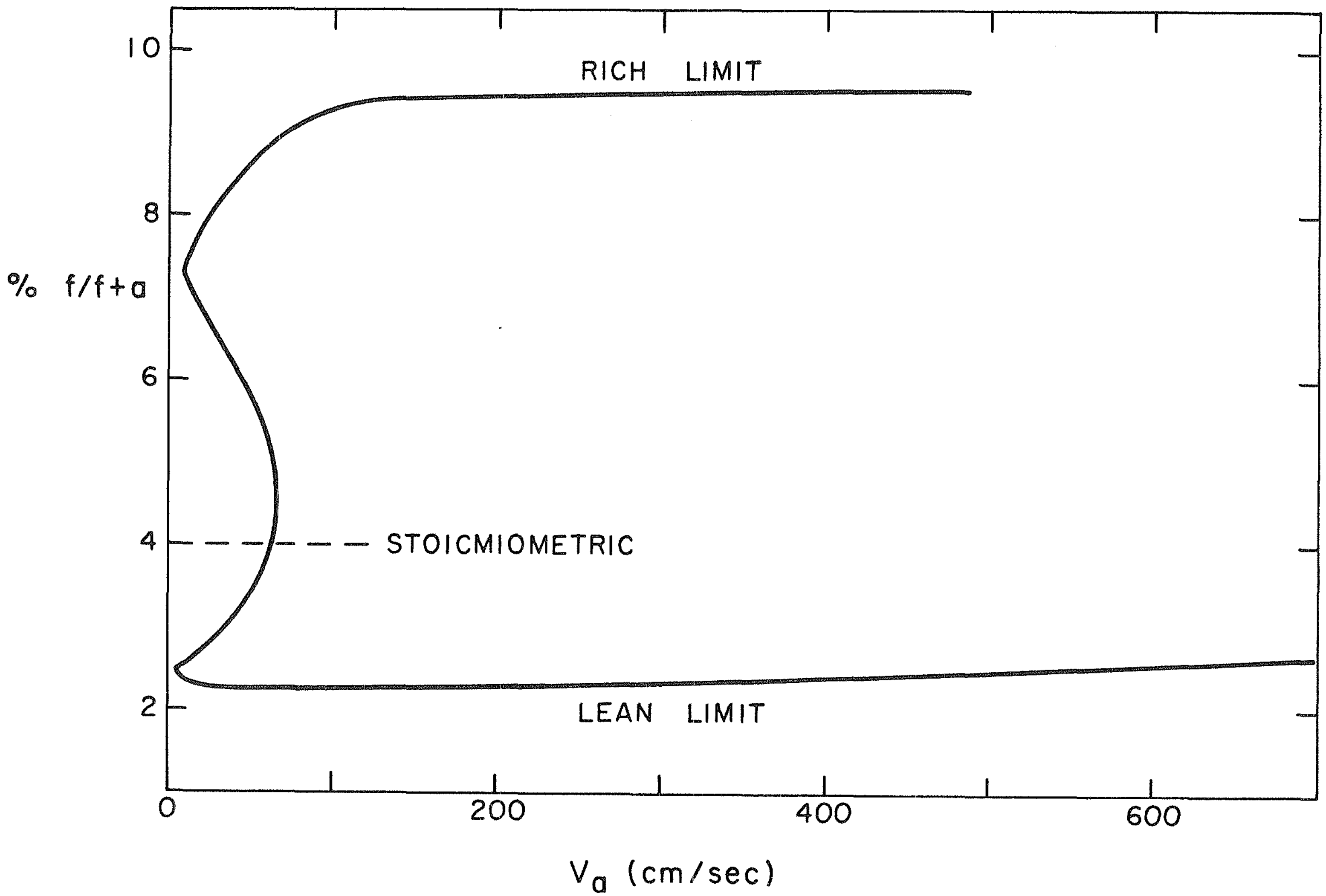

Figure 22. Stability curve of bluff body flameholder.

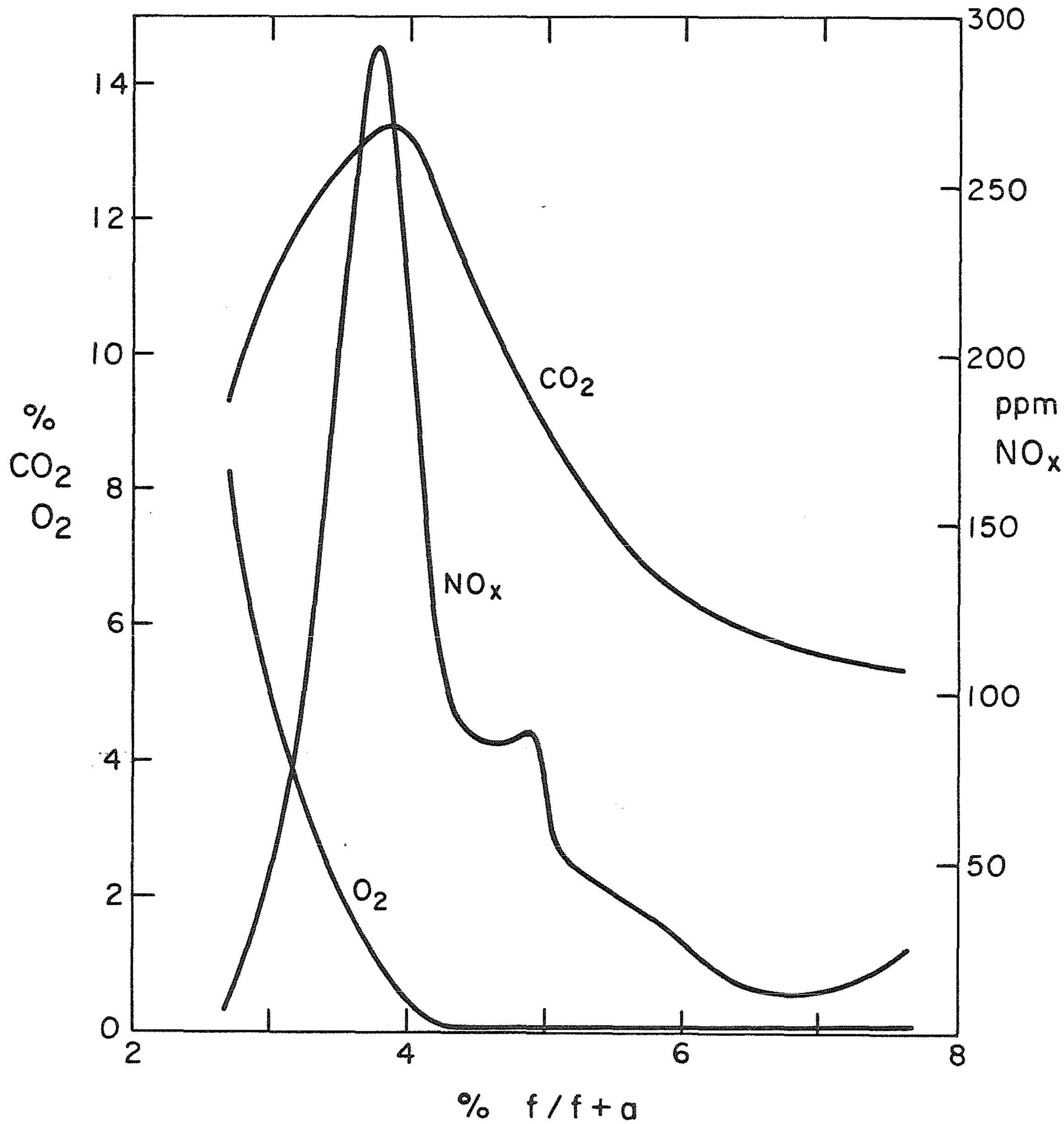

Figure 23a. Combustion product profile for bluff body flameholder, NO_x, CO_2, and O_2. v_a=160 cm/sec.

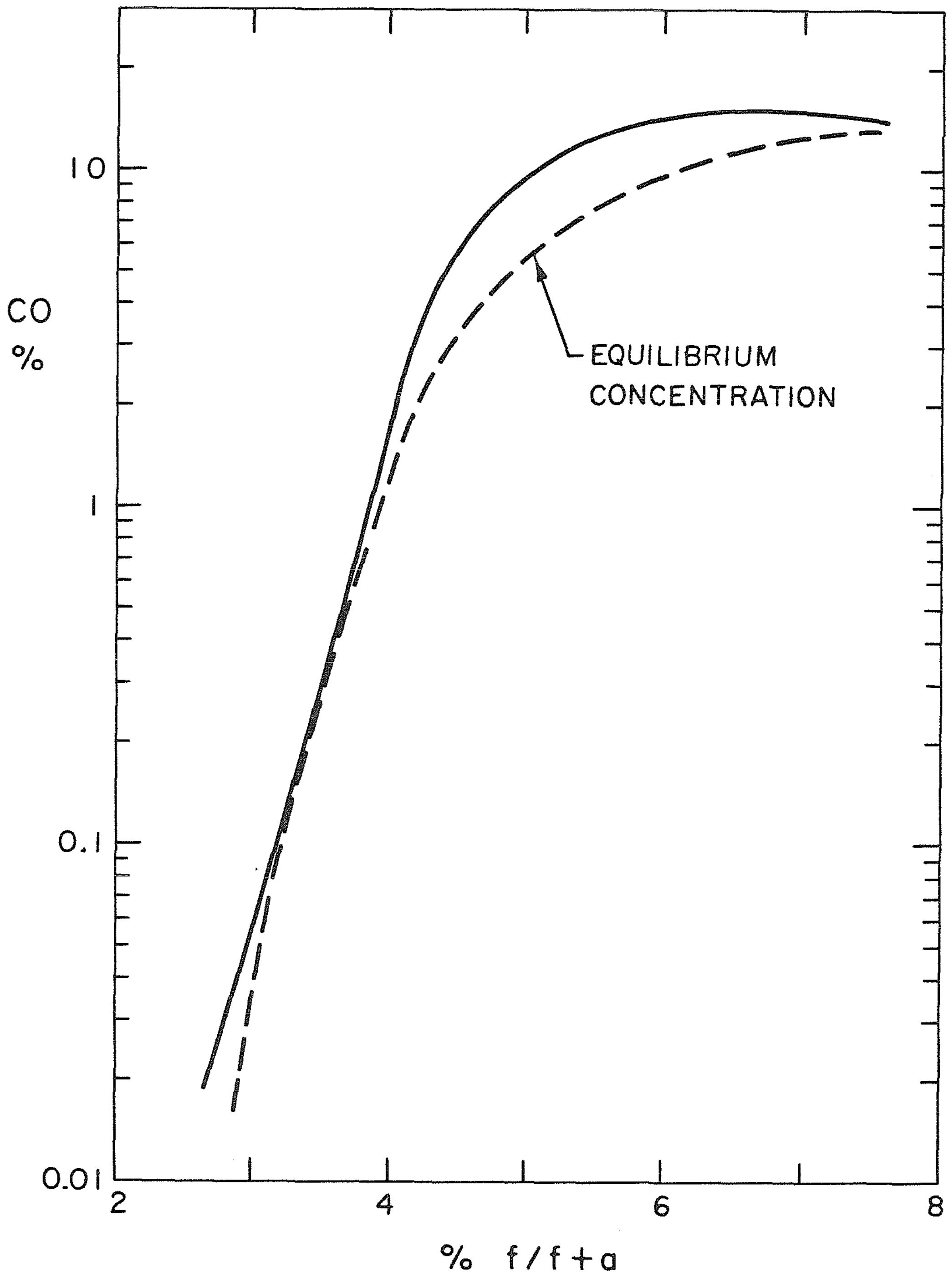

Figure 23b. Combustion product profile for bluff body flameholder, CO. v_a=160 cm/sec.

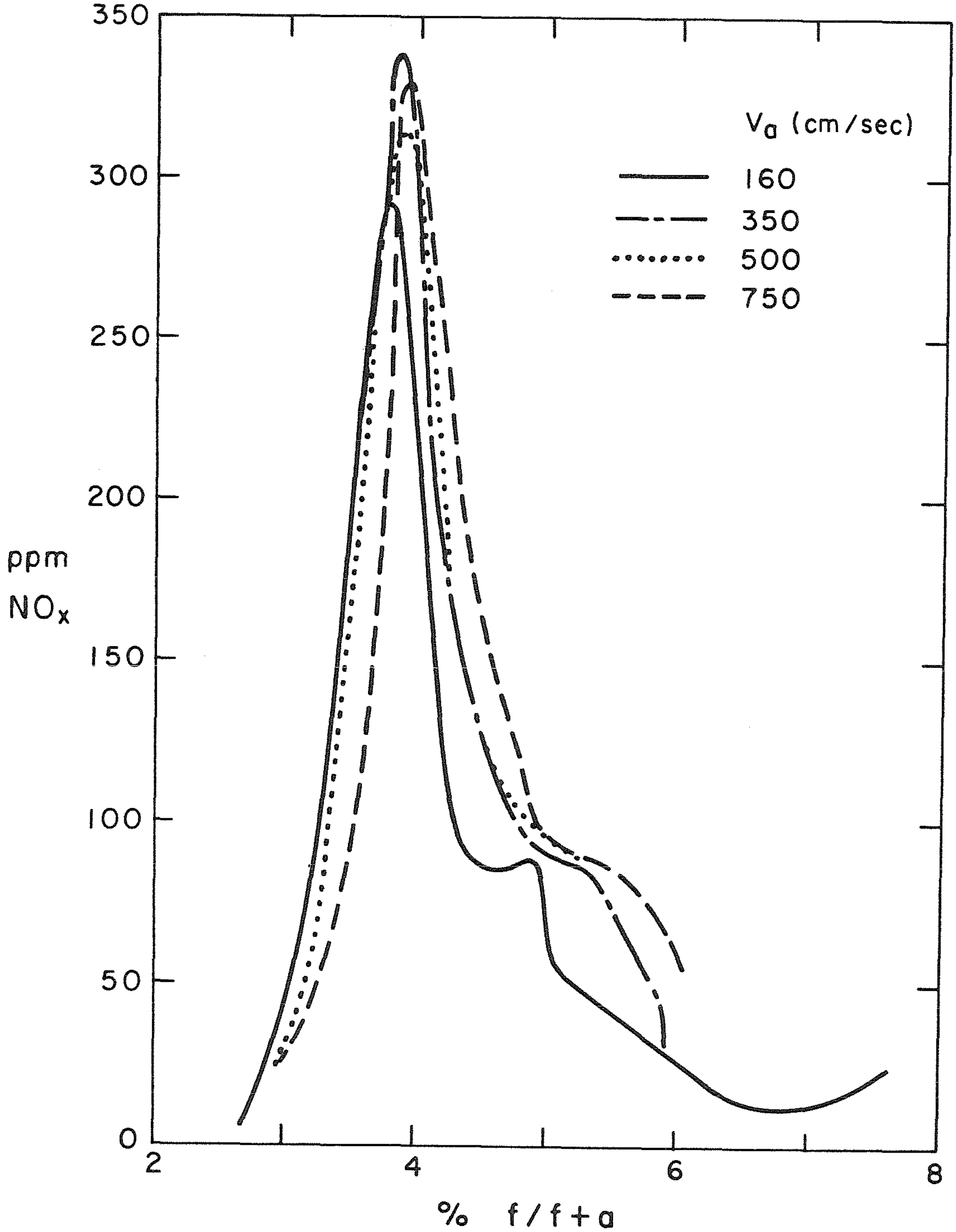

Figure 24. Comparison of NO_x curves for bluff body flameholder.

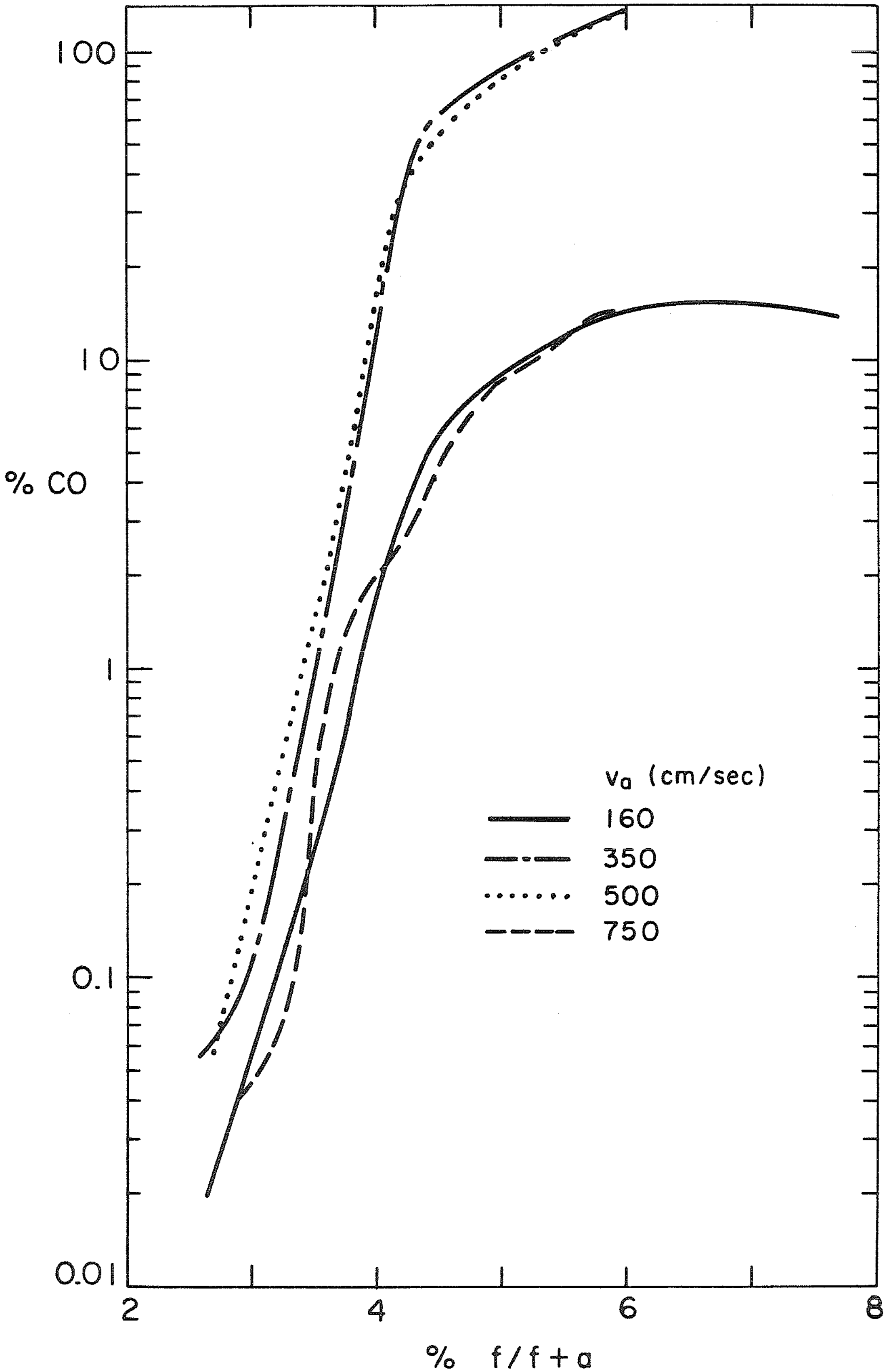

Figure 25. Comparison of CO curves for bluff body flameholder.

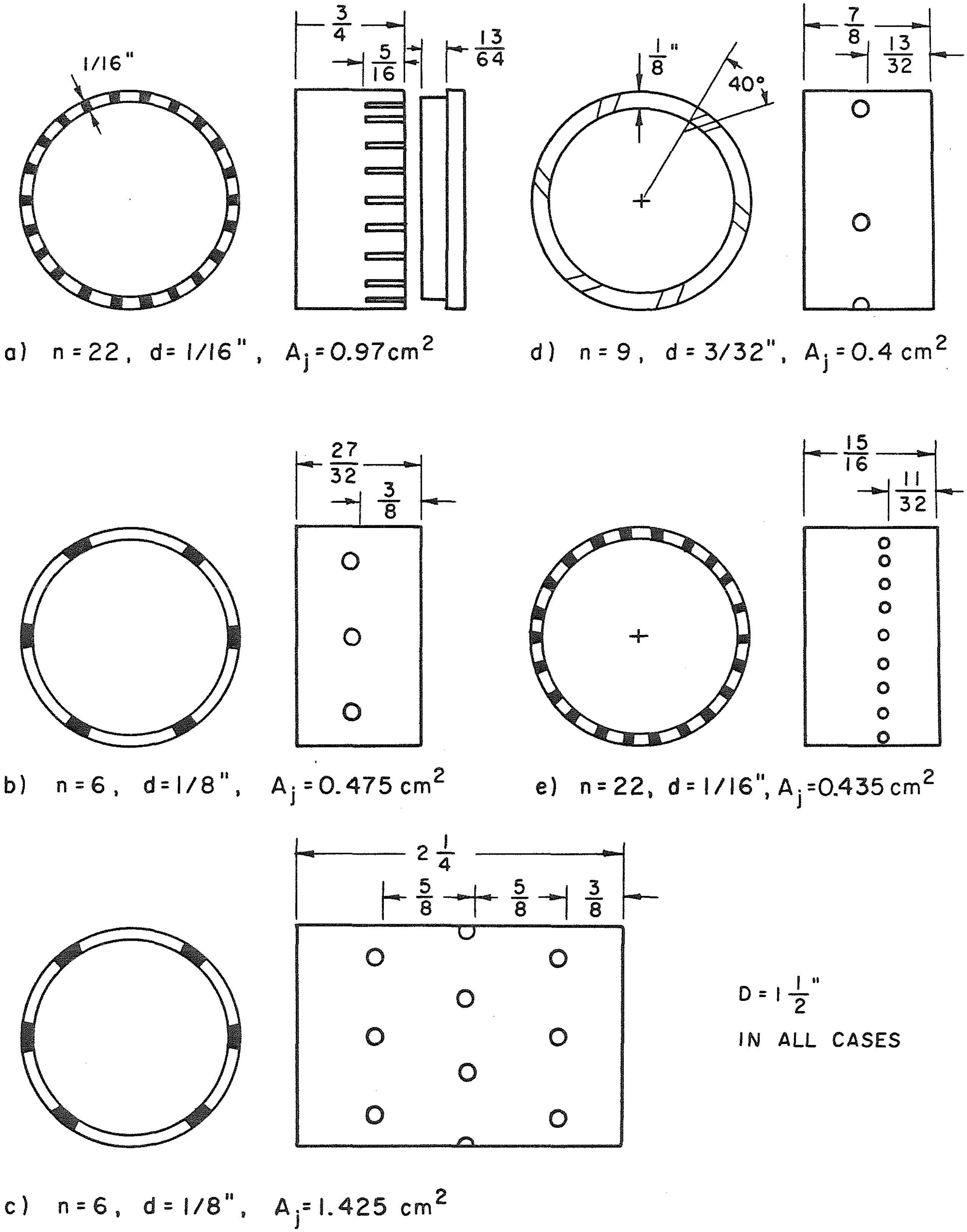

Figure 26. Can type flameholders used in measurements.

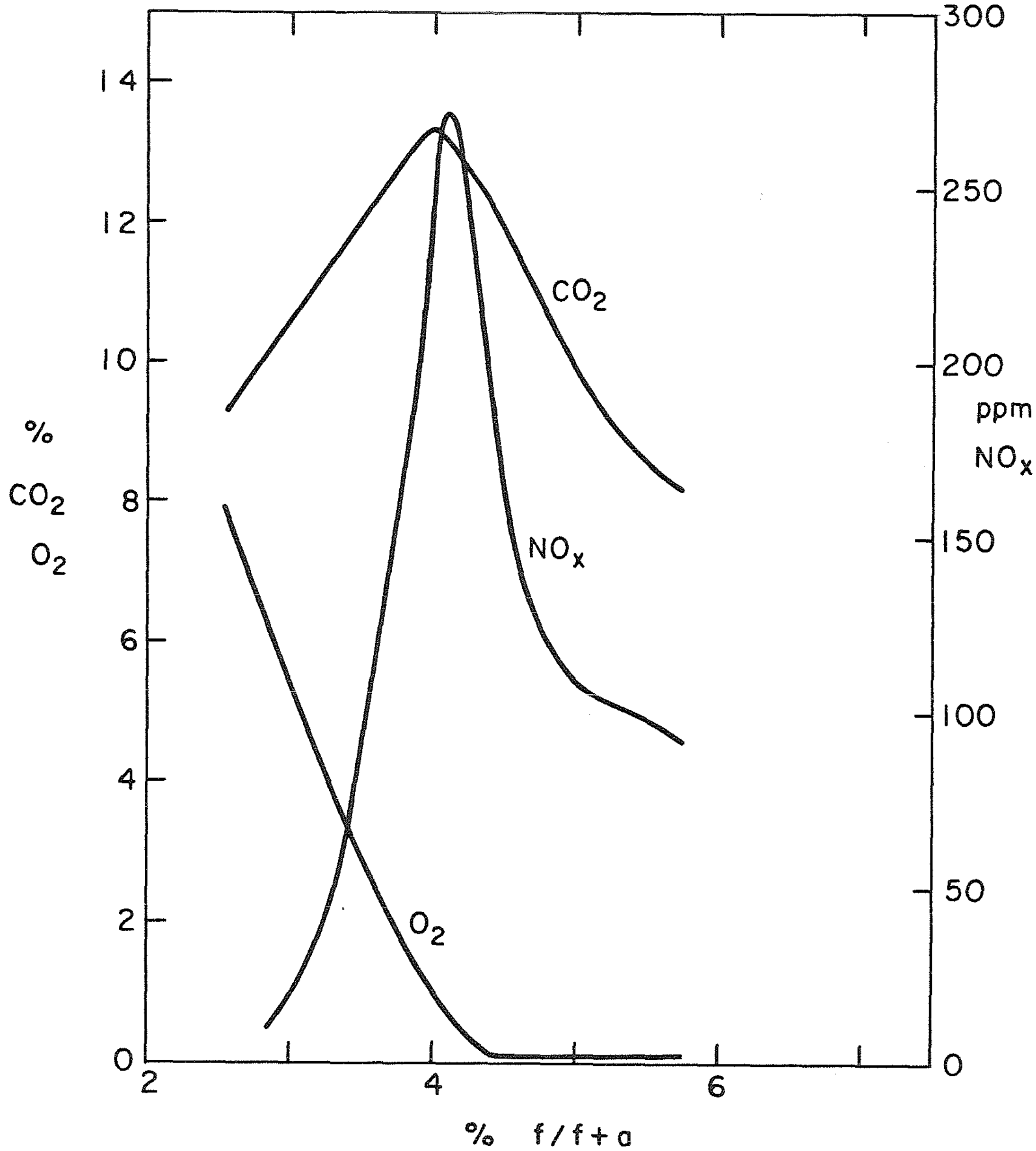

Figure 27a. Combustion product profile for slot flameholder, NO_x, CO_2, and O_2. v_j=500 cm/sec.

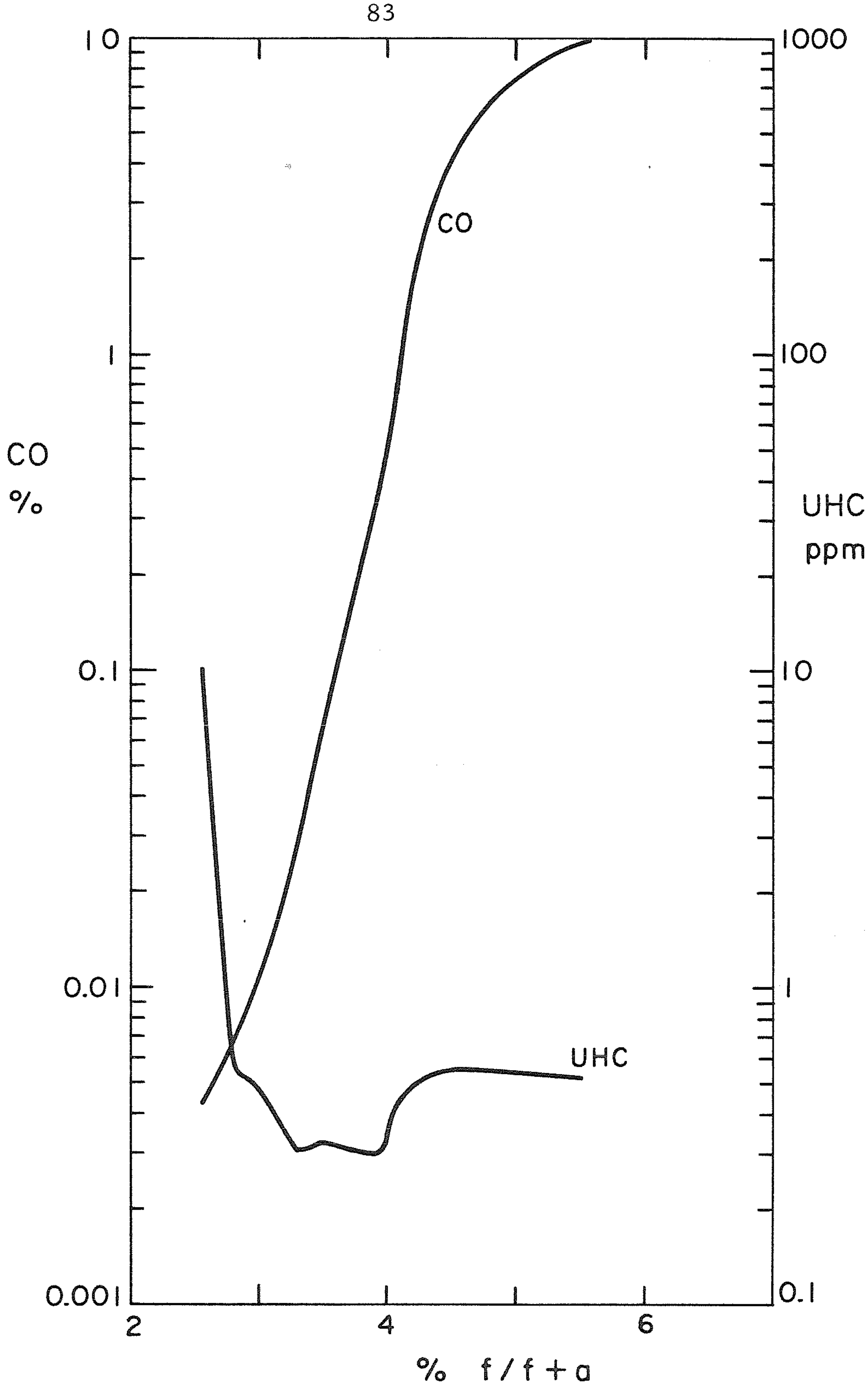

Figure 27b. Combustion product profile for slot flameholder, CO. v_j=500 cm/sec.

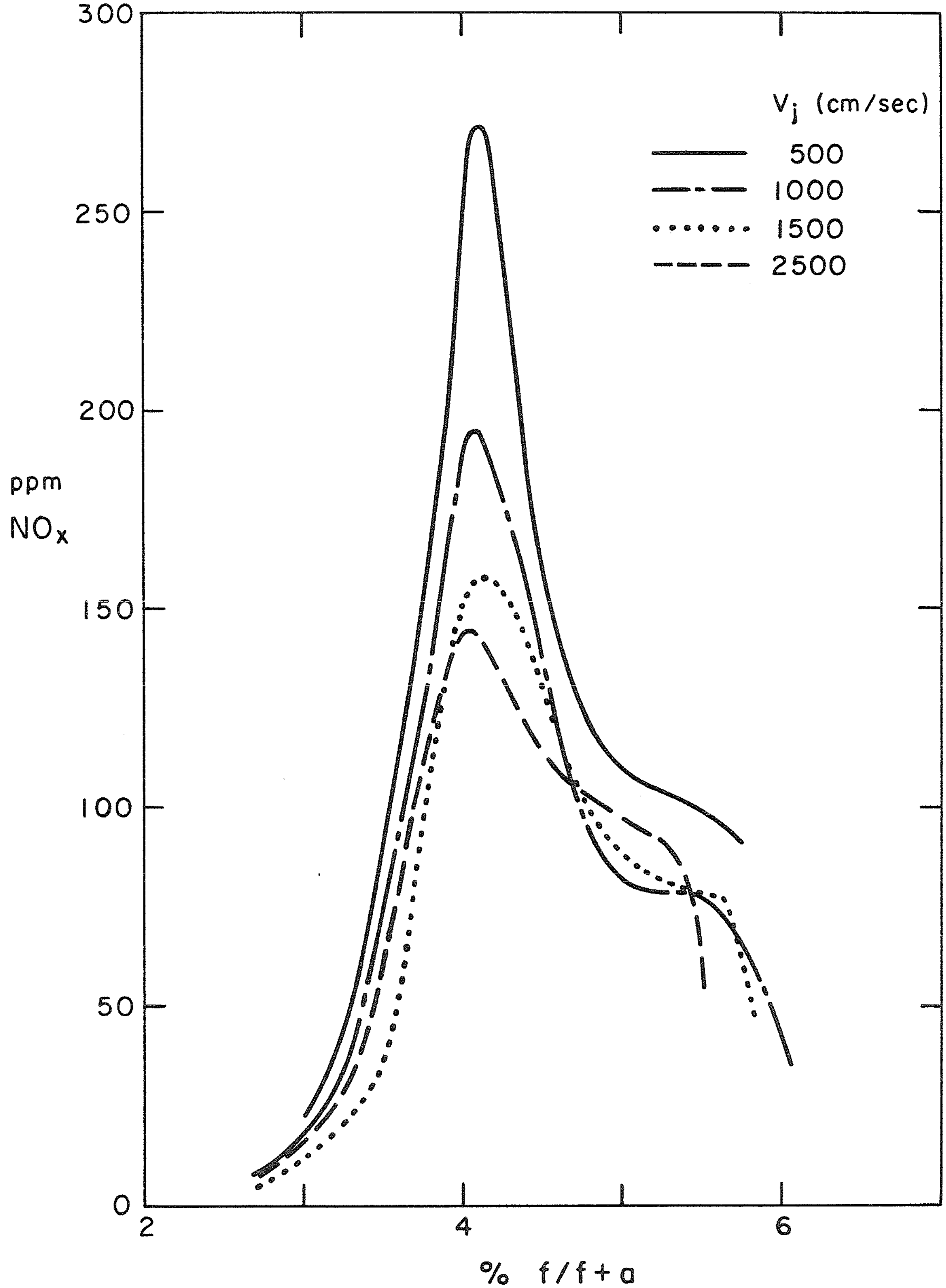

Figure 28. Comparison of NO_x curves for slot flameholder.

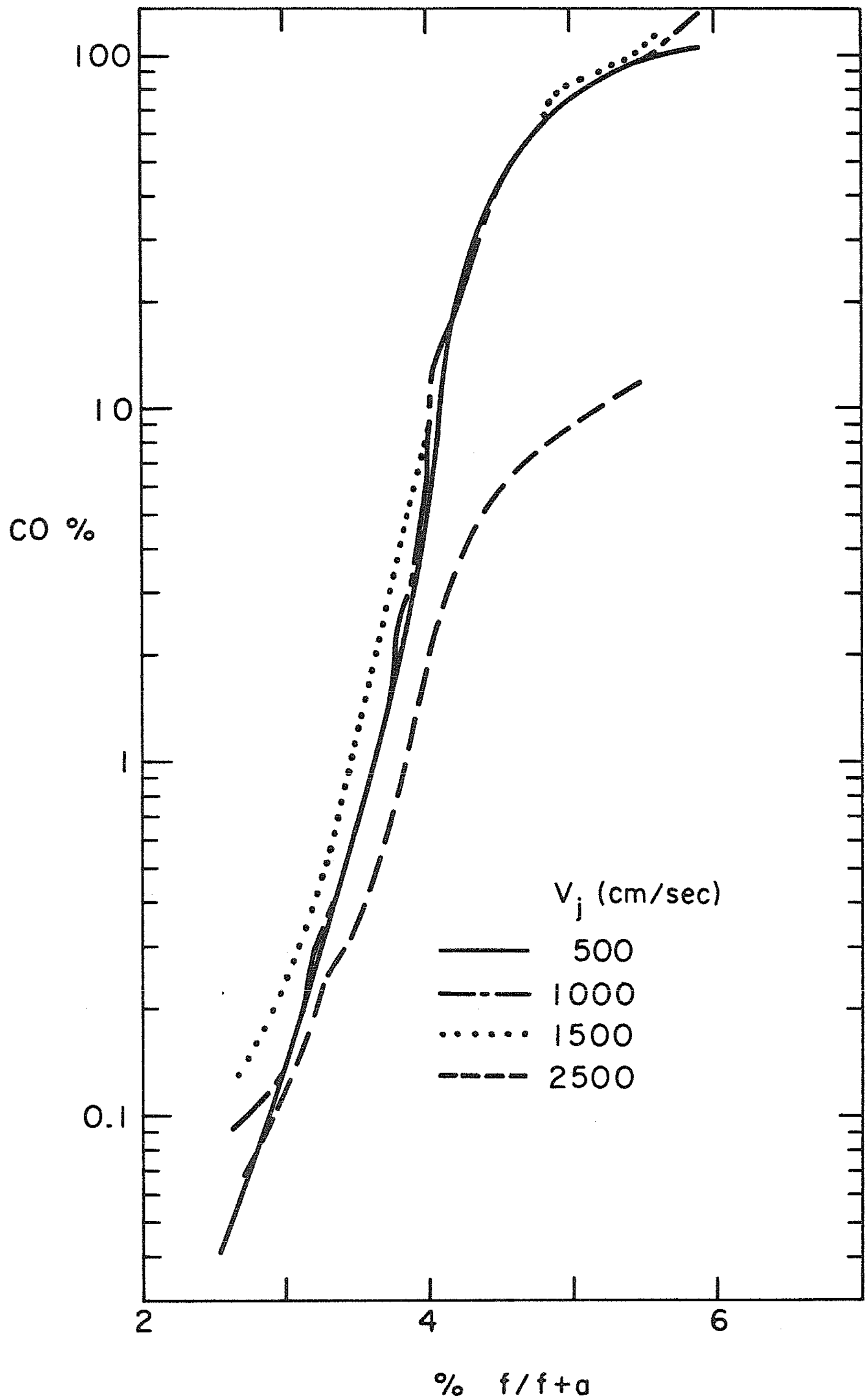

Figure 29. Comparison of CO curves for slot flameholder.

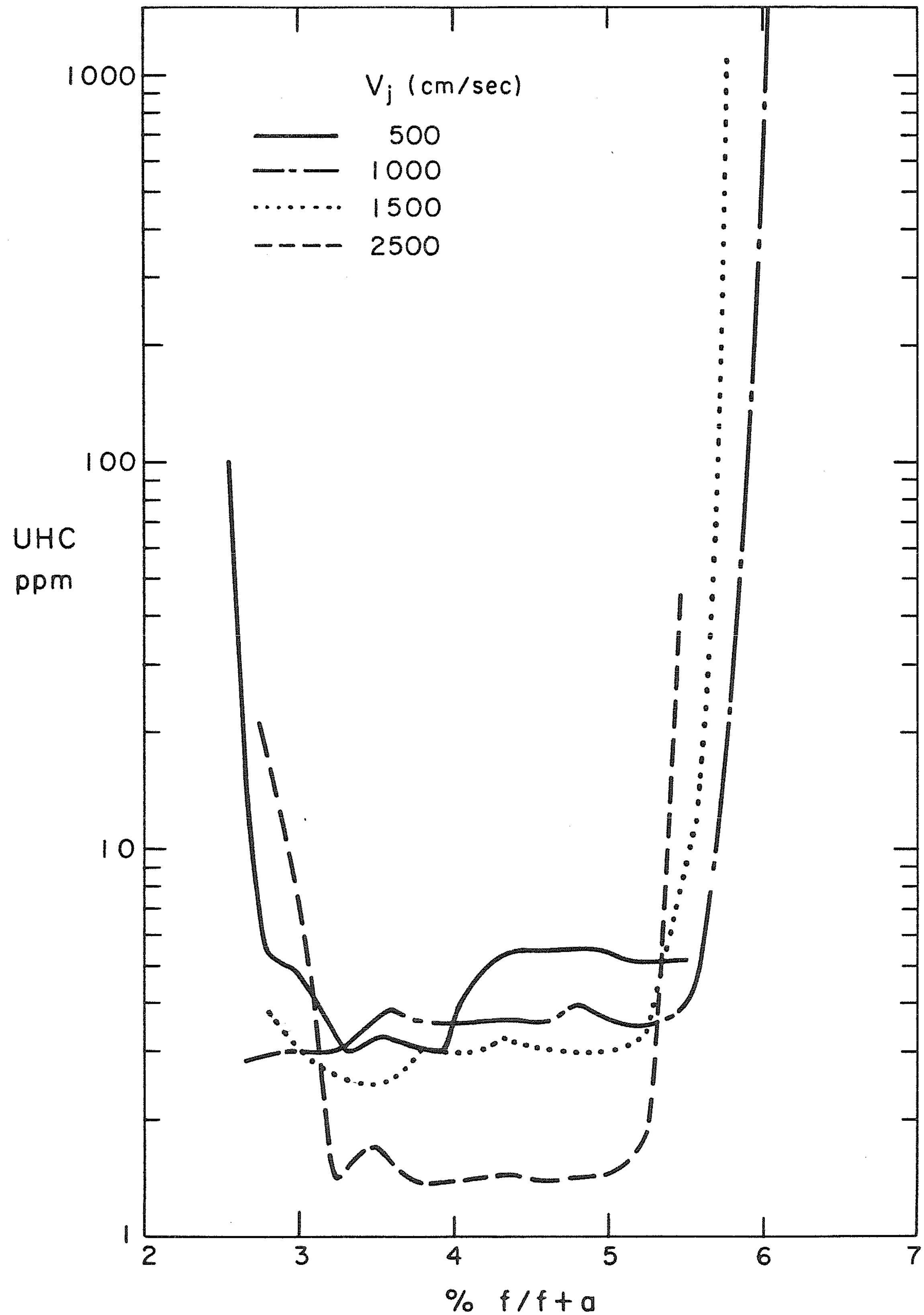

Figure 30. Comparison of UHC curves for slot flameholder.

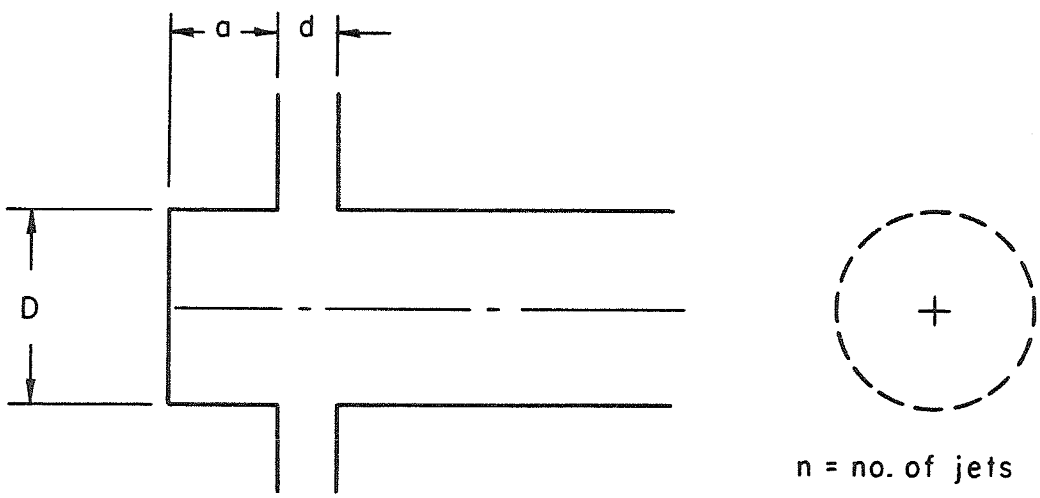

Figure 31. Generalized can type flameholder geometry.

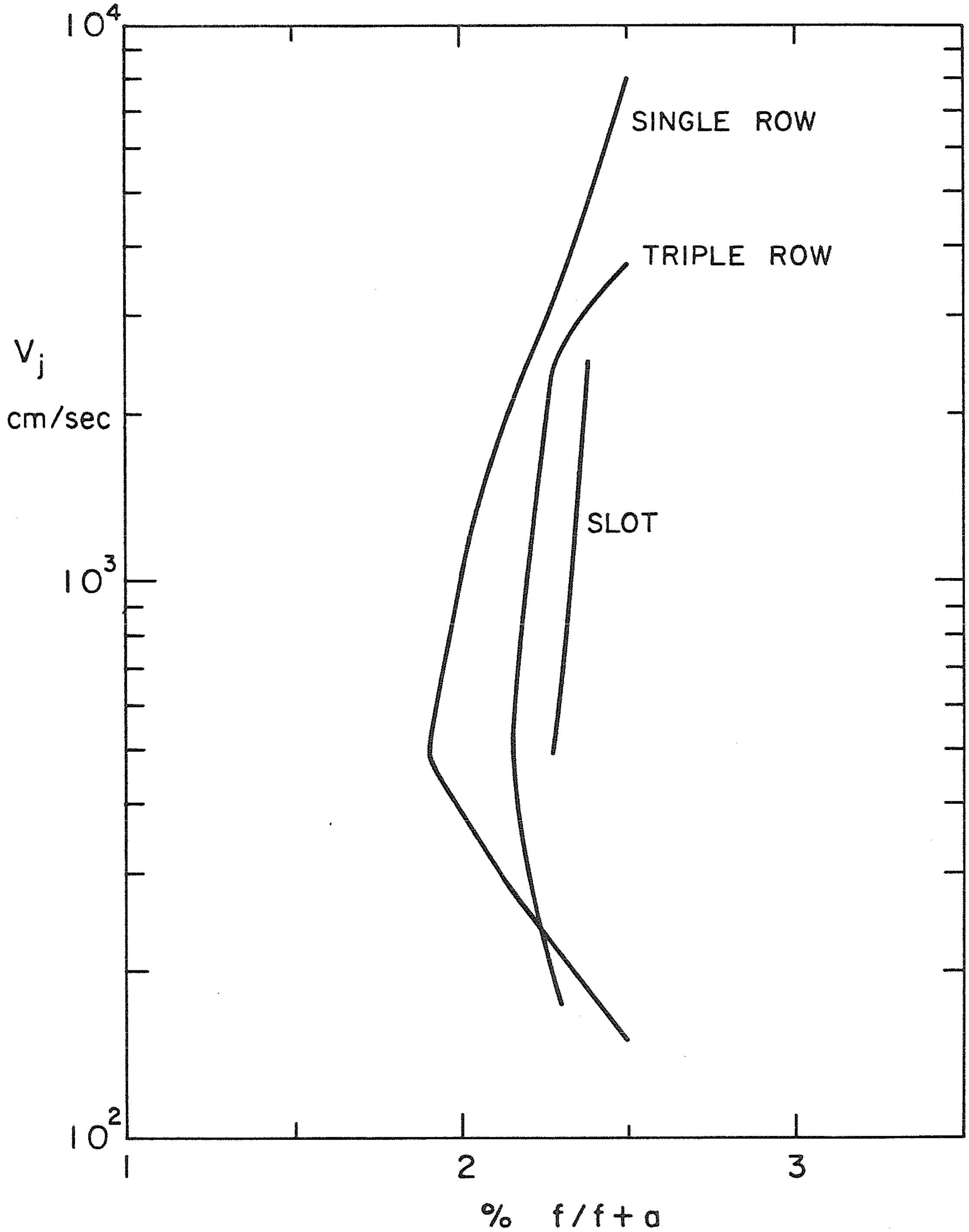

Figure 32a. Comparison of lean blow off performance based on v_j.

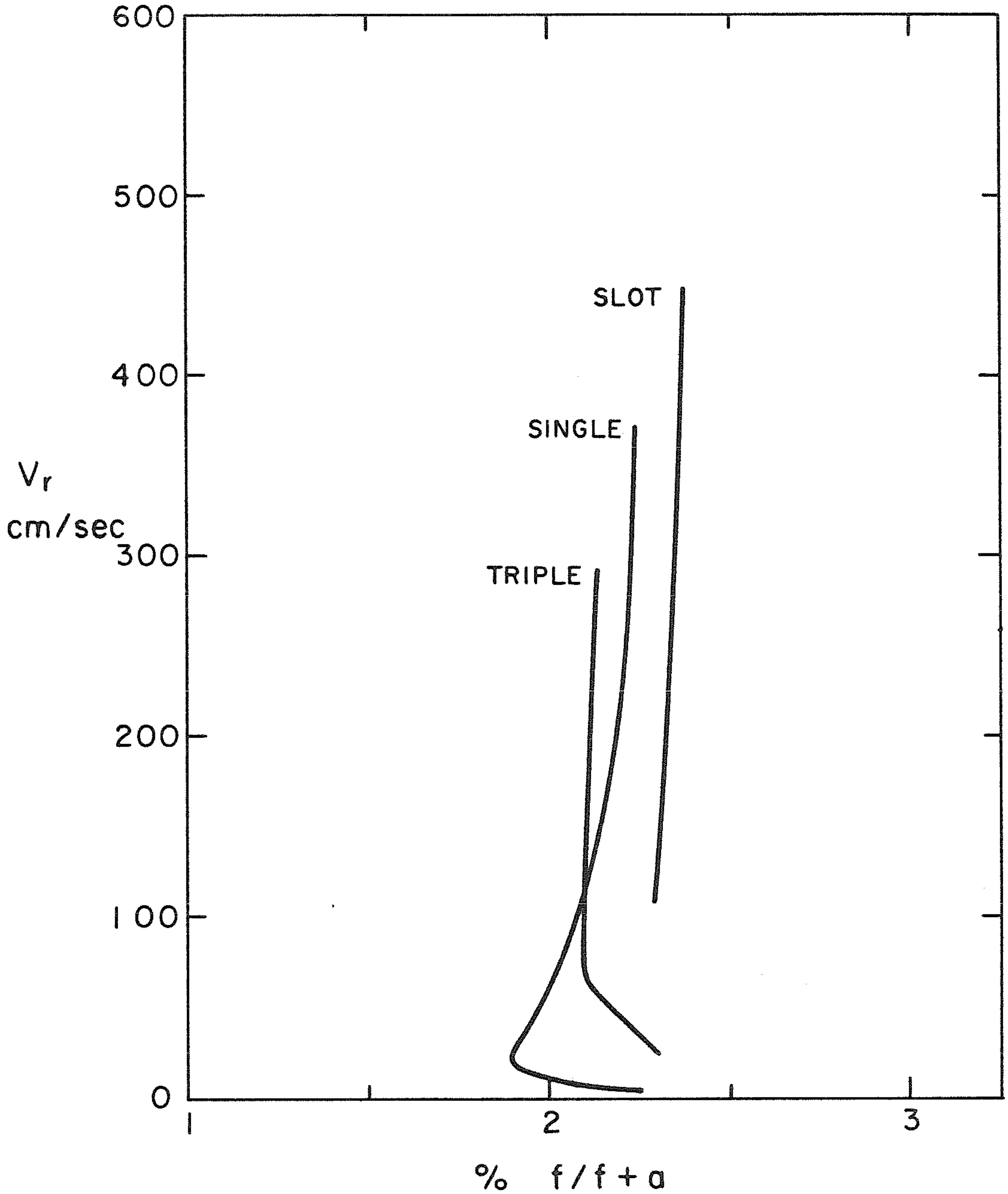

Figure 32b. Comparison of lean blow off performance based on v_r.

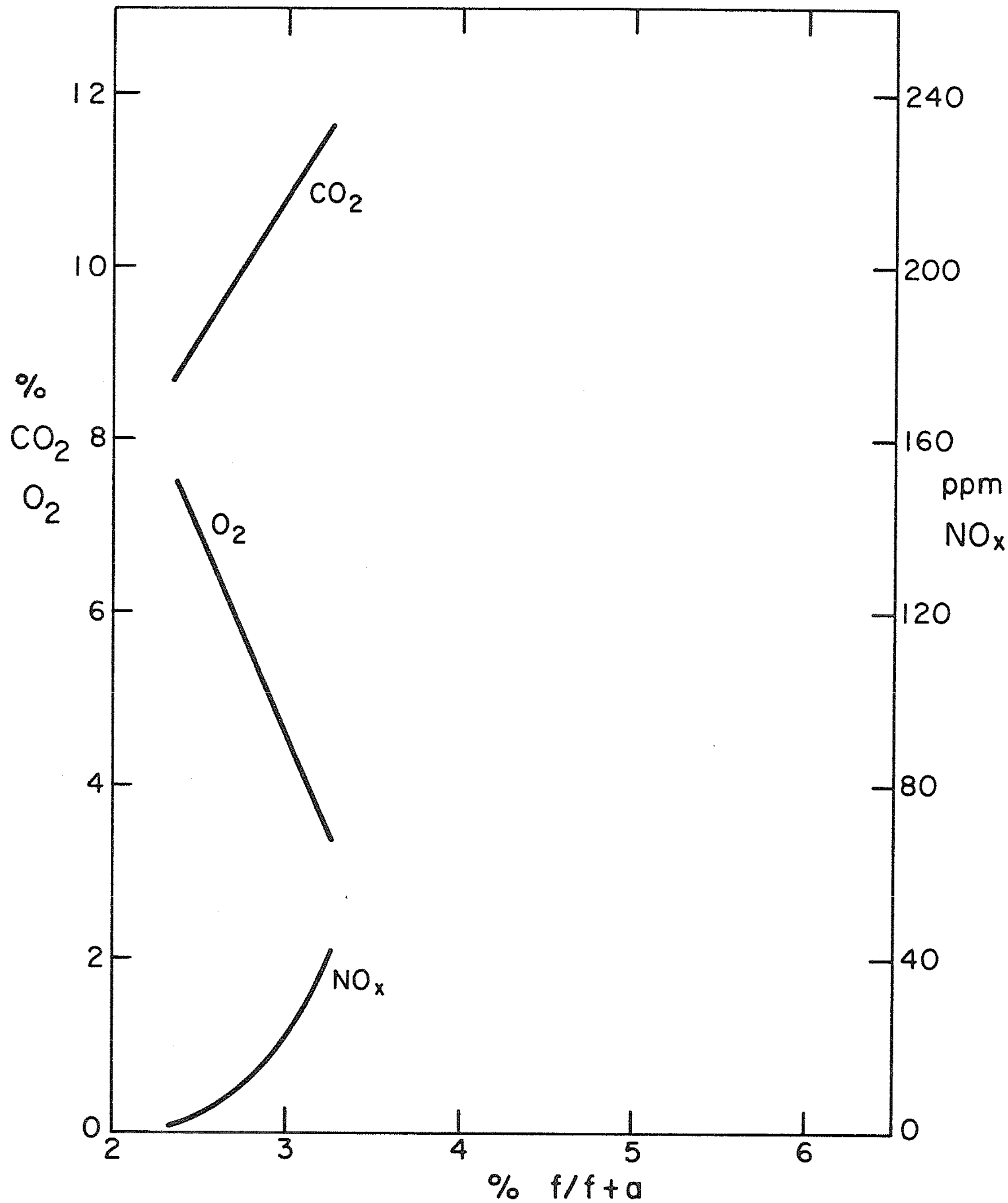

Figure 33a. Combustion product profile for triple row flame-holder, NO_x, CO_2, and O_2. v_j=2500 cm/sec., v_r=340 cm/sec.

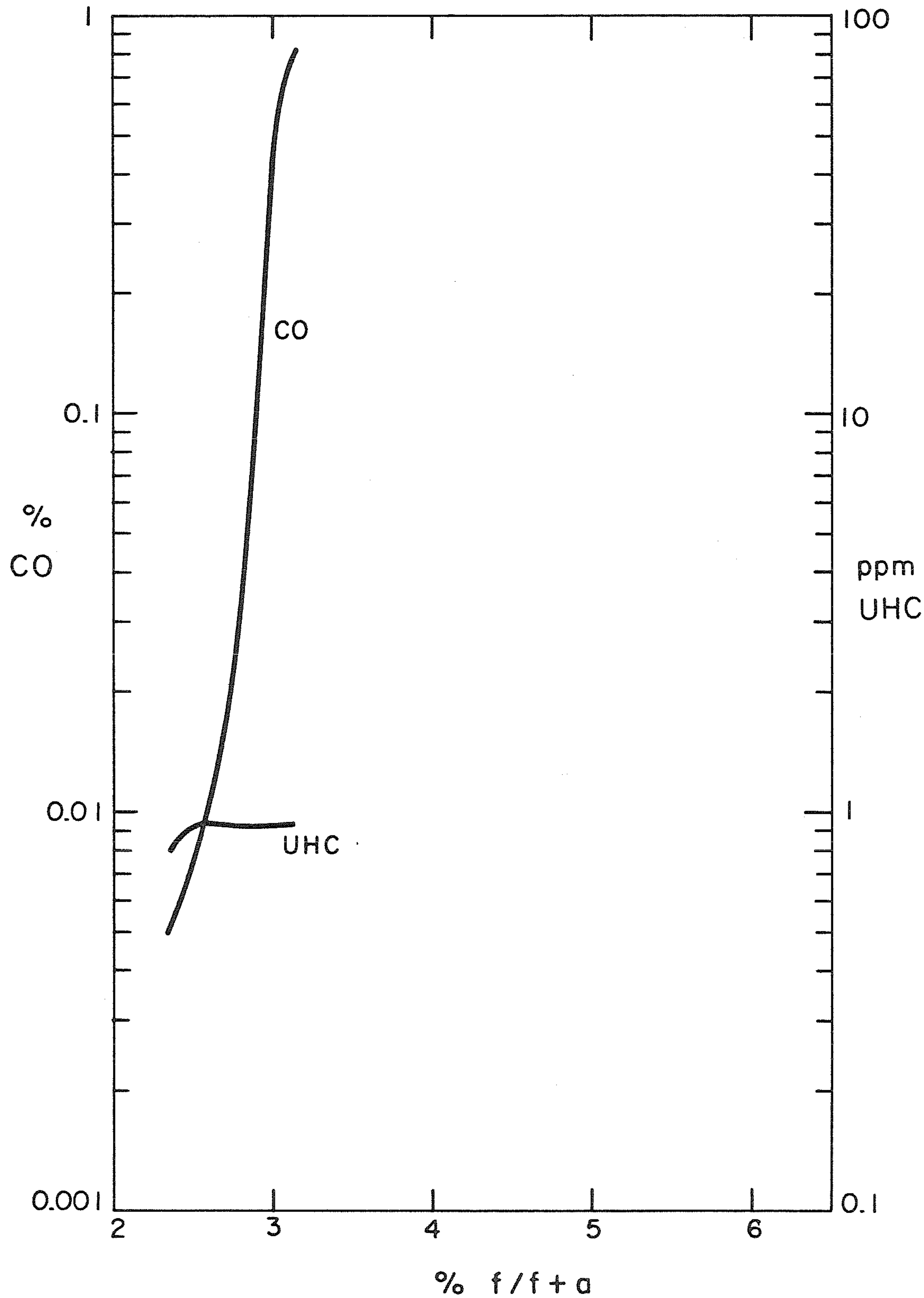

Figure 33b. Combustion product profile for triple row flameholder, CO and UHC. v_j=2500 cm/sec., v_r=340 cm/sec.

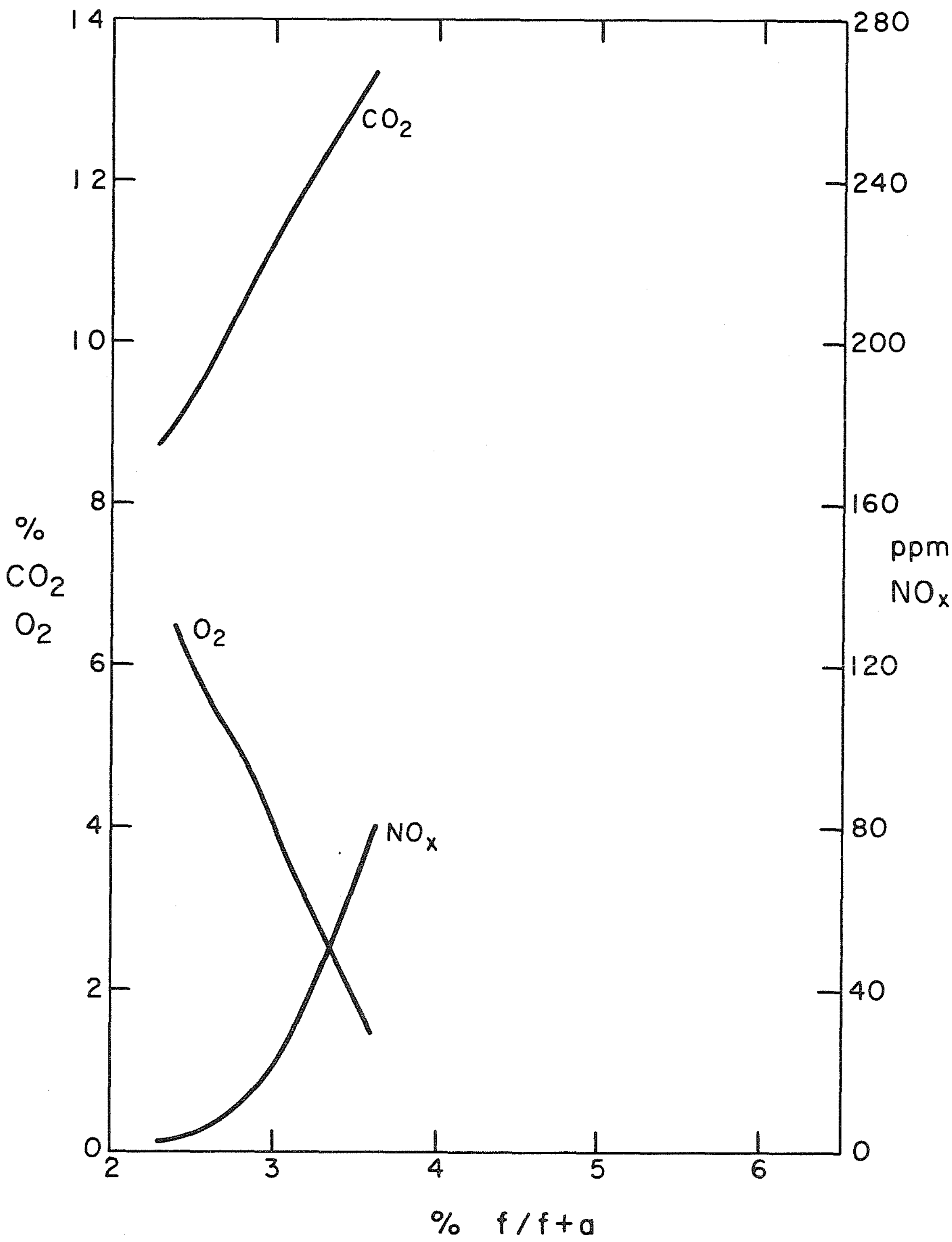

Figure 34a. Combustion product profile for single row flameholder, NO_x, CO_2, and O_2. v_j=2500 cm/sec., v_r=115 cm/sec.

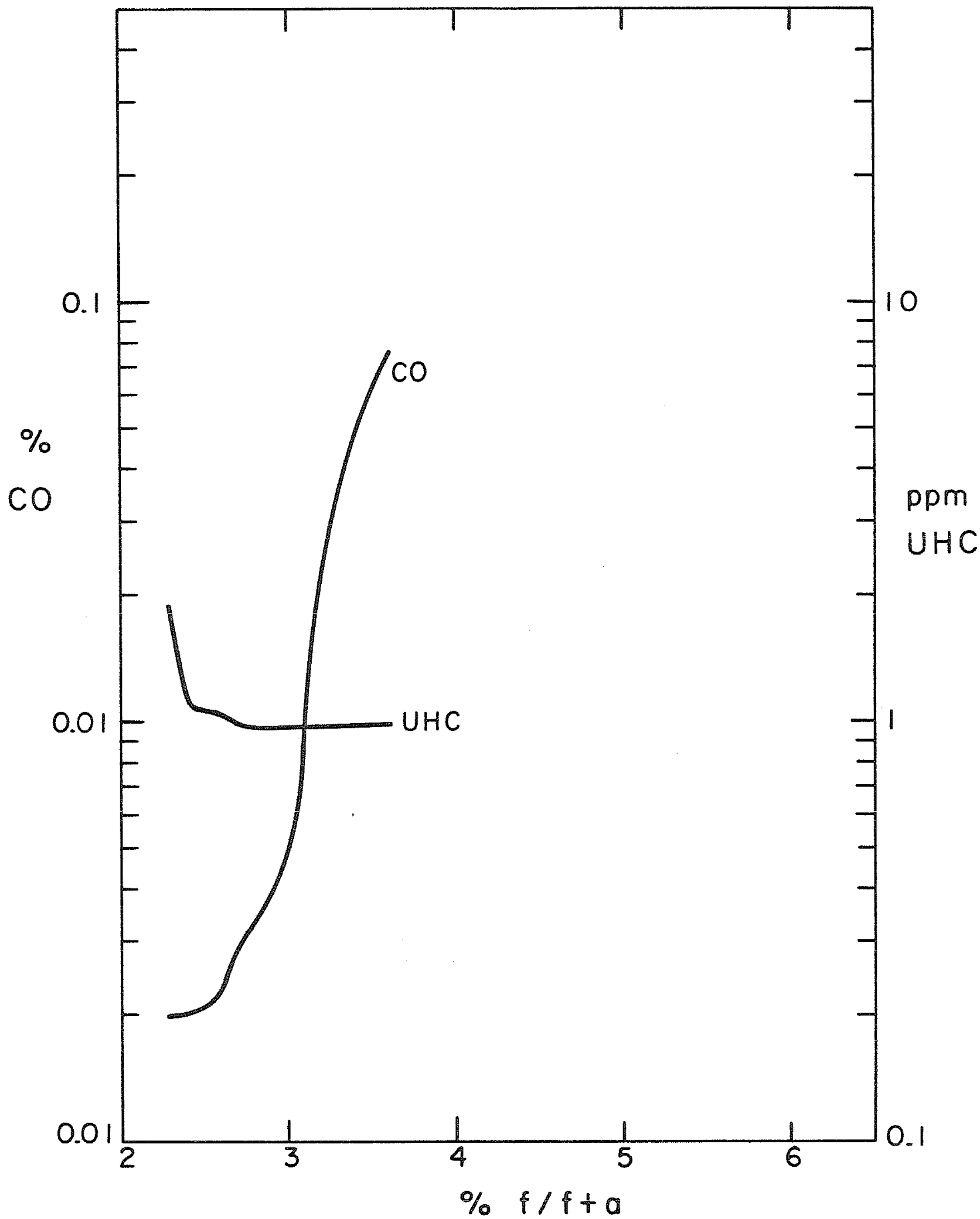

Figure 34b. Combustion product profile for single row flameholder, CO and UHC. v_j=2500 cm/sec., v_r=115 cm/sec.

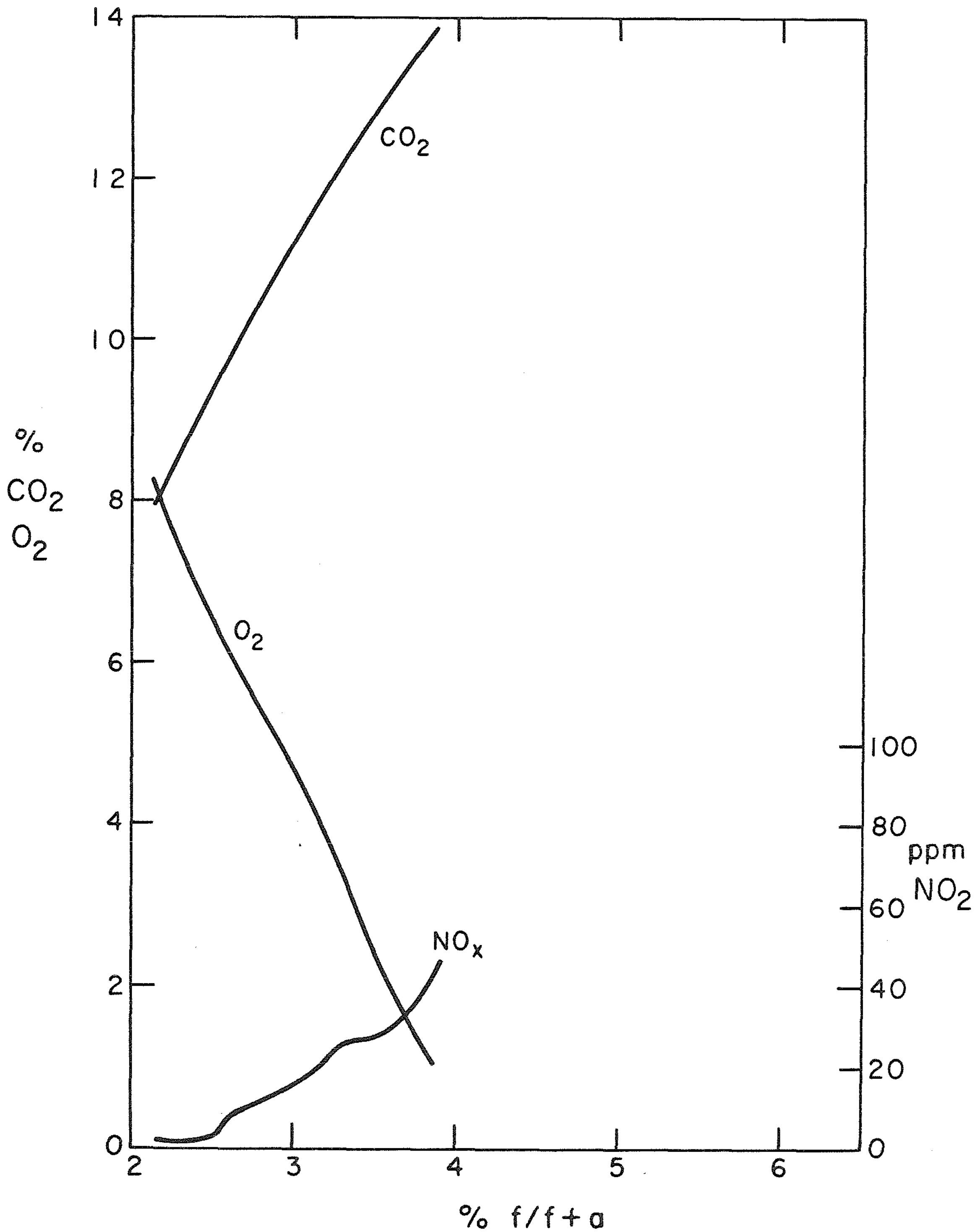

Figure 35a. Combustion product profile for n=22 flameholder, NO_x, CO_2, and O_2. v_j=1100 cm/sec., v_r=50 cm/sec.

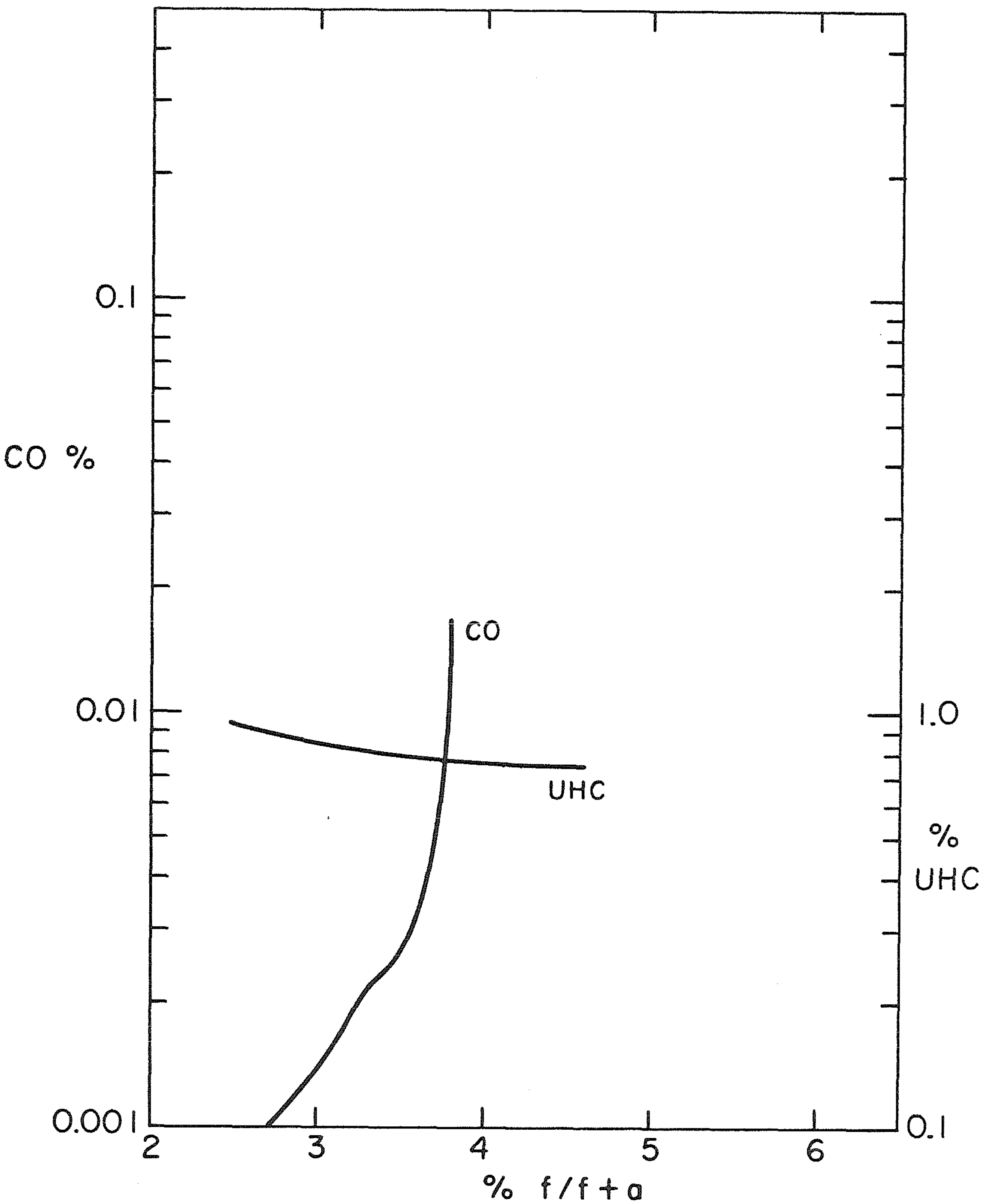

Figure 35b. Combustion product profile for n=22 flameholder, CO and UHC. v_j=1100 cm/sec., v_r=50 cm/sec.

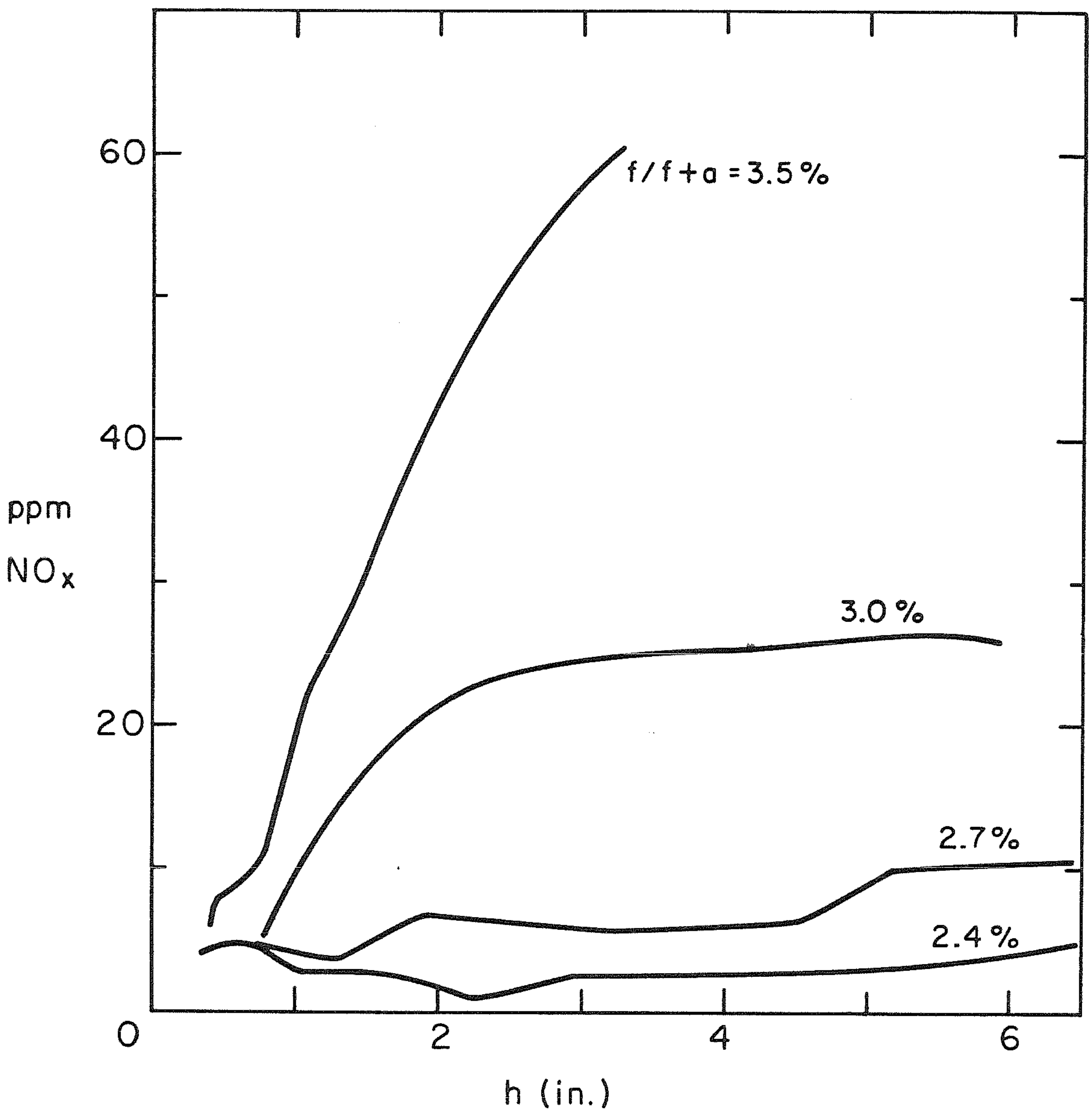

Figure 36. NO_x levels downstream of flameholder.

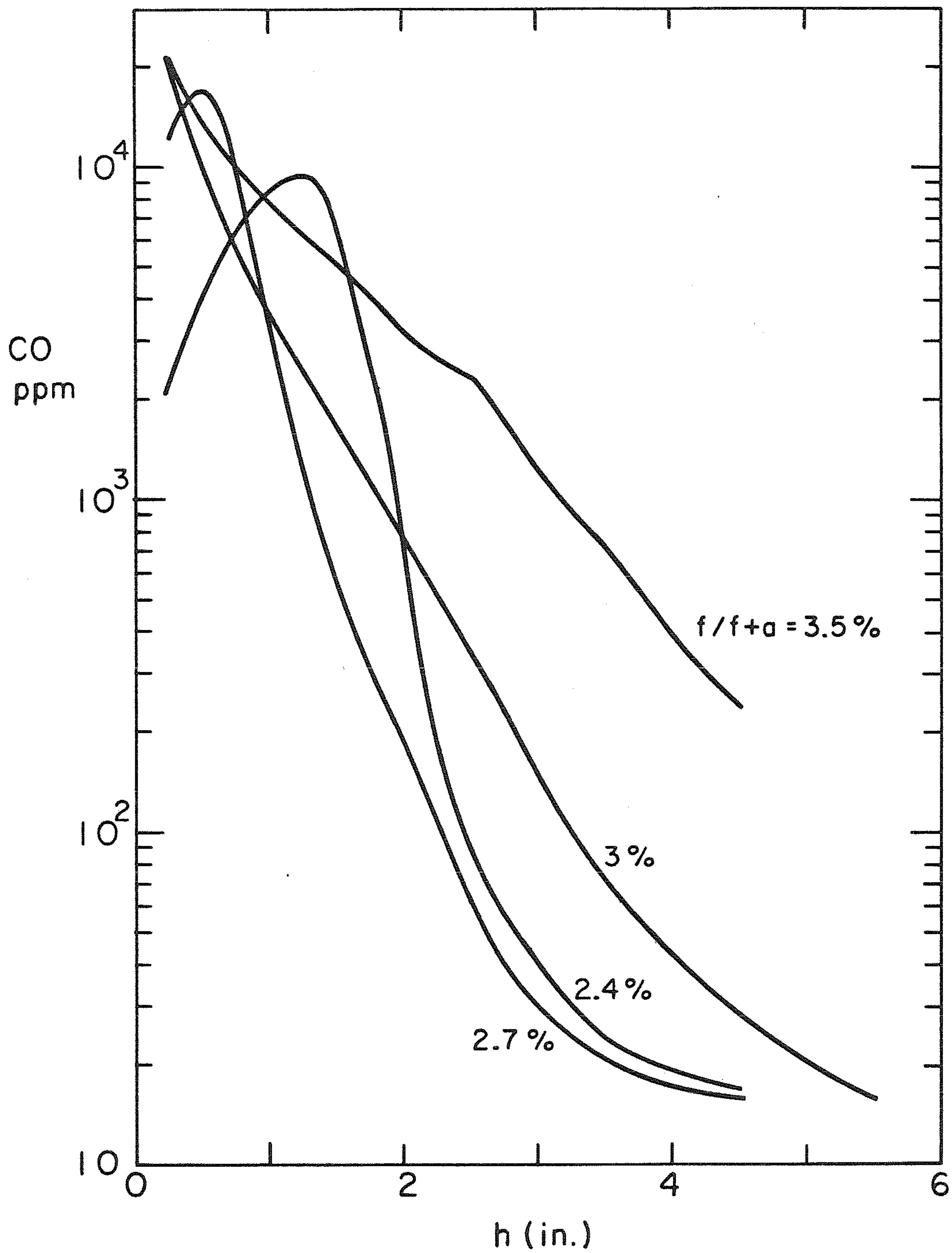

Figure 37. CO levels downstream of flameholder.

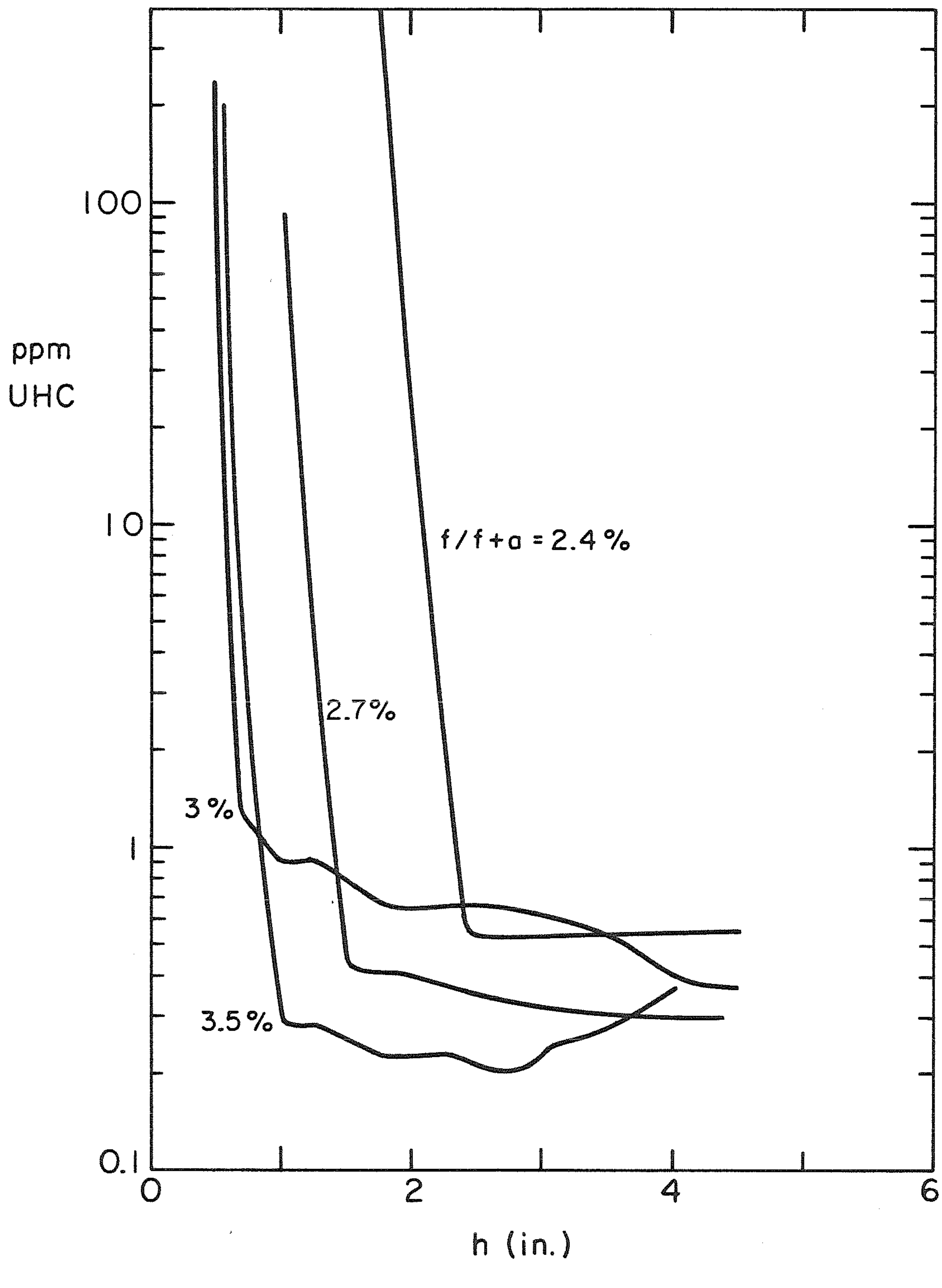

Figure 38. UHC levels downstream of flameholder.

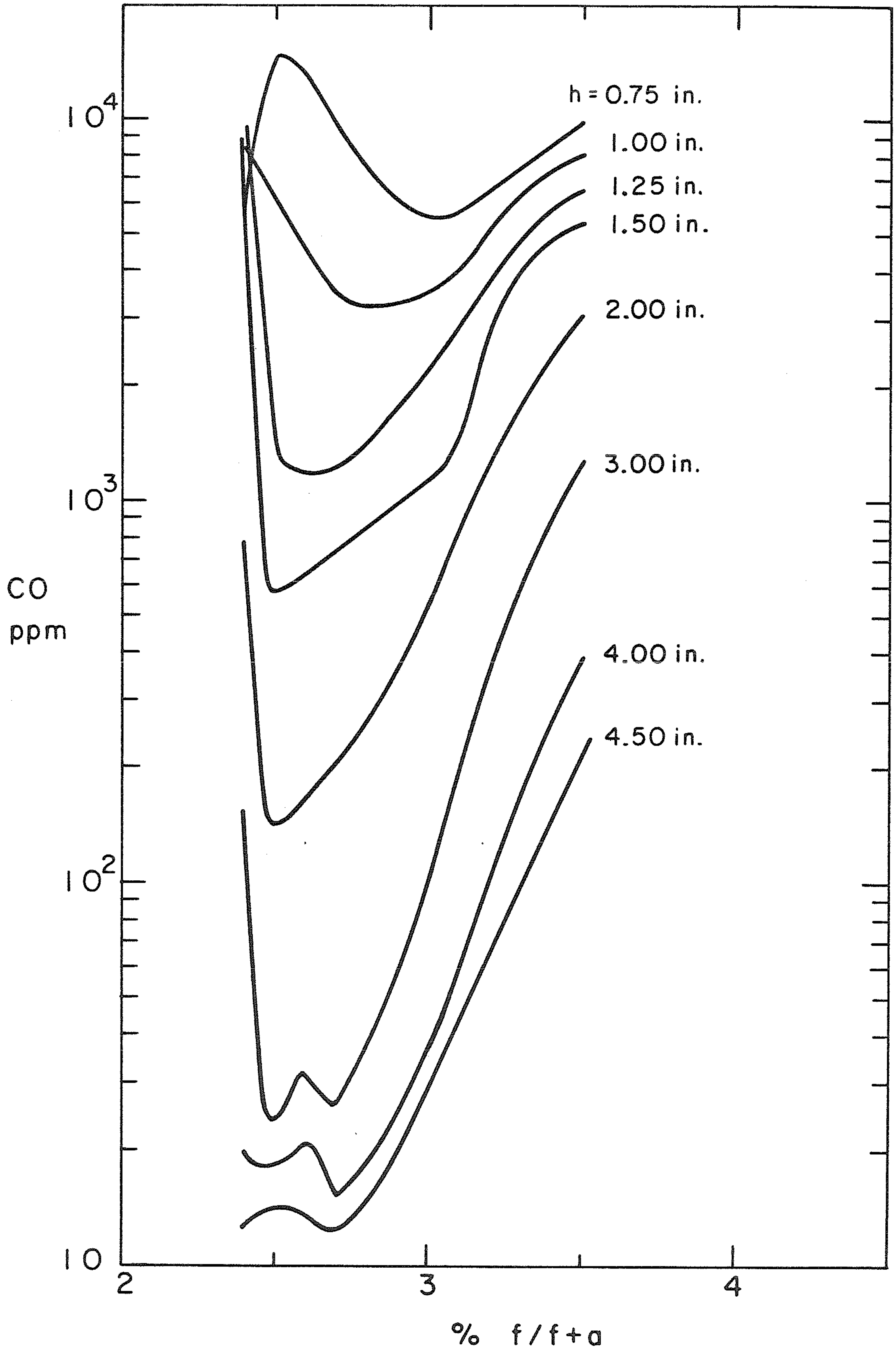

Figure 39. CO levels for various stations downstream.

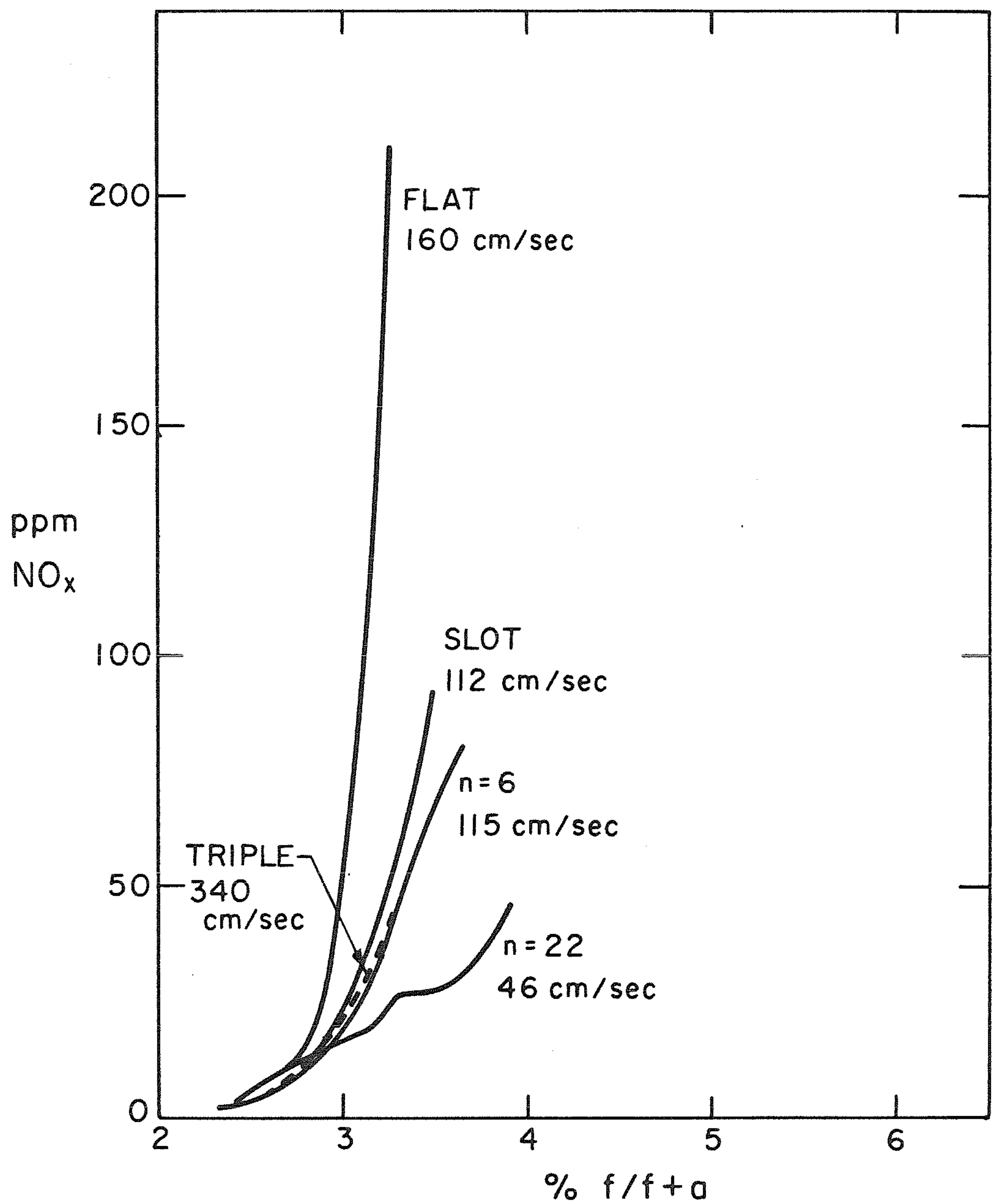

Figure 40. Comparison of NO_x curves for all flameholders.

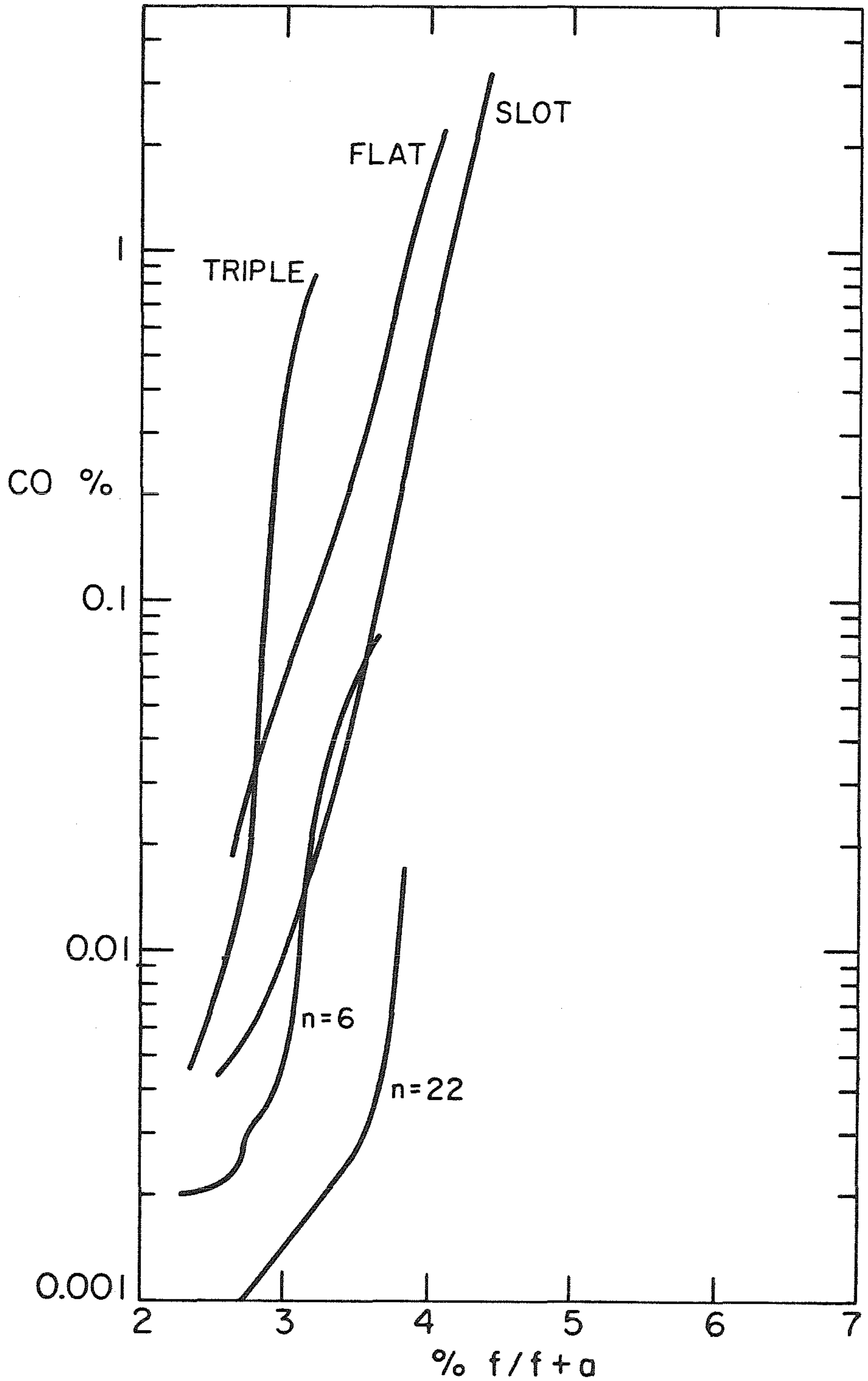

Figure 41. Comparison of CO curves for all flameholders.

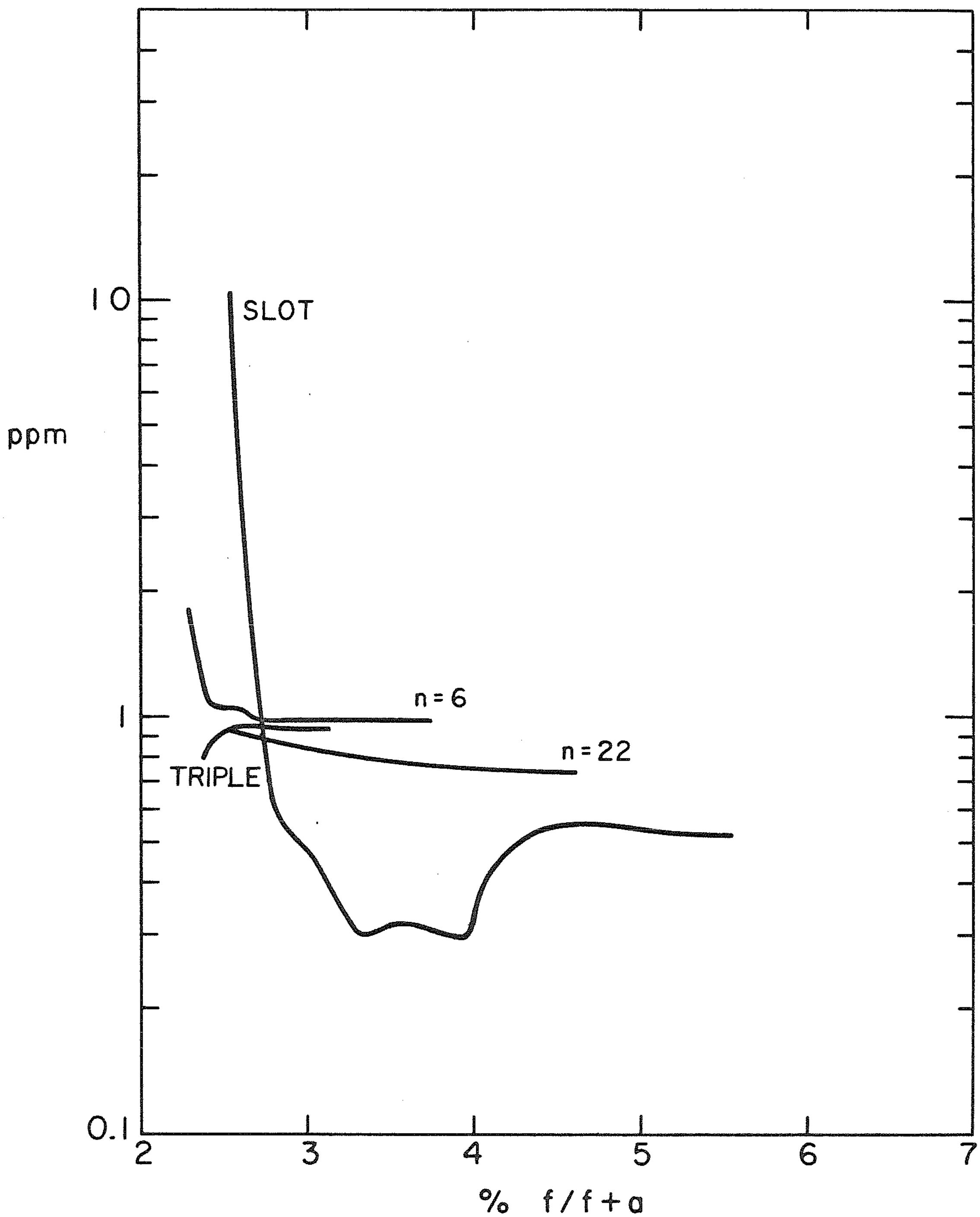

Figure 42. Comparison of UHC curves for all flameholders.

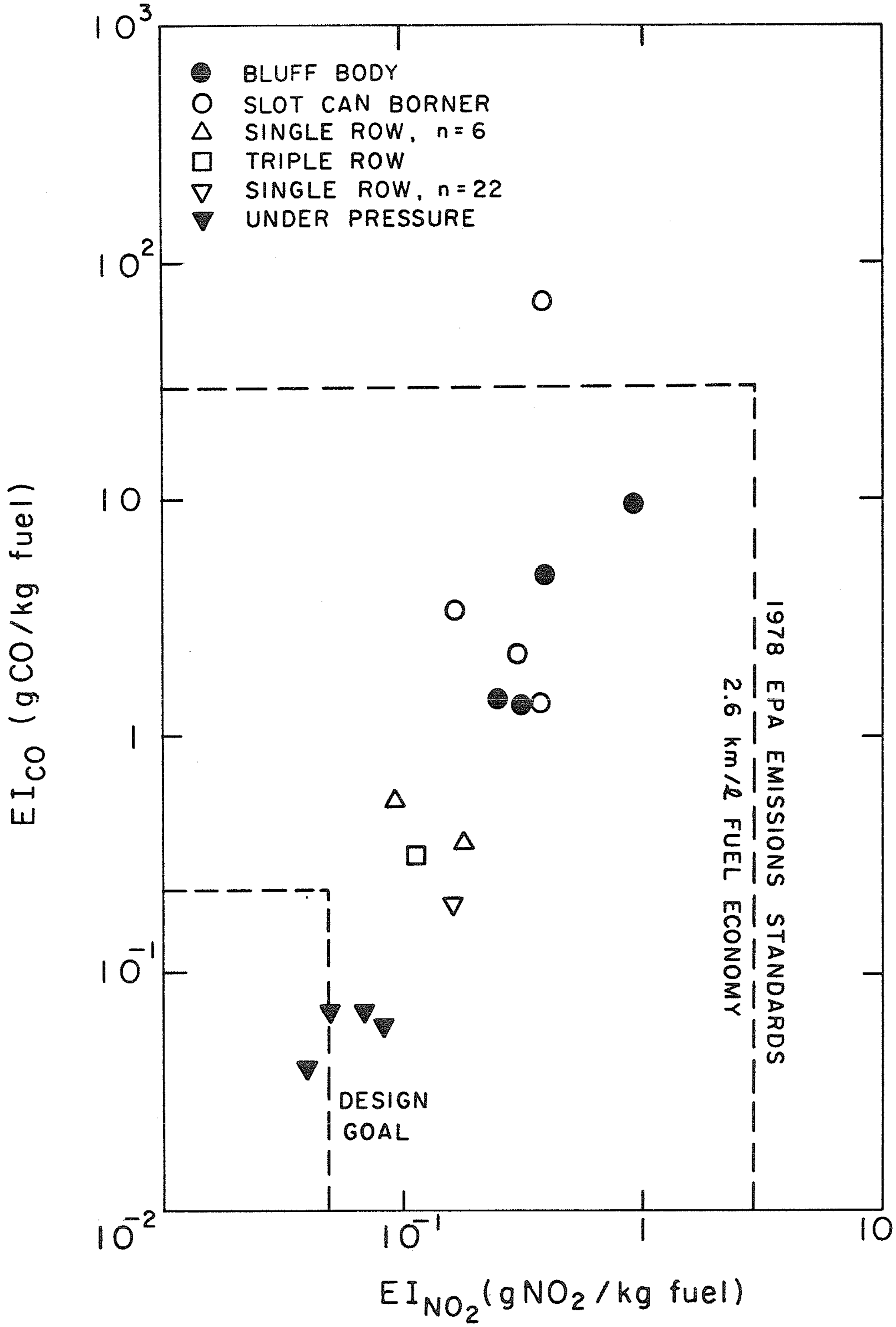

Figure 43. Summary of emissions performance of flameholders developed.