# THE FACTORS GOVERNING STABILITY AND COMBUSTION INTENSITY IN "MECHANICALLY" ATOMIZED OIL FLAMES.

A Thesis Submitted by

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TO MY WIFE.

## SUMMARY

Problems of combustion instability and the need for wider range in firing rates have drawn attention to the need for more fundamental knowledge of the performance and the characteristics of pressure-jet atomized oil flames. A secondary problem is the complication and inconvenience of measurements on full scale equipment.

In the present study various operational conditions for a typical modern burner with a pressure-jet atomizer and a swirled air supply were investigated. The variables chosen were fuel rate and spray angle. Data obtained consisted of gas composition, velocity and temperature measurements.

The development of a laboratory unit was considered desirable to provide a means of facilitating measurements on flames of this type and a rig was constructed which consisted of a suitable combustion box and $^{3}/_{10}$ths scale burner.

It was shown that the matching of the spray angle to the air register aerodynamics affected the flame characteristics very markedly. Regions of high unburnt fuel occurred due to fuel penetration and poor mixing outside

and downstream of the main part of the flame. The effect of widening the spray angle which usually occurs with wide range types of pressure-jet atomizers at the lower fuel rates was shown to be particularly unsatisfactory, and to effect the black smoke limit. The general course of mixing and the progress of combustion through the flame was deduced from calculation from the gas composition results. The distribution and effects of recirculation were shown from the velocity measurements. These were related to data for simple jets.

In particular, it was deduced that internal recirculation in the core was largely responsible for flame stabilization. Theoretically derived scaling criteria were investigated by comparing full scale and model results. These showed that reasonable comparisons were possible between the two, with flames scaled either on the basis of similarity in velocity and relative momentum between spary and air, or on similarity in residence times in the flame. The former was thought probably to be better. Cold flow measurements were made, and showed differences in the size of the central recirculation core between hot and cold but not in the peak velocity to peak velocity diameter early in the flame. Combustion oscillation measurements on a marine boiler produced data which suggested strongly that these were of the "non-acoustic" type. None of the theories so far suggested appear to account with much accuracy for the observed data. In the course of experimental work various special instruments for flow measurement were developed.

It was concluded that the choice of spray distribution to suit a particular aerodynamic pattern was vital to the optimum performance of the burner. In all the cases investigated this was found to be to some extent imperfect. It is considered that decided improvements could be made to the air distribution in burners of this type. The following possibilities are suggested solutions, the use of a higher degree of swirl and a confining quarl, upstream air injection or the use of multiple oil nozzles and by avoiding devices where the spray angle varies widely over the operating range.

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## Acknowledgements.

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# CHAPTER 1.

## Introduction.

### 1.1 The Function of Oil Burners.

In all modern oil burning appliances, the most vital part of the process is the subdivision of the fuel into fine enough droplets for burning adequately under the conditions of use. The size requirement will vary with the type of oil fuel and its application. Three methods are generally available for this purpose:-

(1) Vaporization - only suitable for distillate fuels.
(2) Blast atomization using an air or steam jet to atomize the fuel.
(3) Mechanical atomizers in which a higher angular momentum is imparted to the oil stream resulting in a thin conical or circular film being formed which disintegrates into droplets. This high angular momentum can be achieved in two ways, either by forcing the liquid through a swirl chamber and through a small orifice under high pressure or by spinning the liquid fuel off the edge of a spinning cup or disc.

The selection of the type of atomizer is largely dependent on the application and the ancillary equipment considered economically justifiable. Thus for applications like the open hearth furnace, a long flame is required and steam is readily available for atomization. For marine purposes as indicated later (Section 2.[illegible]) the reverse is the case, and for this and many land boilers, pressure jet atomizers (and to a much lesser extent spinning cup atomizers) are widely used.

The choice of the method of atomization is only the first consideration for the flame performance considered desirable. The heat transfer requirements from the flame make its shape and hence the design of the rest of the burner components, often known as the "air register", almost equally important. This is particularly true in the case of the pressure-jet, which in the absence of an atomizing fluid has a low momentum compared with the air supply (under the air inlet conditions customarily employed in this field.). The result is that the behaviour of the flame is to a very large extent controlled by the burner aerodynamics and in fact by the aerodynamics of the whole combustion chamber system. It is with the study of the behaviour of pressure-jet burners and the attendant air supply arrangements that this thesis is concerned.

## 1.2 The State of the Art.

In the case of the blast atomized flames, a great deal of work has been done in this field in Great Britain and at the International Flame Research Foundation, IJmuiden, Holland, in the past decade, and much has been established about the performance and physical characteristics of this type of flame both in the aerodynamic and thermochemical field as well as in the field of radiant heat transfer.

In mechanically atomized oil flames the only information available until very recently was limited to studies of the atomization process and fundamental studies of actual flames. This was almost entirely restricted to the special case of distillate fuels in gas turbine combustion chambers, an arrangement very dissimilar to conventional oil burning equipment, which seems, so far as almost all published information is concerned, to have received no attention. The result is that for nearly all air directors or air register inlets, the arrangement appears to be almost entirely an "ad hoc" design to give by trial and error a stable flame for the range of throughput desired, and moderately good $CO_2$ figures with the increasing pressure for burner efficiency.

Only a very vague idea seems to have existed as to the pattern of flow established by the flame. The following aspects seem to have received very scant attention up to 1957: -

(1) The attainment of high combustion intensities. Ideas were taking shape but had not been applied to practical systems.

(2) Flow pattern studies almost entirely limited to jet cans.

(3) Gas composition studies - no data available.

(4) Little data on factors controlling aerodynamic patterns in other than very simple systems. This particularly applies to the field of air swirl.

(5) The matching of atomizer and aerodynamic pattern had received no attention except for the effect of droplet character in ignition delay.

(6) The control of flame performance over a wide range of inputs.

(7) The study of combustion instability in liquid fuel fired systems, in particular in large water tube boilers with high possible combustion intensities. No work had been reported.

(8) The interaction of a number of burners in a combustion chamber of complex shape. The available knowledge derived from this field is reviewed in the literature survey.

## 1.3 Scope of work covered by this thesis.

The development of improved combustion equipment can only take place along proper scientific lines if the fundamental principles and their application to existing types of burners and flames are understood. The problem has been divided into the following sections:-

(1) The determination of the detailed performance characteristics of a typical modern burner with a pressure-jet atomizer in relation to some of the most prominent variables and the assessment of this data as compared with the theoretical ideal.

(2) The determination of a satisfactory basis for scaling flames of this type, in order to provide a laboratory rig capable of covering a wide range of variables without having to carry out the work on full-size plant.

(3) The determination of possible lines of improvement of pressure-jet burners, in particular the achievement of higher combustion intensities, better control of combustion instability and the avoidance of other undesirable side effects.

(4) The characterizing of the types of combustion instability observed in water-tube boilers and the correlation of experimental data with combustion oscillation theory.

# CHAPTER 2.

## Survey of Literature

### 2.1 Introduction

The use of petroleum oils has a very long history. The genesis of the modern development of oil burning was the invention of the steam atomizer in Russia due to Spakowski, Lenz and others in the period from 1866 onwards[1]. This was followed by similar developments along the same general lines in the U.S.A., France and Britain, and went hand in hand with the realization that as far as heavy fuel oils are concerned, "Atomizing" is the only way[2].

Although the use of steam atomizing burners spread quite quickly to locomotives and coastal and river shipping, particularly in Russia, loss of freshwater by the use of steam for atomizing was a major disadvantage as far as marine uses in the Western world were concerned. The greatest single advance in this field was the introduction in 1902 by Korting of an atomizer in which oil was given a rapid swirling motion before passing through the final orifice. Pumping the oil at high pressure (100 psi upwards) produced a good quality wide angle spray and eliminated the use of steam or compressed air as the atomizing agent. This development lead rapidly to a very wide field of marine uses for oil. With the exception of spinning cup atomizers, nearly all modern oil burners are based on one or other of the above principles. The atomizer is however, only part of the burning equipment and air director arrangements may vary enormously from a simple quarl to an elaborate arrangement of air ports similar to those in the gas turbine combustion chamber(3) or the Urquart burner(4).

The main difference between air atomizers and the mechanical type is that the jet in the latter having no assisting primary air has a much lower momentum. This requires much greater care in design so that the air and oil are mixed without loss of fuel from the combustion region.

In the realm of pressure jets the combustion intensities which can be obtained have increased enormously in the last three decades. It is surprising that with a few exceptions very few details of this development have ever been published(5-9). Until recently properly conducted scientific studies have been left almost solely to the gas turbine field with its different emphasis(10-13).

The general observations on combustion requirements which follow are common to all the forms of liquid fuel combustion.

## 2.2 Overall Theoretical Combustion Requirements for Liquid Fuels.

### 2.2.1 Physical Processes.

The first requirement is adequate heat transfer to raise the temperature of the incoming fuel to the ignition point. In the following sections, possible ways of doing this are considered.

2.2.1.1 Preheat. Although preheat obviously will assist the process of ignition, preheat to the ignition temperature is out of the question in most practical systems.

2.2.1.2 Compression. The use of compression as in the diesel engine is also obviously out of the question.

2.2.1.3 Conduction. A number of types of laminar flames such as the Bunsen burner are stabilized by conduction of heat back into the incoming fuel. However, data given by Longwell(14) and Rappeneau(15) indicate that the greatest flame velocity and hence permissible supply velocity, although obtained at stoichiometric mixtures, would still be less than a quarter of the velocity required by modern high intensity equipment even allowing for increases in conduction with turbulent flow.

2.2.1.4 Radiation. Radiation only really becomes important in cases where there is high air preheat, hot walls and loading considerably less by 2 or 3 times that obtained in marine boilers(16).

2.2.1.5 Recirculation. In view of the limitations of these other methods, combustion systems of the type under discussion have to rely on the recirculation of hot products from the tail of the flame back to the burner where they are re-entrained.

The effect of the amount of recirculated products on the temperature of the combined jet is shown in Fig.1. for the case of the Kerosine fuel with negligible wall losses.

Bragg and Holliday(17) predicted that a maximum value of the reaction rate would be obtained for a ratio of recirculated products to fresh material of 4.9:1. Experimental values obtained by Longwell & Weiss(18) were somewhat lower at 3.2:1 probably because of the special characteristics of their system. The excessive dilution caused by larger quantities of recirculated products tends to reduce the reaction rate despite the higher temperature.

2.2.2 Chemical Considerations.

It is now necessary to consider the chemical implications of the above. Firstly, the results obtained by Mullins(19) have shown that the predicted maximum combustion intensity obtainable in liquid fuel sprays should be as high as $3.9 \times 10^9$ B.t.u./cu.ft.hr.atm. and is almost the same for all hydrocarbon fuels. This is apparently only slightly affected by

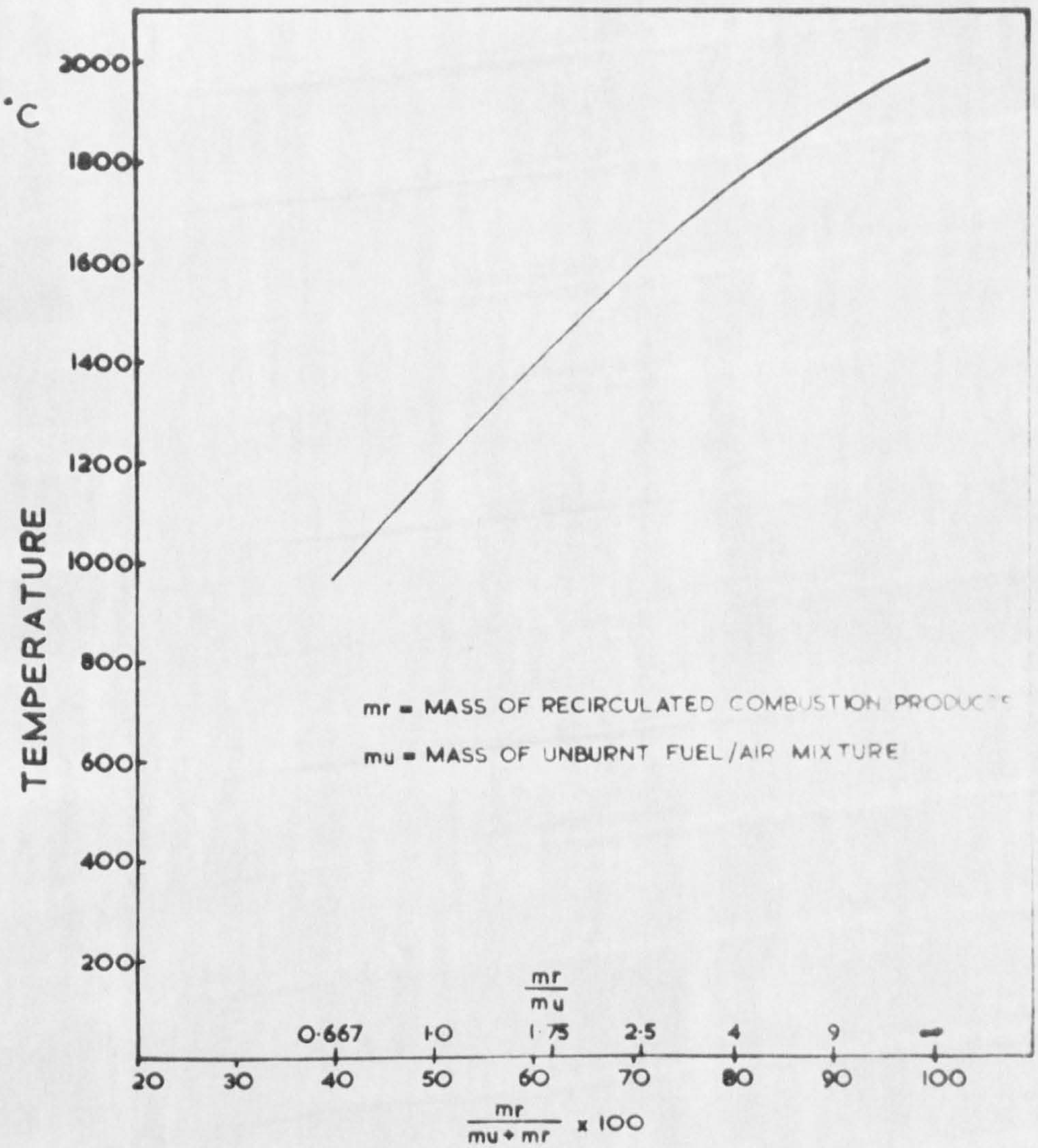



Fig. 1. —TEMPERATURE TO WHICH COMBUSTION PRODUCTS (mr) WILL HEAT INCOMING FUEL/AIR MIXTURE (KEROSINE).

vaporization, droplet size, turbulence or air/fuel ratio. The results obtained from Longwell's apparatus indicated that under the high temperature low heat loss conditions of his experiments the overall order of reaction was 1.8 and the activation energy was $4.2 \times 10^4$ cals/mole. The limiting heat release rate attainable with instantaneous mixing was about $3 \times 10^8$ B.t.u./cu.ft.hr.$(\text{atm.})^{1.8}$ for stoichiometric mixtures. It is of interest in the present context that Longwell states that this is very susceptible to alteration of the air/fuel ratio and to heat loss from the flame. See table 1. below.

TABLE 1.

| Limiting Heat Release Rates B.t.u./cu.ft.hr.$(\text{atm.})^{1.8}$ | | | |
|---|---|---|---|
| Equivalence Ratio.' | (No heat losses) | Heat Losses | (Equivalence ratio = 1) |
| 1 | $3 \times 10^8$ | 0% | $3 \times 10^8$ |
| 1.4 } Lean | $9 \times 10^7$ | 5% | $< 2.1 \times 10^8$ |
| 2.0 } Lean | $1 \times 10^7$ | 20% | $4.5 \times 10^7$ |

Thus, for high intensity, combustion as close as possible to stoichiometric is essential and a compromise is necessary between the combustion intensity desired and the heat transfer to be allowed in the combustion chamber.

Applying normal kinetic theory, the reaction rate is given by

$$\Omega = kp^2 T_m^{-3/2} C_f C_o \exp\left(-\frac{E}{RT_m}\right)$$

where

$\Omega$ = combustion intensity (heat release/unit volume)

$k$ = Collision constant.

$p$ = Pressure in chamber.

$T_m$ = Local temperature of mixture.

$C_f$ = Concentration of fuel.

$C_o$ = Concentration of oxygen.

$E$ = Activation constant.

$R$ = Gas constant.

## 2.3 Ideal Combustion Systems.

### 2.3.1 The Premixed Ideal.

Consideration of the kinetics, as well as the physical properties, naturally lead to the consideration of what constitutes the optimum combustion system. Reference to the combustion efficiency in Longwells experiments shows that for very high reaction rates the combustion efficiency drops substantially so that at the highest rate only 80% efficiency is attainable. This is shown in Fig. 2. Plotting this result in a different way in Fig. 3. shows the effect of this. Curves A and B are for stoichiometric and lean mixtures at

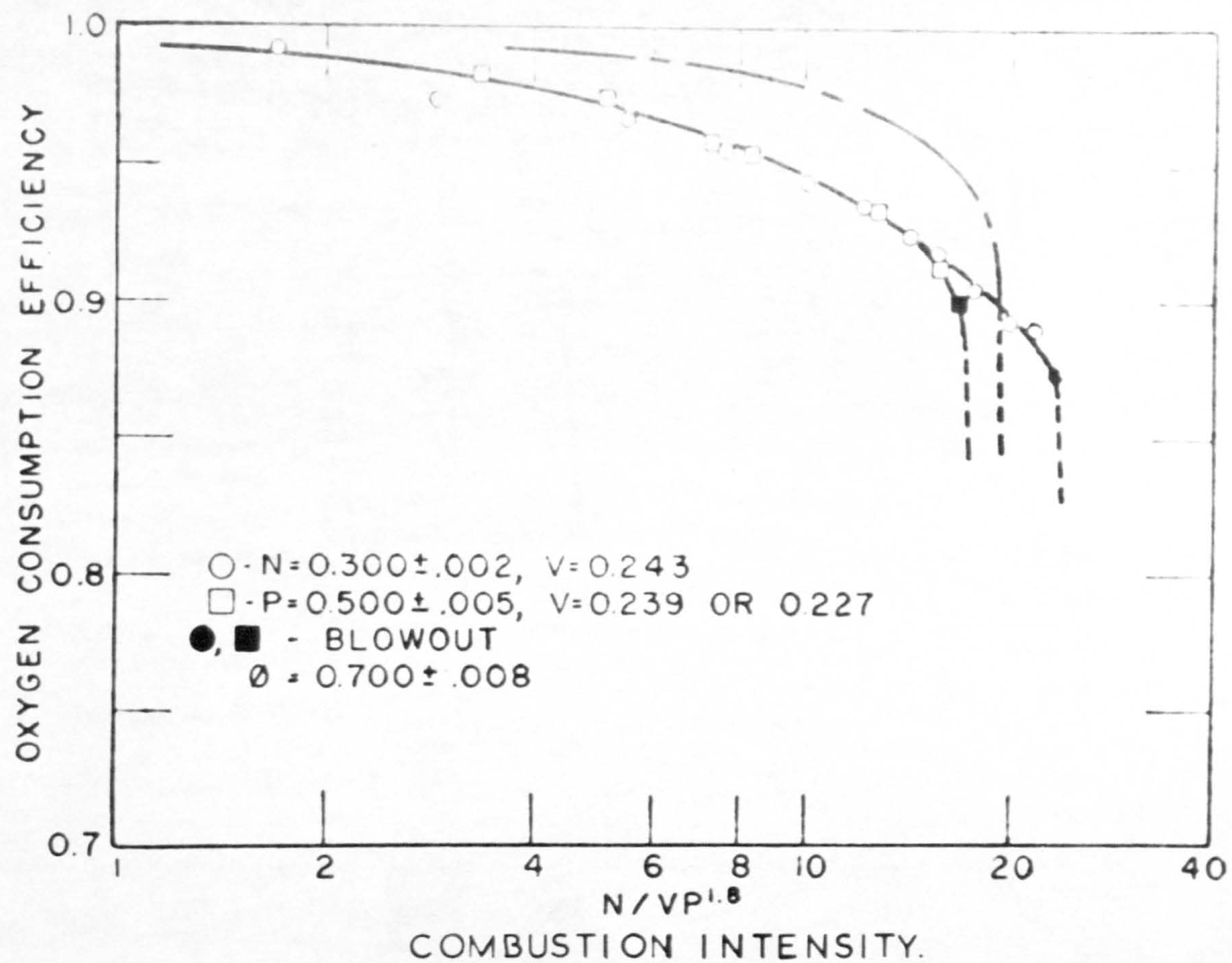



FIG. 2. PLOT OF EFFICIENY AGAINST COMBUSTION INTENSITY.

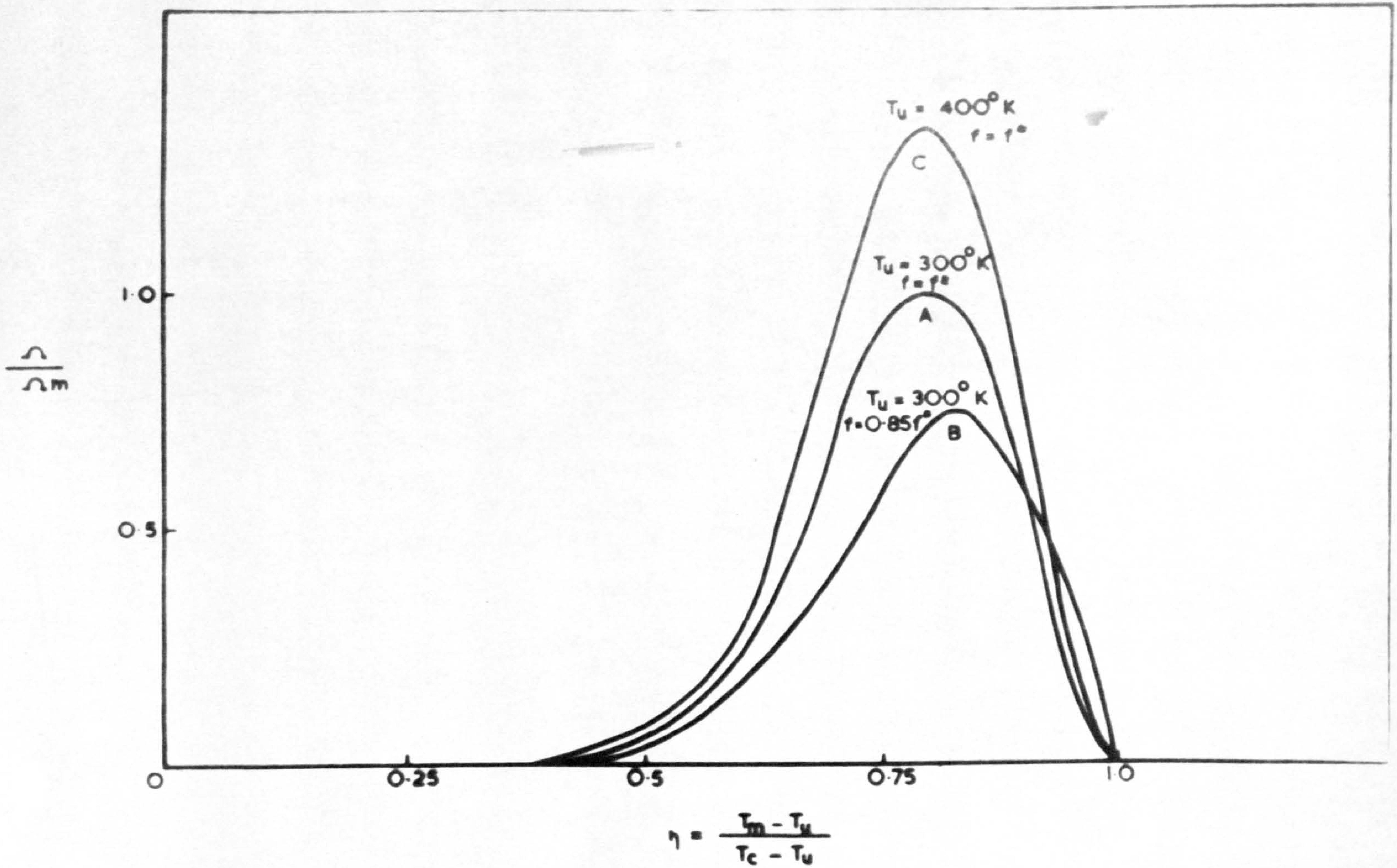



FIG. 3. PLOT OF RELATIVE REACTION RATE AGAINST MIXTURE REACTEDNESS.

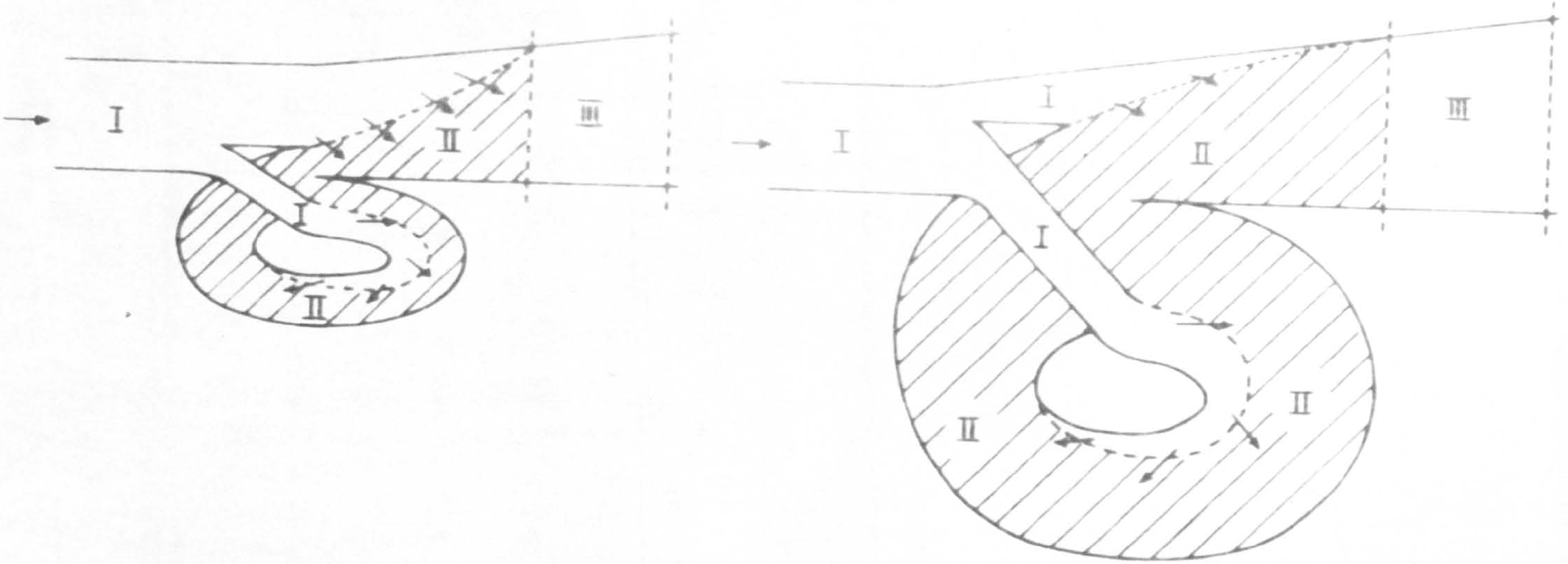
room temperature. Departures from the optimum proportion of fuel burnt which is about 85% decreases $\Omega$, increasing air/fuel ratio also decreases $\Omega$. Since the relative concentrations of fuel and air change as the fuel is exhausted, this latter curve shows some advantage over the stoichiometric case. Curve C also shows that the only other gain that can be achieved is by an external supply of heat by preheating the reactants.

This means that to achieve an acceptable figure (say 99%), the system must be run at a much lower loading or with a secondary combustion chamber in which combustion is completed independently of the main system. The extra volume required can be obtained from the right hand side of the curves using the integral

$$\Omega = \Delta\eta \Big/ \int_{\eta}^{\eta_1+\Delta\eta_1} \frac{d\eta}{\Omega} \quad . \qquad \text{where } \eta = \text{the combustion efficiency.}$$

Bragg[12] has shown in this way that for a premixed system a secondary volume equal to half the primary volume is necessary for 99% efficiency, and equal volumes for 99.9%. (That is providing that this region does not recirculate into the earlier zones).

The ideal state of affairs is represented diagramatically in Fig.4, which also demonstrates the necessity for the correct amount of recirculating products.




LOW RECIRCULATION RATIO

HIGH RECIRCULATION RATIO

TWO STREAM REACTION SYSTEM

FIG. 4. IDEAL COMBUSTOR (RECIRCULATION DIAGRAM).

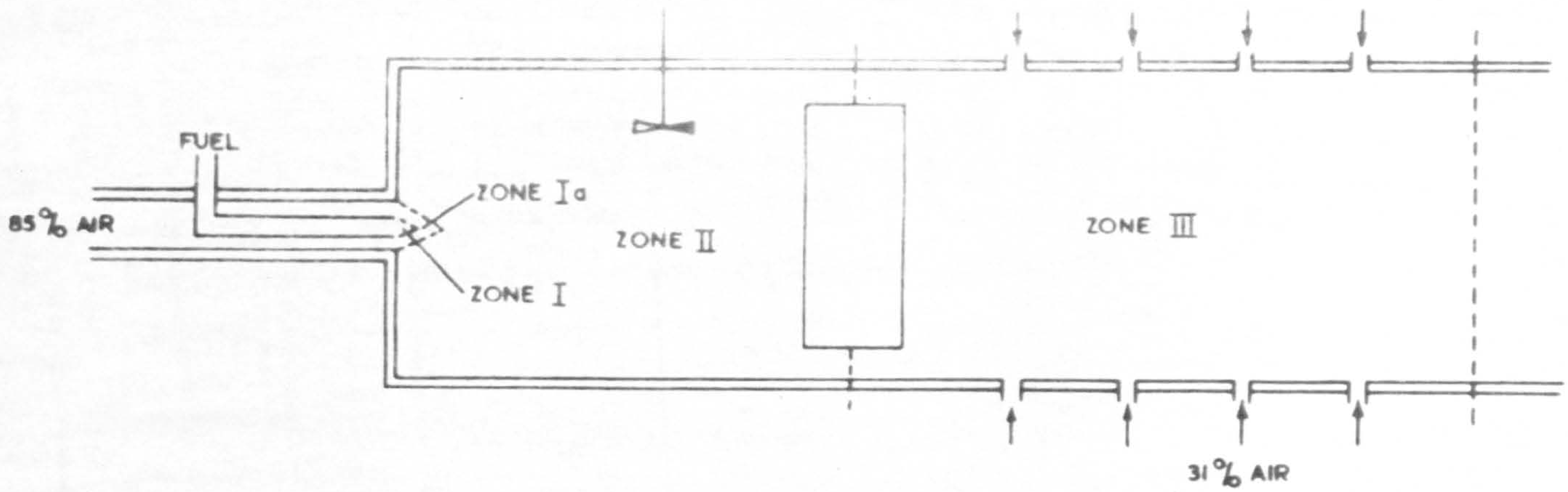
### 2.3.2 Practical non-premixed systems.

The foregoing argument applies as stated to the unusual cases of a premixed system. For most applications this is not possible and the combustion chamber must do duty also as a mixing chamber. In consequence there are additional difficulties in achieving this optimum satisfactorily in line with the combustion requirements of the system. The non-uniformity of mixture does allow however, for wider extinction limits, important in the gas turbine and for "grading" the fuel/air ratio from rich to weak along the flame to obtain a higher and more uniform combustion intensity.

Translating this into actual systems, two optimum systems have been proposed by Bragg[17] and Thring and Masdin[20]. These are shown in Figs. 5. and 6.

#### 2.3.2.1 Two stage process - Fig.5.

Reference has been made already to this argument in the consideration of premixed systems (Section 2.3.1) with the addition in this case of a mixing zone. The first combustion stage is fuel + 85% air, equivalent to Longwells homogeneous reaction and a second in which the final 20% of fuel remaining unburnt, is burnt with 31% air (to give 16% excess overall in this example) each stage requiring about an equal volume to achieve 99% efficiency.




ZONE I. – FUEL MIXING WITH PRIMARY AIR.
ZONE Ia – FUEL-AIR MIXING WITH 4 MASSES OF COMBUSTION PRODUCTS.
ZONE II. – HOMOGENEOUS REACTION ZONE.
ZONE III. – REMAINDER OF AIR MIXED IN & REACTION PROCEEDING FROM 80% COMPLETION TO 99·5% COMPLETION.

FIG. 5. IDEAL NON-PREMIX COMBUSTOR. FOR MAXIMUM COMBUSTION INTENSITY.

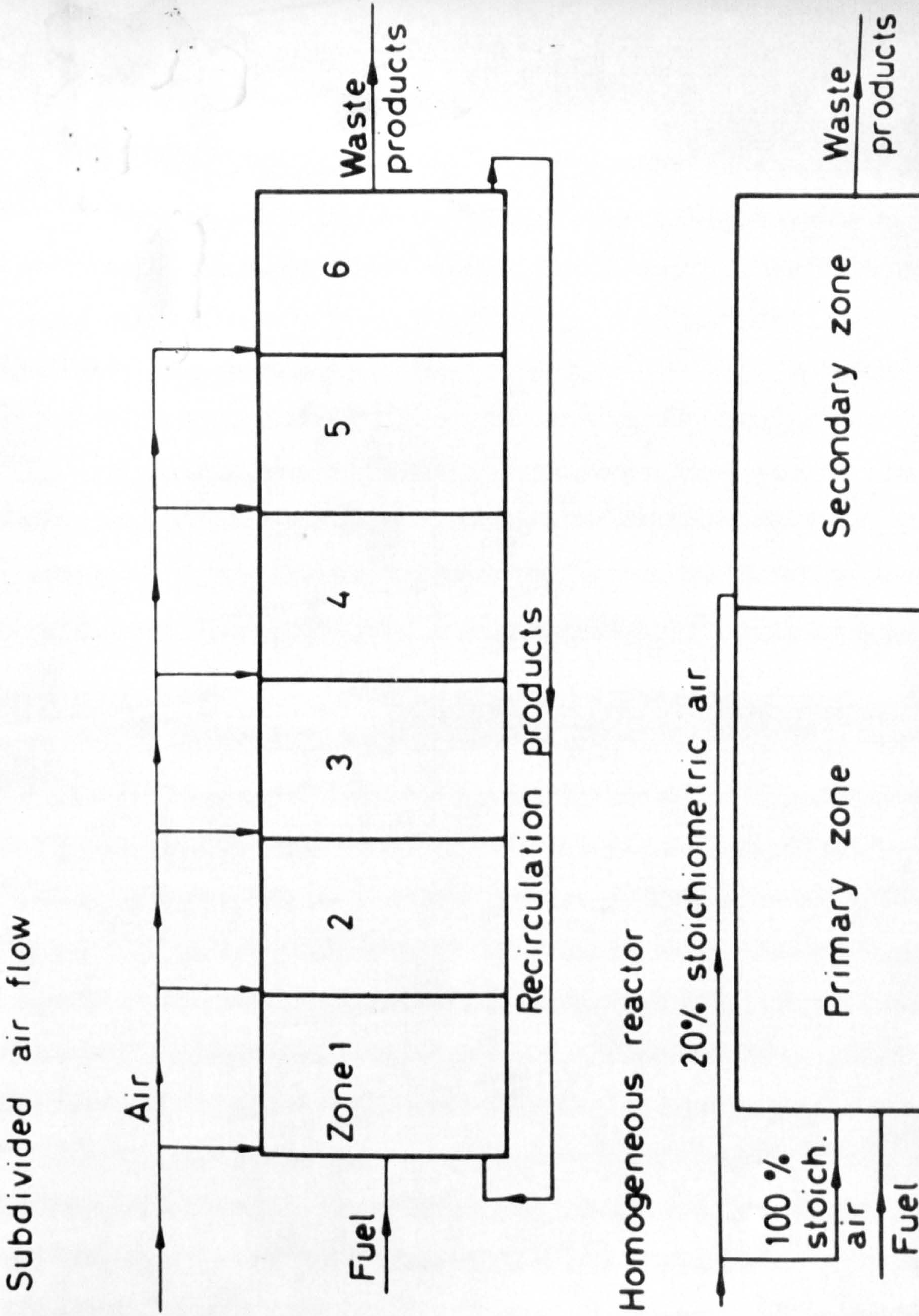



FIG.6. MULTISTAGE COMBUSTOR (HOMOGENEOUS REACTOR SHOWN FOR COMPARISON).

#### 2.3.2.2 Multi stage process - Fig.6.

In this case it has been calculated that by introducing air in stages down the chamber each time the reaction rate begins to fall off, even higher intensities could be obtained. Here also, recirculation is kept to a minimum since each stage provides hot products for the next and only sufficient need be recirculated from the end for the initial combustion stages. The increase in intensity obtained is of the order of 6% over the homogeneous systems. A good gas turbine combustion chamber might approximate to something like this sort of arrangement.

### 2.4 Actual Burner Systems expressed as an approximation of the Ideal.

The theoretical systems considered in section 2.3 take no account of the mechanics of the system in the spray, mixing, burnout of droplets, unfilled combustion chamber space and heat losses. It is assumed that the combustion intensity is limited by purely chemical factors. It is instructive to consider how far actual systems approach this ideal. For the purposes of this comparison it is assumed as a first approximation that the combustion intensity is proportional to some power of the relative pressure loss across the burner $\Delta p/p$. and this is plotted in Fig.7. The homogeneous reaction is on the right hand side, the $\Delta p/p$ being that quoted by Longwell[18], other systems represent other attempts to achieve different

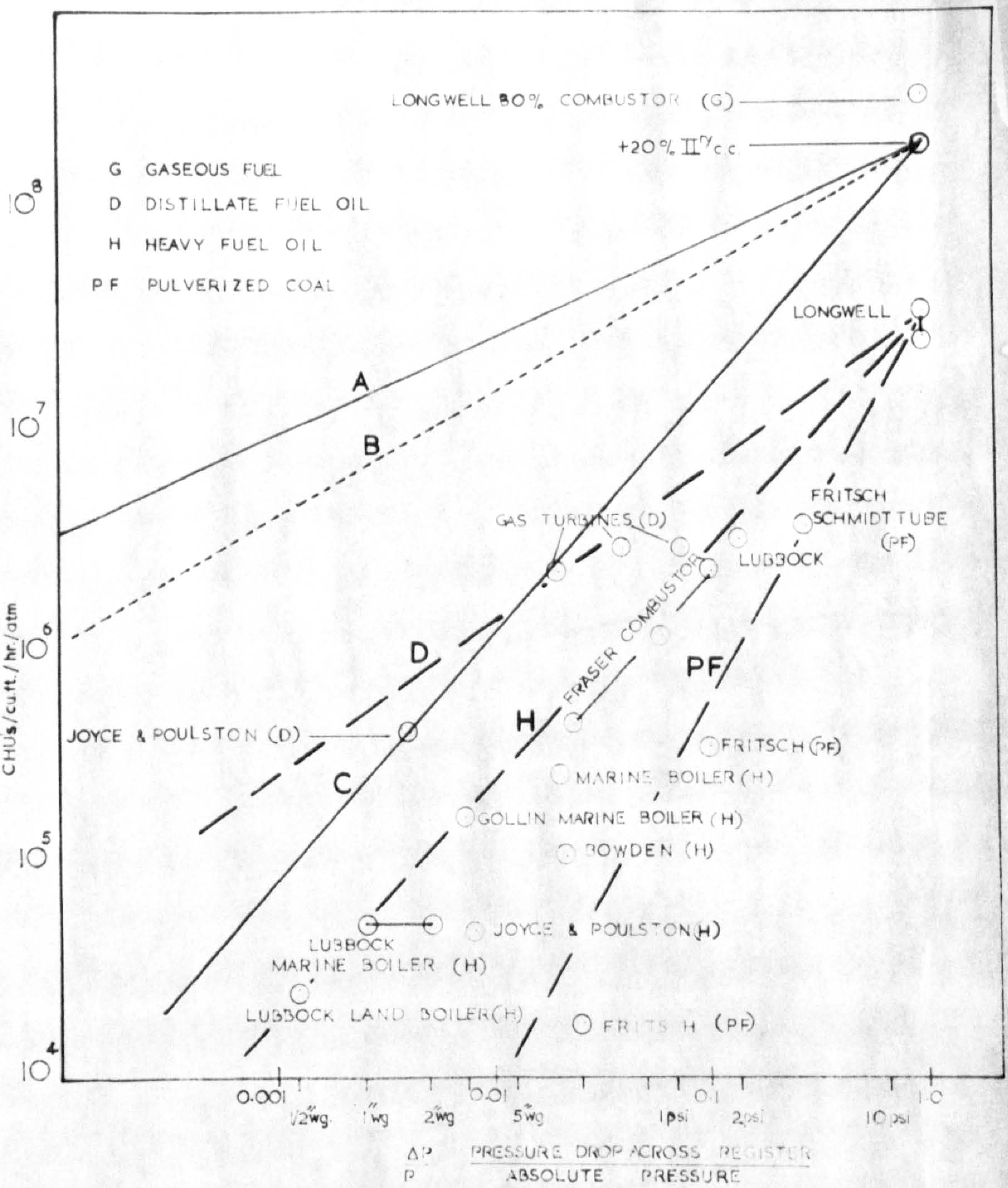



FIG. 7. PLOT OF COMBUSTION INTENSITY AGAINST RELATIVE PRESSURE LOSS FOR SOME REAL SYSTEMS.

compromises between $\Omega$ and pressure loss. In Fig.7. line A represents the ideal, calculated on the assumption that the pressure drop arises entirely from the loss of dynamic head of the gases entering the chamber. On this basis combustion intensity is proportional to $\sqrt{\Delta p/p}$ . The curve B corresponds to Lubbock's Empirical Rule $(\Delta p/p)^{2/3}$ (21), while the curve C which might be regarded as "best performance curve" is nearer to $\Delta p/p$. This last curve is drawn from the theoretical Bragg combustion chamber (equivalent to the Longwell + second stage for remaining unburnt fuel - see sections 2.3.1 and 2.3.2). This however assumes that it is possible to run approximately at this two stage ideal, and that there is no thermal load on the combustion chamber. The curves D,H, and PF are in fact probably more realistic best practice lines for distillate fuels, heavy fuel oil and pulverized coal respectively, and each curve converges on the Longwell system as operated as a homogeneous reactor at 99% efficiency.

The curve D for distillate fired systems with a low thermal load is approximately the same slope as those predicted by curves A and B. The greater energy necessary to produce the same heat release rate for heavy fuel oils and pulverized coal is accounted for by the increasing difficulty in each case in burning all the fuel and by the fact that relatively higher thermal load is placed where it is "seen" by the flame, for boilers as opposed to gas turbines and again for P.F. fired land boilers as opposed to the usual run of marine boilers. There is also probably a scale effect which will mean that large units will consume relatively more energy for the same results as the smaller ones.

The conclusions that can be drawn from this diagram are that improvements in combustion intensities can be achieved in some cases by, (i) design of a system with the proper mixing stages (i.e. as point 2.3.2.2). This should result in considerable improvements in all cases and push the curve D,H and P.F. nearer A and B. (ii) by improved local aerodynamics to improve the rate of combustion of the slow burning carbon residues from the heavy fuel oil and coal. (iii) by reducing the thermal load on the combustion chamber and making as much of the heat transfer as possible "out of sight" of the flame. This in fact would help (ii) as well. These last two points would push the curves H. and P.F. towards D.

## 2.5 Generalized Theory.

On the basis of the consideration of the basic chemical and physical processes, it is possible to indicate the important parameters related to the process and to construct heat and mass balances. Bragg[12] has carried out an analysis of this type by assuming that if the net heat gain for a short period is equated to zero and taking into account conduction and convection and the instantaneous heat release corresponding to the local conditions, the pattern and performance of the burning system is then completely defined by the dimensionless groups, Reynolds No., and "Reaction Rate No.", and the fuel/air ratio distribution. The "Reaction Rate No." is defined as

$$\frac{\bar{\Omega}\, D}{\rho\, v\, C_p\, \Delta T}$$

where $\bar{\Omega}$ is the mean average value of the heat release rate in the combustion chamber,
D is the diameter of the chamber,
$\rho$, v, $C_p$ are the density, velocity and heat capacity of the gases in the chamber and
$\Delta T$ the temperature rise.

How far this generalized theory is applicable to actual systems is discussed later.

## 2.6 Component parts of the dynamics of oil burner systems.

In the consideration of flames, departure from premixed systems to actual atomized oil flames introduces a number of extra variables, which as yet can only partially be put together to form at best a somewhat incomplete picture.

### 2.6.1 The Atomizer.

The detailed study of atomizer systems is outside the scope of this work; the theory of their operation is discussed in a number of reviews[22-26] and an illustration of a very simple pressure jet is given in Fig. 8. showing the principle of operation, the swirl chamber and the nozzle. Sections of actual nozzles are shown in Fig. 9.

The disintegration of the spray is due to the convergence of the two faces of the hollow cone by virtue of the mechanics of the system. The sheet develops holes and subsequently breaks down to filaments and droplets[22][27] having a high frequency instability of its own of perhaps 10,000 cps. This part of the process is of only second order importance as far as subsequent processes in the flame are concerned except in so far as this constitutes part of the delay before the fuel ignites. However, the flame itself may be affected considerably by the characteristics of the spray such as the droplet size, distribution and momentum.

Various relationships have been proposed for the through put of pressure-jet burners, the S.M.D. of the droplets and the spray angle.

### 2.6.2 Throughput.

The normal relationship for fuel throughput Q through an atomizer is usually taken as

$Q \propto d_o^2 \, P^{0.5}$ .......(4) From this the flow number (FN) is defined as $\frac{Q}{P^{0.5}}$ gal./(psi)$^{0.5}$.

In more accurate dimensionless terms due to Radcliffe[28]

$$\frac{Q'}{d_o^4 \, P \, \rho_f \, g} = 0.884 \left( \frac{\mu_f d_o}{Q} \right)^{0.22}$$

where $P$ = atomizing pressure.

$d_o$ = orifice diameter.

$\rho_f$ = density of fuel

$\mu_f$ = viscosity (absolute).

### 2.6.3 S.M.D. and spray angle.

The S.M.D.* has been defined by Fraser[26] on theoretical grounds as

$$d \propto \left( \frac{FN}{\theta} \frac{s}{P} \right)^{\frac{1}{3}} . \left( \frac{\rho_f}{\rho_a} \right)^{\frac{1}{6}} \qquad \ldots\ldots\ldots\ldots\ldots (5)$$

where $s$ = surface tension of the liquid.

$\theta$ = sine of half the spray angle.

$\rho_a$ = density of atmosphere.

and a number of empirical relationships have been reported for different atomizers. Their use however is of somewhat limited applicability, being only really true for the atomizer tested.

* Sauter Mean Diameter. This is the normal method of obtaining an average size and is the diameter of a uniform droplet with the same surface to volume ratio as the actual distribution.

A discussion of the problem of calculating and measuring droplet sizes for the atomizer used is carried further in Appendix III.

The spray angle is a function of the geometric dimensions of the atomizer and is normally reckoned to remain fairly constant for a given atomizer, providing the spray can be fully developed. The exceptions to this are the Duplex and Spill atomizers where this varies with input, port size and relative pressures in the system.

The relationship:-

$$d \propto \frac{\mu_f^{0.20}\ s^{0.50}}{\rho_f^{0.10}} \qquad \frac{\mu_a^{0.25}\ Q_f^{0.25}}{\rho_a^{0.25}\ P^{0.40}} \quad \ldots\ldots\ldots \qquad (6)$$

obtained by Radcliffe(28) is used for comparative calculations.

2.6.4 Spray penetration.

Data on the penetration of sprays is scarce, but Bayley(29) is reported as having carried out an analysis of an evaporating droplet into a combustion chamber and to have obtained a graphic solution which gave the distance of penetration

$$S \propto d^{1.8}\ P^{-0.14}\ V^{0.9}\ T^{-0.055} \quad \ldots\ldots\ldots\ldots \qquad (7)$$

where $V$ is the velocity of the fuel spray.

Over the range of interest, i.e., with $\lambda$ the evaporation constant not too small, this can be reduced to

$$S = 0.1452\ \frac{\rho_f^{0.469}}{\mu^{0.075}\ \lambda^{0.531}} \qquad \frac{V^{0.836}\ d^{1.836}}{\rho^{0.164}} \quad \ldots \qquad (8)$$

which appears to account for all the relevant variables.

### 2.6.5 Droplet size distribution.

So far all proposed distributions are only approximations. One of the simplest is the Rosin-Rammler[30] distribution although that proposed by Mugele and Evans[31] is slightly more accurate. A knowledge of these distributions can be of value in assessing the behaviour of the spray in the air stream.

### 2.6.6 The Hydrodynamic performance and combustion of oil droplets.

The course of evaporation and combustion of the droplet and the effects of turbulence and the hydrodynamics of droplet clouds are for practical purposes very inadequately understood.

Frössling[32] has accounted fairly well for the evaporation of droplets of fairly low volatility under nearly uniform conditions. There is as well very little correlation between the results obtained from the flow patterns produced by freely moving particles. The most satisfactory results from the behaviour of droplets so far has been based on rather drastic simplifying assumption. For example, to calculate droplet paths Clarke[33] and Poulston[34] both assume that the droplets behave as solid spheres which allows the use of the standard results of the drag coefficient. Presumably the actual behaviour of the droplet will be profoundly influenced by the decreasing mass due to evaporation and combustion. Other workers including

Godsave[35], Spalding[36] and Masdin[37] have investigated the combustion of fairly large single droplets (e.g. 1000 $\mu$). The quantitative application of their data to practical combustion problems is still rather uncertain. Unfortunately as Putnam et al[38] point out, actual combustion processes involve a large number of variables many of which are very difficult to determine due to the problem of establishing the precise conditions at any point.

A plot of burning time against droplet diameter due to Essenhigh[39] is given in Fig. 10. This illustrates two important points, firstly that the smaller droplets (i.e., with the greatest surface to volume ratio) burn very much faster than the larger droplets in the spray. The stabilization of pressure jet flames is often attributed to this. Secondly it shows that for any combustion system to be satisfactory, the burning times of the largest drops must be taken into consideration and the residence time of the gas must be long enough for the process to be completed. Thus for a boiler with a combustion intensity of $10^6$ Btu/cu.ft.hr. the burning time of the largest droplet must be less than the residence time which will be about 0.06 secs. This is particularly important with residual fuels with long burn out times for carbon particles of the order of ten times that of the volatiles from the same drop which have been

FIG. 8.

Illustration Showing Principle of Pressure-jet Atomizer.

FIG. 9.

Actual Types of Pressure-jet Atomizer.

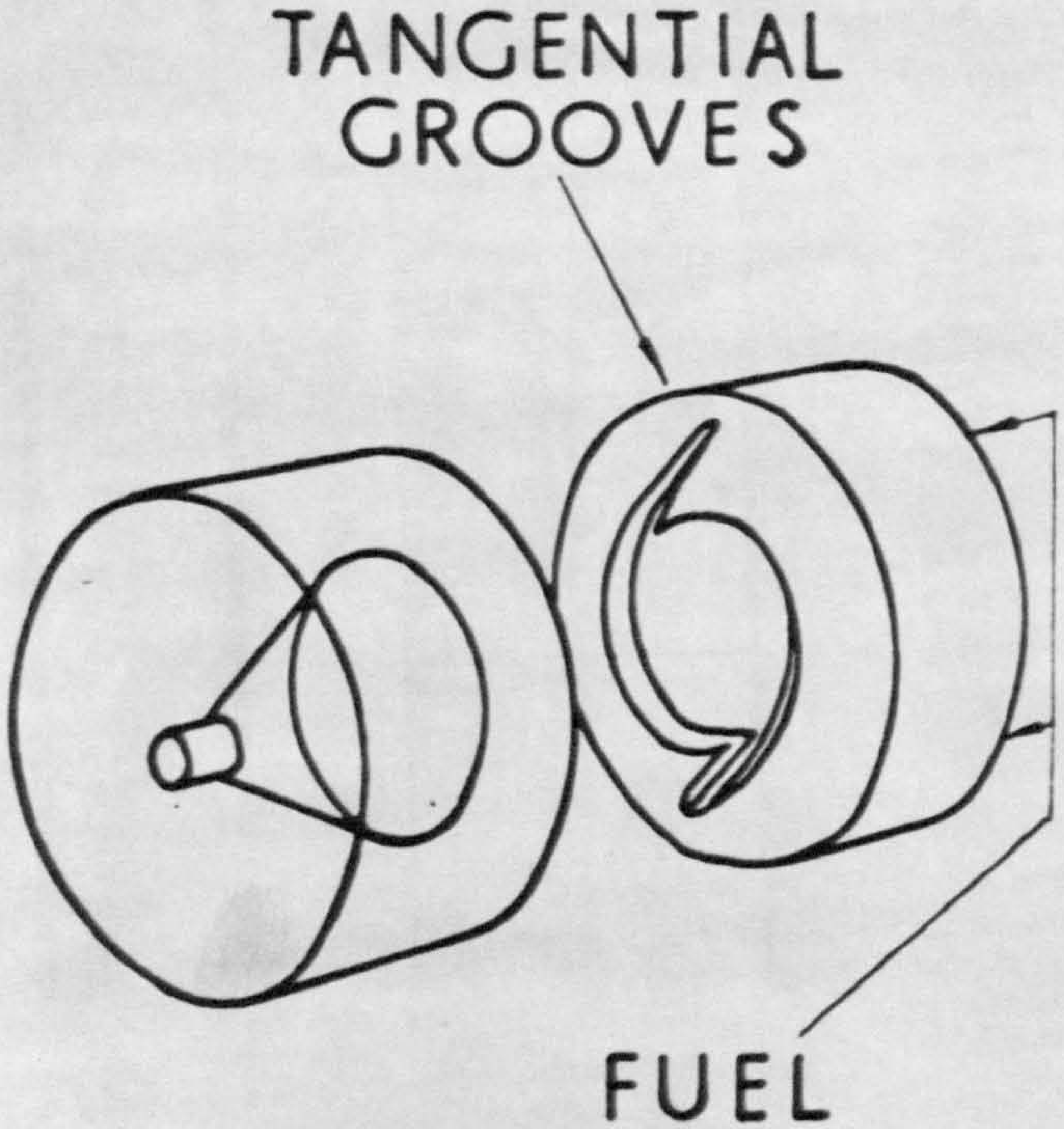
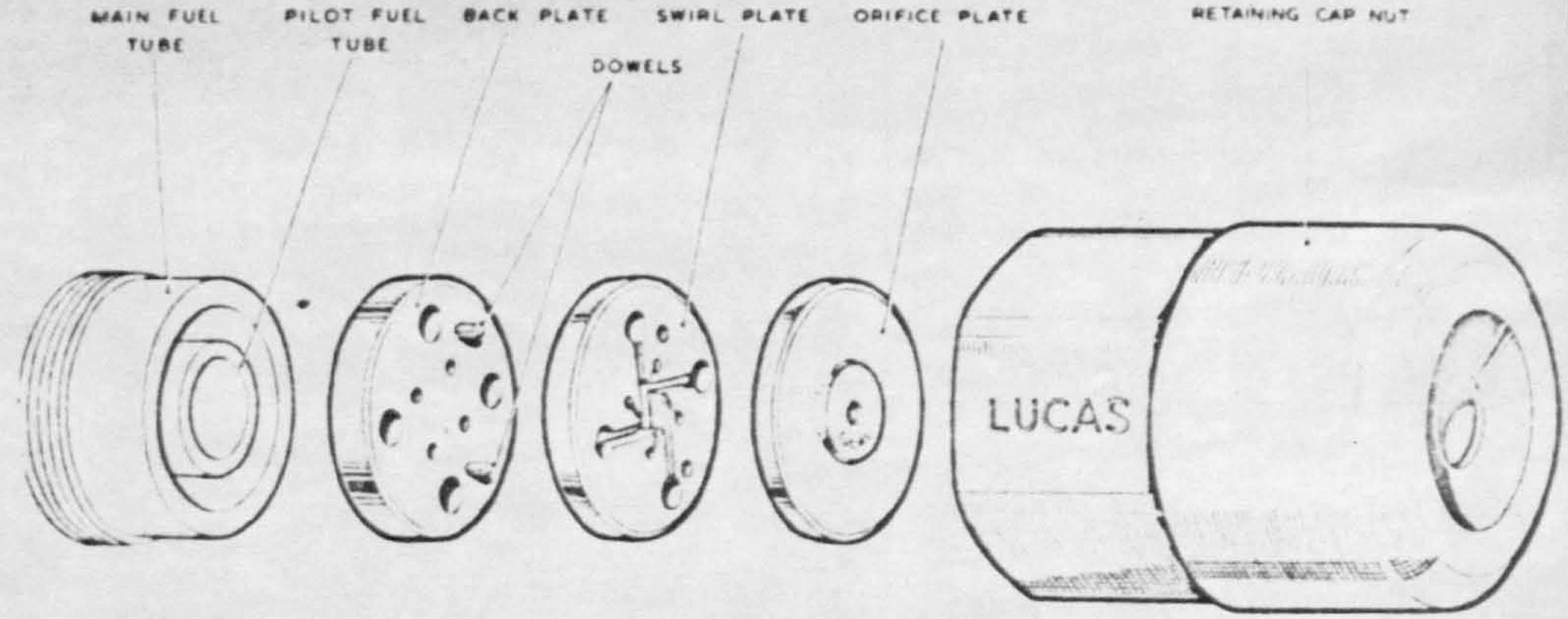
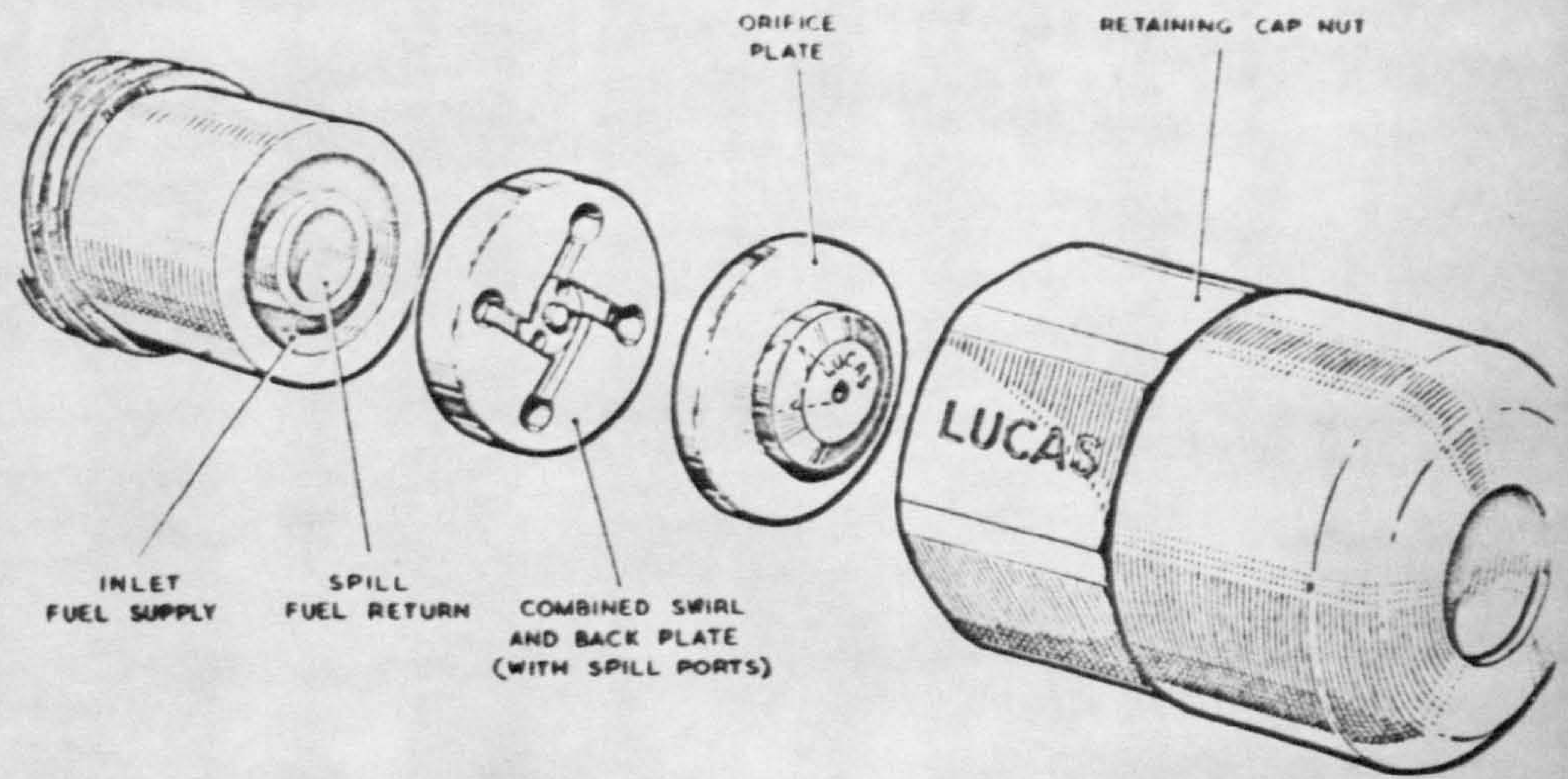



FIG. 8.




DUPLEX ATOMISER & CAP NUT ASSEMBLY




DISMANTLED VIEW OF SPILL ATOMISER

FIG. 9.

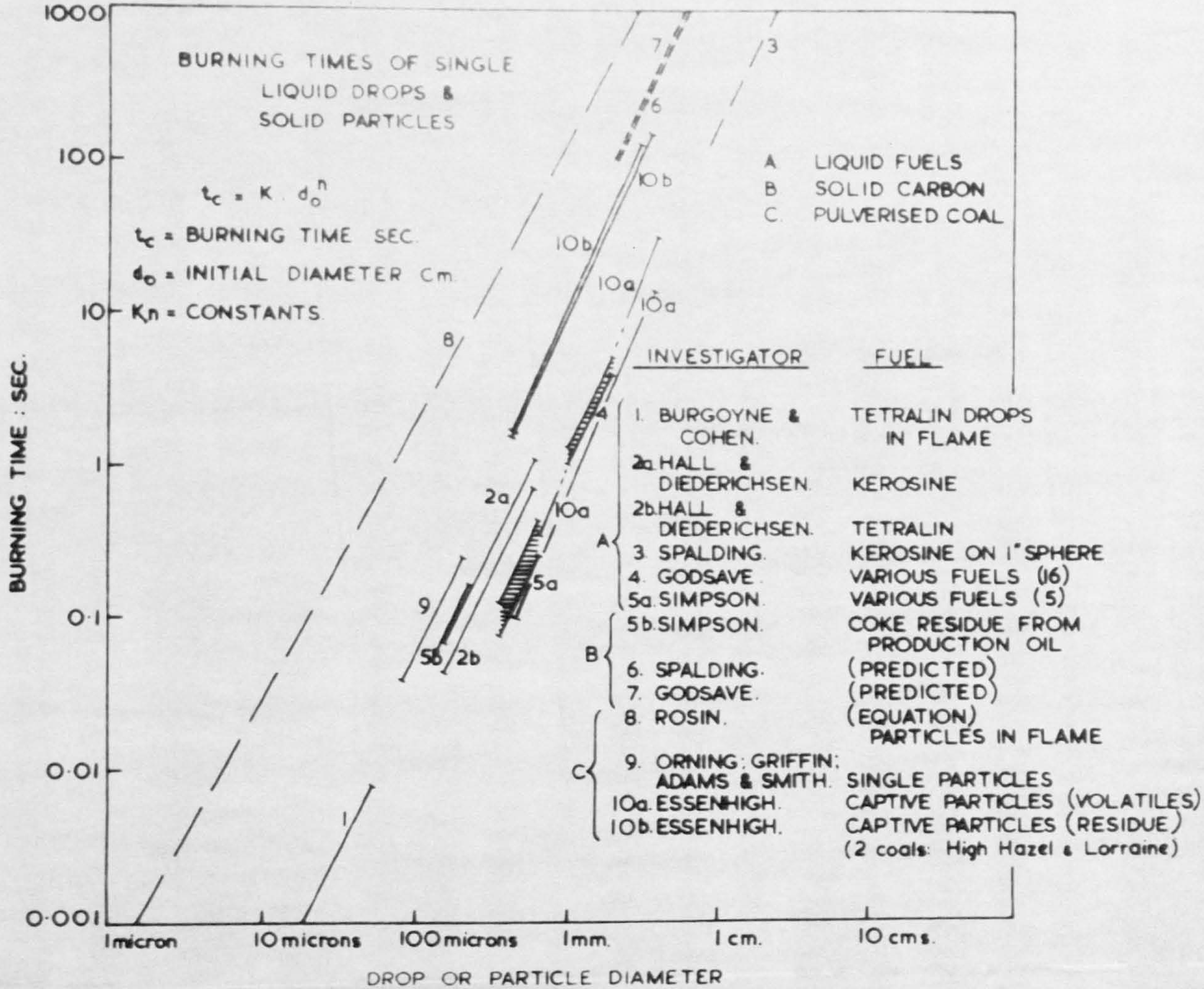



FIG. 10. PLOT SHOWING BURNING TIMES FOR VARIOUS LIQUID AND SOLID PARTICLES (COLLECTED BY ESSENHIGH)

reported by Masdin & Foster[40]. Thus the combustion intensity governs the maximum droplet size which can be accommodated.

### 2.6.7 Flame Propagation Rate.

Data on the rate of flame propagation in oil sprays is somewhat sparse. It may be presumed that this will depend on similar factors to the reaction rate, i.e., oxygen concentration and temperature. There is need for data for oil parallel to that obtained by Beer[41] for a pulverized coal cloud in a controlled mixing history furnace. In general flame speeds for "quiescent" and "stationary" flames, where heat transfer back is largely by radiation, seem to be of the order of 3 ft./sec. However, as Essenhigh and Fells[42] point out, the introduction of recirculated products and turbulent mixing may increase this by an order of magnitude. This suggests that turbulent flame speeds are a function of the gas velocity and the Reynolds number, a conclusion that has been put forward by several workers[43][44].

### 2.6.8 Ignition Delay.

The ignition delay is dependent like the reaction rate and rate of flame propagation on the chemical kinetics, it is particularly important in that it controls the processes up to the point of ignition. Mullins[19] and Masdin[35] found that delay times could be defined in terms of temperature and pressure.

Both these workers were able to determine delay times for a range of temperatures and air vitiation. The chief disadvantage in applying the results is that the delay times have been determined for constant values of $p_{O_2}$ and T during the process of heating up and ignition. In the case of Masdin's any interaction effects from other droplets were also absent.

In practice, a droplet leaving an atomizer will encounter a whole range of conditions in terms of temperature, gas composition and gas velocity between leaving the atomizer and ignition. This means that the calculation of absolute ignition delays is very difficult unless the conditions are known with great precision. This is particularly true in the case of the temperature term which appears in the exponential term. It is perhaps a little more satisfactory to use this to calculate comparative ignition delay terms for different fuels, although it is still necessary to choose the temperature with care (see Section 3.3.3.8).

## 2.7 The Aerodynamics of Jets

### 2.7.1 The Theoretical basis of Jet Work.

The performance of plain jets, especially with respect to momentum, heat and mass transfer have been studied widely and the work reviewed in a number of places for instance by Forstall and Shapiro(45) and Krzywoblocki(46).

The theory of turbulent jets has been attacked in a number of different ways, but the most widely used basis is Prandtl's momentum exchange concept. An example of the use of this type of approach is Newby and Thring's(47) application of it as applied to flame length and scaling criteria. The conservation of momentum implies that although exchange takes place between velocity and local pressure the total momentum does not alter.

#### 2.7.1.1 Simple Jets.

The behaviour of single jets injected into free space has long been understood fairly generally. The entrainment properties of a jet have been shown to follow the relation(48)

$$m = m_o \left(0.40 \frac{x}{d_o} - 1\right) \quad \ldots\ldots\ldots \quad (9)$$

where $m$ = mass flow at distance from the nozzle.

$m_o$ = initial mass flow.

$d_o$ = nozzle diameter.

and the axial velocity after a critical distance is known to decrease inversely as the distance travelled by the jet. The velocity distribution also has been shown to be of a Gaussian distribution form.

#### 2.7.1.2 Complex Jets.

More recently because of their importance in relation to practical problems, interest has focussed on annular, multi and concentric jets. The effect of the first of these is to introduce the equivalent of a bluff body into the stream with a consequent low pressure zone and local recirculating eddies. Studies of these have been carried out by Ullrich[49], Cohen de Lara[50], Beer, Chigier & Lee[51], and Patrick[52]. The results have varied somewhat depending on the presence of thick or thin walls between the jets and the relative velocity of the two streams. It has been shown that after about 6 diameters downstream, the jet profile usually progressively approximates to the normal jet distribution for a corresponding single jet.

### 2.7.2 Recirculation.

The control of recirculation in confined jets has already been shown to be a very important part of flame aerodynamics and has formed the subject of a considerable amount of study. Two main theories of recirculation have been suggested, the first by Thring and Newby and the second by Craya and Curtet.

Thring and Newby suggested[47] that recirculated material should be studied as a function of the parameter θ which for a twin fluid system is defined as $\frac{m_o + m_a}{m_a} \cdot \frac{r_o}{L}$ They assumed that for $r_o << L$ the jet could be assumed to behave as a free jet until it comes into contact with the wall (so far as behaviour inside the jet is concerned). Using the results of Hinze and van der Hegge Zijnen for the mass $m_1$ entrained in distance $x$ which is from equation 9:-

$$m_1 = m_o \left(0.2 \frac{x}{r_o} - 1\right) \quad \ldots\ldots\ldots\ldots \quad (10)$$

where characteristic length $x_1$ at which a "free jet" would touch the walls = 4.5L. They propose that since no material is added or lost directly from the recirculated zone

$$m_r = (m_1 - m_a) = m_o \left(0.2 \frac{x_1}{r_o} - 1\right) - m_a \ldots \quad (11)$$

$$\theta = \frac{1}{0.9} \cdot \frac{m_o + m_a}{m_r - (m_o + m_a)} = \sqrt{\frac{G_T}{G_o}}$$ (the ratio of the total momentum flux to that of the fuel). ............ (12)

$m_o$ = mass flow of burner fluid.

$m_a$ = mass flow of burner air.

$m_r$ = mass flow of recirculated gases.

$r_o$ = burner radius.

L = characteristic distance.

G = momentum. (Subscripts T and O for total and oil respectively).

The last named workers developed the idea of the "equivalent" single jet which expresses the individual momenta as a single equivalent momentum. Temperature differences are allowed for by the "equivalent" isothermal jet, i.e., by calculating the equivalent diameter that the stream would need if it left the burner at the mean temperature further down the jet, while maintaining momentum similarity.

This gives rise to an expression for the "equivalent burner diameter" for a twin fluid stream where the final density is $\rho_g$

$$d_o' = \frac{2(m_o + m_a)}{\sqrt{(G_o+G_a)\rho_g}} \qquad \ldots\ldots\ldots\ldots \qquad (13)$$

This "equivalent" burner diameter" is an extremely useful way of equating jets of different size or density as for example in scaling combustion systems and cold models.

Craya and Curtet[53] adopted a more elaborate approach, which unfortunately involves severe approximations.

Three characterisations for recirculation have been proposed, these are complementary rather than alternatives. They are:-

1. Physical limits of various zones in the flow pattern.
2. Mass of material recirculated (usually cited as $m_r$ max.)
3. The mixing pattern as determined from the concentration of temperature profiles.

These may be considered as follows:-

1. This is particularly sensitive to the geometry of the system and from the scaling point of view is only a suitable means of comparing flow patterns in similarly scaled systems.

2. Is a useful overall criterion for obtaining a convenient comparative figure and has been shown[50] to be comparatively insensitive to alterations in the flow pattern shape due to downstream effects ($\pm$ 5% in Cohen de Lara's experiments).

3. The information both 1 and 2 provide, while important, is rather limited particularly as regards information on composition and mixing. For this reason it is necessary to go further than these and study the variation in composition in the system with recirculation not only in bulk flow quantities but in actual local concentrations, which are relevant to what is happening combustion wise. Methods used to study these can be either cold models:-

(a) with stream either of different density or composition to simulate burner fluid[54-56].

(b) with a tracer introduced at some downstream point and determination of "influence coefficients"[57][58],

or secondly by measurements on actual flames using radioactive or inert gas tracers or occasionally the recirculated combustion products themselves[51].

### 2.7.3 Experimental studies of jets.

Experimental studies on jets have fallen into three main sections:

1. Profiles and behaviour of simple free jets and jets issuing into a stationary or slowly flowing secondary stream(47)(48)(50)(54).

2. Profiles of annular and concentric jets and swirling jets.

3. The effect of chamber variables.

The reason for these different approaches is based on special requirements. In case 1, that of determining in overall terms the amount of gas recirculating to provide data for the kind of approach discussed in Section 2.2.1.5 and also the estimation of mixing lengths along the jet, the main variables of interest are the momentum of the jet and the ratio $r_o/L$. In the second case the requirement has been a much more detailed study of the behaviour of particularly the early stages of annular and multiple jets and the behaviour of a number of variables other than momentum e.g., the width of the annulus the cone angle of the quarl $\alpha$ (Quarl is burner term referring to the surround of the air jet usually made of refractory material) - and the angle of the swirl $\beta$. This study is concerned mainly with flame holding, mixing between the jets and combustion intensity. The third section has only received rather scant attention except in the very simplest cases of square or circular box surrounding the jet. Downstream effects have a considerable influence on the flow pattern in the combustion chamber(50)(59)(60).

## 2.8 Burner Aerodynamics.

### 2.8.1 Blockage due to atomizer and register parts and flame stabilization.

Any burner which involves solid parts inside the air inlet will cause some degree of "shadow" downstream of the burner. This will result in a low pressure region and a compensating local eddy. Since flame anchoring depends on the presence of a region where the rate of flame propagation is just balanced by a fairly moderate flow velocity, this type of eddy downstream of the atomizer has often been regarded as helping to produce this condition, and the behaviour of bluff body and other stabilizers have been widely studied(61)(62). However, an important consequence of the presence of a flame is a considerable modification in size in the consequent eddy zone(58)(63), which means that any cold aerodynamic results need to be used with caution.

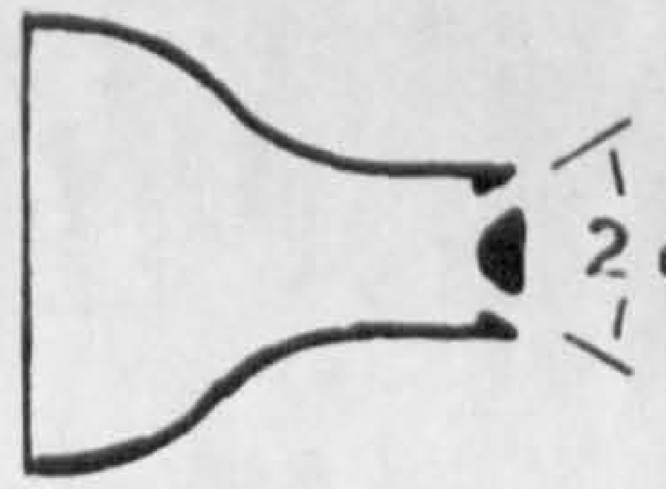
Also, it has been suggested that the presence of swirler blades and other mechanical devices produce discrete jets with weak recirculation between them with similar results. There is however, very little experimental evidence to confirm or deny this and the action of zones of this type is obscure(64).

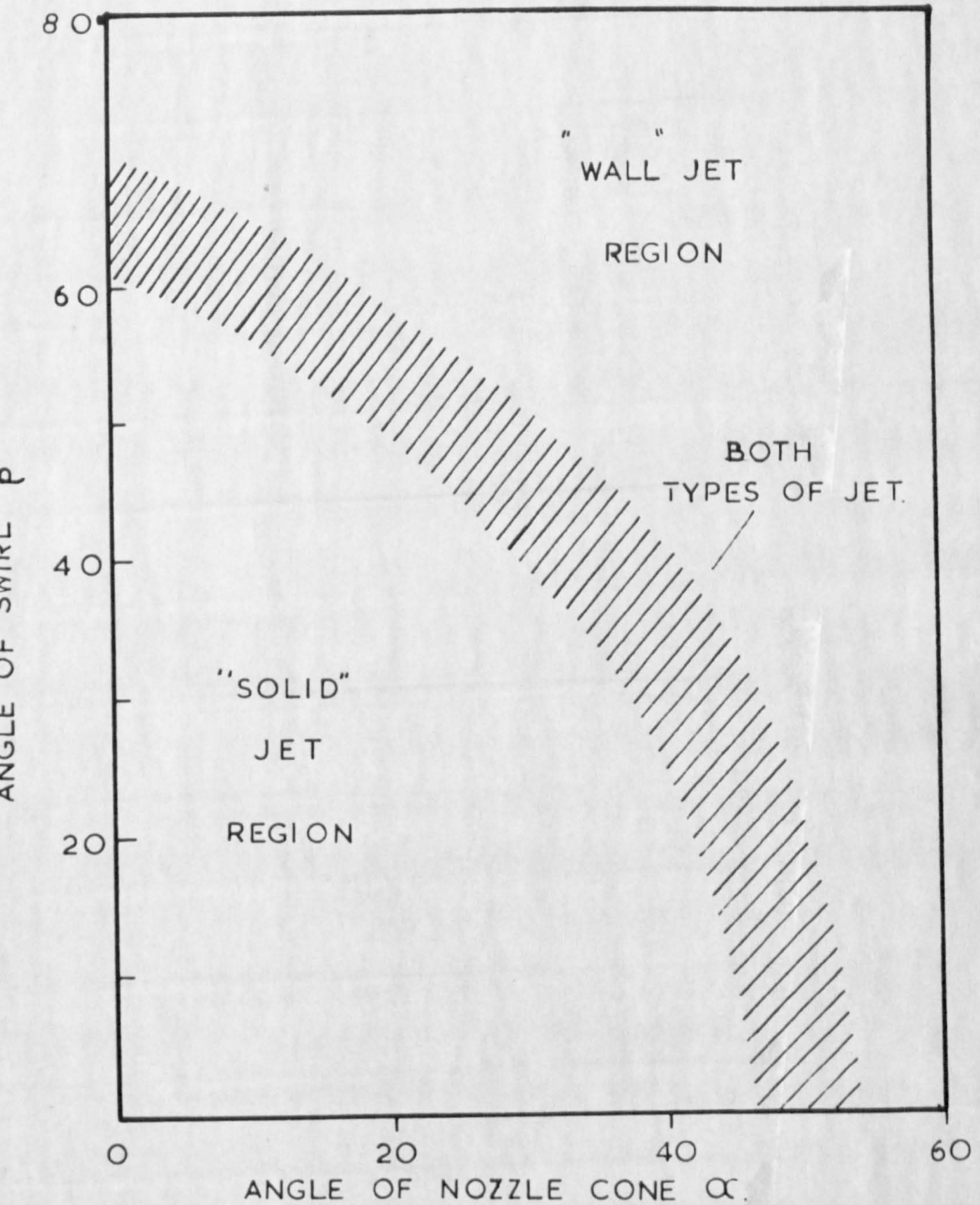
..

2.8.2 Flame length, mixing and jet configuration.

In the field of simple enclosed jets the type of aerodynamics has been widely investigated. Factors affecting combustion length and similarity between different jets have been studied by Thring & Newby in cases where the combustion pattern is precisely governed by the turbulent mixing process caused by the entrainment of the surrounding air and gases by the incoming jet of fuel and air. In the pressure-jet flame, however, the picture is much more complex as the fuel distribution is dictated to a considerable extent by the atomizer characteristics and is consequently not uniform across the jet. The jet itself is also often a complicated configuration due to the burner components. Work on these complicated configurations is still at a very early stage and not a great deal of data is yet available. Some of the most interesting work has been carried out by Ullrich who studied experimentally an annular jet with the cases (1) of a plain parallel nozzle, (2) a diverging nozzle, and (3) the effect of swirl superimposed on this. Unfortunately he did not develop any theoretical background for this part of his work. The three main conclusions of his work were, firstly that above certain values of $\alpha$ and $\beta$ the normal "solid" jet opens out into a "wall" jet with an intermediate region where the one form is stable and the other insipient. The plot of values for these effects are shown in Fig. 11

FIG. 11a. TYPE OF ANNULAR NOZZLE USED BY ULLRICH.




# FIG. 11b. PLOT OF EFFECT OF CHANGING CONE ANGLE AND SWIRL ANGLE.

(FROM ULLRICH'S DATA.)

These results are very important as they show that it should be possible to predict very readily from a preliminary study the points where crucial changes in the jet configuration are likely to occur for various combinations of $\alpha$ and $\beta$. Secondly Ullrich showed that by the use of bluff bodies to split the jet into several sections, it was possible to obtain a stable half way stage between "solid" and "wall" jets. In other words to project concentric jets without coalescence.

This in fact forms the basis of the work so far carried out on swirl, although the work reported in this thesis and the current work of Beer et al[65] and Hubbard[66] on actual flames with a swirled air supply extend some of these ideas.

2.8.3 <u>A discussion of actual pressure-jet burners.</u>

There is in the literature a remarkable dearth of information on the development of actual burners with the exception of those sited in references (5-9). The development of pressure-jet burners can be summarized briefly as follows: Firstly very early development indicated that for reasonable performance, forced draught was almost essential. A minimum of 0.6" w.g. approx. has been quoted as the lowest permissible draught loss[5], although this figure is sometimes considerably lower on small units. Secondly, atomization was found to have a considerable influence on the achievement of short flames without too lower efficiencies. Thirdly the control of the air round the atomized oil was found to have a very profound influence on the behaviour of the flame.

The simplest form, namely pressure jet firing into hot surroundings with an air flow from a simple nozzle or quarl, has been used for applications such as cement kilns. Somewhat similar flames also have been investigated at IJmuiden.

The next stage was to provide a simple convergent divergent throat to the air nozzle as illustrated in Fig.12(a) with the oil spraying close to the divergent section.

More refined attempts at burner design almost always have included either a divergent annulus for the air supply or the introduction of an angular component to air supply usually known as "swirl". A burner of the first type is illustrated in Fig.12(b) which shows a widely divergent annulus with a small amount of air addition through holes inside the annulus presumably to prevent coking. The introduction of swirl in the air supply is usually produced by a bladed device somewhere in the burner, typical examples being shown in Figs.12(c-e). The last of these being a modern marine pattern designed for a high draft loss, wide range of fuel throughput and to provide a flame well away from the immediate region of the quarl, and is consequently known as a "suspended flame wide range register".

FIG. 12.

Various Practical Air Registers in Use.

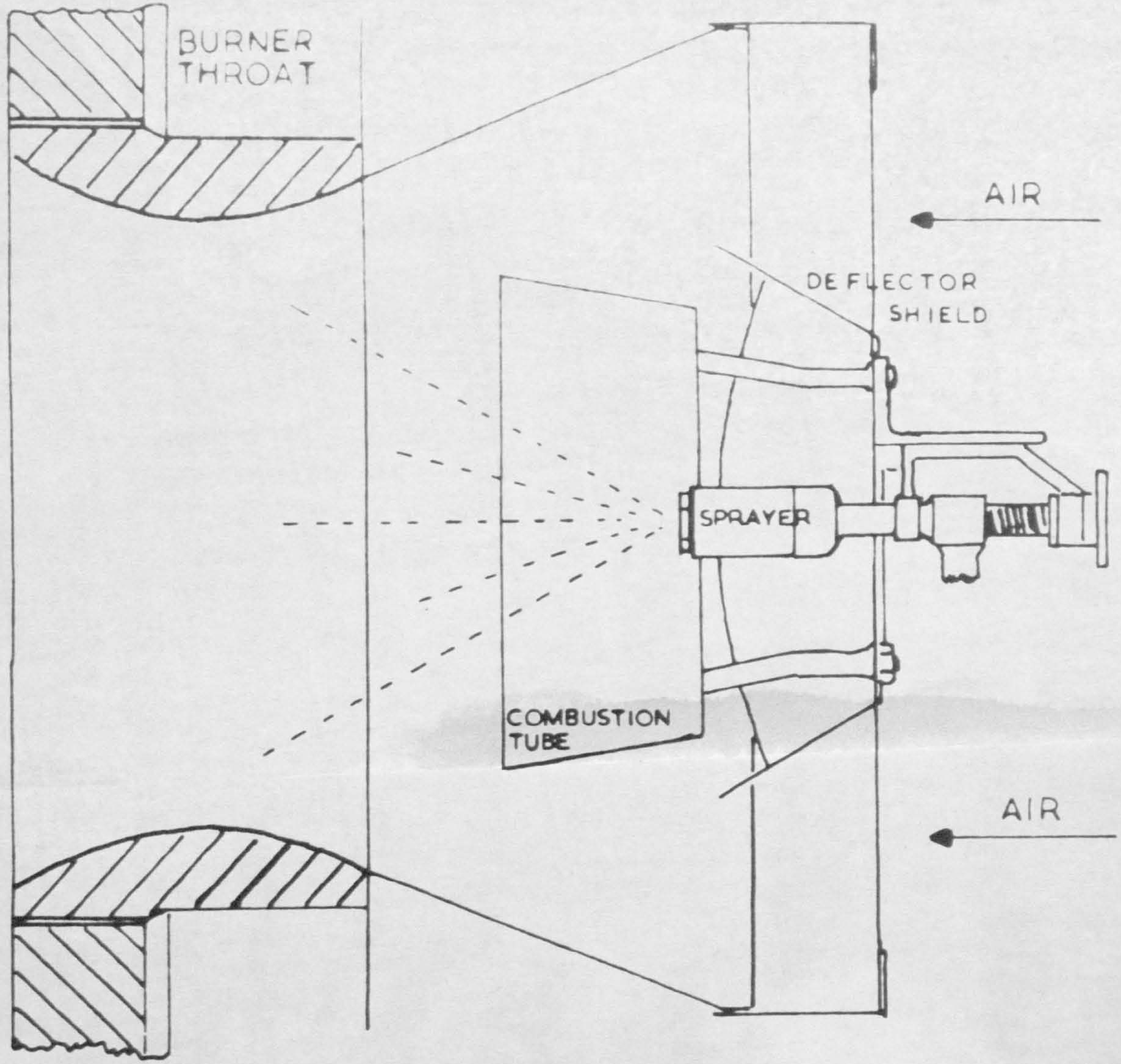



Fig.12.a. 1941 type Admiralty air register
(impingement type)

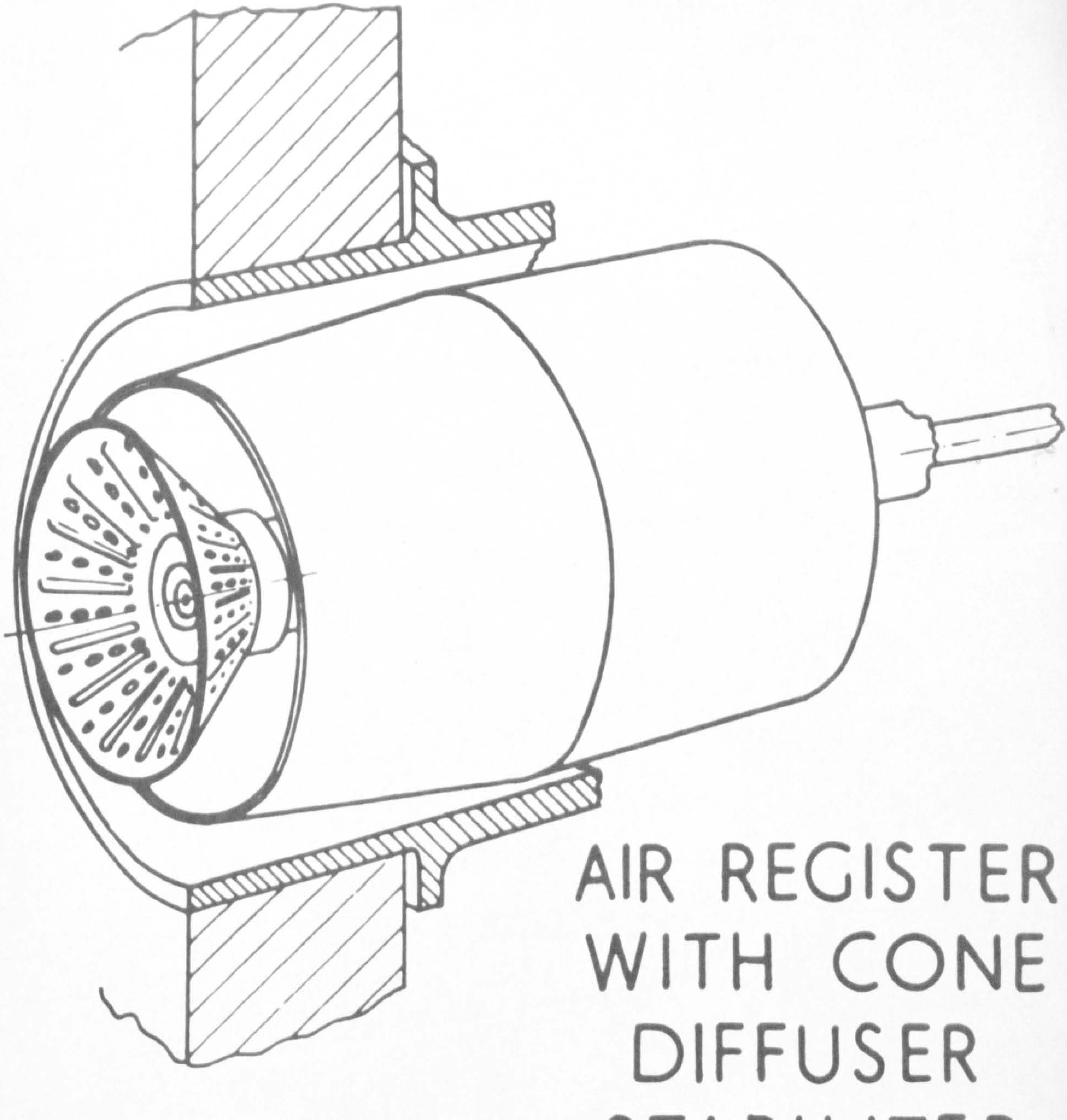



FIG. 12.b. *A diffuser cone stabilized system.*

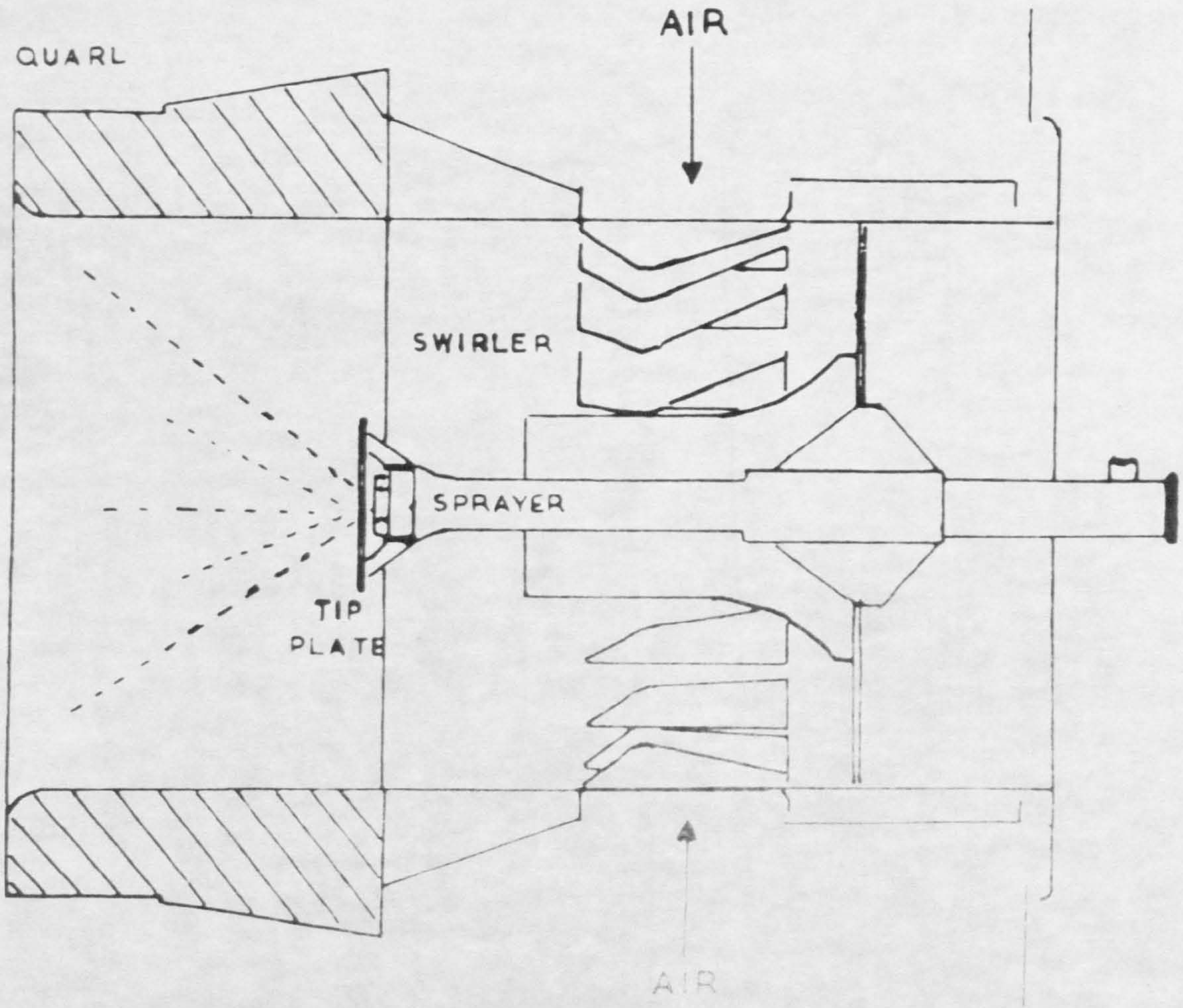



Fig. 12c 1943 type Admiralty air register with swirler and tip plate stabiliser.

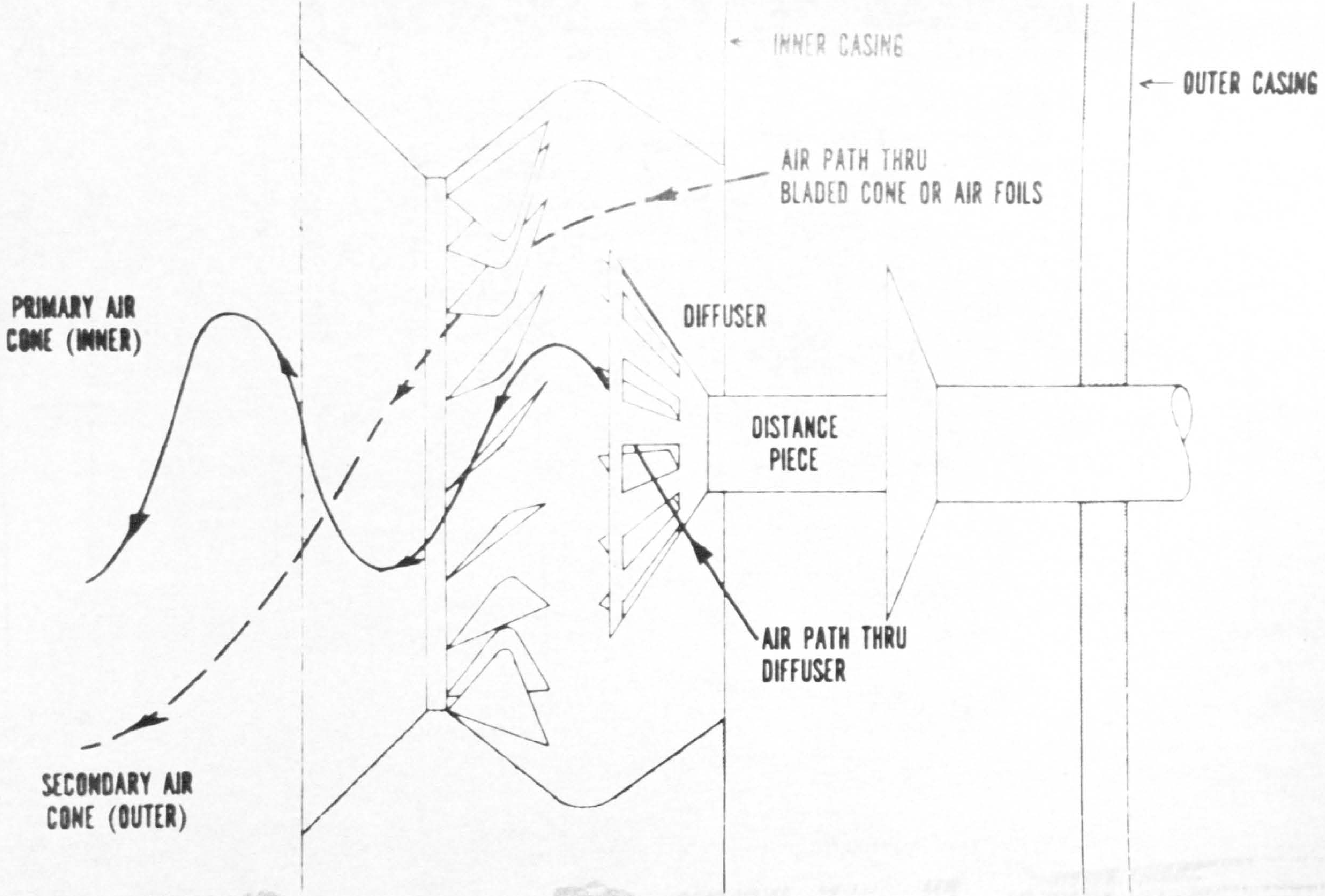



FIG. 12d. BABCOCK AND WILCOX CAROLINA AIR REGISTER WITH CONICAL DIFFUSER SWIRL VANES.

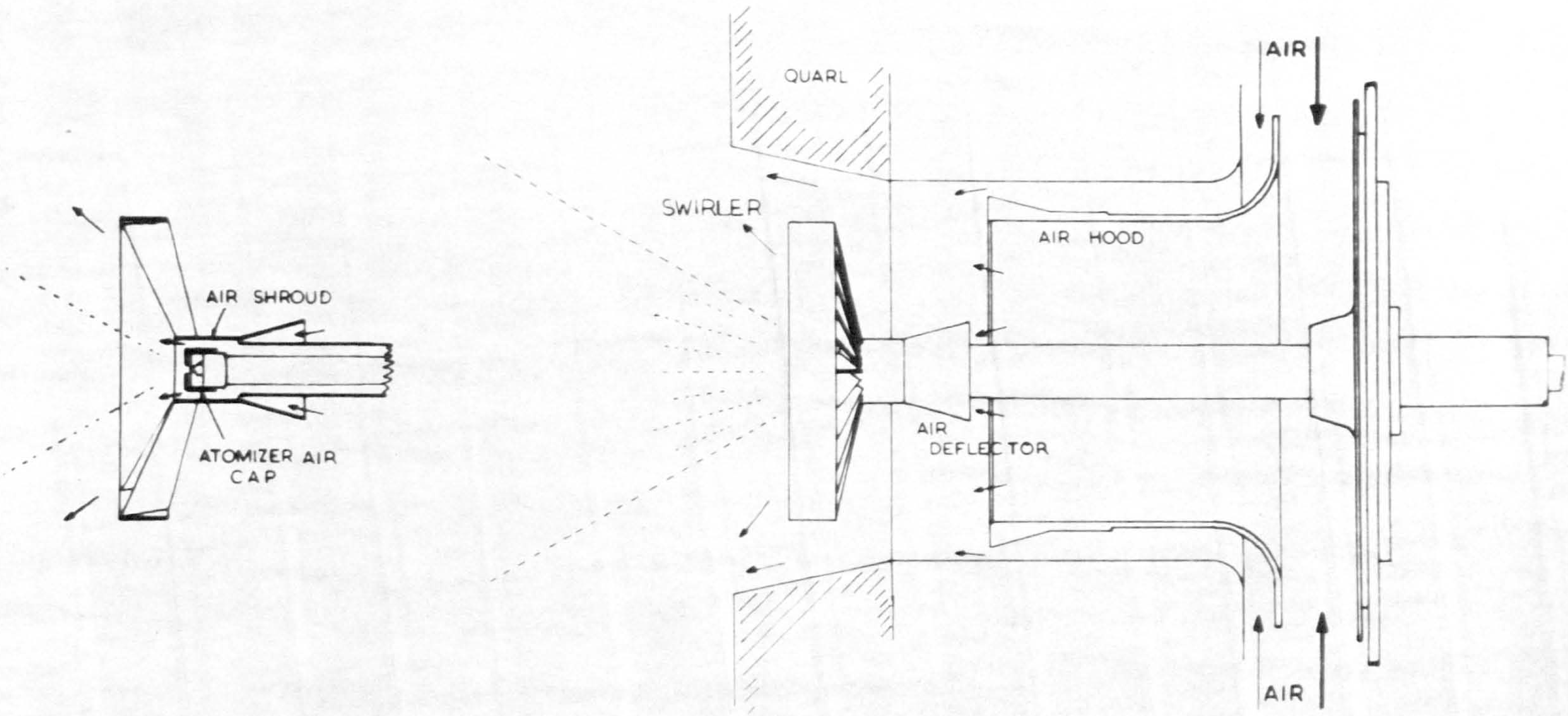



FIG.12.e. MODERN SUSPENDED FLAME AIR REGISTER

Investigations on the performance of pressure-jet flames, apart from the work recently carried out by Beer[65], appear to have been of a very 'ad hoc' nature and to have been concerned with performance factors such as turn down ratio that could be achieved, stack solids and flame shape. These, of course, can be important operating criteria, but no basic data seems to have been collected affecting control and performance of the flame. In the related fields of gas turbine design, a much more extensive study has had to be made and where this work is relevant, it has been mentioned in this chapter and in the review in Appendix I. In some cases work on gas turbine combustion chambers has been concerned principally with practically important quantities such as blow off conditions, but also a good deal of detailed work has been done on gas turbine combustion chambers both in the hot and cold models. Some examples of this are work carried out by Joseph Lucas and Shell Research[34]. Both these groups of workers have been particularly concerned with flow patterns in the combustion chamber and the extent to which cold models could be related to the hot working unit. Pavia & Rosenthal[62] and Romanska & Foster[68] concentrated on mixing and the temperature and gas composition distributions. Poulston & Winter[34] and Binark & Ranz[69] and Clarke[33], also considered the effect of the combustion chamber aerodynamics in the oil spray.

The combustion chambers investigated all had a small swirler and secondary air inlets downstream of the burner and the detailed patterns observed differ substantially from those likely to be produced by an air register operating on a boiler front even with a swirled air supply.

These workers have in the main gone no further than producing patterns of flow and temperature and the same data on combustion intensities and efficiencies. Toone[102] has taken the last type of data further and been able to calculate activation energies. Spalding[70] has pointed out, however, that data on the distribution of velocity is insufficient to carry out a full quantitative prediction of how a given system will behave and has suggested the following alternatives.-

(i) The inclusion of so called "influence factors" representing the influence of temperature and heat release on each other in adjacent elementary sections.

(ii) By integrating the differential equations governing velocity and eddy diffusion if these can be represented in a sufficiently simple manner.

(iii) By complete description of idealized elementary flow fields of which the complete combustion chamber can be regarded as a summation.

(iv) Overall theorems limiting performance.

Unfortunately lack of data on "influence factors" and eddy diffusivity and the aerodynamic patterns produced in many circumstances limits the usefulness of methods i - iii to very simple examples. The type of prediction referred to under heading iv is discussed in the section 2.5 above.

## 2.9 Combustion Oscillations.

### 2.9.1 General Observations on Unstable Combustion.

Due to the tendency to develop special terms to describe particular phenomena in a limited context it is necessary to specify accurately the meaning of the terms used in this field. Unstable combustion, in the sense of combustion oscillations, refers to the development of regular periodic fluctuations in flame characteristics for a flame and combustion chamber that will under other conditions exhibit steady burning characteristics. Thus it is necessary to distinguish it from a flame that is behaving in an erratic fashion due to the inability of the flame front to fall into any fixed position or range of positions perhaps due to the absence of any "stabilizing" characteristics of the burner. Also it is necessary to make a definite distinction to randomness in combustion due purely to turbulence in the flame and the combustion process[71] and the regular cyclic oscillations with a definite characteristic. The former is usually reported as being "white noise", i.e., covering a wide spectrum without definite predominant frequencies. Very recently Smith & Kilham[72] have shown that combustion noise in open turbulent flames can be related to the geometric and flow parameters defining the flame and they have shown that the sound generated may be considered to arise from a statistical distribution of monopole sources throughout the zone of combustion. The peak frequencies were found to be a function of the parameter, burner port size, mixture velocity and flame speed expressed dimensionlessly as the Strouhal number.

Proper cyclic combustion oscillations falling within the definition just given have been known for a long time[73]. The earlier phenomena to be observed were of the so called singing flame type which was analysed in some detail by Lord Rayleigh[74], who was able to specify the basic conditions for combustion oscillations to be possible. This is:- 'a component of the fluctuating heat addition must be in phase with the pressure wave', which in effect means that the rate of heat addition must be a positive function of the pressure drop if the oscillation is to be driven by the combustion process.

The problem has become of considerable interest in the field of liquid and solid propellant rockets in recent years where the phenomena have been described as "chugging" and "screaming" depending on frequency. The distinction between the two being made at about 1000 c.p.s. In this field the problem of high frequency instability is usually the most critical one and probably involves complex modes of oscillation. However, frequencies higher than 100 cps. are very rarely encountered in the general run of industrial and boiler problems and only in very exceptional cases higher than about 700-800 cps.

### 2.9.2 Low Frequency Oscillations - Acoustic Theories.

The somewhat varied literature in the field of low frequency oscillations has been reviewed recently[75] and a discussion of the important differences between some groups of proposed theories have been discussed by Brown & Thring[76]. These are discussed in detail in Appendix II. In outline two sets of theories have been proposed. The first group of these is based on the acoustic properties of the system, these assume that the oscillation is the acoustic resonant mode of the system. This is excited by fluctuations in the combustion process, so that the resonant frequency with the organ pipe or Helmholtz frequency of the system depending on the dimensions and layout of the combustion chamber and flues.

Examples of this type of approach were proposed by Rayleigh[74], Eliass & Gordon[77], Putnam[78], Merle[79] Sanders & Lawrie[80] (see Appendix II, Table II (b)). The problem with many of these theories is unfortunately that they do not relate the combustion process properly to the oscillation produced. The exceptions are Rayleigh and Merle both of whom are at some pains to show how energy from the combustion process is able to drive oscillations. If the oscillation were of the acoustic type the only significant variables should be linear dimensions of the different sections of the system, the temperatures at certain stages and the value of $\gamma$, the ratio Cp/Cv for the gases concerned. These last two variables of course affecting the speed of sound. $\gamma$ does not vary very widely between air and normal combustion products and the speed of sound varies therefore as $\sqrt{T}$. Thus an acoustic oscillation is characterized effectively by a frequency which is only a function of dimensions or $\sqrt{T}$. Thus for a given combustion chamber, frequency should be nearly constant, in the ordinary operating range. Francis et al[81] have shown recently that there is a tendency for there to be a preferred frequency range even when dimensions are varied widely, a different harmonic being adopted when the natural frequency departs too far from this preferred range of about 75-150 cps.

### 2.9.3 Low Frequency Oscillations - Non-Acoustic Theories.

The theories that consider oscillations as independent of the acoustic properties of the system include those proposed by Crocco & Cheng[82], Tischler & Bellman[83], Thring[84] and Fritsch[85] (See Appendix II, Table II (a)).

In principle these non-acoustic theories suggest that the oscillation is governed by the ability of the pressure fluctuations in the combustion chamber to modulate the heat release by controlling the flow admittance and egress to and from this combustion chamber. This makes factors such as ignition time delay, inlet and exit resistances and residence times the important parameters. The non-acoustic theories vary as to how far they are able to bring the combustion process properly into account. Unfortunately to make any of this then manageable it has been necessary to assume a fairly idealized system. The most relevant of these to boilers are those proposed by Thring and Fritsch. These are in their main idea fairly similar. They both assume that the mode of oscillation is dependent on mixing conditions. This is particularly intended to apply to the type of case, such as a pressure jet fired system where pressure fluctuations in the combustion chamber of probably a few inches w.g. are sufficient to have a marked metering effect on the air supply, but not on the fuel supply which is supplied at a much higher pressure.

This will obviously alter the equivalence ratio and mixing conditions during each cycle. This is an important difference as compared with Crocco & Cheng and Tischler & Bellman's approach both of which assume that the fluctuations in absolute pressure are sufficient to affect substantially the chemical reaction rate without necessarily affecting the mixture ratio.

### 2.9.4 Low Frequency Oscillations - Experimental Data.

While all these theoretical ideas are very interesting the validity of the arguments is shown by experimental data. In this field the amount of experimental data actually available to compare with the corresponding theoretical ideas are surprisingly small and in the published literature usually insufficient data are available on the results reported to make wide spread comparisons possible.

In the case of gas flames both premixed and unpremixed there is some body of evidence to support an acoustic mechanism(78)(81)(86)(87). It is possible to correlate reported observations with the acoustic properties of the burner combustion chamber and stack and sometimes the room in which the combustion chamber is operating. This, however, does not apply to all cases as for instance the data of Howland and Simmonds(86), which showed apparent departures from this type of mechanism.

From the frequency ranges obtained in a number of liquid fuel fired units that have been investigated the oscillations appear to be no longer dependent on acoustic properties of the system. In fact in some cases such as a water tube boiler it is very difficult to see how an acoustic effect would be possible since the form of construction involves a large number of tube covered surfaces or large bundles of tubes and is bound to be very heavily damped acoustically. This type of oscillation has already been mentioned from the theoretical point of view and as already indicated there is not very much experimental evidence to go on. As far as boilers are concerned some results obtained by Fritsch have been reported and these were shown to be fairly close to the value he was able to calculate, although there was insufficient data to show any trends that could be used to compare theoretical predictions with practical results. Papadakis[88] has been able to demonstrate oscillations of this type in a small oil fired experimental unit. These showed that frequency changed very markedly with features such as air-fuel ratio resistances in the system etc., to a much greater extent than could be accounted for by the change in temperatures of the gases. His results are parallel to those reported in this thesis, namely that oscillations of this type occur only under conditions of air deficiency, that mixing had an effect on the oscillation limits and that there were sometimes more than one band of frequencies possible.

Recent results of Schalla[89] also have produced results indicating oscillations of this type which seemed to be due to changes in the heat release rate either due to composition or duct section changes. In general experimental data seems to be more complex than the results predicted from the various proposed theories. This is particularly true in the case of the non-acoustic types of oscillation where phenomena like the different frequency bands are not explained by any of the theories. The ingenious suggestion of Stewart & Nesbitt[90] that phenomena of this type can be accpunted for by pressure waves modulating the flow through different restrictions in the system, so that the period of oscillation becomes the time for pressure wave to travel upstream to say the burner or other restriction and the time for the flow modulation to reach the flame, does not seem in fact likely to give answers of the right order of magnitude. It seems likely that a complete answer to the problem can only be found in a much better estimate of the effect of pressure fluctuations on the ignition delay of the droplets, which is very sensitive to changing conditions, and the instantaneous mixing in the flame than has been achieved so far.

# CHAPTER 3.

## The experimental approach adopted.

### 3.1 Scope of the investigation.

#### 3.1.1 Simplification of Problem.

In order to study combustion equipment at all satisfactorily it is necessary to decide what aspects of the system can be considered reasonably without obscuring the main issues involved. The first problem arises from the fact that unlike some other forms of combustion chambers most boilers are asymmetric in shape. In most marine boilers the burners are placed at one end and are usually in some arrangement which makes it difficult to select any elementary section of circular or hexagonal shape to represent the field of action of each flame. Unfortunately the field of each burner usually includes a wall and two or three other adjacent burners which means that flow patterns probably will be distorted to a greater or lesser extent.

The only solution to this problem is to consider the system in the following sections:-

(1) Combustion chamber flow as a whole.

(2) Single burner operating under conditions comparable to that of a single burner in an actual case.

(3) Interaction between several adjacent burners.

(4) Combustion instability.

### 3.1.2 The objectives for this study.

The topics selected for study developed from problems occurring in pressure-jet burners with combustion pulsations. It was apparent, however, that the real problem went further than the occurrence of combustion instability. This was the lack of basic knowledge on the performance of burners of this type, except for what had been gained by individual groups developing actual burners to meet particular operating requirements. For this reason the main emphasis has been placed on the investigation of non-oscillating flames in order to provide basic data from which to tackle combustion oscillation problems. Some of the justifications for this are discussed with the experimental results later. Basic data of this type are also required in order to provide a foundation from which more novel ideas could be built. It was, therefore, considered desirable for work in this field to start from investigations on an actual type of burner. This study comprises the largest section of this work.

The construction of a laboratory rig for investigating pressure-jet flames on a small scale made it necessary to undertake a study of the scaling process and to show how far flames of this type could be reasonably characterised and scaled down. While this provided a study on its own, it also served to check and expand to some extent data obtained on the full size combustion equipment.

In addition, attention was necessary to consider lines that looked promising in altering the flame in the direction of positive advantage. Particular directions of interest were the achievement of higher combustion intensities, the avoidance of combustion instability, the control of flame shape and performance for a wide range of operating conditions, widening the smoke limits and satisfactory combustion with minimum excess air. It was not possible to carry out specific investigations on all these topics, but the data obtained were reviewed carefully with these aims in view.

The background to this problem of combustion oscillation made it necessary to obtain some data on combustion instability and for this purpose oscillation measurements were made on actual marine boilers. This in fact covers item (2) and part of item (4) of the division of combustion chamber problems given in Section 3.1.1. Other sections of the problem have been tackled by other members of the author's team(59)(60)(87)(88).

## 3.2 The apparatus used and the measurements made.

### 3.2.1 Type of Apparatus.

The method of study required a hot test facility. The full size burner under investigation has a fuel consumption of up to about 1 ton per hour, that is about $5 \times 10^6$ B.t.u./hr., a heat output much too large to cope with easily on a laboratory scale. It was therefore considered necessary to design a suitable apparatus in which experiments could be carried out with a scaled down burner.

The two alternatives for combustion chamber size were to consider a section of the combustion chamber which would be occupied by one flame with the disadvantages mentioned above, or to consider the case of one flame in a combustion chamber the same size as a typical boiler. This was also convenient since a suitable test facility for comparative work was available at the Admiralty Fuel Experimental Station at Gosport (See Section 4.1.). For measurements of combustion instability an actual ships boiler was used. (See Section 4.3.)

### 3.2.2 Variables Studied.

The following variables were selected for study:-

#### 3.2.2.1 For a combustion chamber with unstable combustion.

(i) The effect of fuel rate on observed oscillation frequencies of a near limiting condition.

(ii) The variation of limiting condition with pressure drop across air register.

#### 3.2.2.2 For full size and model burner.

(i) Throughput rate.

(ii) Spray angle.

(iii) Important scaling parameters.

3.2.3 Measurements made.

Measurements made were as follows, excluding normal fuel and air flow, pressure drops and stack analyses.

3.2.3.1 For combustion instability.

1. Frequency measurements at different points in the system.

2. Temperatures in the combustion chamber.

3.2.3.2 On the full size and model burners.

1. Gas composition.

2. Gas temperature.

3. Some components of gas velocity.

## 3.3 Scaling.

### 3.3.1 The development of ideas in Scaling.

The earliest attempts to scale combustion were the work of Thring[91] in relation to open hearth furnaces, and Lloyd[92] with gas turbine combustion chambers. Thring showed that attainment of momentum similarity and reduction of burner dimensions to an "equivalent nozzle diameter" gave a reasonable means of comparison even between a hot system and a cold model. Lloyd's work showed also that provided excessive chamber chilling did not occur, reasonable correlation of combustion chamber efficiency could be obtained by operating with similar chamber pressure and velocity. The present status of work in this field is reviewed by Spalding[93].

### 3.3.2 Scaling parameters.

The factors most likely to be of importance in deriving a reasonable scaling for oil flames are jet momentum, turbulent mixing, reaction rate and spray characteristics.

Most of the scaling parameters suggested, apart from the obvious one of Reynolds number, derive from work in connection with the gas turbine combustion chamber. Scaling of pressure jet flames for ordinary combustion chambers (i.e. with no secondary air) does not seem to have arisen previously. A detailed theory of scaling for gas turbine chambers was originally proposed by D.G. Stewart[94]. This worker reviewed, and reduced to convenient dimensionless forms, a large number of relevant parameters affecting scaling. This is sometimes referred to as "pL" scaling as the normal method of operation is to maintain the group pL constant so that as size decreases the pressure in the combustion chamber increases. This holds groups such as Reynolds number, Mach number and the chamber loading constant.

The main problem is that complete modelling of a combustion process is impossible at one and the same time as reference to Fig. 13. will illustrate. A second consideration is that a number of very important parameters involving for instance jet configurations and similar spacial effects do not reduce easily to dimensionless analysis and are apt to be overlooked by mechanical application of scaling rules. The selection of the right rules to apply or ignore, depends largely on the result required. For example the result required may be just an end effect such as a blow off limit or an outlet temperature, where intermediate stages are of little importance as long as the final result is reasonable. On the other hand some detail of the process may be the important one, in which case quite a different set of considerations might be relevant.

FIG. 13.

Plot of Scale Factor against Various Dimensionless Groups.

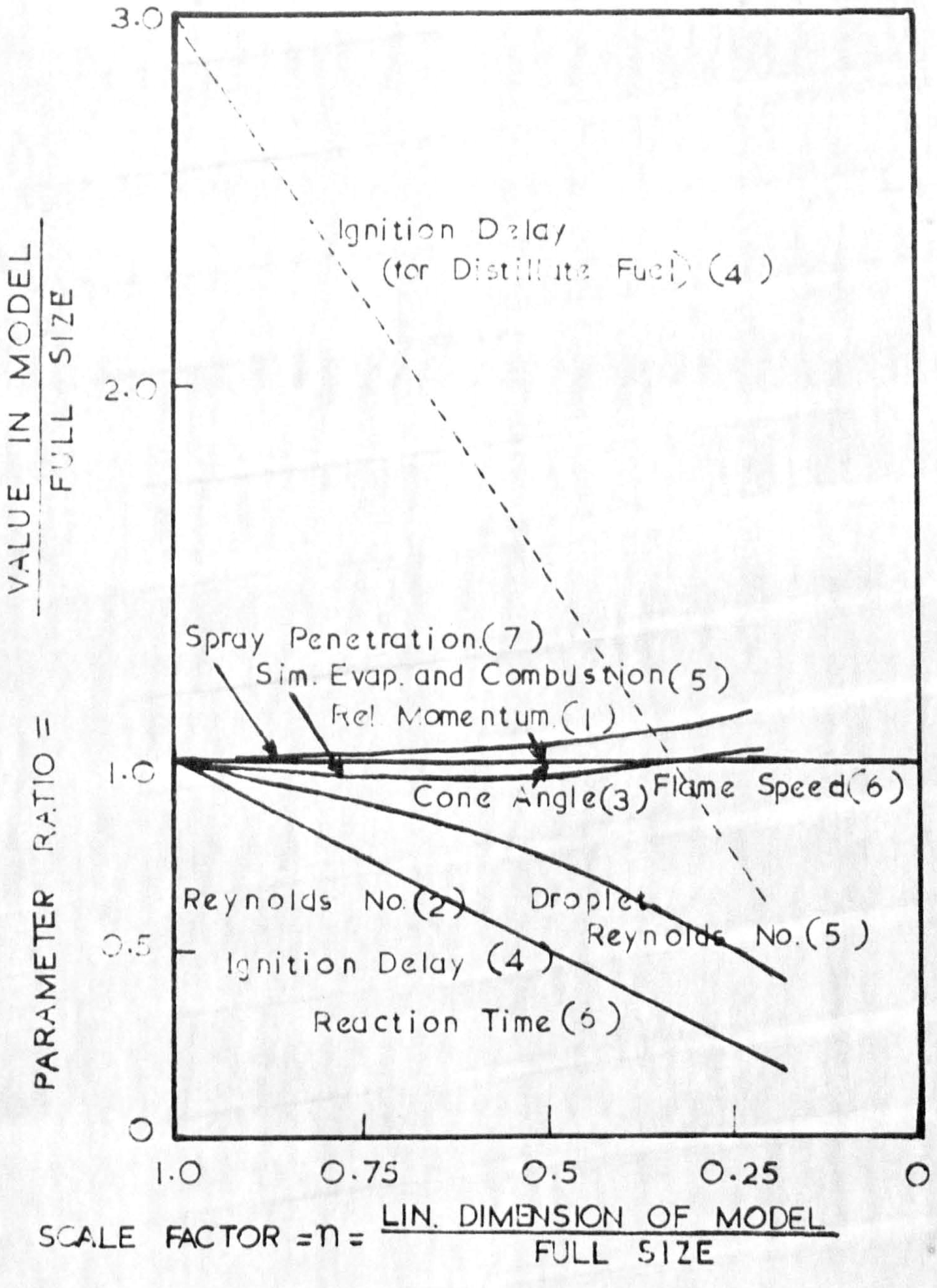
(a). Air velocity, relative momentum and air-fuel ratio constant. Reynold's number varied.

(b). Residence time, relative momentum, air-fuel ratio constant. Reynold's number varied.

(c). Reynold's number, air-fuel ratio, atomizing pressure constant. Velocity varied.

(d). Air velocity, air-fuel ratio constant. Reynold's number varied. Low atomizing pressure.

(e). Air-fuel ratio x $\sqrt{\text{atomizing pressure}}$ and velocity constant. Diameter of burner and Reynold's number varied.



3.0
2.0
1.0
0.5
0
Ignition Delay
(for Distillate Fuel) (4)
Spray Penetration (7)
Sim. Evap. and Combustion (5)
Rel. Momentum (1)
Cone Angle (3)
Flame Speed (6)
Reynolds No. (2)
Droplet
Reynolds No. (5)
Ignition Delay (4)
Reaction Time (6)
PARAMETER RATIO = VALUE IN MODEL / FULL SIZE
1.0
0.75
0.5
0.25
0
SCALE FACTOR = n = LIN. DIMENSION OF MODEL / FULL SIZE


FIG. 13. a.

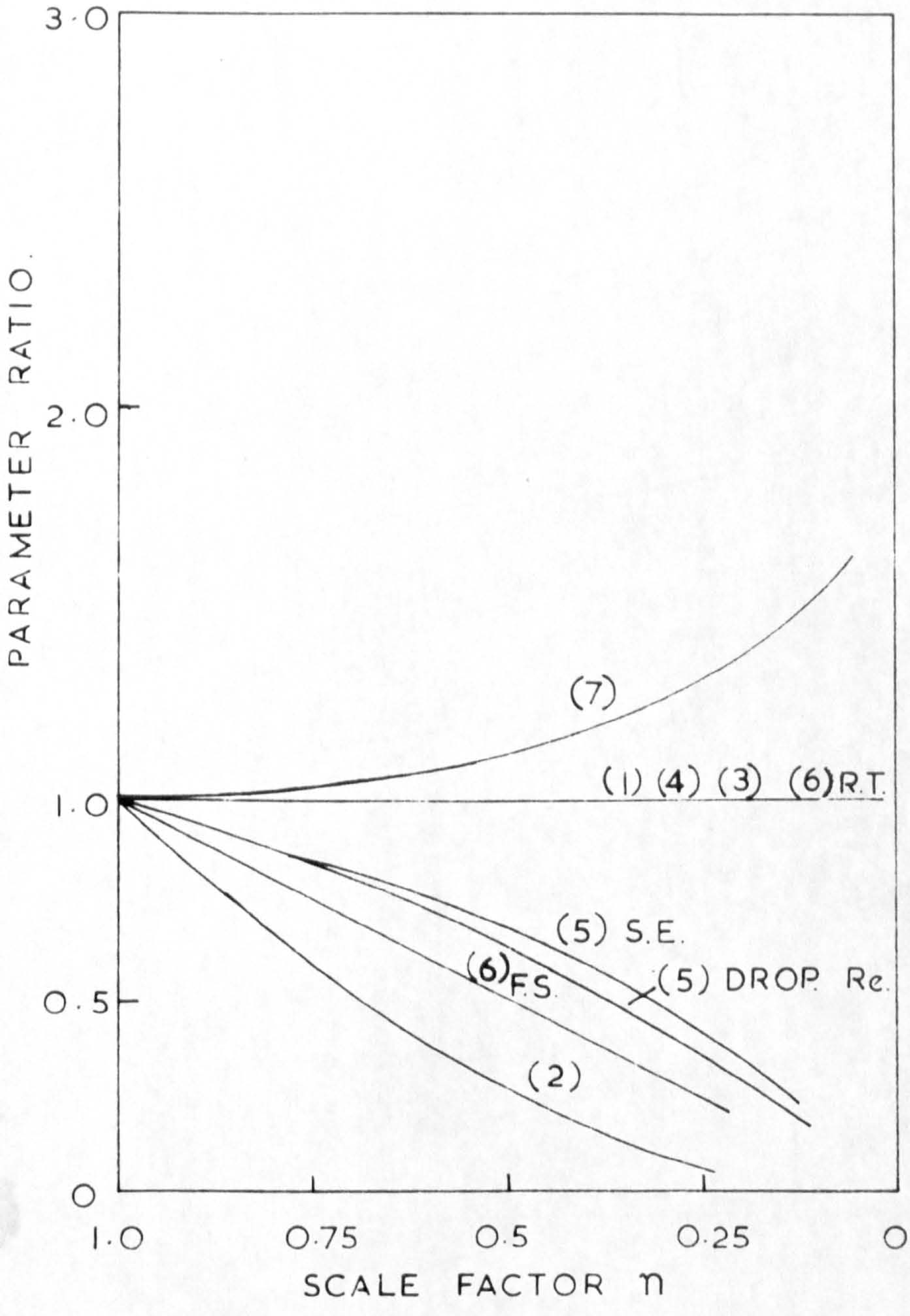

3·0
2·0
1.0
0.5
0
PARAMETER RATIO.
(7)
(1) (4) (3) (6) R.T.
(5) S.E.
(6) F.S.
(5) DROP. Re.
(2)
1.0
0.75
0.5
0.25
0
SCALE FACTOR n

FIG. 13.b.

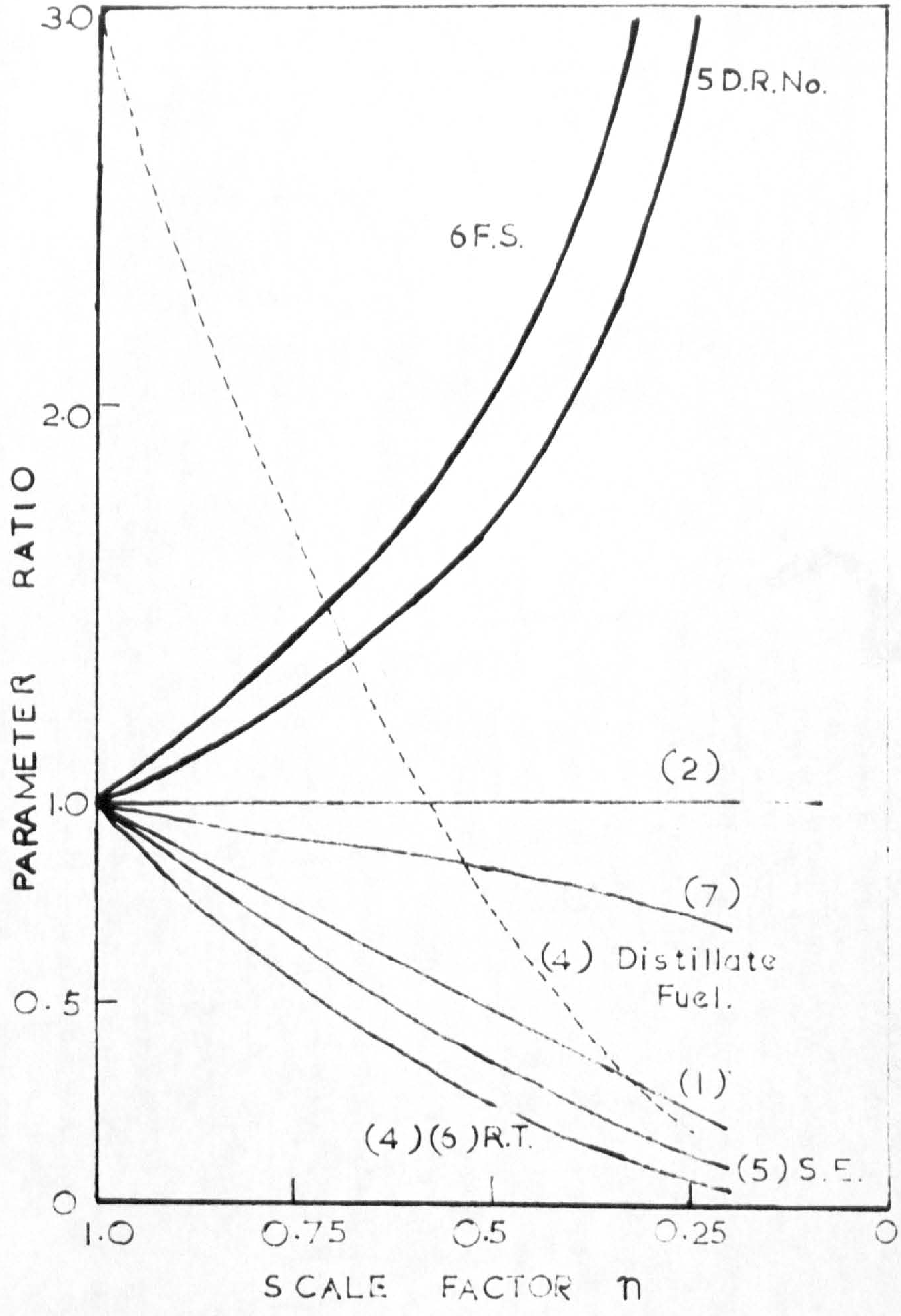
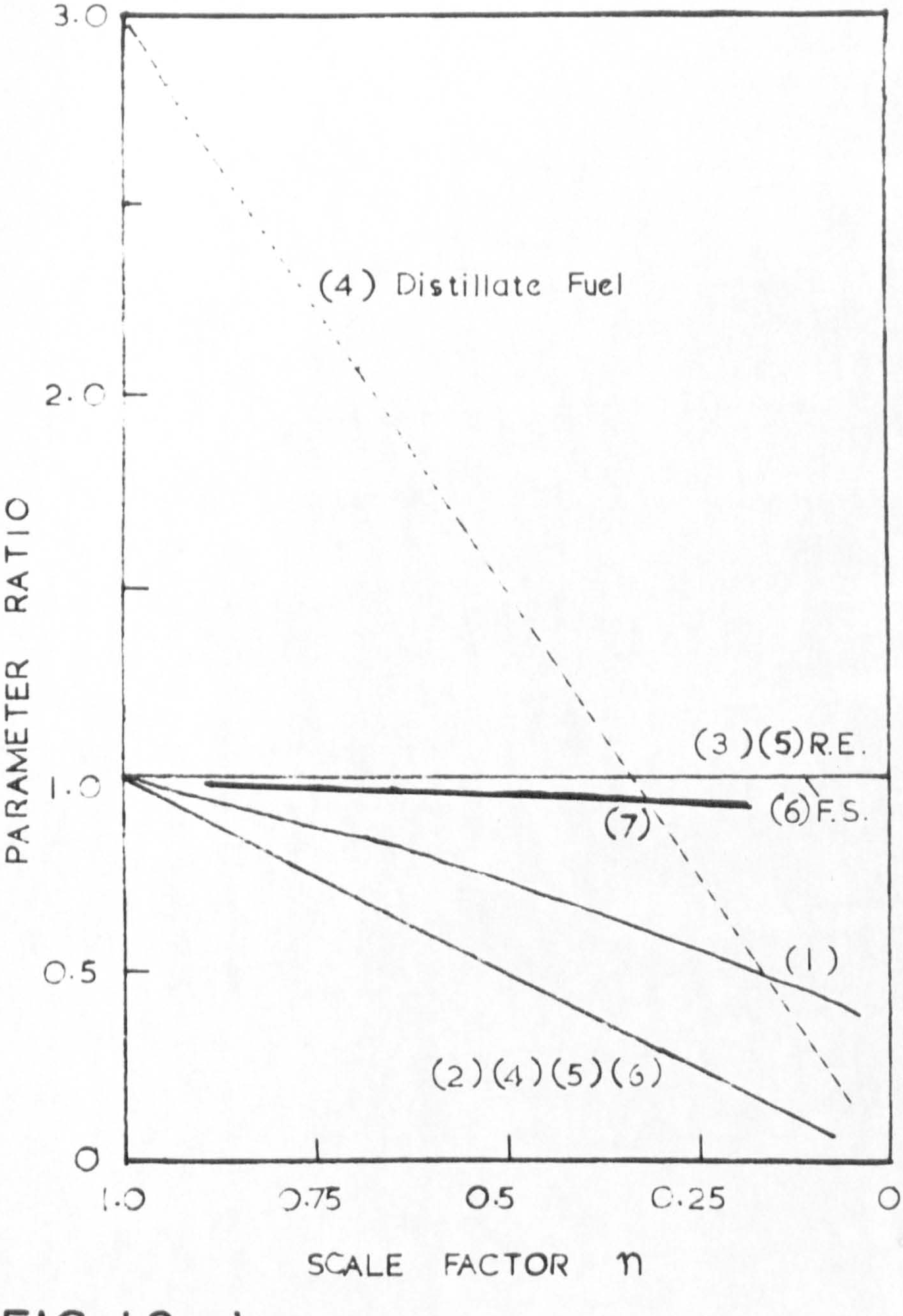

3.0
2.0
1.0
0.5
0
PARAMETER RATIO
5 D.R. No.
6 F.S.
(2)
(7)
(4) Distillate Fuel.
(1)
(4)(6) R.T.
(5) S.E.
1.0
0.75
0.5
0.25
0
SCALE FACTOR n

FIG. 13.c.



3.0
2.0
1.0
0.5
0
PARAMETER RATIO
(4) Distillate Fuel
(3)(5) R.E.
(6) F.S.
(7)
(1)
(2)(4)(5)(6)
1.0
0.75
0.5
0.25
0
SCALE FACTOR n


FIG. 13. d.

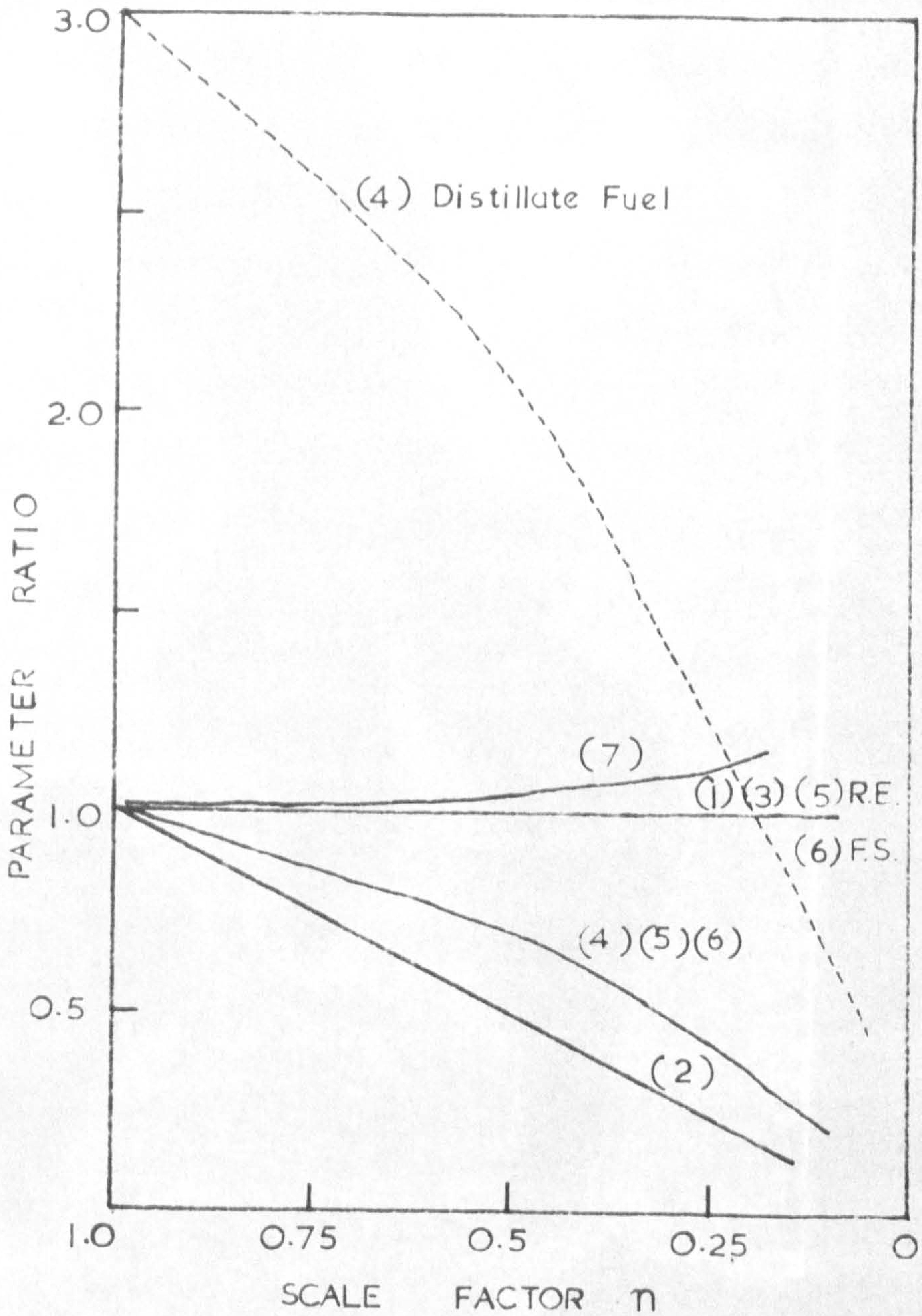

3.0
2.0
1.0
0.5
PARAMETER RATIO
(4) Distillate Fuel
(7)
(1) (3) (5) R.E.
(6) F.S.
(4) (5) (6)
(2)
1.0
0.75
0.5
0.25
0
SCALE FACTOR n

FIG. 13e.

The original objectives for which Stewart's criteria were proposed included development work on gas turbines. The parameters usually compared were the blow off limits, variation of air fuel ratio, combustion efficiency for different air-fuel ratios, simulated altitude conditions and temperature traverses in the outlet duct. In the present work the emphasis necessarily is not only on the final products, but also the processes taking place in the flame. These require local measurement of air-fuel ratio, combustion efficiency and velocity to establish the special characteristics of the flame. In this case blow off limits and variations in pressure are of no immediate interest. There is also the difference that a gas turbine combustion chamber is much more nearly an approximation to a perfectly stirred reactor than a flame in the relatively large combustion space of a boiler. It may be anticipated that flow configuration factors affecting the interaction of the air and oil, such as spray penetration and reaction time may have an enhanced importance in this particular case.

The effect of flow configuration and the fuel distribution seem to be those least clearly represented by Stewart's results. The development of the correct three dimensional flow patterns and material distributions are most important in this case.

Representation of the correct spacial similarity, in fact implies geometric similarity of the appropriate burner parts and this must be regarded as the most important criterion. This is assuming that the density change between input air and combustion gases remains the same for the prototype and the model. The effect of this with a cold model is discussed later.

An important difficulty that arises in the present case is that scaling on the normal pL basis is not itself possible without a very elaborate test facility capable of running at several atmospheres pressure. This means that the groups mentioned above in this connection can be no longer maintained constant, but as is discussed later this is not of over-riding importance. Taking Stewart's form of analysis, Table II. represents the various groups reduced to their simplest form, assuming that combustion chamber pressure p, air fuel ratio f, and the fuel properties remain constant. The order given is Stewart's assessment of relative importance. Bragg[12], Herbert and Bamford[95], Lefebvre and Halls[96] amongst others have developed and clarified some of these ideas.

## TABLE II

| | | Simplified expression assuming p,f, and fuel properties constant. |
|---|---|---|
| 1. | Spray momentum relative to air momentum | $P^{\frac{1}{2}}/v$ |
| 2. | Reynolds number | vL |
| 3. | Spray cone angle | $\phi$ |
| 4. | Ignition delay | $L/_{v\tau}$ |
| 5. | a) Simultaneous evaporation and combustion | $L/_{vd^2}$ |
| | b) Droplet Reynolds number | vd |
| 6. | Reaction kinetics and flame propagation:- | |
| | a) Reaction time/Residence time | $L/_{v}$ |
| | b) Turbulent flame speed/Air velocity | v |
| 7. | Spray penetration | $\dfrac{\mu^{0.25}\, s^{0.05}\, L^{0.051}}{\rho_f^{0.051}\, v^{0.258}\, d^{0.259}}$ |
| 8. | Dilution and mixing | Complex (Geometric and temperature similarity) |
| 9. | Surface heat transfer | $\dfrac{\text{Heat transfer rate to the surface}}{\text{Overall heat release}}$ |

P = atomizing pressure

v = velocity of air stream

L = characteristic length

$\tau$ = ignition delay

d = droplet diameter

$\mu$ = viscosity

s = surface tension

$\rho_f$ = density (fuel)

f = air-fuel ratio

### 3.3.3 Discussion of the Importance and Validity of the different scaling factors.

#### 3.3.3.1 Geometric Similarity.

It is now necessary to discuss the validity and relative importance of these different criteria. The question of geometric similarity has been touched on already in order to preserve the correct distribution of fuel and air. This is assuming that the density changes between the incoming air and the combustion chamber gases are the same in the prototype and the model. If there are wide differences this might need some modification and this is discussed later in connection with the use of a cold model analogue of the system.

#### 3.3.3.2 Spray Cone Angle.

Although the spray momentum will be very much lower than that of the air, the effect of the cone angle on the spray distribution may well be considerable and it is necessary to maintain strict angle similarity between the prototype and the model. This in effect is an extension of the maintenance of geometric similarity.

#### 3.3.3.3 The Choice of Suitable Representative quantities.

One of the big advantages of dimensional analysis is that it is not always necessary to know absolute values for many of the quantities, as long as their trend can be related to some conveniently measured components of the system. An example of this is the simplification of temperature terms since the theoretical outlet temperature and velocity are simply related to the easily measured input quantities thus:-

$$\frac{T_{Th}}{V_{Th}} \propto \frac{T_a}{V_a} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (14)$$

The only other important criterion is that the components chosen should be significant variables. The selection of a representative linear dimension is the most important example of this. In the gas turbine work already cited this is quite clearly the chamber width or length. In the case of pressure-jet burners in water tube boilers, the considerably increased value of the ratio of chamber diameter to burner diameter makes the chamber dimensions rather less significant, especially as most boilers are sufficiently asymmetric to make these rather difficult to define accurately from the flow point of view. As the region of greatest interest lies close to the burner and the combustion chamber is too short for anything approaching plug flow to develop the burner diameter has been taken as the most significant dimension. The dimensions of the combustion chamber have been kept in proportion as fqr as practical considerations would allow.

The choice of a suitable velocity for the scaled combustion chamber is dictated largely by its effect on the main groups and it is impossible to change velocity to satisfy all of them simultaneously. The usual practice is to maintain v constant between prototype and model, but it could be varied to achieve specific effects and this is demonstrated in some of the diagrams in Fig.13.

#### 3.3.3.4 Momentum Similarity.

Stewart pointed out that where it was necessary to consider the interaction of two streams as well as the overall aerodynamics, it was necessary for the momentum ratio between the two to be constant between prototype and model, i.e.,

$$\frac{V}{f\, v_a} = \text{const.}$$

assuming constant atomizer efficiency

$$V \propto \left( {}^{P}/_{\rho_f} \right)^{\frac{1}{2}}$$

and hence

$$\frac{1}{f\, v_a} \left( {}^{P}/_{\rho_f} \right)^{\frac{1}{2}} \text{ const.} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \qquad (15)$$

The relative momentum of the oil is very small compared with that of the air. This means that this parameter is unlikely to affect to any great extent the general velocity and mass flow patterns in the system, but almost certainly will have a very considerable effect on the gas composition and efficiency patterns.

In some of the work cited in this section[94][95] matching the atomizer seems to have been the major problem and it was considered unlikely that the need to do this would be diminished in the type of system under consideration.

3.3.3.5 Reynolds Number.

Up to certain limits, processes such as turbulent mixing and hence the physical transport of reactants is proportional to the Reynolds number of the air stream.

Thus for complete comparison

$$\frac{p\,v\,L}{\nu} = \text{constant.} \qquad \ldots\ldots\ldots\ldots \qquad (16)$$

where $\nu$ = kinematic viscosity

in the present case p is atmospheric and of course constant and under combustion conditions $\nu$ will remain constant as well, provided the temperatures are the same. This means that to maintain strict Reynolds similarity the velocity would need to increase as the scale decreased. This is contrary to the requirements for several other groups and is liable to introduce sufficiently high velocities into the system to interfer with the combustion processes. This would also mean that the flow rate at the system would only vary as $L$ instead of $L^2$ which would eliminate the main reason for modelling by producing a heat release rate too high to cope with in the laboratory.

Fortunately, the effect of Reynolds number is considerably less as the flow becomes more turbulent, and providing the flow is well into the turbulent region it has relatively little effect on the mixing processes.

The droplet Reynolds number is a function of relative velocity and droplet diameter and can be varied independently of the jet Reynolds number.

3.3.3.6 Spray Penetration.

Equation 8. derived by Bayley[29] has been mentioned in section 2.6.4 already. Stewart makes the penetration term dimensionless by taking it as proportional to the linear dimension term. By a series of substitutions to eliminate terms that are difficult to measure he derives the following rather unwieldy expression for the scaling group.

$$\left\{ \frac{f^{0.20} \; s^{0.05}}{\rho_f^{0.051}} \right\} \left\{ \frac{L^{0.051}}{f^{0.25} \; v_a^{0.258} \; d^{0.259}} \right\} \left\{ \frac{(T_{Th} + 120)^{0.042} \; T_a^{0.258}}{T_{Th}^{0.164}} \right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots (17)$$

The bracketed terms represent fuel. properties, operating conditions and the operating temperature respectively.

For constant temperature and fuel properties this would reduce to:-

$$\frac{L^{0.051}}{v_a^{0.258} \; d^{0.259}} = \text{const.} \quad \ldots\ldots\ldots\ldots\ldots\ldots (18)$$

As will be seen from the plots in Fig.13. the effect of this is usually a good deal smaller than many of the other groups.

#### 3.3.3.7 Temperature.

The importance of maintaining the gas temperature constant has been mentioned already and the effect of this is considered on some of the other groups in later sections. The question of heat transfer from the walls to the flame is a rather complicated one as it involves calculation of convective and radiant components from hot walls and thermal load surfaces. The term derived by Stewart is in this case rather an oversimplification. According to Lefebvre[96] heat transfer processes are subject to scaling effects of their own, and unless the combustion intensity is very high or a large proportion of the heat transfer surface is directly exposed to the flame it is probably sufficient to concentrate on temperature similarity in the jet.

#### 3.3.3.8 Ignition Delay.

The determination of ignition delays has been discussed in Section 2.6.9 and is separated here because it controls what happens up to the point of ignition and as already pointed out like flame propagation, and reaction rate, is dependent on the chemical kinetics of the combustion reaction. It is made dimensionless through division by the residence time in the combustion chamber.

Thus $\tau_{(Th,\ p)} \frac{v_{Th}}{L} = \text{const.}$ .......... (19)

i.e. $\frac{v_a\ T_{Th}\ \tau_{(Th,p)}}{L\ T_a}$ which reduces to L/v for similar fuel properties and temperature.

In the curves calculated in Fig.13. use is made of the data of Masdin[37] for similar fuels to those under consideration and this shows that the ignition delay group can be adjusted by replacing the residual fuel of the prototype by a distillate fuel.

Use of $\tau_{Th}$ avoids the necessity to use an exponential term in the expression as it is extremely difficult to predict the temperature conditions encountered under actual burning conditions.

3.3.3.9 Simultaneous evaporation and combustion.

The rate of burning away of a droplet derived from Godsaves[33] work gives the relation:-

$$\frac{dm}{dt} \propto \frac{kd}{C_p} \log \left\{ 1 + C_p \left( \frac{T_{max} - T_a}{E} \right) \right\} \quad \text{..........} \quad (20)$$

where $m$ = mass of fuel

$k$ = constant

$C_p$ = specific heat at constant pressure.

$T_{max}$ = the maximum flame temperature

$E$ = total heat of evaporation.

and for similarity the evaporation rate must be proportional to the original droplet diameter and its rate of passage through the chamber $= \rho_f \, d^3 \quad \frac{v_{Th}}{L}$

and thus

$$\frac{k \, LT_a \log \left\{1 + \frac{C_p}{E}(T_{max} - T_a)\right\}}{C_p \, v_a \, \rho_f \, d^2 \, T_{Th}} = \text{const.} \quad \ldots\ldots\ldots \quad (21)$$

for fuel properties and temperature constant this reduces to $\frac{L}{vd^2}$.

This assumes that the droplet Reynolds number remains constant. If this is not so an additional factor must be taken into account. From the results of Agoston et al[97] it has been shown that in a moving stream relative to the droplet

$$\frac{dm}{dt} = \frac{dm_0}{dt}(1 + \gamma \, Re^{\frac{1}{2}}) \quad \text{where } \gamma \doteq 0.24 \quad \ldots\ldots\ldots \quad (22)$$

Thus $\frac{dm}{dt}$ in equation 22 is now $\frac{dm_0}{dt}$ and the scaling parameter becomes

$$\frac{k \, L \, Th \log \left\{1 + \frac{C_p}{E}(T_{max} - T_a)\right\}}{C_p \, v_a \, \rho_f \, d^2 \, T_{Th}} (1 + \gamma \, Re_d^{\frac{1}{2}}) \quad \ldots\ldots\ldots \quad (23)$$

However in the case of fine atomized spray $Re_d$ is likely to be a good deal less than 1 (Bamford quotes $Re_d > 0.5$). This means that errors involved in ignoring this are likely to be only small and it is therefore sufficient to ignore minor changes in droplet Reynolds number.

### 3.3.3.10 Reaction Kinetics.

The main problem in reproducing the combustion conditions is the scaling process in the correct control of the way the reaction proceeds. The actual reaction occurring in the flame will be very complex and it is necessary in the first instance to postulate an equivalent simple homogeneous 2nd order reaction in place of these.

This means that it is possible by derivation from equation (1) to get the proportional rate at which the fuel burns away.

$$- \frac{1}{Q_f} \frac{dQ_f}{dt} = k\,T^{-\frac{1}{2}} \exp(-E/RT) \qquad \ldots\ldots\ldots\ldots \qquad (24)$$

where k incorporates the collision factor and concentration factors.

If the reaction is to have proceeded to the same degree at a given scale distance it follows that the scaling factor will require the reaction rate to increase as the residence time in the flame decreases.

Thus the group becomes

$$k \frac{L}{v_a} \frac{T_a}{T_{Th}^{3/2}} \exp(- E/RT_{th}) \qquad \ldots\ldots\ldots\ldots\ldots\ldots \qquad (25)$$

Now if the proportional reaction rate remains constant it follows that the factor will be inversely proportional to the residence time. Because of the complexity of the process it is necessary to assume the actual reactions taking place will follow the same kinetic form as this single "global" reaction rate.

Lefebvre and Halls[96], working on the basis that under suitable conditions the process can be defined in terms of a function of the turbulent flame speed and velocity, have produced a rather complex equation for the combustion efficiency, which reduces in this instance to a similar result. This result is in fact the consequence of varying the mass flow as $L^2$ into a volume that varies as $L^3$.

### 3.3.4. Maintenance of physical and combustion similarity.

As already indicated the most important physical similarity criterion is geometric similarity providing the same temperature is maintained. The other groups which are also likely to be important from the point of view of physical mixing are items 1,2,3 and 6 in Stewart's list in Table II . These are the momentum ratio between the air and the oil spray, turbulent mixing processes which can be represented by the Reynolds Number, the geometric similarity and spray shape and its penetration. Correct modelling of these criteria should result in the establishment of the correct velocity profiles and the correct distribution of fuel and air as represented by local air-fuel ratios.

The second important set of groups affect the combustion processes. Apart from the maintenance of the correct temperature profile the actual chemical reaction is immaterial as far as the physical processes are concerned, but for complete representation it is necessary to add the chemical aspects. These are in fact usually the considerations of most importance in the modelling of combustion processes and the stage that it is very difficult to replace with cold models or analogues of some kind. To obtain chemical similarity it is necessary for the model to reproduce the important stages affecting the chemical process. These are represented by the groups 4,5, and 6 in Table II. and are the ignition delay, simultaneous evaporation and combustion reaction kinetics and flame propagation.

As has been made clear already it is not possible to maintain correct similarity for all these groups simultaneously, especially as the present requirement eliminates the maintenance of a constant pL factor. This means that scaling must be based on a careful selection of conditions to give correct values for specific groups. The practical variations in conditions are fairly limited, some of the most directly controllable of these are listed below. The effect of each set of conditions is illustrated by the plots of parameter variation against scaling factor in Fig. 13.

The most important information that is not obtainable from this analysis is the "rate determining step" in the process.

It might be that the modelling of physical features and chemical features is necessary for a complete picture, on the other hand, one or other may have such an overriding effect that certain departures in the others are of relatively little importance.

The particular variables that have been considered in Fig. 13. are:-

Air-fuel ratio, velocity, relative momentum and Reynolds number. The practical permutations possible are set out below:-

1.
- a. Air velocity constant.
- b. Relative momentum constant.
- c. Air-fuel ratio constant.
- d. Reynolds number allowed to vary.

Fig. 13. a.

2.
- a. Air-fuel ratio constant.
- b. Relative momentum constant (this could not in fact be achieved in practice but nearest values taken.)
- c. Velocity allowed to vary to give residence time constant.
- d. Reynolds number allowed to vary.

Fig. 13. b.

3.
- a. Reynolds number constant.
- b. Velocity varying in accord with Reynolds number.
- c. Air-fuel ratio constant.
- d. Relative momentum dictated by fixed atomizing pressure.

Fig. 13. c.

4.
a. Air velocity constant.
b. Air-fuel ratio constant.
c. Change in relative momentum to accommodate lower atomizing pressure.
d. Reynolds number allowed to vary.

Fig. 13. d.

5.
a. Air-fuel ratio x $\sqrt{\text{atomizing pressure}}$ constant and hence relative momentum constant.
b. Velocity constant (but diameter of burner altered).
c. Reynolds number allowed to vary.

Fig. 13. e.

Briefly the consequence on scaling parameters of these various permutations are reviewed below.

1) This is probably the most theoretically correct of the five. Only the Reynolds number in the selected variables is not maintained constant between prototype and model. The resulting diagram shows that the droplet Reynolds number is somewhat low, and as already discussed this is probably not very important. It also shows that the groups for ignition delay and the reaction time-residence time ratio are also much too low. The former can be improved by adjusting the fuel properties and the selection of a suitable distillate fuel will bring the figure back to the correct value. The remaining lack of similarity in the last group suggest that this should give poorer efficiencies in the model than the prototype.

2) In view of the impossibility of balancing the reaction time and residence time ratio without lowering the velocity, this selection of variables gives the correct residence time. This means that velocity and Reynolds number are reduced very seriously. The maintenance of the relative momentum of the oil and air is very difficult to achieve at this level and a substitute is to maintain a constant S.M.D. figure which helps to maintain all the terms related to the combustion process constant. This should give the correct chemical requirements, but this may give rise to noticeable departures in the physical processes.

3) This selection of variables shows the consequences of maintaining strict Reynolds similarity, without a corresponding pressure term to compensate for the effects. As can be seen, all the other groups depart very markedly from a constant ratio. It might be anticipated that even if a flame could be lit at all under these circumstances the result would be a very distorted model.

4) This is the same as 1,with the exception that the momentum ratio between air and fuel is no longer constant and is replaced by constant droplet size. This might be a convenient adjustment of the operating conditions from the practical point of view. The consequences however, are worsening of the terms affecting the combustion process for the

minor advantage of maintaining the correct droplet Reynolds number.

5) With this group of variables the effect of using the air-fuel ratio as a means of adjusting the scaling parameters is investigated. This shows some slight improvements in ignition delay and reaction time to residence time ratio and also a big improvement on droplet Reynolds number. On the other hand, the simultaneous evaporation and combustion term has been altered. The main disadvantage is that the burner inlet size is changed from the strict geometrical model with a consequent possibility of introducing errors on this account. Although superficially this is nearly as attractive as 1., it involves changing several extra variables between prototype and model with a consequent increased risk of introducing errors.

On this basis 1. and 2. seem to give from different points of view, the most nearly correct set of scaling ratios, and these two sets of conditions were therefore selected for experimental study.

### 3.4 Design of Oil Burner.

#### 3.4.1. Type of burner selected for study.

The terms of this investigation required the use of an actual piece of modern combustion equipment. The type selected was an Admiralty suspended flame register - (so called to distinguish it from flames that come very close to the air quarl), with pressure-jet atomization for the oil. This type of burner was developed by empirical methods for Admiralty needs after the Second World War, and was aimed to achieve good 'stability' (in the sense of flame holding) and high turn down. This register design is of the 'swirler' type. The full size prototype is illustrated in Fig.14. Three main air paths exist.

(I) Between the quarl and the swirler, which by itself will direct almost all its momentum as an axial component.

(II) The swirler, which is actually a series of vanes which will direct the part of the air passing through outwards as indicated, although once clear of the quarl the jet will spread outwards until it meets the outer air supply I.

(III) The third air stream is a small flow actually round the atomizer nose cover mainly to avoid carboning up.

The wide range atomizers normally used either are 'Duplex' atomizers with a double set of swirl grooves into the swirl chamber or rather more usually 'spill' atomizers in which part of the oil is allowed to spill back into the feed tank from the back of the swirl chamber.

In the full scale experiments the register used was as shown and both Duplex and Spill atomizers were used to give the various spray characteristics required.

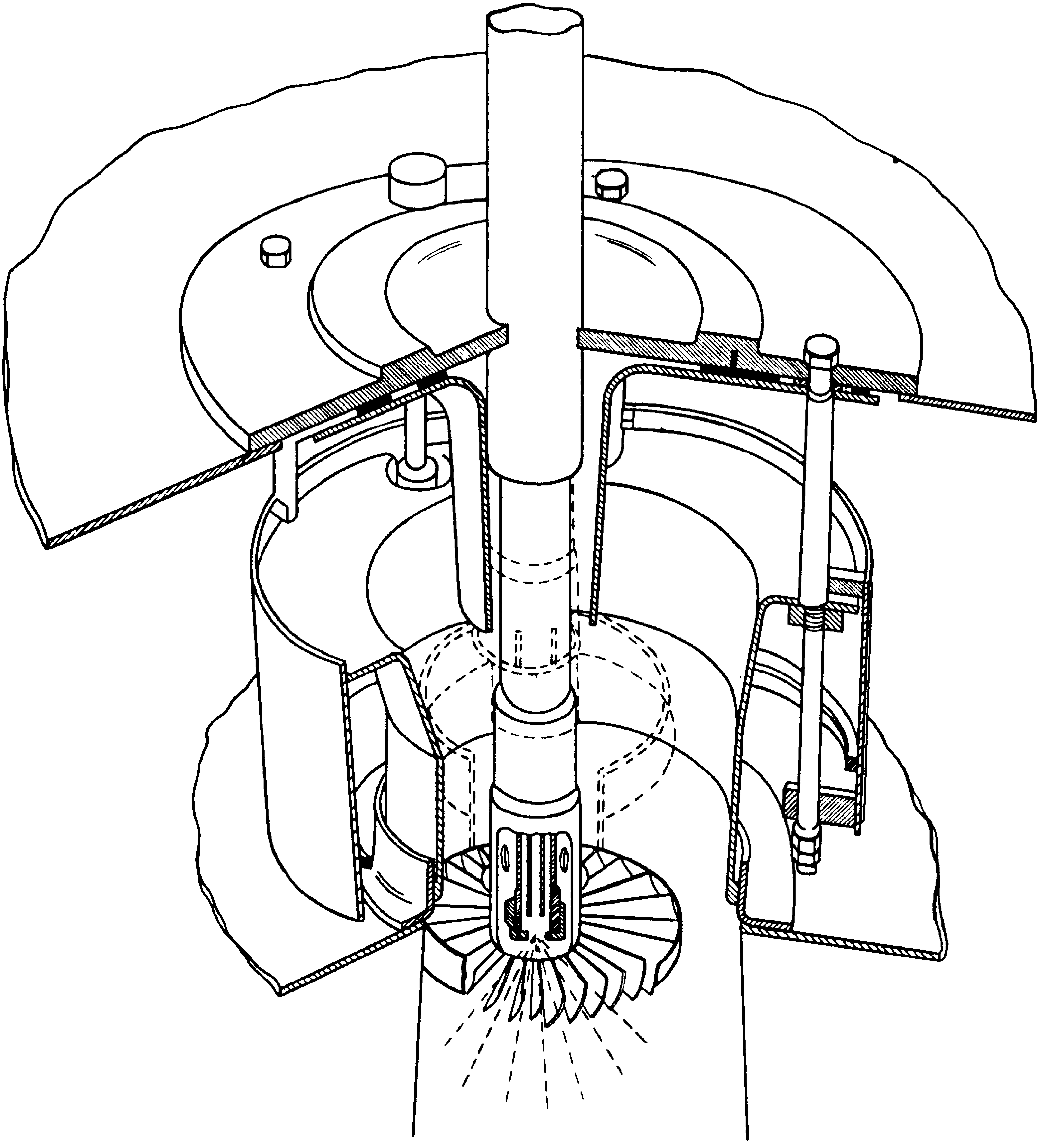


Fig. 14. A SUSPENDED FLAME REGISTER.

The actual burner used in the experimental work was designated a 12" register. This specifies the burner tube size, the air swirler inside having an outer diameter of 10".

Three operating conditions were chosen for study:-

1) Register with normal Duplex atomizer running at full-power.

2) As above but at 1/4 power.

3) Register running at full power rate but with wider angle oil spray. For convenience a spill atomizer was used to obtain the wider angle.

These three conditions gave full power condition as a reference point against which the effect of turn-down ratio and the effect of using a wider spray angle could be compared.

The use of the wider spray angle was chosen in order to obtain information on the effect of atomizer matching as it was thought that the operating results would vary with spray angle and that this might have some bearing on the problem of combustion oscillation limits.

The inclusion of a low turn-down ratio was designed to provide information on why combustion conditions deteriorate badly at low power levels in flames of this type. This deterioration is marked usually by increasing air-fuel ratios at the black smoke limit, until the air requirements are two or three times the stoichiometric value, and pulsation troubles.

These two factors usually form the practical limits of operation as far as air-fuel ratio is concerned.

### 3.4.2 Modelling the Burner.

The three important aspects of modelling a burner of this type are similarity, heat release rate and the feasibility of modelling the register parts on the scale selected.

On the question of similarity, the subject of scaling has been discussed in the previous sections. As the air flow through all parts of the register is at the same temperature for prototype and model there is no reason to adopt anything other than a geometric model of the original. Fortunately, the values taken for the possible variables in cases 1. and 2. of section 3.3.4, result in the fuel rate varying as $L^2$ and $L^3$ respectively. As a result it was still possible to maintain a reasonable burner size for conveniently low fuel rates. The scale ultimately chosen was $\frac{3}{10}$ths. This was the smallest scale in which available atomizers could be fitted without distorting the burner parts and with slight modification allowed the parts to be rigid enough to stand up to operating conditions. The burner components are illustrated in Figs. 15. and 16.

The atomizers were changed to give the desired throughput and spray angle for the different sets of scaling criteria.

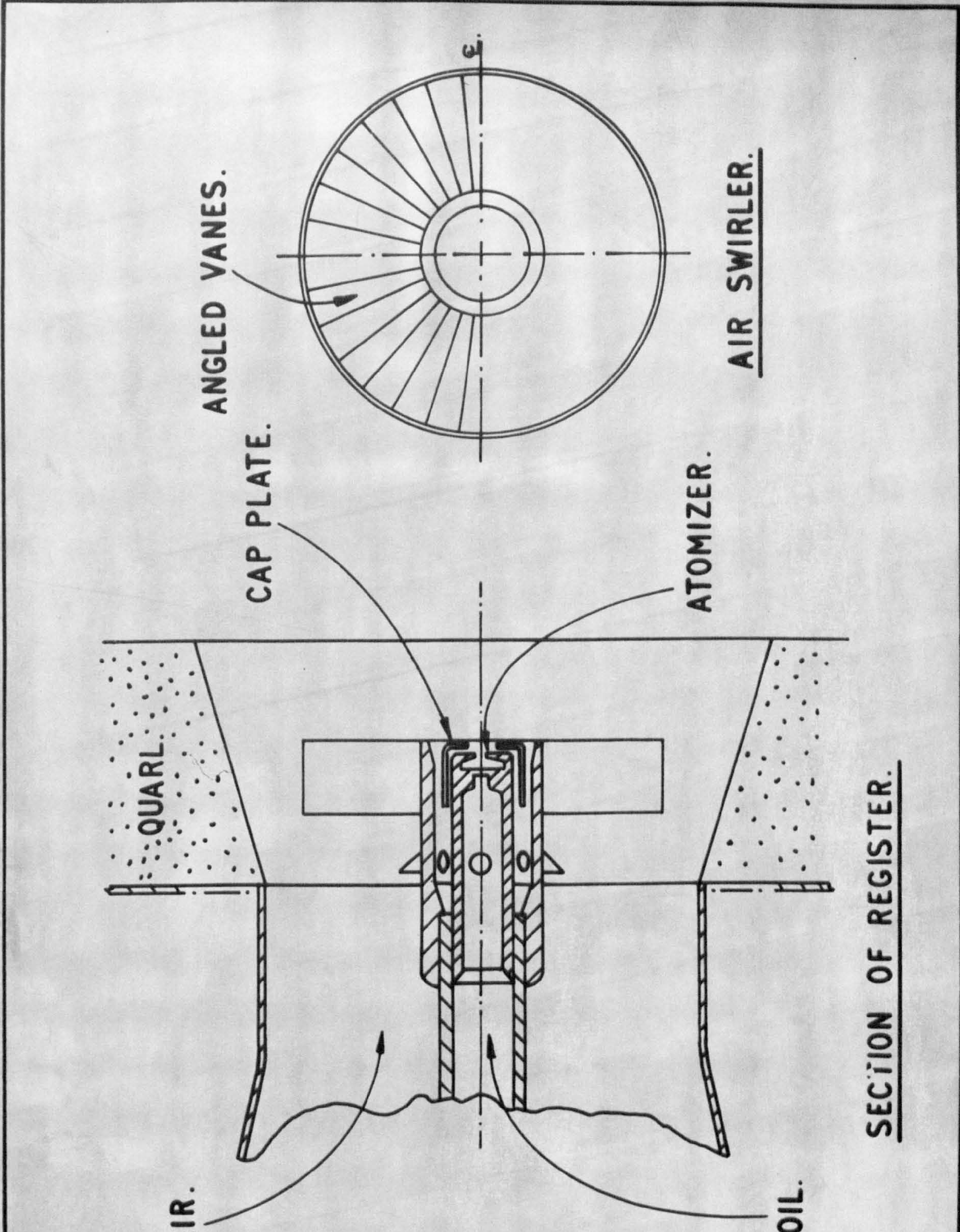
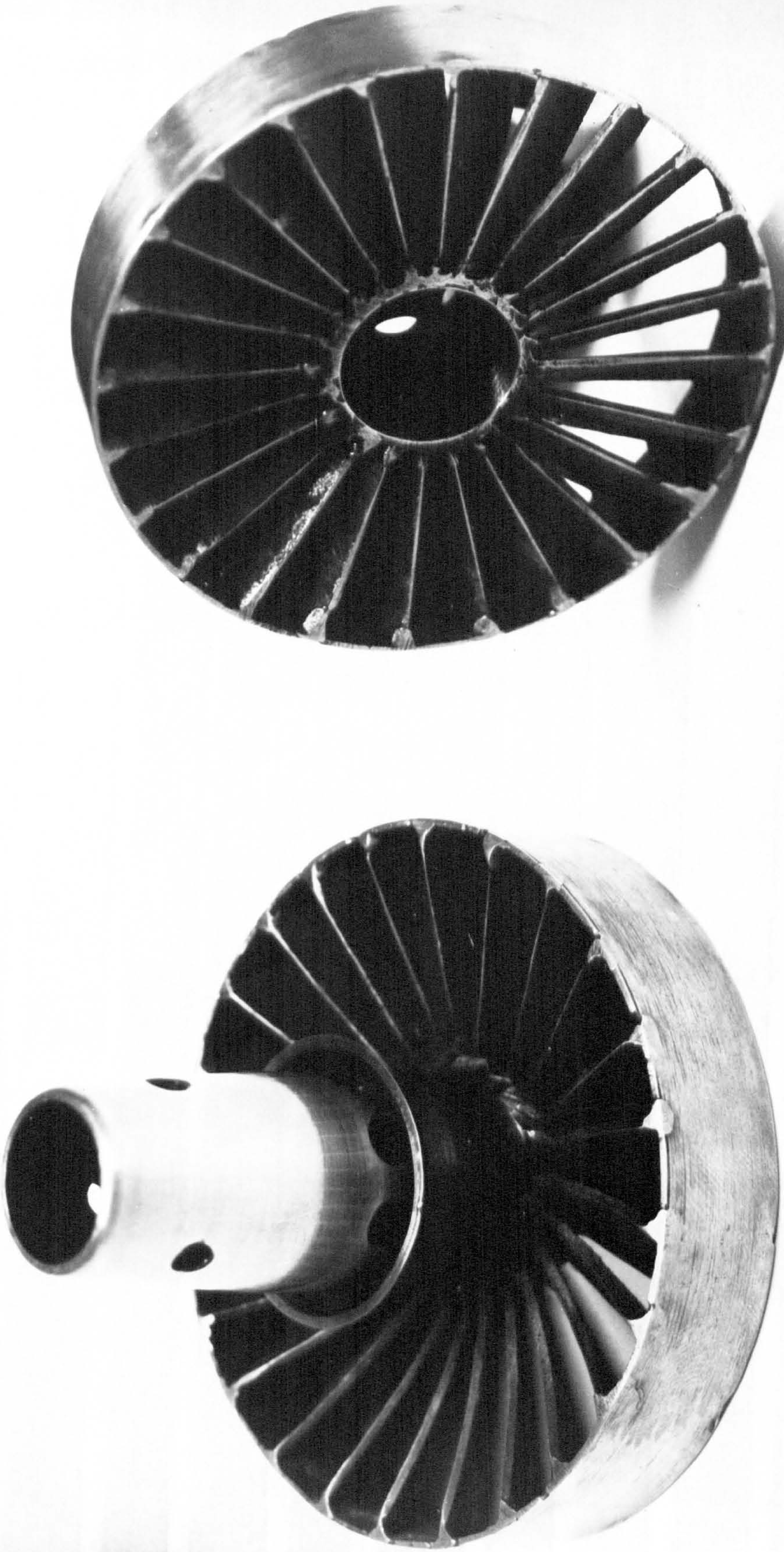



FIG.15. MODEL REGISTER

FIG. 16.

Vane Swirler for Model.



A second burner was built and tried. This had characteristics corresponding to case 5, where the burner size was distorted, but for the reasons mentioned in section 3.3.4. it was considered to be based on more dubious principles and no detailed experimental data was obtained.with it.

### 3.5 Combustion oscillation experiments.

As is already apparent from the review of the literature (Section 2.9. and Appendix II),the apparent wide range of combustion phenomena reported made it desirable to carry out this section of the investigation under actual operating conditions on ships of the type on which the maximum trouble had been experienced. These were three drum boilers with two waste-gas passes and a ducted air supply. The nearest land based boilers in type had only a single gas pass and were known to behave differently in certain respects.

The two characteristics of oscillating combustion that it is possible to measure are the oscillation frequency and the amplitude. The low frequency can be measured with simple apparatus without much difficulty. The determination of the frequency is important since this will have some fairly direct relation to the driving phenomena. (Thus for example,in an acoustic type oscillation a knowledge of the frequency makes determination of the mode of oscillation possible, i.e., longitudinal, transverse, etc.). All the theories in the literature review make use of the oscillation frequency as the main characteristic. A knowledge of the amplitude is also important. There are, however, a number of problems in the measurement of this which have yet to be sorted out[98]. The amplitude becomes important once the conditions have moved away from the limit of oscillations. Under these conditions the amplitude probably has a value of the same order as the pressure drop across the register.

Unfortunately the theoretical computation of actual amplitudes is at the moment out of the question due to the number of factors involved.

For the experimentscarried out in connection with this study the limiting frequency was measured against fuel input rate and characteristic pressure drop across the air register. An approximate analysis of the combustion products were also included,although there were some uncertainties about the actual results. Apart from the actual collection of data it was hoped to show what effects influenced the oscillation frequency and the limiting condition for oscillation.

### 3.6 Cold Model Work.

The main objectives of this work were taken up with establishing the behaviour of existing flames and methods of scaling. Some supplementary work was planned using the burner running cold in order to determine the initial flow configuration at the burner, to compare the hot and cold flow patterns and to determine pressure drop data.

The question of similarity arises in this case when the flow patterns are compared with the hot results. Thring[47] has shown that similarity can be maintained even for systems running at substantially different temperatures by arranging the momentum in the flame or simulated flame to be the same and if necessary distorting the geometric scaling of the burner until it is achieved. This leads to the concept of an equivalent nozzle diameter (see section 2.7.2. ) which may if necessary include both fuel and air which can be used to reduce dimensions in the combustion chamber to a dimensionless form (i.e. $x/d_o'$).

Unfortunately the early stages of a jet with swirled flow are so strongly controlled by the geometrical features of the system that this is still likely to give misleading results. These are discussed with the experimental results.

# CHAPTER 4.

## Description of Apparatus used.

### 4.1 Boiler and Full size Burner.

The boiler used for the full size burner work was a Foster Wheeler type two drum boiler set up as a test boiler at the Admiralty Fuel Experimental Station. Fig. 17. shows the general arrangement of two refractory walls and floor, the thermal load being provided by a water wall and tube bank. It was modified by eliminating some tubes on the side water wall and positioning five ports for probes on the side, at the following distances from the front wall - 7", 26½", 46", 83" and 102". The maximum loading for the boiler was about 4,500 lb./hr. of oil, but for this experimental work fuel rates of up to 2,200 lb./hr. on one burner only were used.

### 4.2 Laboratory Furnace.

For simplicity in laboratory work a refractory lined combustion box was used, despite the fact that this imposed some limitations. This was built to give approximately the same residence time as a boiler, although any attempt at an accurate copy was out of the question. The furnace shape and flues were arranged to give a symmetrical flow pattern and hence any problems arising from the rather special shapes of most marine boilers were avoided.

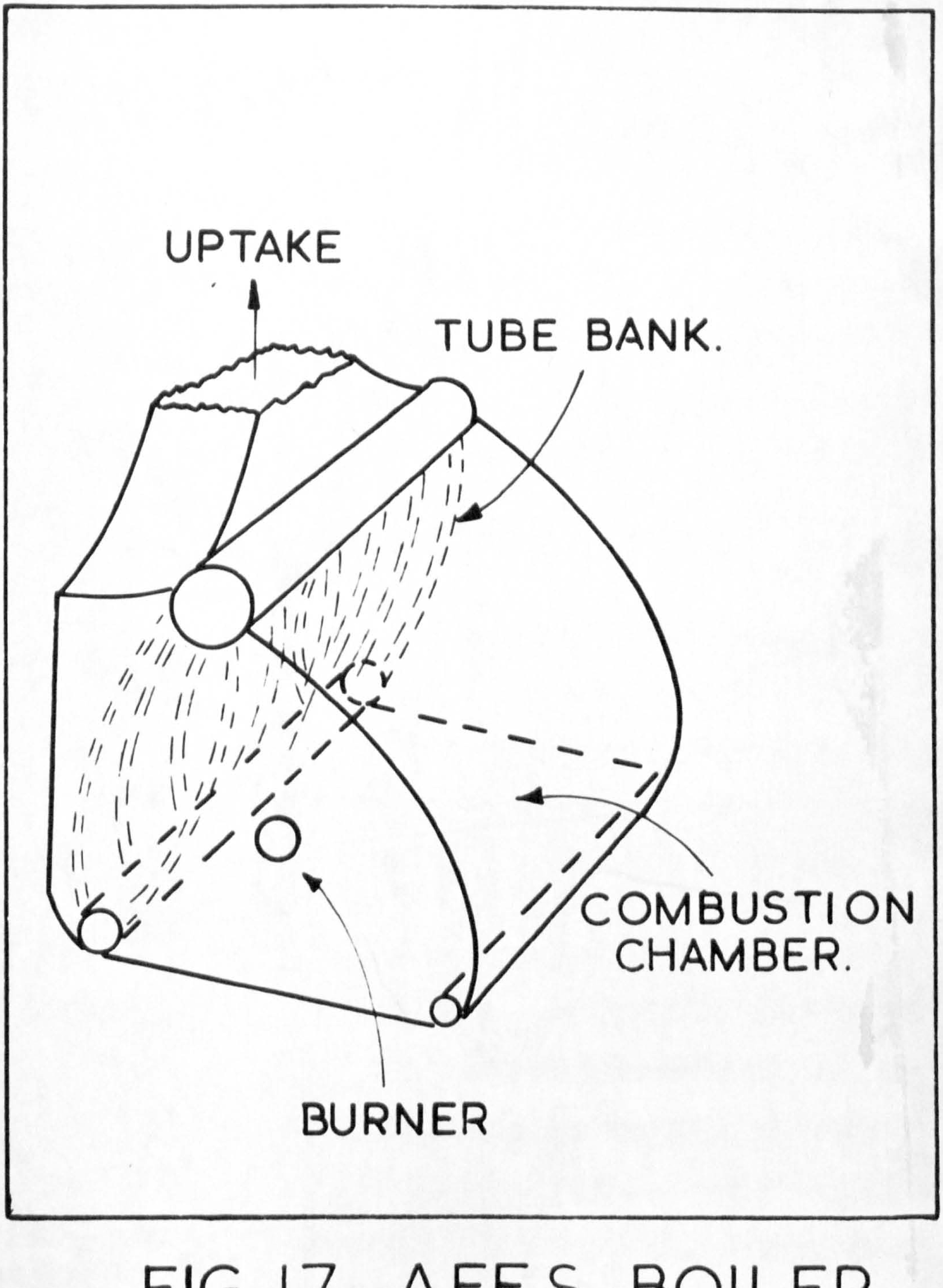



FIG. 17. A.F.E.S. BOILER.

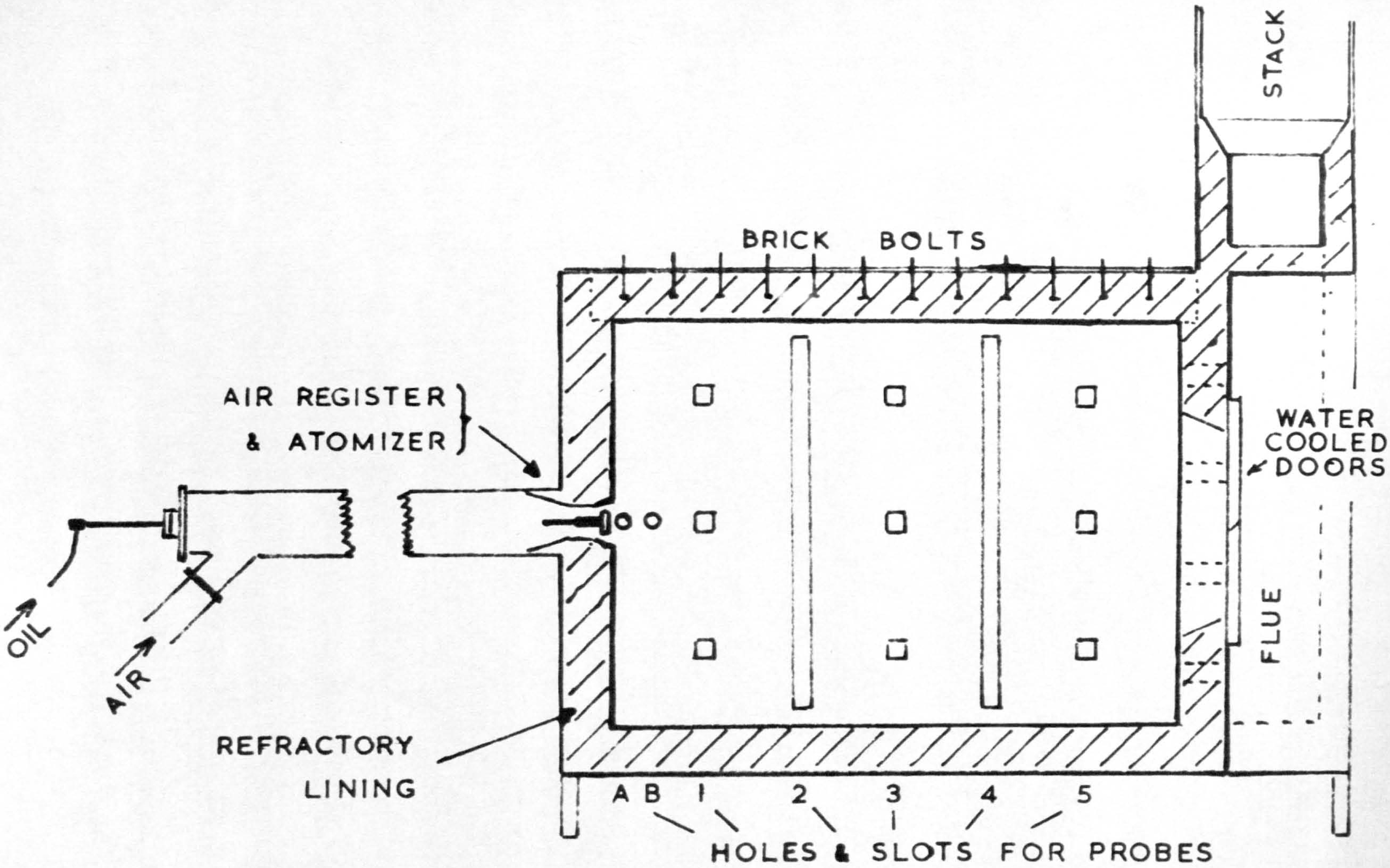



FIG. 18. SECTIONAL DIAGRAM OF LABORATORY FURNACE

The actual construction was in the form of a steel case to prevent excessive air leakage inwards and a $4\frac{1}{2}$" H.T.I. refractory brick lining with a silliminite coating. The internal dimensions were 3'3" across x 3'3" high x 4'3" long. The general arrangement is shown in Fig. 18. and photographs in Figs. 19. and 20.

Special features of construction were:-

(i) The removable "suspended" roof which consisted of a 1/4" steel plate with 12" x 12" x 4"HTI tiles. The tiles were secured by 3/8" bolts recessed into the brick and bolted to the plate. Spring washers were provided to allow for any thermal effects.

(ii) To keep the total length of the furnace to a minimum, the flue was split up into four sections connected to the combustion chamber by a series of ports. This allowed probes to be inserted upstream.

(iii) For maximum flexibility the furnace was provided with sampling ports and slots with water cooled doors and two holes and a water cooled slot between the flues on the back wall.

The furnace was initially set up with a short stack through the roof. Later it was moved to the Department's new building and connected with an induced draught duct system. Dilution air was used to cool the stack to a point well above the gas

FIG. 19.

Interior of Furnace under Construction, Showing Sampling Ports and Flues.

FIG. 20.

External View of Furnace.

sampling point. The details of the air and oil supply systems are given in Section 4.4.2.

## 4.3 Boiler used for combustion oscillations.

A few preliminary results were obtained on the boiler described under Section 4.1. However, much of the air ducting arrangement and stack were not in line with current practice in the Fleet. It was therefore considered desirable to make measurements on an actual boiler installation on a ship. In view of the difficulties involved in coping with the large volume of steam produced while the ship was alongside in the harbour, it was necessary to carry out these trials at sea. The work was carried out while other boiler trials were in progress, and the ship was steamed as far as possible to meet with trial requirements. The boiler in this case was one of a pair of three drum Yarrow boilers connected to common blowers and stack. Each was fired by six burners of a similar type to the S.W.F.R. air register used for the static work. Because of the high stresses placed on the boiler by pulsation at high powers, oil throughput was limited to 1,700 lb./hr. on each burner. The boiler front and measuring gear are shown in Fig. 21. For purposes of this work the second boiler remained out of action.

FIG. 21.

Front of Boiler Used for Pulsation Experiments,

Showing Position of Microphone Detector.

## 4.4 Instrumentation of Test Facilities.

### 4.4.1 A.F.E.S. Boiler.

Certain limitations had to be accepted in this case, since accurate metering of the air to this particular pressurised boiler room was not available. This was overcome by analysis of the flue gases.

Metering of oil was carried out roughly by setting the burner to run accurately at a given pressure, with a known oil temperature. This was checked by timing the consumption of small quantities of oil (usually 10 gal.) with a stop watch. This process was necessary to avoid the rather complicated procedure of working back from the fuel viscosity when this was subsequently determined by laboratory test.

Flue gas analysis was carried out by three Mono $CO_2/CO$ recorders, sampling from grids in the stack for control purposes. Occasional crsat checks were made to verify these readings.

Boiler room air pressure and register draught loss were also checked and recorded at regular intervals.

### 4.4.2 Laboratory Furnace.

Much greater control was possible on this apparatus. The general arrangement of supply services is shown in Fig. 22.

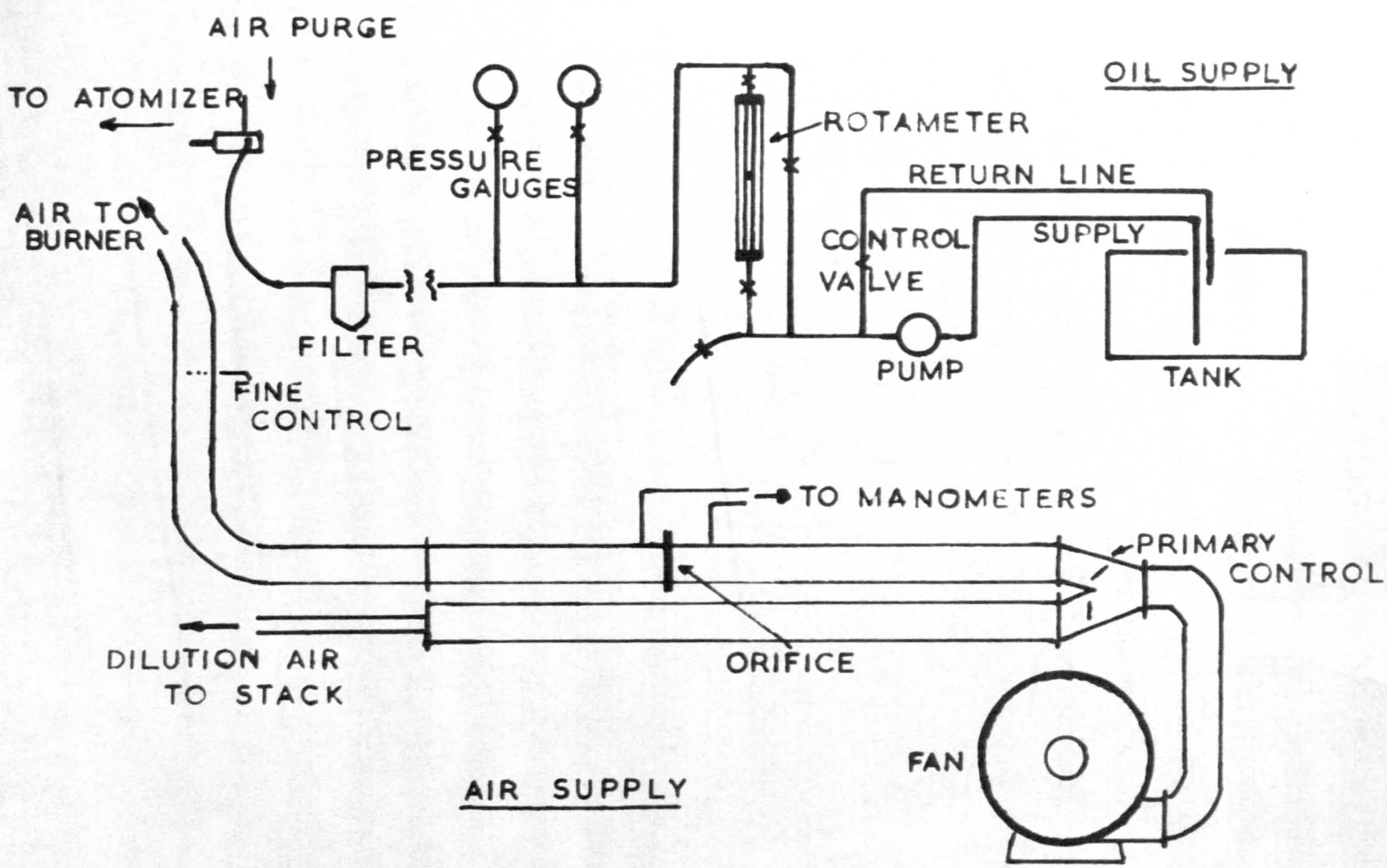



FIG. 22. DIAGRAM SHOWING ARRANGEMENT OF SERVICES FOR LABORATORY FURNACE

The air supply was provided by a Keith Blackman fan in the basement with a manifold arrangement as air off-take. The main air supply line was provided with an orifice metering device with D. and D/2 tappings designed according to B.S.1042. For the orifice plate used flow rates in lb./hr. were calculated from the relation 2942 $\sqrt{\Delta h}$ $\sqrt{\rho_{or}}$ determined from B.S.1042 data. ( $\Delta h$ = pressure drop across the orifice

$\rho_{or}$ = density of air at the orifice).

Coarse control of the air supply was carried out by regulating of the air supply to the various off-takes at the manifold, and fine adjustment by a gate valve at a suitable position between the orifice and the burner.

A long approach tube to the burner was provided with flow straightening vanes to ensure uniform air supply at the register.

The oil supply system consisted of a 40 gallon supply tank which could be refuelled as required from 40 gallon drums stored outside the building. The supply system from the pump allowed for spill back to the tank if necessary. Interchangeable rotameters were provided for metering the oil supply. Duplicate pressure gauges were used to meter the oil supply pressure. As the pressure gauges were still some distance from the burners, calibration runs were carried out with the burner discharging into a suitable receiver. An extra pressure gauge was mounted just behind the atomizer to determine the pressure drop from the measuring point.

Static and dynamic tappings were provided just behind the burner at each end of the combustion chamber, in the stack, and also at a suitable point near the bottom and near the top of the furnace. In order to ensure reasonable reproducibility of furnace conditions, Chromel Alumel thermocouples were provided at most of these points and also at two places on the back wall.

Sampling in the stack was provided by two L-shaped gas sampling grids. These were arranged to cover the stack in four quadrants, each limit having a number of small gas sampling holes. Separate leads were provided from each grid and taken via suitable condensation traps to the stack analysis stand; this was provided with a twin range Kent oxygen analyser and a Mono $CO_2$/CO meter. The oxygen analyser was identical to the type on the gas analysis rig and a full description appears under section 4.5.2. Suction was provided by a small Dymax diaphragm pump.

Some complications were introduced by the ducted gas disposal system fitted in the new laboratory. However, the draught break provided by the dilution air duct which surrounded the lower part of the stack substantially reduced the suction on the furnace. Control was effected by means of a damper in the stack pipe after the dilution air had been mixed in. In order to avoid trouble from overheating in the latter stages of the ducting where this formed part of the laboratory ceiling, a second forced dilution air supply was added after the damper.

#### 4.4.3 Instrumentation for combustion oscillation.

Instrumentation for the combustion oscillation work was carried out in two parts, measurement of input variables (mean values only where these were subject to oscillation) and measurement of oscillation frequency and amplitude.

Unfortunately on operational ships, very little instrumentation is provided which is really capable of exact measurement of quantities. It was, however, possible to meter the oil input accurately, to carry out the combustion gas analysis by normal plant testing methods, and to observe the register draught loss immediately prior to the onset of unstable combustion.

Metering of the oil throughput was carried out by measuring the supply and spill pressures to the burners on accurately calibrated gauges. The actual flow rate corresponding to these pressures could be determined from the maker's calibration data, once the oil viscosity was known. Gas analyses were made as soon as possible after the onset of oscillations. However, due to the necessity for restricting oscillation to the minimum period possible to avoid damage to the boiler, it was not possible to let the air rate assume a steady value. This meant that the exact point at which gas samples were taken was difficult to control. They have provided, however, indication of the apparent "air/fuel" ratio inside the oscillation limits. The samples were analysed for $CO_2$, CO and $O_2$ as these were the only combustion gas constituents likely to occur. It was not known, however, if appreciable quantities of unburnt fuel were lost as unburnt carbon under the black smoke conditions that usually occurred along with the combustion oscillations. This introduced a further uncertainty. A subsidiary analysis for oxygen was made with a D.C.L. oxygen analyser sampling the flue gases continuously. This instrument works on the basis of the variation of deflection of a dumbell suspended in a magnetic field due to the magnetic susceptibility and hence oxygen concentration of the gas passing through it. It was possible from this to determine the minimum oxygen concentration occurring during pulsation, this was particularly useful at low fuel rates. Here again the time lag in sampling meant that this was just the minimum reached and it was not possible to correlate it exactly with any specific point in the oscillation data.

## 4.5 The Measurements made in the Combustion Chamber.

### 4.5.1 Temperature Measurement.

Local temperature measurements were made in each flame investigated in order to obtain the temperature distribution and to allow the gas density to be determined for the velocity measurements.

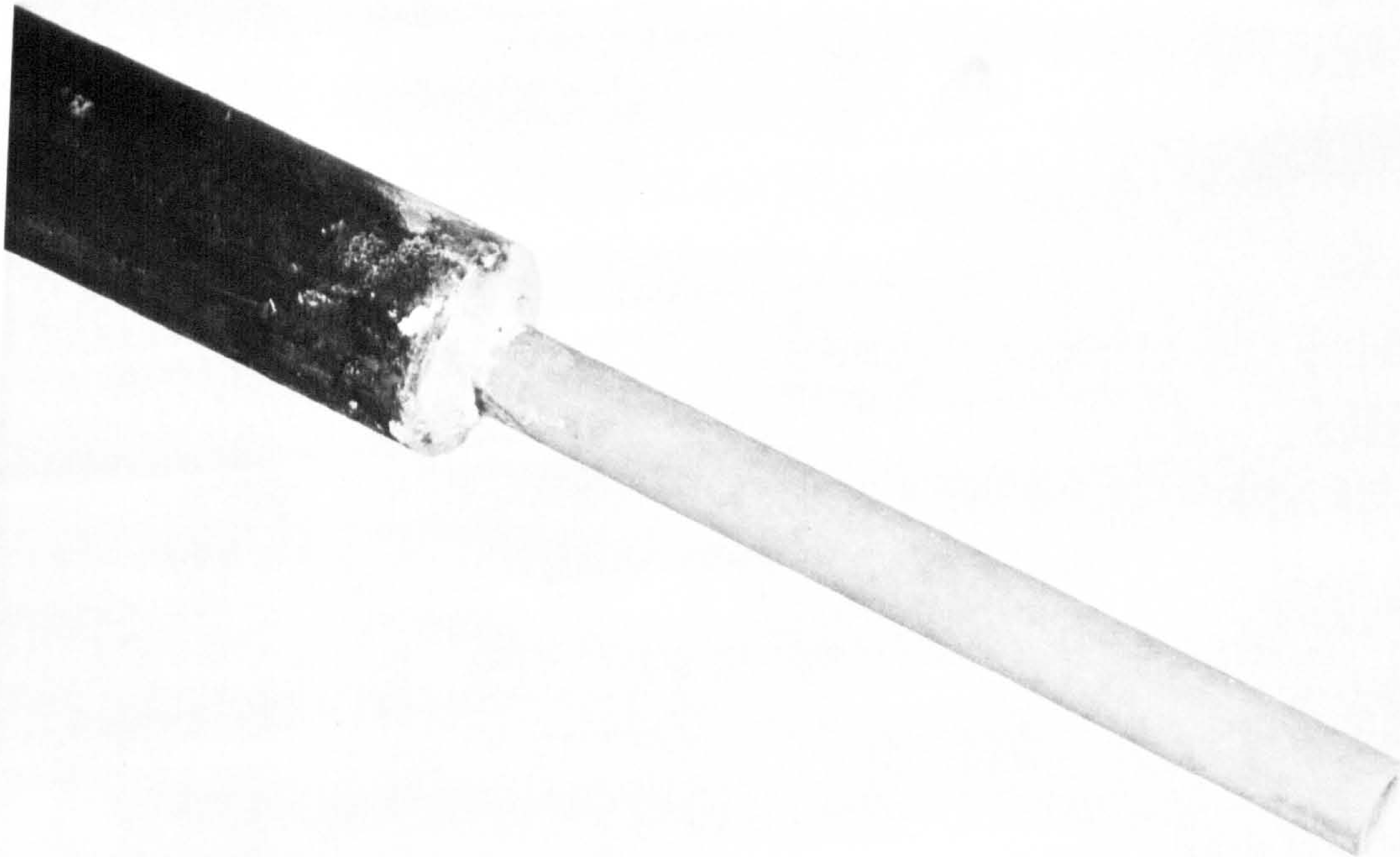
Two satisfactory techniques are readily applicable to local temperature measurement in the combustion chamber. These are the suction pyrometer and the venturii pneumatic pyrometer. Other techniques exist for high temperature measurement such as the line reversal technique, or from Schmidt type measurements, but these are not easily used when substantial temperature gradients are encountered in the gas stream, as in the work described in this thesis. Of the two workable techniques the suction pyrometer is considerably easier to construct and is a good deal less complicated to use.

The suction pyrometer used in the laboratory furnace consisted of a refractory radiation shield enclosing an inner sheath and Pt/Pt 13% Rh. thermocouple. This is shown in Fig. 23. A similar but slightly larger device was used for some of the

FIG. 23.

Operating End of the Suction pyrometer

Used in the Laboratory.



temperature measurements made at the Admiralty Fuel Experimental Station. A diagram showing the construction of this type of pyrometer is given in Fig. 24. For the other measurements on the A.F.E.S. boiler standard pyrometers, manufactured by Land Pyrometers, were used with either blackened metal or refractory shields depending on the temperature. These are shown in Fig. 25.

The suction pyrometer presents some problems in use, one being the rather fragile nature of the refractory shields used and the second the problem of correcting the results to give a true gas temperature. This latter stage is usually carried out by determining the suction pyrometer efficiency.

The "efficiency" is defined as the difference between the maximum recorded temperature with suction past the thermocouple and the temperature with no suction, divided by the actual gas temperature minus the no suction temperature. Thus, for a measured difference in temperatures with and without suction of say 180° C. and an efficiency determined as 90%, the actual gas temperature would be:

$$\left( \frac{100}{\text{efficiency}} \times T_{max} - T_0 \right) + T_0$$

thus the true figure to be added to $T_0$ to give the correct gas temperature would be

$$100 \times \frac{180}{90} + T_0 = 200^\circ + T_0{}^\circ \text{ C}$$

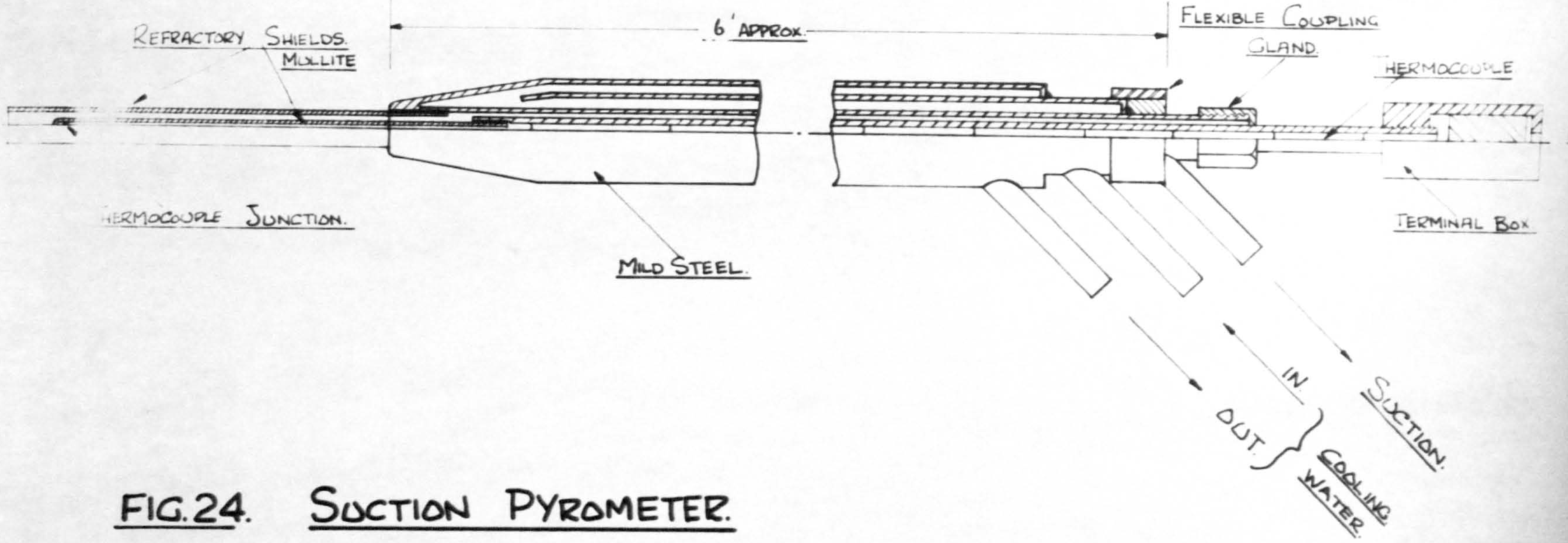



FIG. 24. SUCTION PYROMETER.

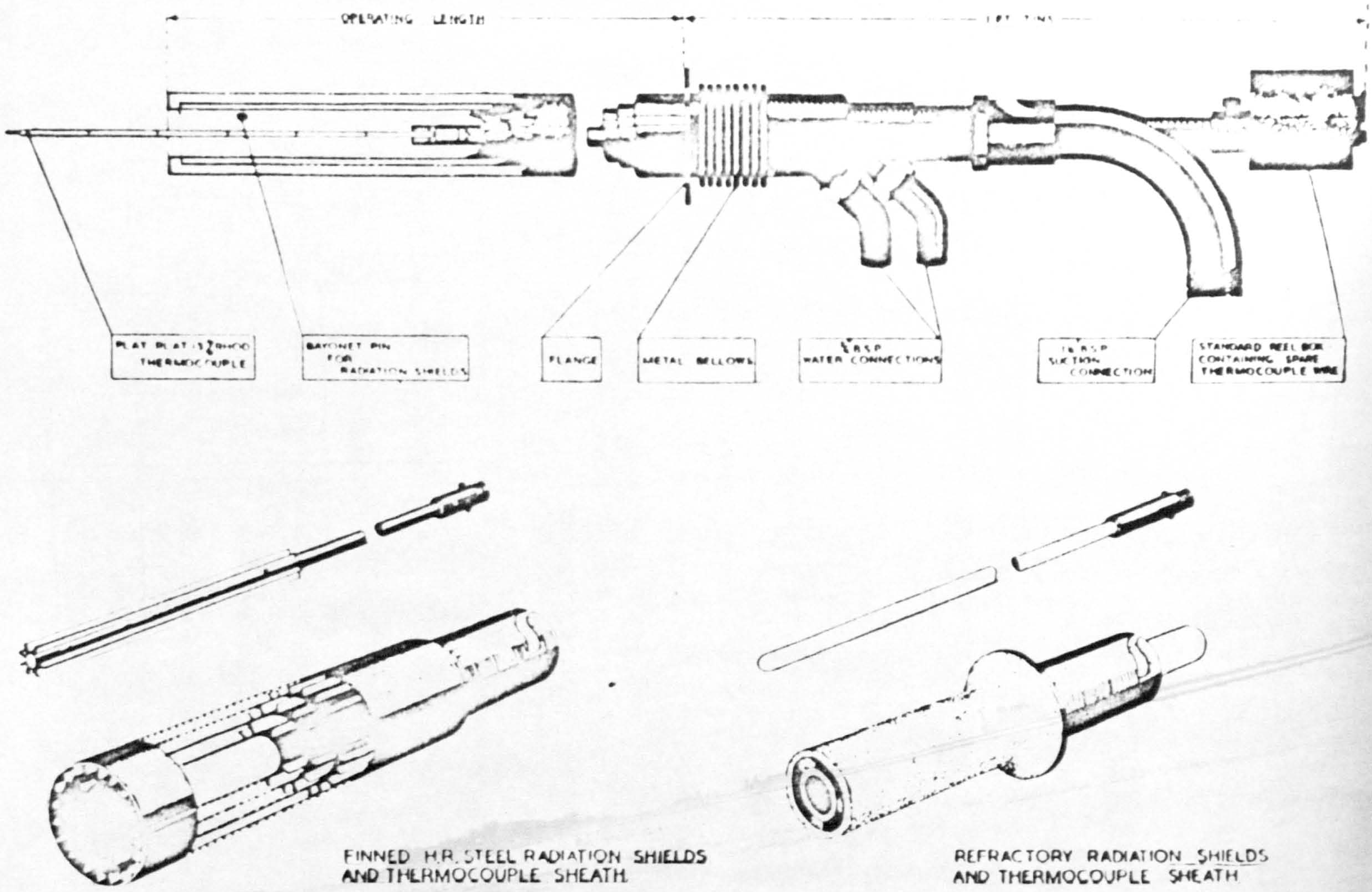



FIG. 25. MULTI - SHIELD PYROMETER.

The "efficiency" for a suction pyrometer can be determined according to Land and Barber[99] in three ways.

(1) By using the effect of suction velocity on the temperature measured to determine a "shape factor" for the pyrometer shields.

(2) By use of the ratio of the time lags that occur when the suction is turned on and off.

(3) By direct calculation from the thermal properties of the suction pyrometer shields.

Usually standard curves calculated by Land and Barber are used to relate calculated factors such as the "shape factor" to the efficiency. These methods of correction are discussed further in Appendix IV.

Wall temperature measurements were made at a number of places on the laboratory furnace using embedded Chromel-Alumel thermocouples for control purposes.

### 4.5.2 Gas Analysis Scheme.

The principal way of following the progress of combustion is by analysis of the combustion gases at various points. Two quantities may be determined - the air-fuel ratio and the combustion efficiency. The first gives the proportion of fuel products to air products at any given point, the second the extent to which combustion has actually proceeded at that point.

These quantities, usually are determined from a fuel analysis of the permanent gases up to the lowest members of the $C_xH_{2x+2}$ and $C_x H_{2x}$ groups. That is to say $CO_2$, $CO$, $O_2$, $H_2$, $CH_4$, $C_2H_4$ and $N_2$ (by difference).

From these results, Carbon and Hydrogen balances are made and from these the air-fuel ratio can be calculated (see Appendix V.). Extracting the figure of unburnt fuel in all forms, the combustion efficiency also can be calculated.

Suitable methods for the estimation of these are by:-

1. Orsat.
2. Gas Chromatography.
3. Gravimetric determination.
4. Mass Spectrometer.

None of these methods is very rapid and except for the last are subject to fairly serious inaccuracies, particularly as far as components in low concentrations are concerned.

To provide a much more rapid means of analysis than that of the conventional 'wet' means using a "Dutch" type orsat[100], it was found at a fairly late stage in the work that it was possible to develop a 'mechanical' analysis train. This is designed to measure only essential components, to determine the concentration plot of $CO_2$, $CO$ and $O_2$. These figures can be used to calculate air-fuel ratios and combustion efficiencies except in the region where appreciable quantities of hydrocarbons and hydrogen occurred ,when full analysis was necessary.

In general the ideas for this were drawn from the work of Macfarlane[101] and Toone[102].

The instruments used for $CO_2$ and CO measurements were standard IRGA's (Infra-red Gas Analysers) manufactured by Grubb Parsons & Co. Ltd. The principle on which these work is illustrated in Fig. 26. Measurement is by differences in adsorption produced on the two sides of the so called 'Luft Cell' by the passage of two 'chopped' infra-red beams from a pair of low voltage Nernst-Glowers through a sample tube and an air filled reference tube. The difference is produced by absorption of radiation by the gas in the test cell. The detector cell is made sensitive to certain absorption bands only appropriate to the selected gas by filling it with the gas in question. Sensitivity with this apparatus can be as high as 0.05% of sample for full scale for CO and 0.01% for full scale for $CO_2$ using the maximum length of cell.

Two oxygen analysers were used, one as part of the analysis train and one for final analysis of waste gases from the stack. In the latter case the $CO_2$ and CO figures were checked by occasional use of the analysis train. Both analysers were identical and were of the paramagnetic type ,being manufactured

Fig. 26.

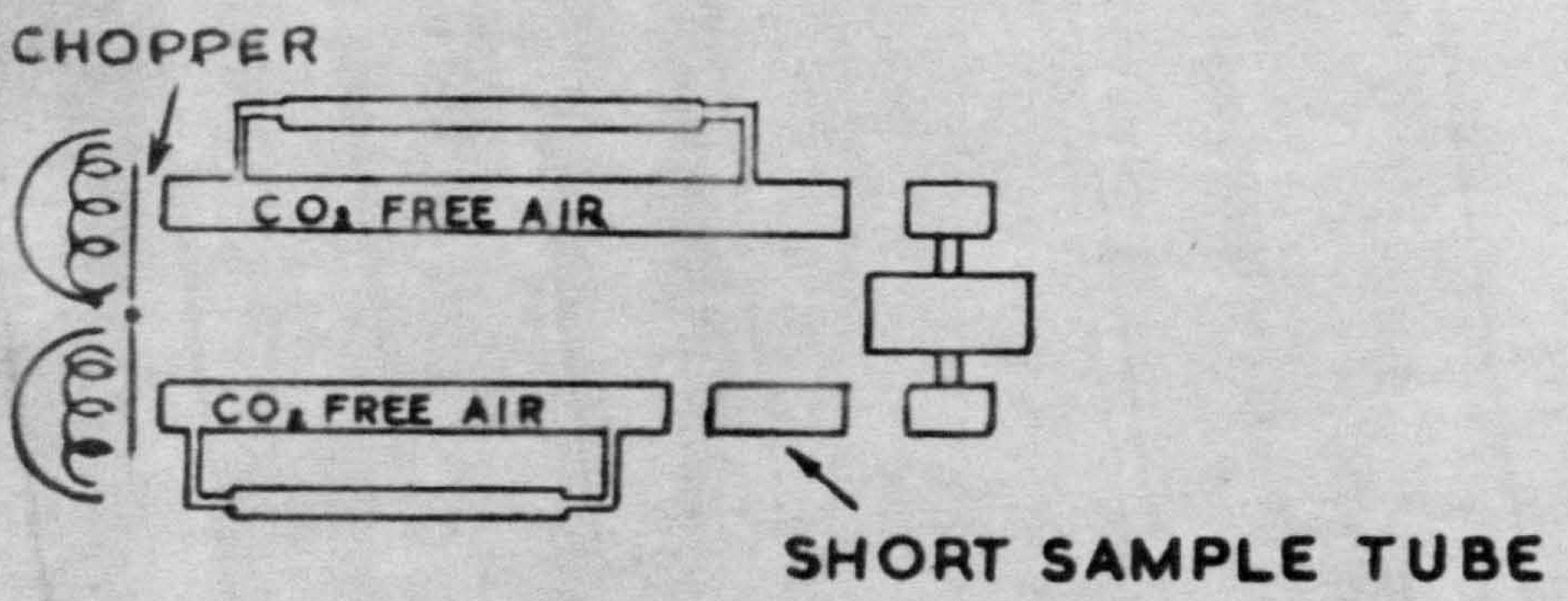



Fig. 27.

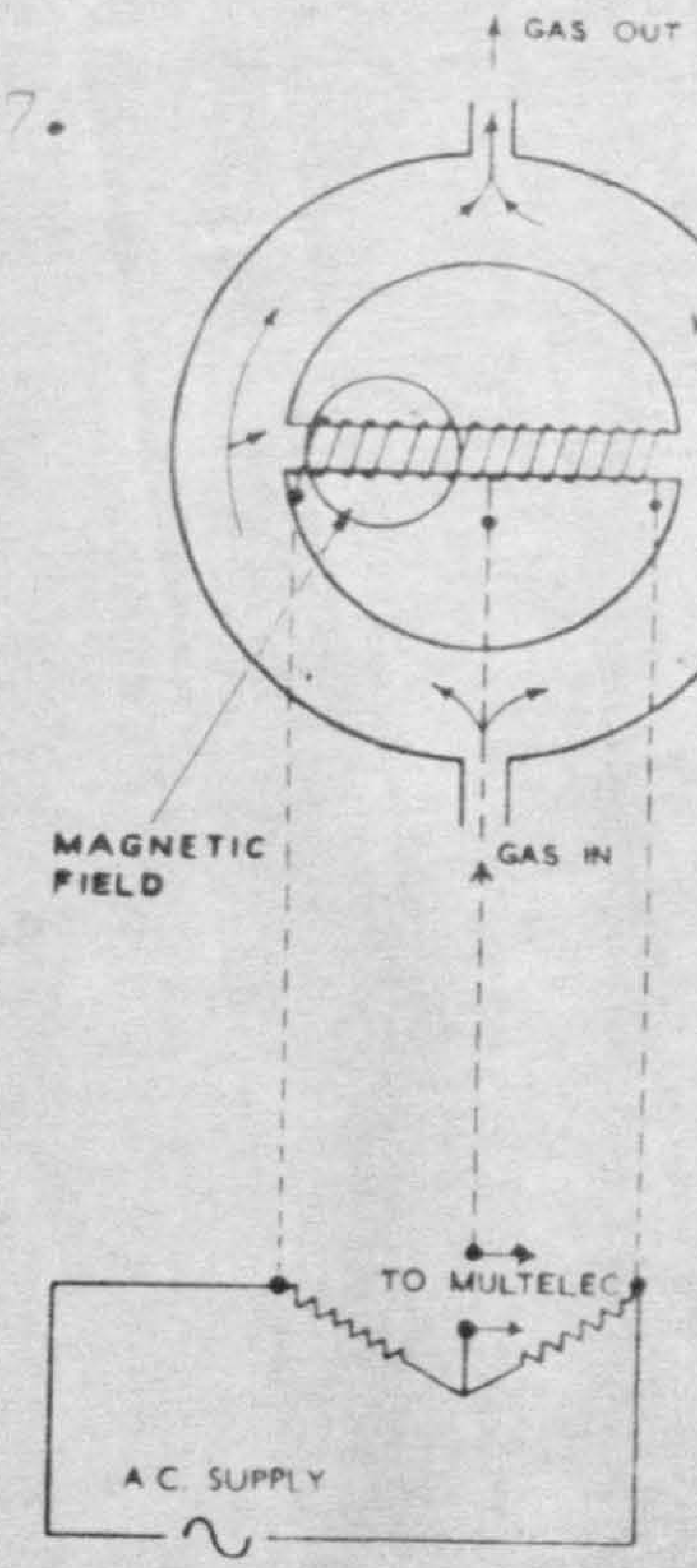



Fig. 26. Diagram of the principle of operation on an infra-red gas analyzer.

Fig. 27. Diagrammatic representation of measuring cell of oxygen analyzer.

by George Kent Ltd. These depend on the magnetic wind effect due to changes in thermal conductivity in the magnetic fields of paramagnetic gases. Gases are divided into two classes as far as magnetic effects are concerned namely, the diamagnetic gases which seek the weakest part of the field and the paramagnetic which seek the strongest part of the field. Nearly all gases are diamagnetic (typical volume susceptibilities $\chi$ are $N_2 = -0.342 \times 10^{-6}$; $CO_2 = -0.423 \times 10^{-6}$ CGS Units at $20^{\circ}$ C). The exceptions are $O_2$ and NO ($\chi$ values $O_2 = +106.2 \times 10^{-6}$; $NO = +48.65 \times 10^{-6}$). The form of the instrument is as shown in Fig. 27. The measuring cell has a horizontal link with two identical adjacent windings connected across the Wheatstone bridge and is heated by a voltage across the bridge. A permanent magnet is set up with the field acting across one end of the link. The oxygen from the stream passing through the cell is drawn into the magnetic field and heated, which reduces its susceptibility, i.e., $\chi$ (vol. suscept.) $\propto \frac{1}{T^2}$ and is therefore displaced by fresh cool gas and passed along the tube from left to right. The loss of energy from the l.h. coil results in the bridge being unbalanced, and with calibration, this can be used to measure $O_2$ concentration. The theory of this type of instrument has been worked out by Lehrer[103].

It is necessary to consider sources of error with this type of instrument which can be due to:-

1. Effect of carrier gas composition.
   (a) On calibration.
   (b) On zero shift.

2. Temperature and conductivity.
   (a) Temperature changes due to changes in gas flow rate or drift in the instrument temperature.
   (b) Changes in thermal conductivity of the gas due to change of gas composition.
   (c) Changes in gas density due to changes in pressure. This can give rise to large errors and a standard correction procedure is necessary for all readings.

The methods employed to deal with these sources of error are discussed in Appendix Vl.

With this instrument, the chief disadvantage is that although the magnetic susceptibility is unaffected by other physical effects, the detecting device is very sensitive to temperature, gas conductivity and errors from altering the inclination of the measuring section. From this point of view most of the above corrections would have been unnecessary in a "dumbell type" instrument such as the Servomex,had one been available.

The actual gas sampling from the furnace was carried out with water cooled gas sampling probes. These were 3/4" o.d.

probes with a 1/16" bore gas sampling line in the centre, 3' long for laboratory work and 7'6" long for work at A.F.E.S. The sampling end of the one constructed for laboratory use is shown in Fig. 28. Gas sampling rates were kept as high as possible to provide rapid quenching of the hot gas sample in order to stop further reaction between the constituent gases.

### 4.5.3 Design and operation of Velocity Measuring Instruments.

#### 4.5.3.1 General Considerations.

The problems involved in this research gave rise to the need to develop instruments suitable for this particular piece of work. The instruments described fall into three groups. All were pitot type probes coupled to a suitable micromanometer and were potentially capable of reading down to about 2 ft./sec. at 20° C or about 6 ft./sec. at 1500° C, this being the reasonable limits of measurement of the micromanometer. Because of the high turbulence and steep pressure gradients encountered, the most vital feature in the design of suitable instruments was to keep the pressure tappings as close together as possible. This requirement excludes conventional Pitot Static tubes of the N.P.L. standard type, as the dynamic and static tappings are some way apart (i.e. 6 tube diameters in the standard N.P.L. design). There is a further disadvantage with this type of pitot in that the static readings become suspect

FIG. 28.

Water-cooled Gas Sampling Probe,

Showing Sampling End.

if the pitot is badly aligned. The practical minimum diameter for a water cooled pitot would be about 1/4" which would still leave about 1.3/16" between the two tappings.

A further problem in the system under consideration is that some parts of the flow field are not by any means parallel to the combustion chamber axis which means that either only certain components of the flow can be measured or that recourse to some form of 3 dimensional instrument is necessary.

4.5.3.2 <u>Design of Single directional pitots.</u>

For much of the large scale furnace work it was not feasible to carry out proper 3-dimensional measurements with the time apparatus and equipment then available. The solution adopted was to design a pitot with upstream and downstream tappings about 2 D apart and with a fairly wide acceptance angle. The disadvantage of this type of pitot is that in the relationship

$$\Delta p = K \tfrac{1}{2} \rho v^2 \qquad (26)$$

the factor K does not remain constant. For the standard N.P.L. pitot, K is designed to be K = 1 except at very low velocities. For upstream and downstream pitots this factor can vary according to design from about 1.05 - 1.3. This factor is a function of Reynolds number and it is necessary to apply an appropriate correction. This was obtained by calibration of the pitot against an N.P.L. pitot on a suitable calibrating rig. There is one advantage in the increased K factor, as this gives a larger differential reading.

Data given by Winternitz[104] shows that the effect of yaw on a tapping is a function of ratio of tapping size d to outside diameter of the tube D. By adopting a fairly large size for d/D very small changes occurred in the readings for yaws of up to $20^{\circ}$.

In order to make the probe sufficiently robust for the length of its 7 ft., its outer diameter was 1.1/16" and the width of the T head was kept as small as possible. The two pressure tapping connections were carried down inside the main water cooled tube which had concentric water passages. A plot of the variation of the factor K with Reynolds Number is given in Fig. 29.

In operation, the most satisfactory way to establish the main flow pattern was to take measurements in the horizontal plane parallel to the axis. Further readings in the vertical plane were used to establish the presence of any appreciable vertical component. In use in heavy fuel oil flames, it was necessary to keep a close watch on the measurements for signs of the tappings being blocked by partially burned fuel.

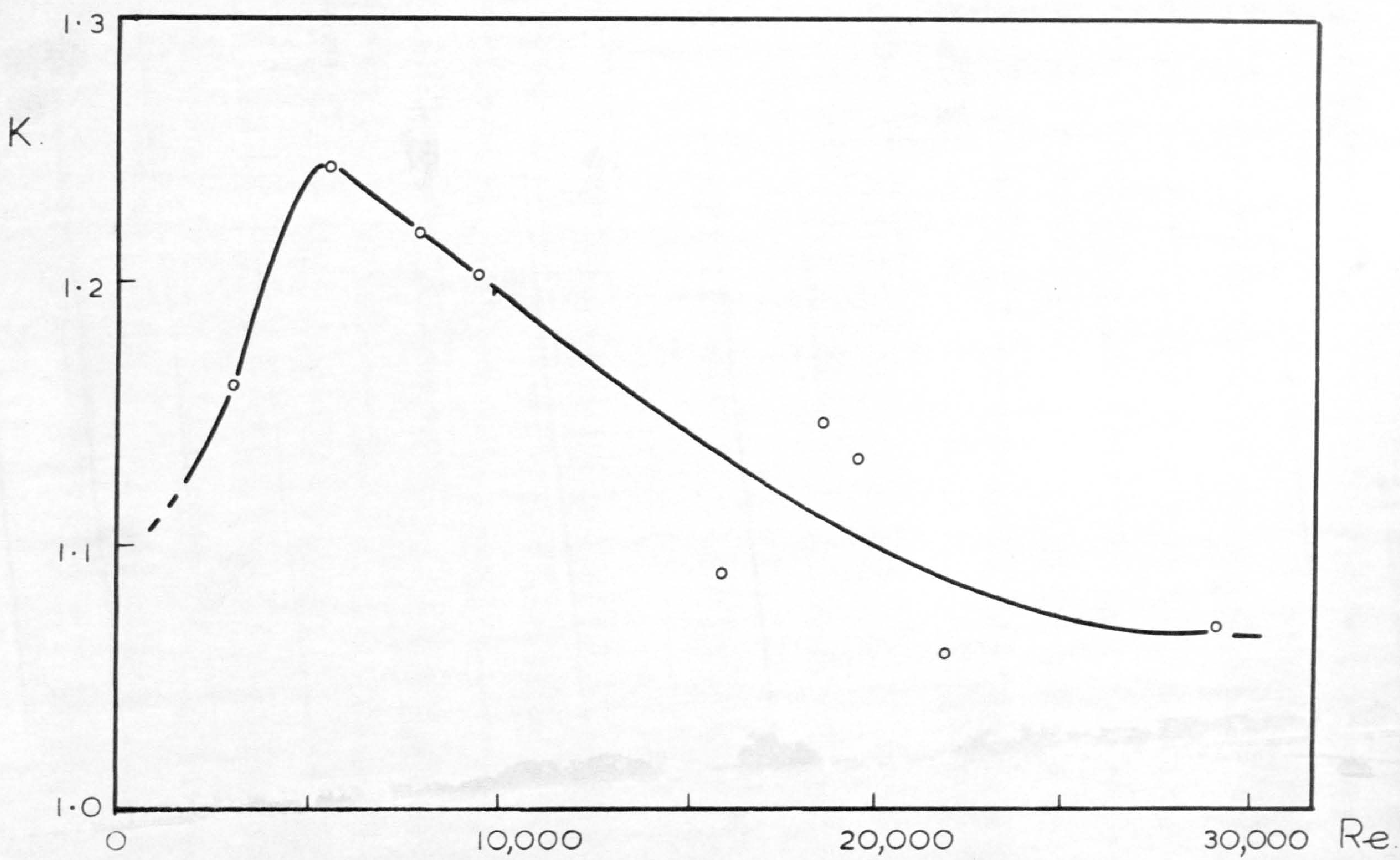



FIG. 29 PLOT OF K-FACTOR AGAINST REYNOLDS NUMBER FOR T-SHAPED 1 1/16" DIAMETER WATER-COOLED PITOT.

FIG. 30.

Water-cooled T-shaped Pitot.

To determine exactly the velocity and direction at any point, a third component in the horizontal plane at right angles to the axis would have been necessary. Measurement of this was not possible with this pitot, but it was considered that except near the front and back walls and inside the recirculation core in the centre of the flame, the component of flow in this third direction was not likely to be very significant.

A somewhat similar pitot with upstream and downstream tappings was used in the laboratory furnace. The opportunity was taken in this connection to design the instrument round special 6 mm. diameter stainless steel tube. This had an internal division to provide the out and return water way for cooling and also included two pressure tapping leads. The use of this small tube gave a final tube diameter of less than a quarter of that necessary for the pitot for full size work. Because of the small dimension of the tube a second water cooled jacket was mounted over it leaving about 6" clear at the end, see Fig.32.

#### 4.5.3.3 The design of pitots for three- directional measurements.

The problem of full three-dimensional flow measurement is a rather complex one and it has only been possible to deal with it in a rather limited way in the present work. One of the chief problems is that of satisfactory design and operation, particularly

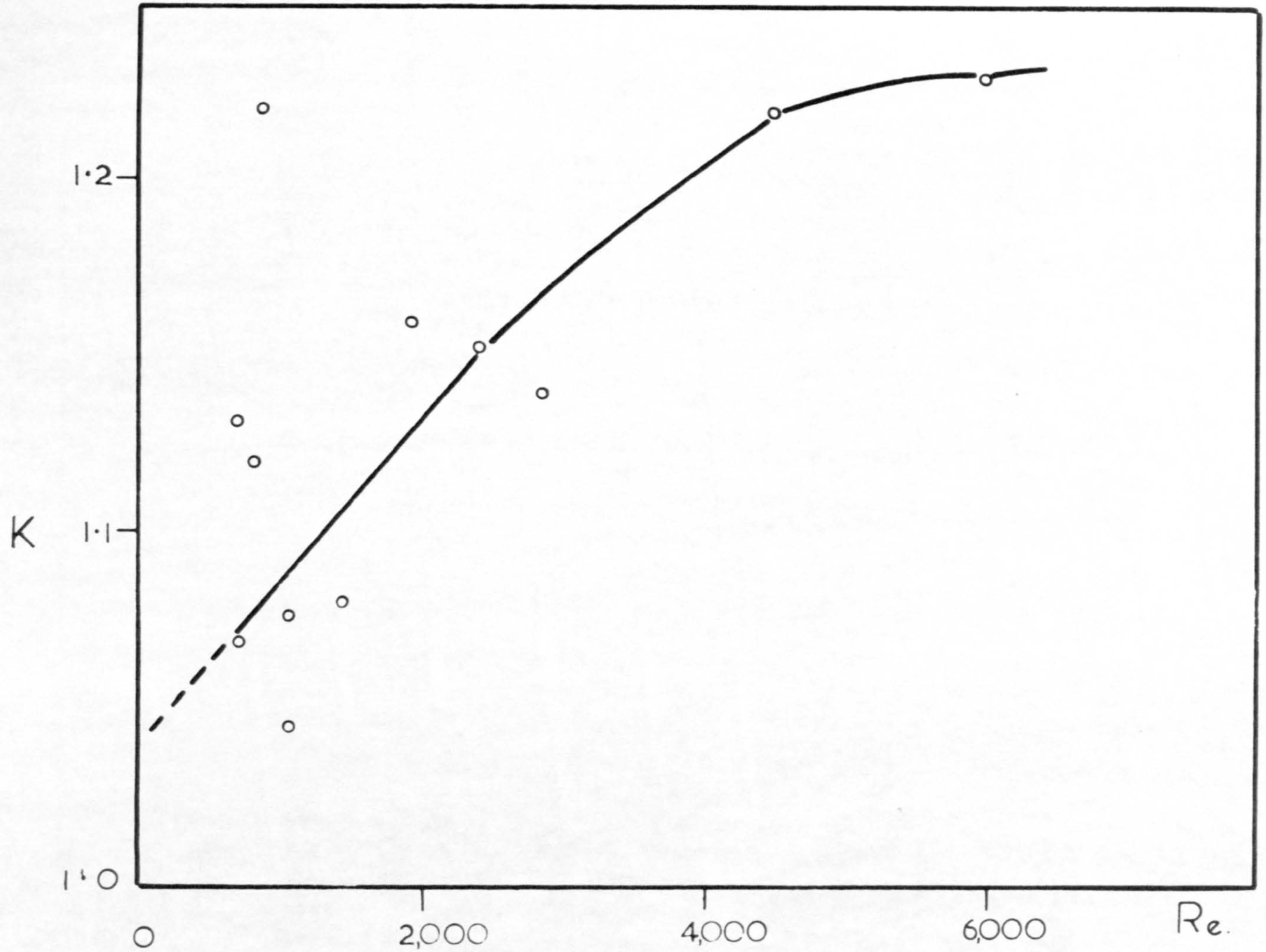



FIG. 31.a. PLOT OF K–FACTOR AGAINST REYNOLDS NUMBER FOR 1/4" DIAMETER WATER–COOLED PITOT.

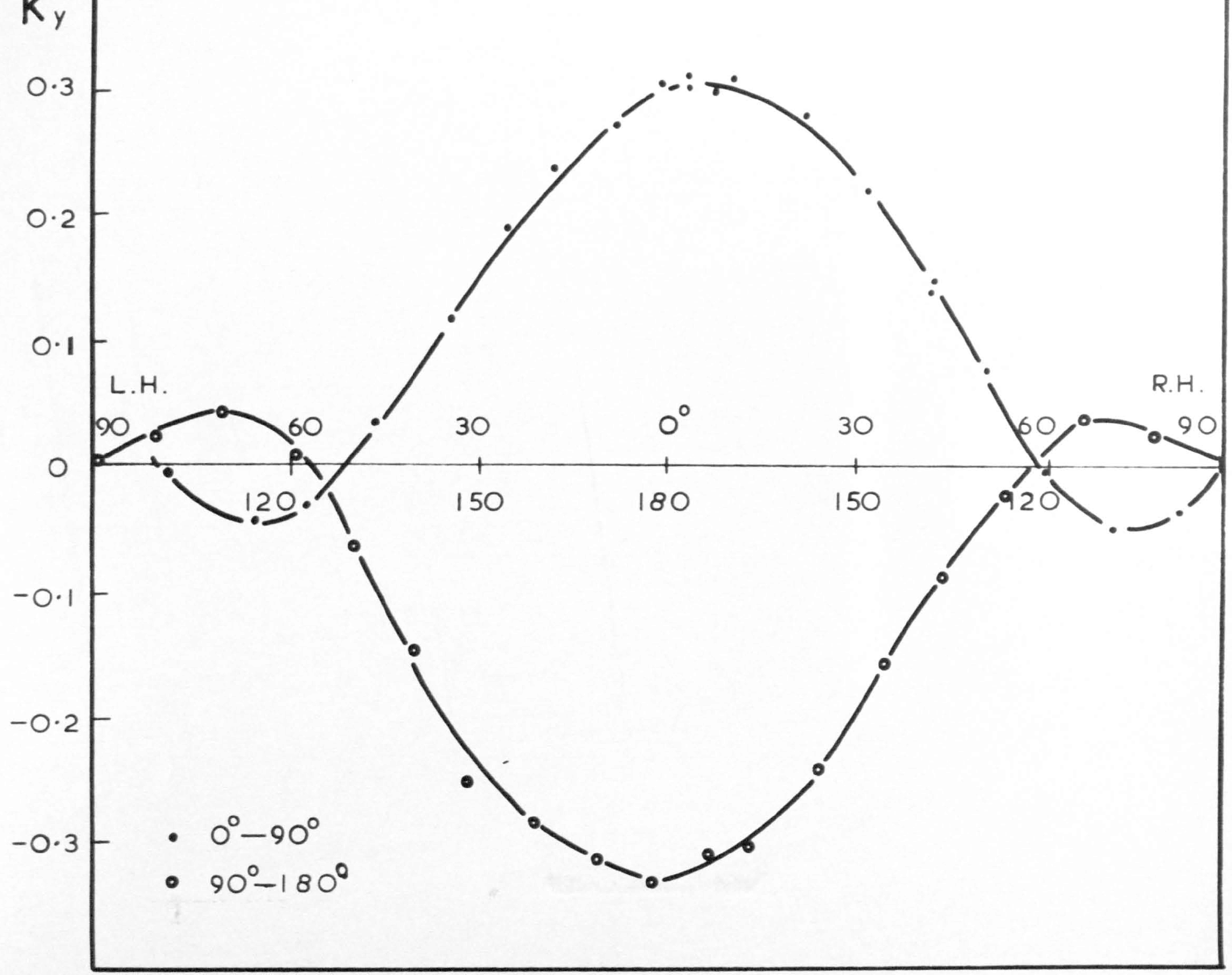
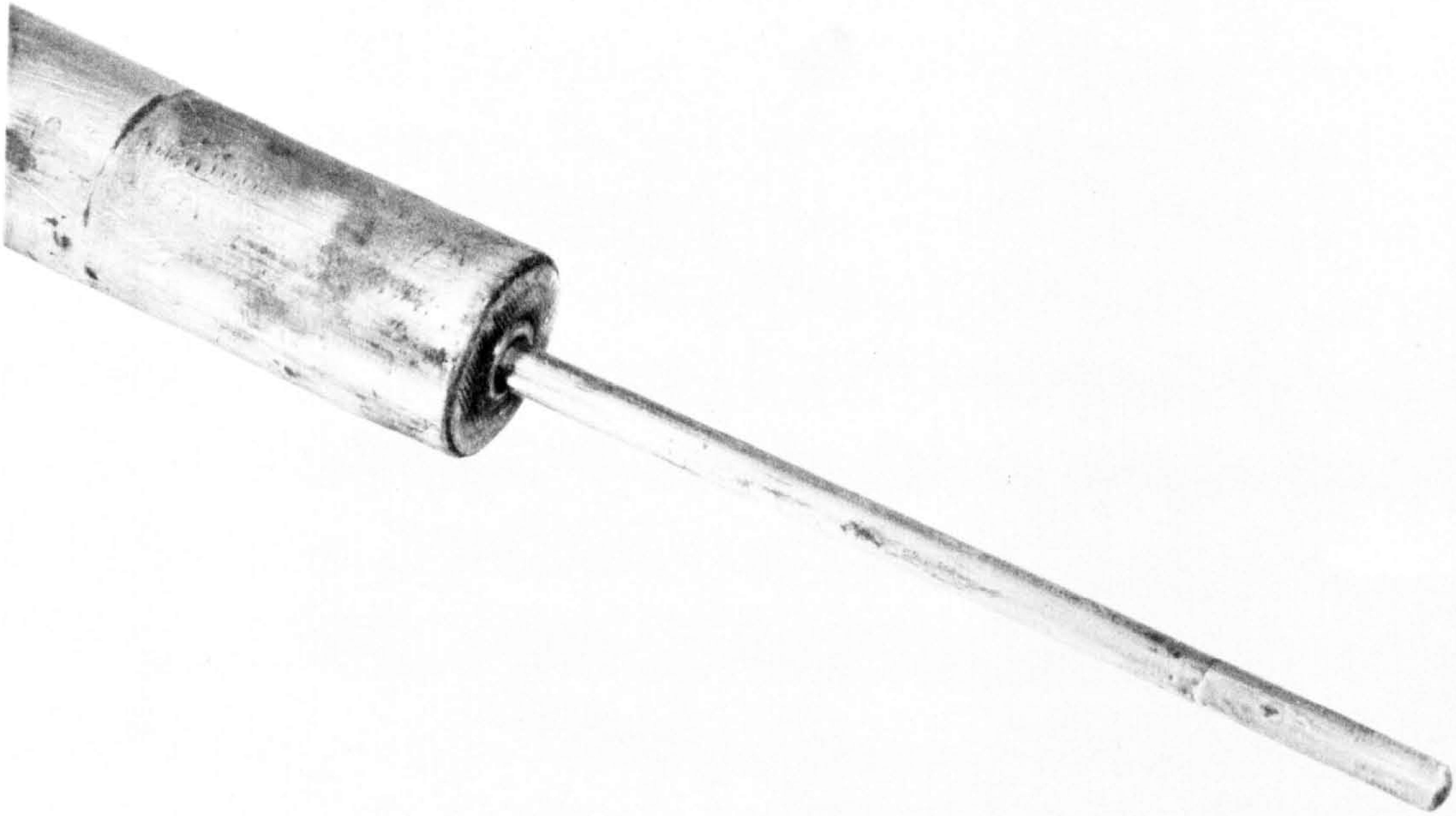



FIG. 31b. PLOT OF $K_y$ FACTOR AGAINST YAW ANGLE FOR 1/4" DIAMETER WATER - COOLED PITOT.

FIG. 32.

Water-cooled $\frac{1}{4}$" Diameter Pitot.



for water cooled instruments for measurements under hot conditions. Three pitots suitable for measurements in three dimensions have been designed and constructed. These were all of the 5-hole type, one was designed for cold measurements, one with a fairly novel conical head for the laboratory furnace work and one for use on the full scale boiler measurements.

A review of the design of these pitots is given in Appendix V11. along with the method of operation and calibration data.

4.5.3.4 Calibration.

Both the single direction and 3-dimensional pitots require to be calibrated. K factors vary substantially between different instruments and sometimes with Reynolds number. In addition the yaw and pitch characteristics need to be known.

The rig used for calibration work consisted of a 3.3/4" diameter tube 3 ft. long connected to the air supply. The output profile from this was arranged to be almost flat by the use of suitable baffles in the tube. In front of the end of the tube was a working table fitted with a 180° scale and interchangeable arms for carrying different sizes of pitot. The probes were clamped at the outer end so that they could be yawed on their own axis, the movement of the mounting arm provided the required value for the pitch angle.

A small pitot constructed to N.P.L. standards was mounted so that it could be located in place of the probe head and this was used to set the calibration rig for the required velocity. A temperature measuring point was provided in the air tube.

Calibration curves for the various probes used are shown in Figs. 29 & 31. and Appendix VII.

In the case of the single direction pitots the factor K is plotted in terms of Reynolds number. Although the variation of K over the range of calibration appears fairly large, it is only equivalent to a difference of ± 4% in the velocity about the mean value of K, in the range up to a Reynolds number of $10^4$. This means that an approximate Reynolds number can be computed from an appropriate "average" value of K in the range of the measurement and the correct value read off from Figs. 29. and 31.

4.5.3.5 Errors due to fluctuating flow.

Some error in readings may be expected due to fluctuating flow. Normal practice was to take a mean value for this reading. Measurement of velocity on the calibration rig allowed a check to be made against a calibrated orifice. This showed that a mean value could be taken with a less than 2% error. A second check was made using an oscillograph with a response of the order of 500 cps to ensure that the ordinary recorder was not making any more substantial variation in reading due to its fairly slow response time.

Ower[105] has shown that in a fluctuating flow the average pressure due to the flow

$$= p_0 + \tfrac{1}{2}\, p\, v_0^{\,2}\,(1 + \alpha^2/2)$$

Thus the percentage error on the average velocity

$$= 100\,[\sqrt{(1 + \alpha^2/2)} - 1]$$

where $\alpha$ = the amplitude of velocity pulsations.

The error in most of the readings reported which had been computed by this method was found to be less than 6 %.

To avoid any errors due to unequal line capacitance effects, the connections and rubber leads between the tappings and the pressure transducer were kept of equal length and size for both connections. As far as possible changes in diameter along the length connections were also avoided for the same reason.

4.5.3.6 Differential pressure measurement.

In view of the range of velocities expected,-that is much less than 150 ft./sec., - a sensitive form of manometer system was necessary. For this purpose it was desirable to have an instrument that would respond to pressure fluctuations as well as mean pressures, one with high sensitivity but with readily variable range and, to avoid as far as possible errors in reading, a device which could be made to give a permanent record of the reading.

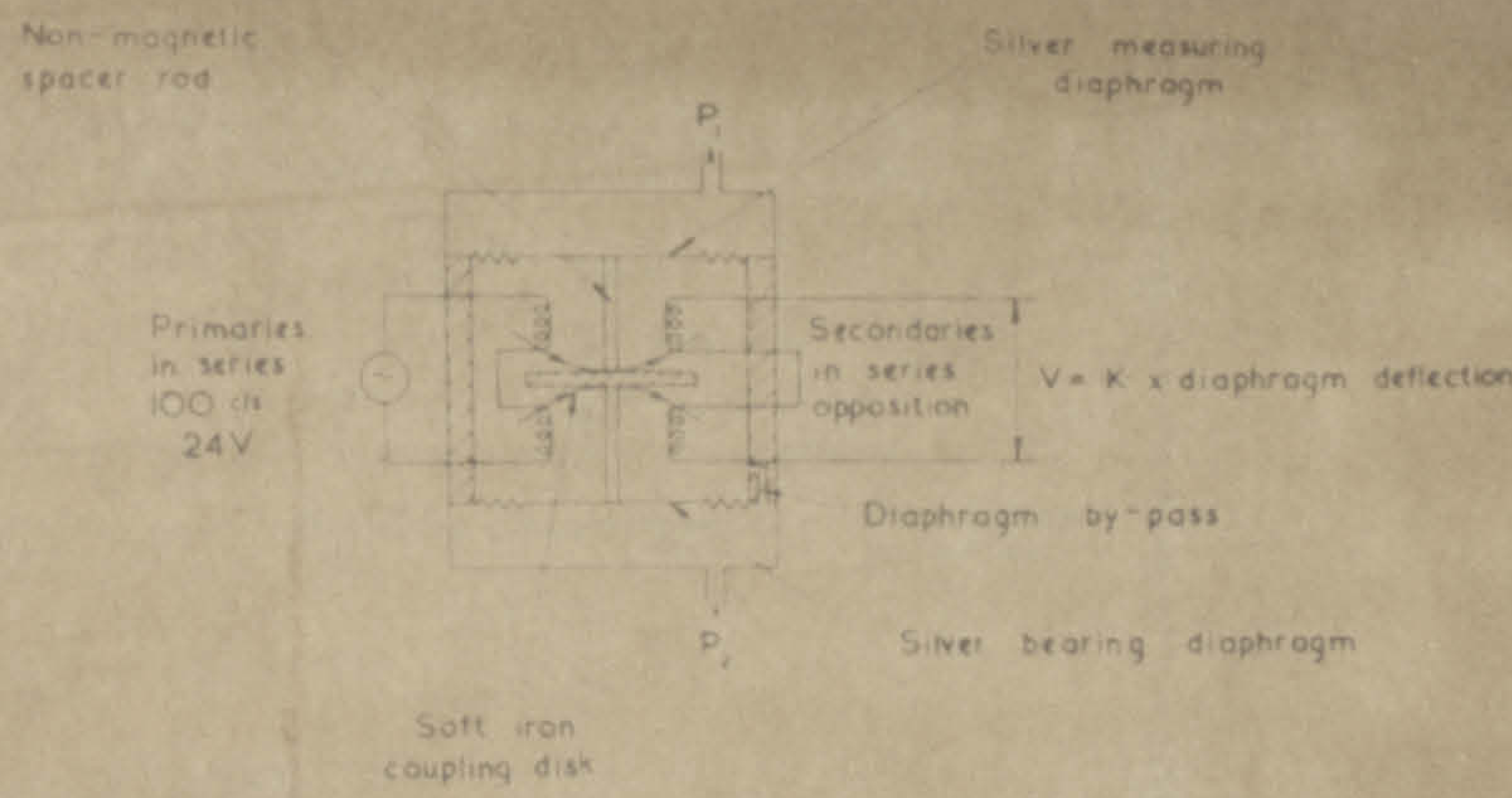
For most pressure differential measurements a mutual inductance pressure sensing device was used and arranged with a suitable circuit to give an output for driving a standard millivolt recorder, (ranges of 0-2, 0-20 and 0-50 Mv. being used as convenient.) The sensing head and its electrical system which was made by the Ateliers de Construction Beaudouin, Paris, is illustrated in Fig. 33-4. Silver diaphragms are used to move a core inside the mutual inductance windings in each cell. The twin cell arrangement compensates for the reaction of the system to movement and vibration, so that moving the capsule does not affect the calibration.

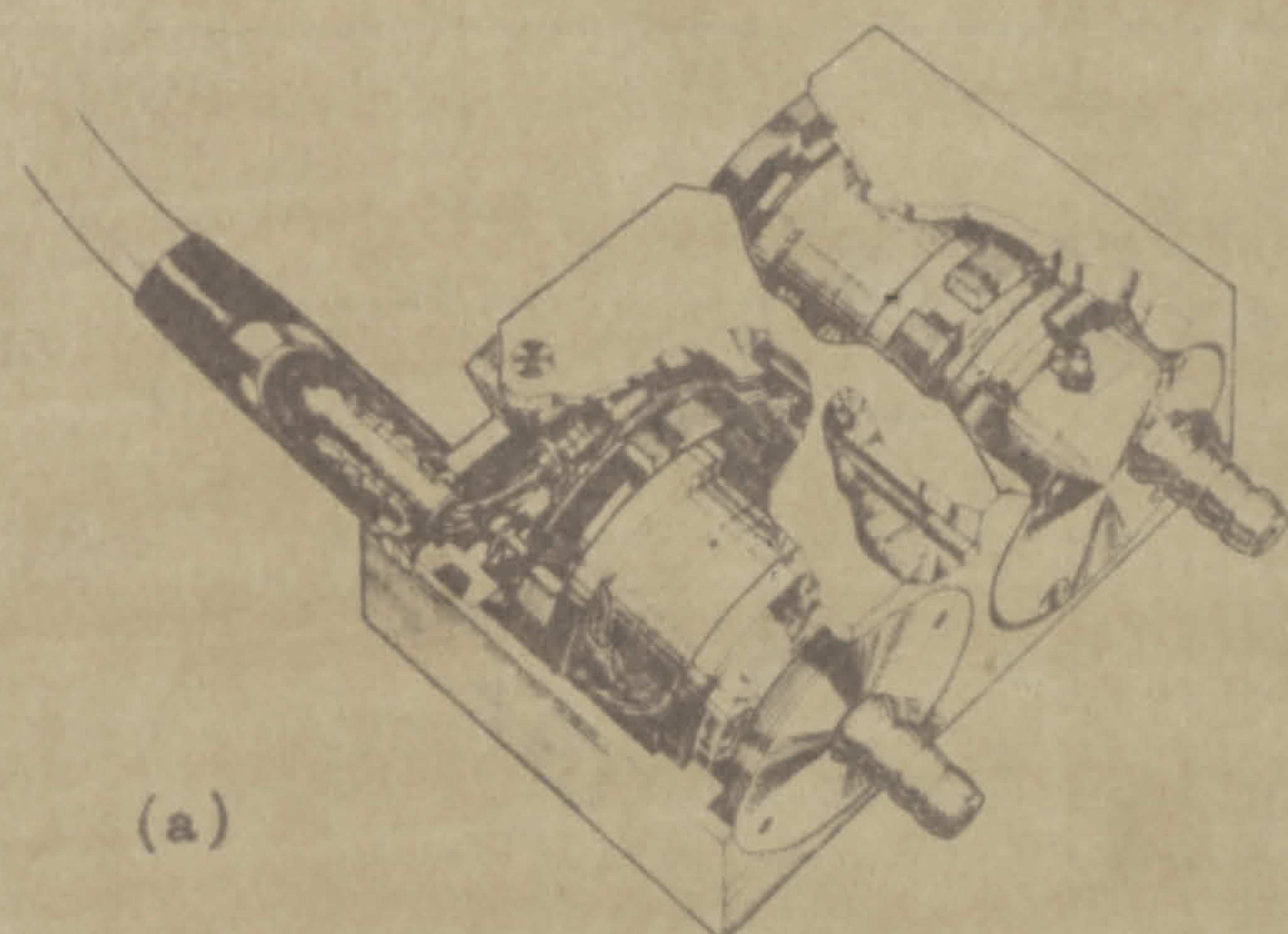
The general arrangement of the circuit is shown in Fig. 35.a. and consists of a 1000 c/s supply at 24 V. rms. Details of the circuit for the generator used are shown are shown in Fig. 35.b. This is claimed[106] to give an output with less than 1% variation with time and up to ± 15% variation in mains input.

The output from the capsule was rectified by two Phillips OA 85 Germanium Rectifiers and fed to a bridge circuit with coarse and fine zero adjustment together with a sensitivity selection switch. The off balance millivolt output from this was then fed straight to a suitable millivolt recorder.

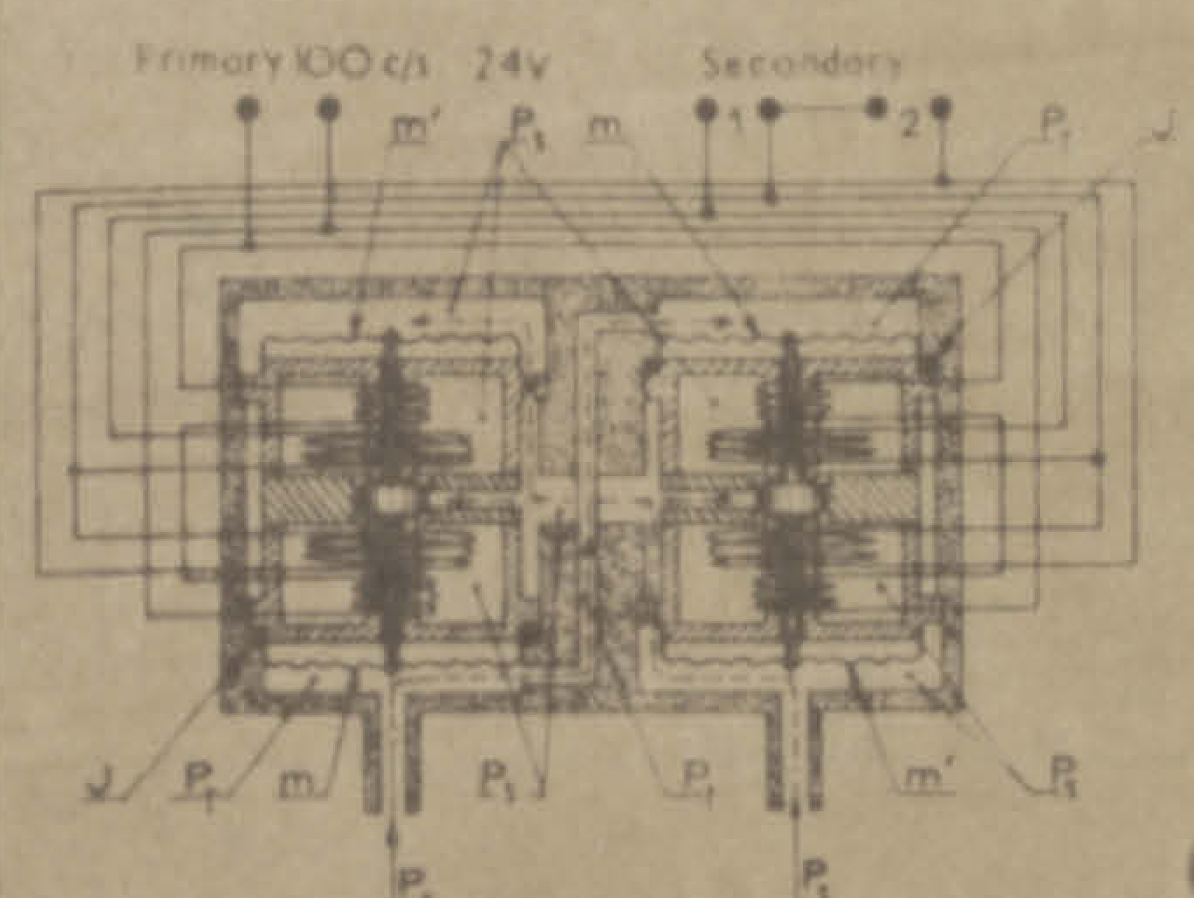
The Beaudouin transducer is of course not an absolute instrument and it was necessary to calibrate the transducer

FIGURE 33



(a)




(b)

$P_s$ = AVERAGE STATIC PRESSURE

$P_t$ = TOTAL PRESSURE

J = GASKET

m = MEASURING DIAPHRAGM

m' = BEARING DIAPHRAGM

FIGURE 34

Fig 33 -Principle Of The Beaudouin Differential Transformer Micromanometer.

Fig 34 -Micromanometer Sensing Head. (a) Detailed Cross-Section. (b) Internal Electrical System.

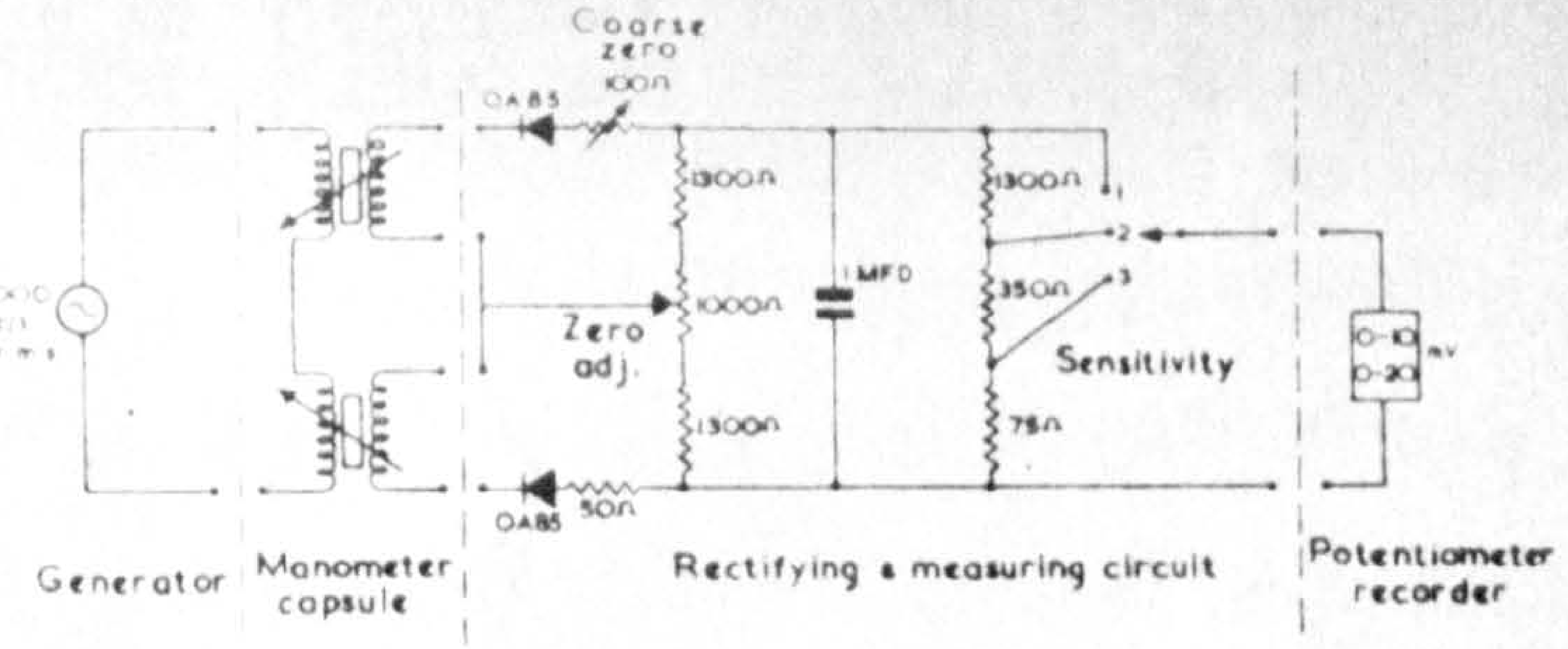



FIGURE 35.a.

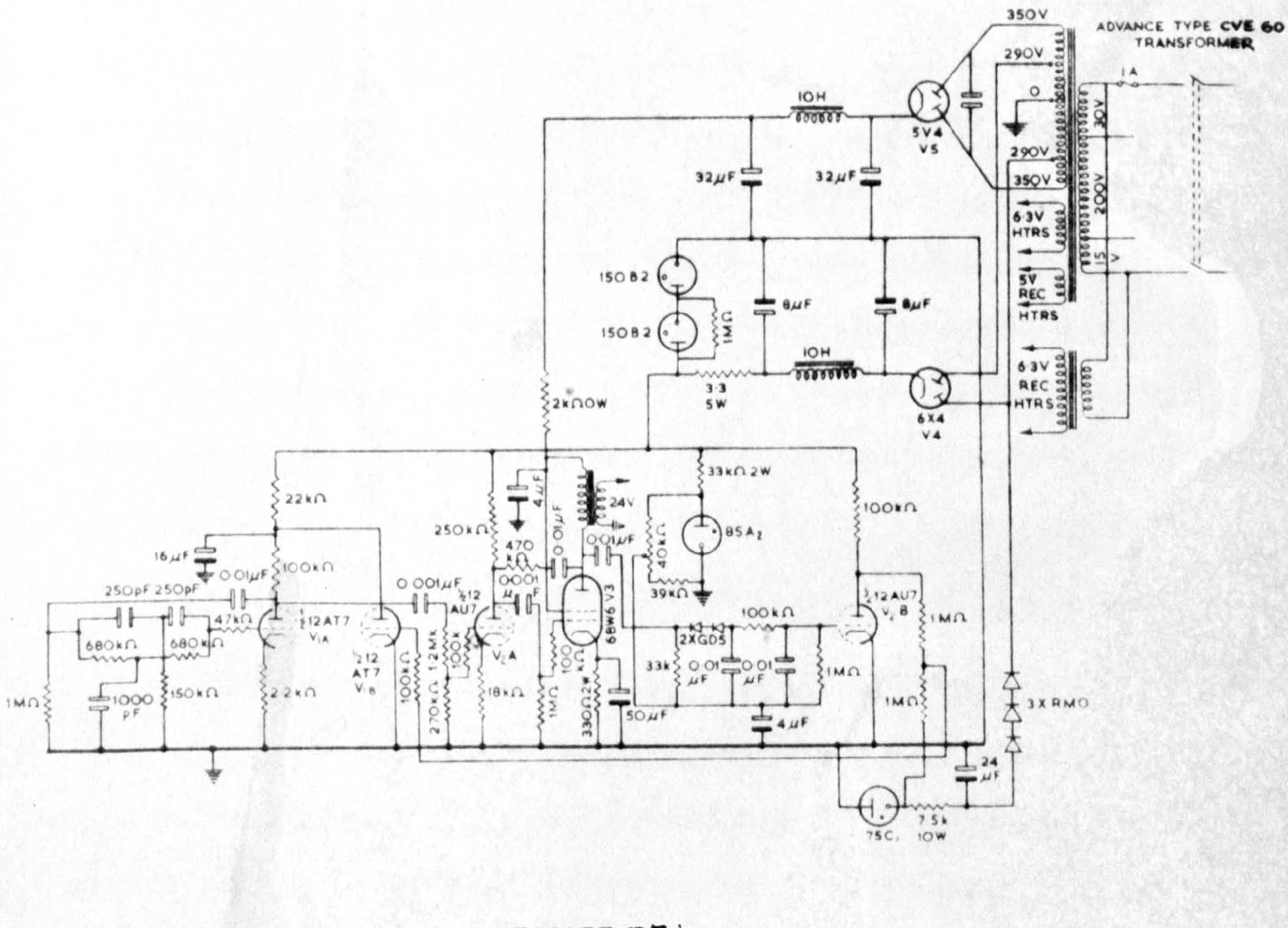



FIGURE 35.b.

Fig35.a.-Circuit Diagram Of Beaudouin Mutual Inductance Pressure Sensing Head.

Fig35.b.-Circuit Diagram Of 1000 c.p.s. Oscillator Unit.

equipment at the beginning and end of a run for each range used. For this the transducer was coupled to an "Oldham" type micro-manometer reading to .001" w.g. The calibration was found in fact to be very stable, it only being necessary to correct for zero drift at suitable intervals between readings.

For certain purposes a capacitance type pressure transducer was used. This was manufactured by Infra-Red Developments Ltd. a circuit diagram being shown in Fig. 36. The disadvantages of this instrument were its relatively slow response time so that it gave only a mean value of the pressure difference, and the need to change the sensing device for difference ranges. It was, however, a very convenient instrument for showing the null point for the yaw measurement with a three-dimensional pitot, and for measuring large differential pressure readings outside the range of the Beaudouin transducer.

### 4.5.4 Measurement of Combustion Oscillations.

Measurement of combustion oscillations was carried out with two very simple pressure detectors originally developed by C.A. Homsy and the author for laboratory scale work. These consisted of a tube from the pressure measuring point connected by a short side tube to a telephone microphone; the main tube terminating in a pressure absorbing chamber packed with glass wool. One of these is illustrated in Fig. 37. The signal

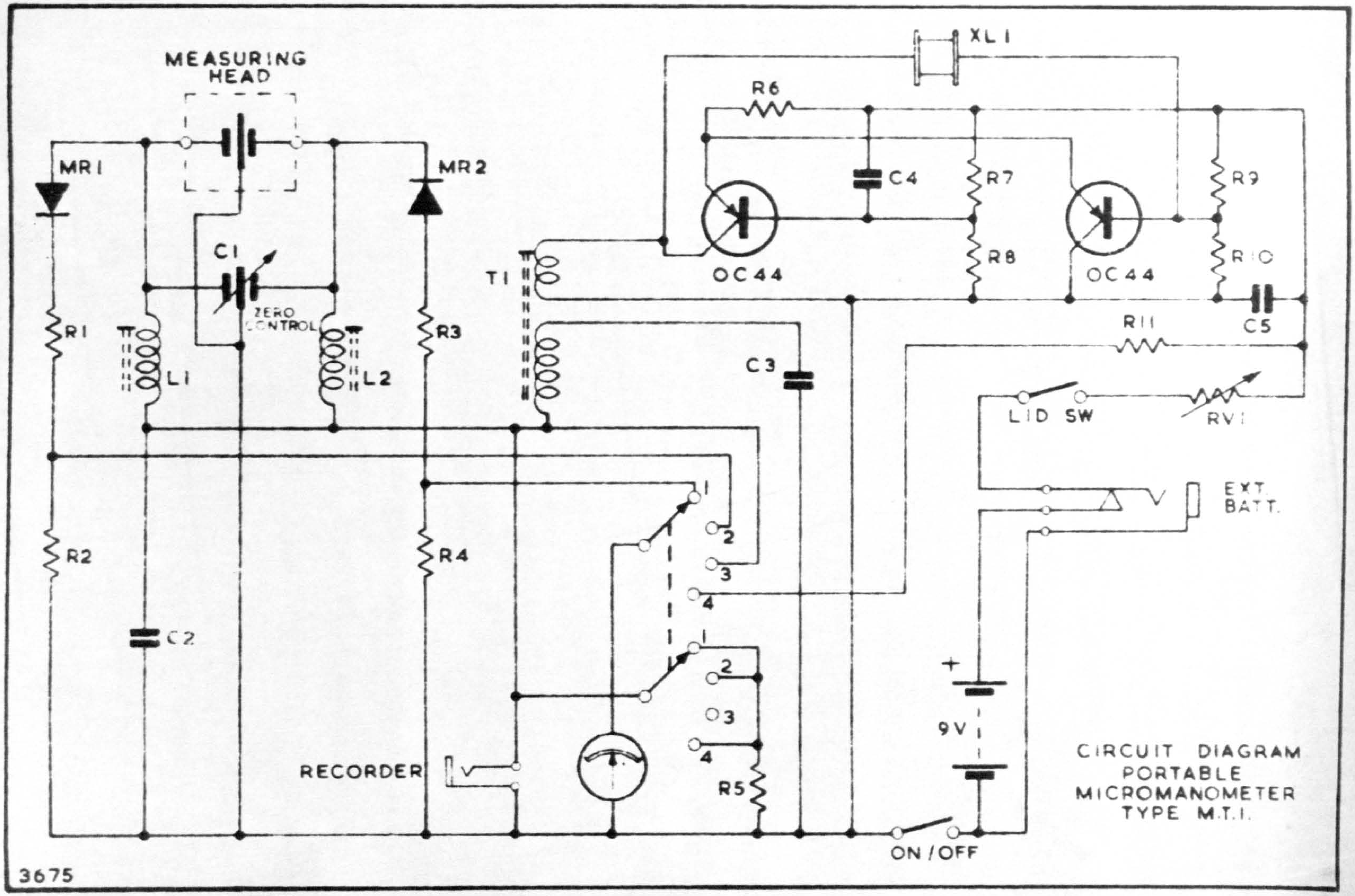



FIG. 36. CIRCUIT DIAGRAM FOR THE CAPACITANCE TYPE PRESSURE TRANSDUCER.

from these was fed through a transformer to a twin beam solatron oscilloscope fitted with a 35 mm. camera. The telephone type detectors proved quite satisfactory for measuring oscillation frequency. The output however, was not sufficiently consistent to provide a satisfactory means of measuring oscillation amplitude, although attempts were made to calibrate them with a small variable speed piston device which provided an oscillating pressure change of 2" w.g. This is illustrated in its most developed form in Fig.38.

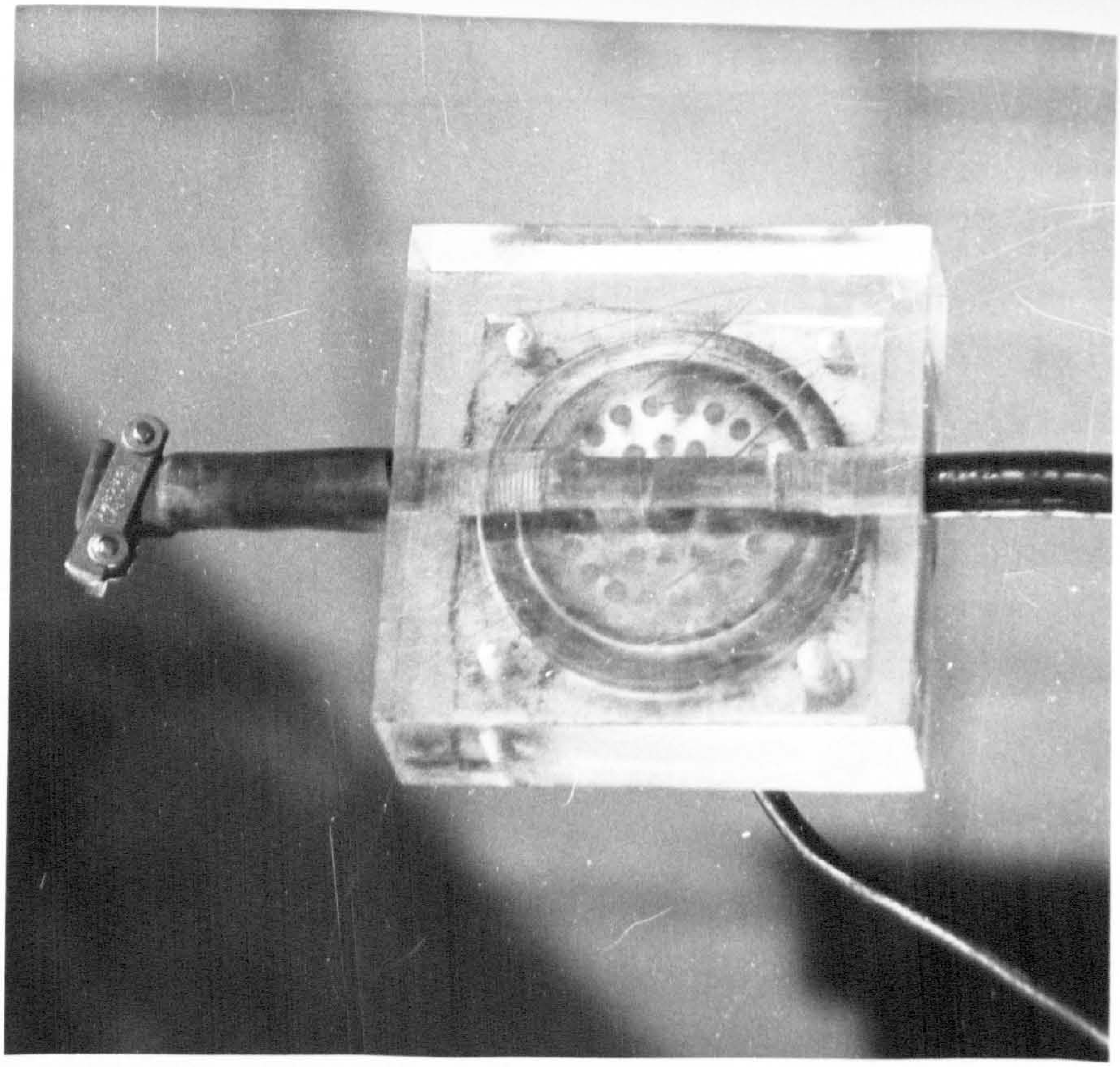
A film record was made of the oscilloscope trace during oscillation. This was analysed subsequently for oscillation frequency against the record of a standard time base signal.

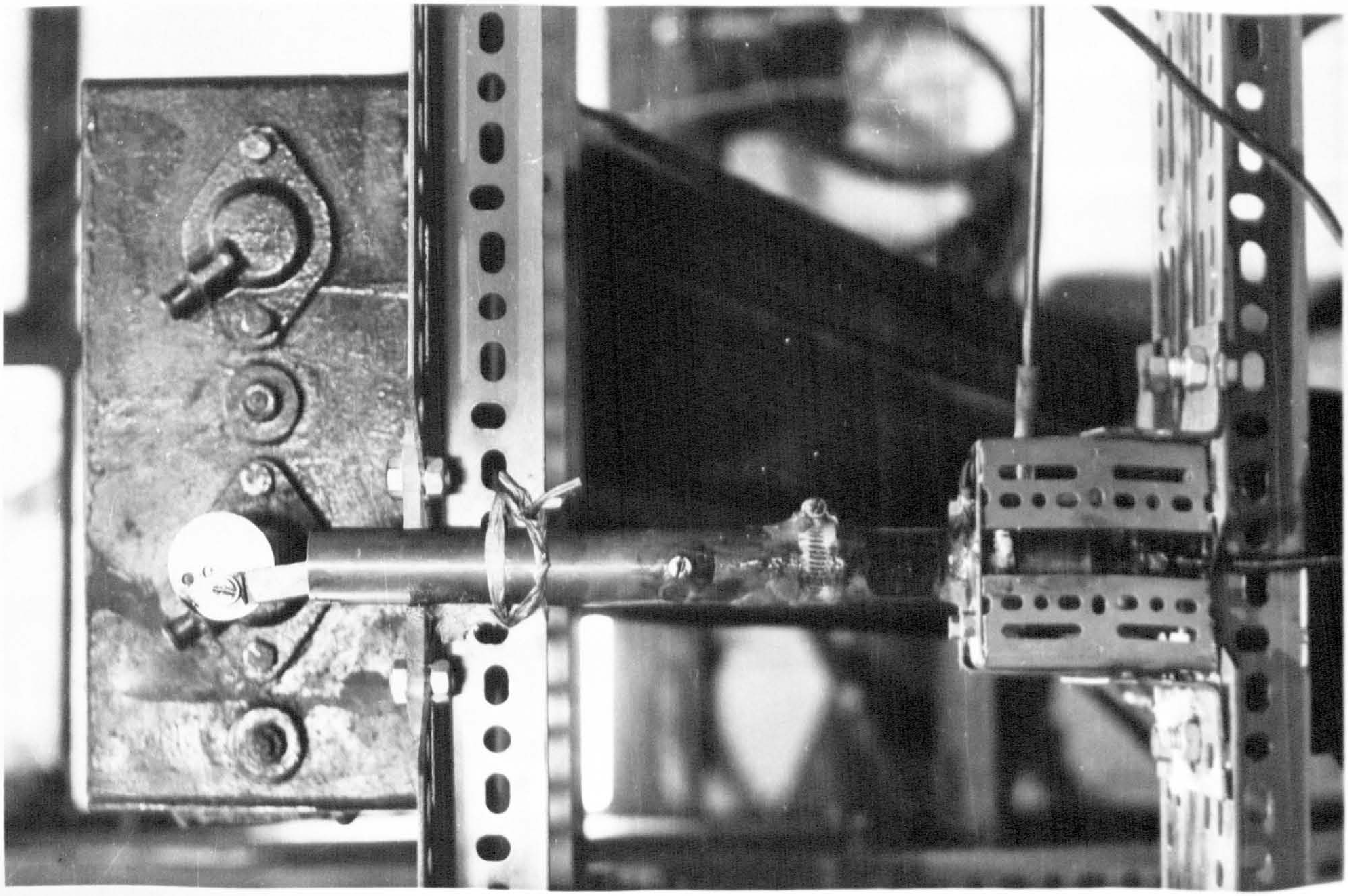
FIG. 37.

Telephone Detector Used for Oscillation Measurements.

FIG. 38.

Variable Speed Piston Device for Amplitude Calibration.





# CHAPTER V.

## Discussion of Experimental Results.

### 5.1 Presentation and Comparison of Results.

#### 5.1.1 Principles Adopted.

Reduction of experimental results to a form where they are strictly comparable with one another, presents a number of problems when three dimensional flow systems are being considered. It is particularly difficult to find satisfactory criteria for comparison on an overall basis. As has been mentioned already, gas turbine work has taken often some specific characteristic such as the blow off limit as a suitable basis for comparison. This method of comparison, however, has no particular relevance to boiler conditions and would be difficult to carry out satisfactorily on the full-size burners. The nearest equivalent would be black smoke limits for different flames. This occurs usually at about the stoichiometric air-fuel ratio. Some preliminary experiments suggested that this was in fact affected by wall temperature requiring very accurate reproduction of full-size conditions. Assessment of the smoke limit is also somewhat subjective. Further, from analagy with data involving soot formation and emissivity measurements, previous experience[107] suggests that such a parameter may be particularly susceptible to scaling effects. As it was desired in this study to obtain more detailed information on the behaviour of the different flames, the following bases of comparison have been used:-

1) Comparison of composition, velocity and temperature in typical horizontal sections as a whole.
2) Comparison of some of the components on the axis and along one or more lines at right angles to the axis.
3) Comparison of bulk flow quantities such as forward and recirculated mass flows.

For simplicity the air-fuel ratio is expressed as the equivalence ratio. This is the actual air-fuel ratio divided by the stoichiometric air-fuel ratio.

For greater ease of comparison the results are presented graphically as iso-concentrationsplots and velocity and temperature profiles. It is convenient to consider linear distance in terms of 'burner diameters' as this is particularly useful in comparison between model and full size. This terminology is used in this discussion and the 'burner diameter' is referred to the quarl diameter which is the end of the diffuser section of the burner at the combustion chamber wall. This is the diameter which affects the downstream flow pattern. (In the full size burner this is 16 inches and for the model 4.8 inches).

### 5.1.2 Plots given from the results.

The gas analyses obtained are presented as:-

1) Oxygen, Carbon Dioxide and Carbon Monoxide concentration plots. These are considered as demonstrating the distribution of oxygen combustion products and gaseous fuel component respectively. Concentrations of $H_2$, $CH_4$, and $C_2H_4$ were almost always very low and no significant information is apparent from comparison of individual concentrations of these components, which is not already apparent from the Carbon Monoxide plot.

2) Equivalence ratio and combustion efficiency plots which were calculated as indicated in Appendix V. . These demonstrate the relative proportions of the original fuel and air components reaching any point and the proportion of fuel burnt at the same point.

These data are presented as isoconcentration plots in Figures 40-43 for full scale work, and Figures 49-50 for the model results.

Solid lines on the plots represent contours which join points at two or more slots. The dashed lines are ones through only one point, or where some interpolation has been necessary. The use of isoconcentration plots rather than a series of profiles was considered preferable, despite some disadvantages mentioned later. These present a clearer picture and make possible a comparison between different plots where the slots are not in the same position.

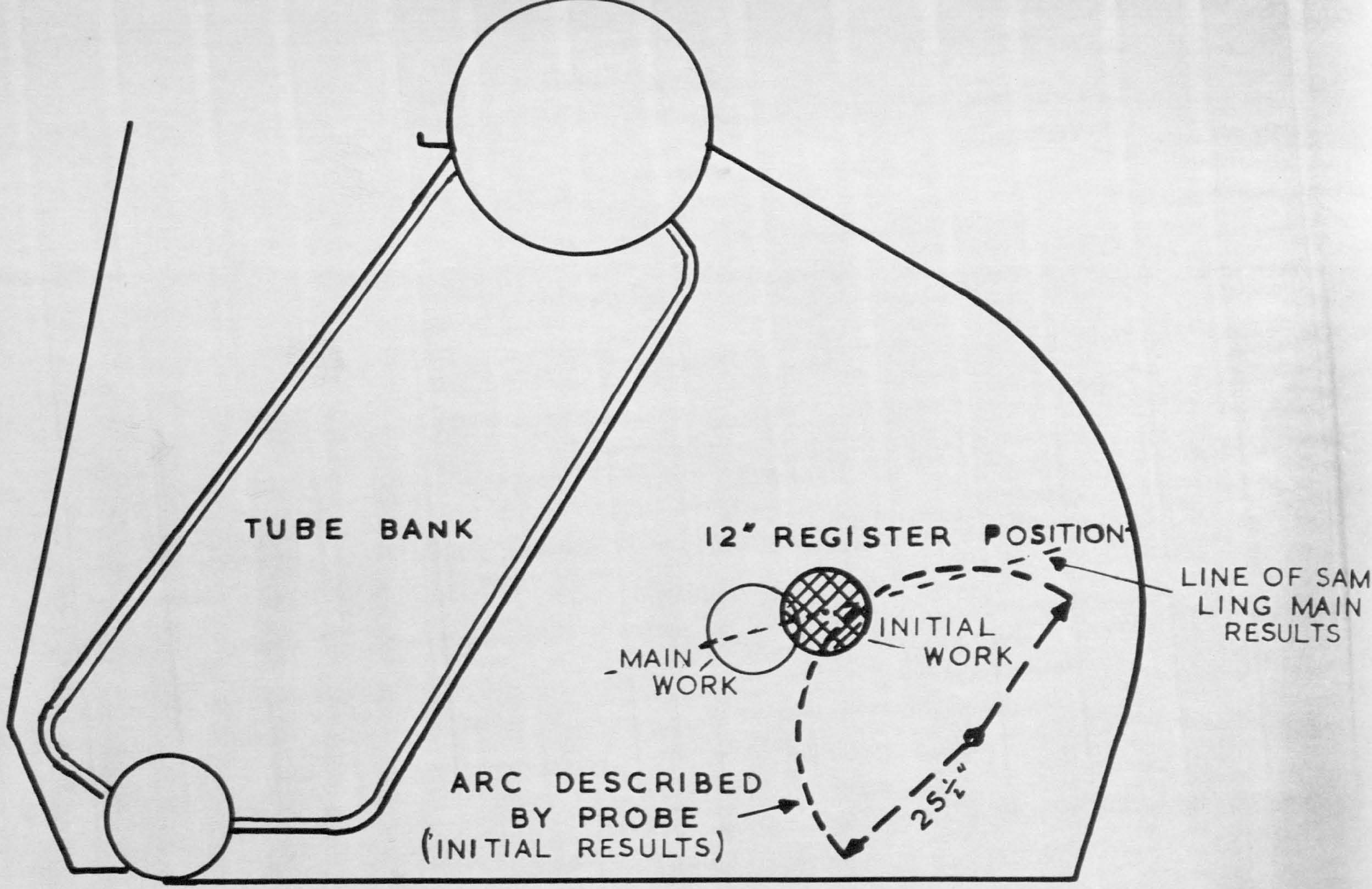
Velocity measurements obtained are presented as plots of velocity profile at each measuring slot with the jet boundary marked (as indicated by zero forward velocity). Figures 44-46. Temperature measurements are similarly represented by plots of temperature and profile at each slot. Figures 47-49.

Most of the data are presented as horizontal sections across one half of the combustion chamber. Some initial measurements were made for the 60° Duplex full-power case (see Table III) on an arc from the sidewall to the furnace floor, using an L shaped gas sampling probe inserted through the front wall. (See Figure 39). These results are plotted separately as the burner position was somewhat different from the subsequent runs. These data are useful because they show that the flame was reasonably symmetrical between the vertical and horizontal directions. They also draw attention to the one disadvantage of isoconcentration plots. This is the problem of detailed patterns between measuring points being overlooked. This is apparent in comparing some features of these plots with the corresponding ones in the subsequent rather more extensive investigation. In the latter instance the slots were further apart than the probe positions in the initial work. Allowing for this, the fact that the burner was in a different position on the front wall and that in the early work the probe did not go through the centre of the flame the results compare satisfactorily.

## TABLE III.

## BURNER VARIABLES STUDIED IN FULL-SIZE FLAMES AND MEAN VALUES FOR OPERATING CONDITIONS.

| CASE | 1 | 2 | 3 |
|---|---|---|---|
| REGISTER | 12" S.F.W.R. REGISTER | | |
| FUEL RATE lb./hr. | F.P. 2175 | F.P. 2175 | $\frac{1}{4}$ POWER 570 |
| ATOMIZER PRESSURE P.S.I. | LUCAS DUPLEX 550 | LUCAS SPILL 780/700 | LUCAS DUPLEX 500 |
| NOMINAL SPRAY ANGLE | $60^\circ$ | $80^\circ$ | $90^\circ$ |
| REGISTER DRAUGHT LOSS "W.G. | 5.2 | 6.0 | 1.4 |
| EQUIVALENCE RATIO | 1.28 | 1.38 | 1.98 |
| RESULTS PLOTTED IN FIGS. | 40, 41, 44, 47 | 42, 45 | 43, 46, 48 |




Scale $\frac{1}{2}'' = 1'-0''$

FIG. 39. ARC IN WHICH GAS SAMPLES WERE TAKEN.

5.2 Experimental work using full-scale burner.

The choice of burner and the reasons for choosing particular variables has been discussed in section 3.4.2. The variables selected for study in these trials were fuel flow rate and spray angle, and the conditions investigated are tabulated in Table III.

The initial experimental results given in Fig.40.a-d. were in fact a rather limited version of the main results given in Fig.41.a-e. for the 60° Duplex atomizer flame.

In the initial experimental work successful measurements were limited to a gas composition survey of the earlier part of the flame obtained from a number of sampling positions between the floor, the centre line of the flame, and the side wall of the boiler, using an L - shaped gas sampling probe.

In the main experimental runs additional access was available and measurements were possible at stations covering the full length of the combustion chamber. In this trial, in addition to gas composition, temperature and velocity were measured. The equipment used consisted of water cooled gas sampling probes, suction pyrometers and the T-head velocity probe described in Section 4.5.3.2.

The use of a spill atomizer was not intended as a comparison of methods of atomization, but as a convenient method of obtaining an $80^\circ$ spray of the right throughput, although this introduced some uncertainty on the question of droplet sizes (see Appendix III).

The $90^\circ$ spray angle quoted for the $\frac{1}{4}$ power case was the normal spray angle for this type of Duplex atomizer at this degree of turn down.

## 5.3 Experimental results obtained from full size burner work.

### 5.3.1 Gas composition.

#### 5.3.1.1 Carbon Monoxide.

The composition plots for the three full size burner cases investigated are given in Figs. 40 - 43. Those for Carbon Monoxide concentration are given in Figs. 40a, 41a, 42a, 43a. The results showed two noticeable features in CO distribution.

Firstly, there was a high unburnt fuel region, with as much as 8% CO in the core of the flame in the two full-power cases. The actual value depended on the air-fuel ratio for the flame, but the shape altered quite markedly with spray angle. This extended to slot 3 (about $2\frac{3}{4}$ burner diameters) for the $60^\circ$ spray and only to slot 2 (about 2 burner diameters) for the $80^\circ$ one, although it was somewhat wider in the second case. For the low-power condition with its $90^\circ$ spray this central core hardly appeared except very slightly in the region of slot 2. This high CO figure will be seen by reference to the velocity plots in Figs. 44-45 to extend beyond the short recirculation zone on the axis of the flame which extends about two burner diameters downstream. The tail of this CO region extended the whole length of the combustion chamber in both full-power cases. Presumably once clear of the recirculation zone, oxygen transport to this region is a turbulent mixing process across the jet similar in form to a simple turbulent diffusion flame.

In the second place an additional detached zone of high CO occurred. In the two full-power cases this region was clear of the main forward flow altogether and was situated near the sidewall in the region of slot 2. about two burner diameters downstream.

For the low-power flame, a similar region occurred in the region of slot 3, but rather nearer the centre of the combustion chamber on the fringe of the jet registering 1.9% CO, for the highest value.

Reference to the oxygen concentration plots in Figs. 40b, 41b, 42b, 43b, shows an apparent anomaly. This was the apparent occurrance of substantial CO figures in regions in which there is an appreciable $O_2$ concentration. This occurred at several places on the walls, notably at slot 2 for the $60^{\circ}$ (Duplex F.P.) case, it also occurred in the $\frac{1}{4}$ power case in the region of highest CO concentration. There are two possible explanations for this, firstly, because of the time mean nature of the sampling process and secondly due to chilling outside the flame. The first, which is the more likely explanation, could occur if the concentration in a particular zone was subject to fairly rapid fluctuations with a resulting mean in which both $O_2$ and CO occur in fairly high figures. The second is unlikely, except near the walls, as gas temperatures in the main body of the combustion chamber are about $1000^{\circ}$ C even in the low-power case.

### 5.3.1.2 Oxygen.

The oxygen results are plotted in Figs. 40b, 41b, 42b, 43b. As might be expected $O_2$ concentrations were generally high where $CO_2$ and CO concentrations were low. This was mainly along the front wall and along the side wall, except in the region of slot 2. In the full power cases, the region of low concentration spreads a good deal more widely than the jet boundary, indicating that most of the oxygen found its way into the combustion region at a fairly early point. One consequence is that the recirculation zone in the region of slot 2 was included also in this low $O_2$ part of the flame. Further downstream higher oxygen figures were recorded, with a distinct bulge in towards the centre. In the low power case, the pattern was somewhat different. The lowest oxygen figures were no longer along the centre line, but in the region of slot 2. just inside the jet boundary and roughly in the region of highest velocity gradient. It will be noted that all oxygen figures recorded are high, above 6%. This makes some of the CO figures rather surprising, but apart from the reasons already mentioned, this seems to indicate that in this case in particular some of the fuel persists as a droplet for a long way into the furnace, hence the arrival of high CO figures well down stream.

5.3.1.3 Carbon Dioxide.

In general, $CO_2$ concentrations were high where the $O_2$ and CO concentrations were low. The high $CO_2$ figures indicate regions where combustion is largely complete and where recirculation products are distributed. These results are plotted in Figs. 40c, 41c, 42c, 43c.

One feature which required explanation, was the apparent decrease in $CO_2$ figures and corresponding increase in oxygen figures near the walls in some places. The characteristics of the air register and the velocity measurements did not indicate any forward flow along the wall as would be produced by a 'wall' type jet (see Section 2.7.3). It seems likely that this variation in figures near the wall was due to local leakage round the gas sampling probe, as the outside of the boiler was slightly pressurized to provide air for the burner.

It was difficult to get a perfect seal, although special shaped brick plugs were used to fit round the probe in the sampling port. Leakage from this source would have occurred only at the slot where the probe was inserted and is likely to have had very little effect on the overall air-fuel ratio.

## 5.3.2 Equivalence Ratio and Combustion Efficiency.

### 5.3.2.1 Equivalence Ratio.

Frequently, equivalence ratio plots are used to determine the bounds of a flame on the assumption that an equivalence ratio of one will indicate the flame boundary. There are difficulties about such an assumption which will be discussed later, but equivalence ratio distributions are useful in determining where the fuel and air go which are being supplied to the system and in providing a means of comparing different flames.

The comparison of the two full power cases (Figs. 40d, 41d, 42d.) shows some quite marked differences. In the case of the Duplex atomizer the 'flame volume' was quite small and compact and high equivalence ratios occurred just to the outside of this. In the spill case, despite a higher initial equivalence ratio the 'flame volume' is much larger and tends to spread outside the main jet boundaries (see Fig. 42.d.), and persists all the way to the back wall.

In the low power case (Fig.43d.) no figures as low as 1.0 were recorded although the general pattern was similar to the others.

The absence of a 'flame boundary' can be accounted for as follows. Firstly, unlike the two full power cases, high $O_2$ figures occur inside the flame, and since the flame is short and really a thin hollow cone, the fuel rich zone will have been very limited and thus not appeared in the measurements. Secondly, with the high overall excess air figure which occurred in this instance, the remarks previously made may apply about the effect of time mean sampling giving an 'average' gas sample, which may differ from the instantaneous value. This averaging effect applies also to the determination of flame boundary and is the reason why it is difficult to use the equivalence ratio of one as an accurate measure of flame limits.

Plots of the equivalence ratio along the axis are made in Fig. 55. This showed, rather surprisingly, that the equivalence ratio on the axis was rather more constant for the 80° full-power case than for the 60° full-power case. In the low power case, the equivalence ratios were as might be expected, a good deal higher.

5.3.2.2 Combustion Efficiency.

The combustion efficiency provides an index of the relative amount of fuel actually burnt at any given point.

These calculated figures reflect, of course, similar trends to theCarbon Monoxide plots, with low values in the cone of the flame and in pockets outside the main jet. (See Figs. 41e, 42e, and 43e.) For the full-power cases, the proportion of fuel burnt had a minimum in these zones of 0.5 - 0.6 and 0.85 in the centre and outside the jet respectively. Outside these areas the figure was usually about 0.95 or better. In the low power case, the principal zone of unburnt fuel was on the fringe of the jet with a minimum of about 0.85. The variation of combustion efficiency along the axis is plotted in Fig.56.

### 5.3.3 Velocity and temperature measurements.

#### 5.3.3.1 Velocity Measurements.

The velocity profiles associated with these flames are given in Figs. 43-45. Due to alterations to the boiler after the gas composition measurements were made, some of the slot positions are slightly different in the case of the 60° full power and low power flames. The measurements given are for the component of velocity parallel to the burner axis, the line marking the traversing position for each slot being taken as the zero coordinate. The approximate jet boundary is also shown.

The main features of the profiles were that the early part consists of an annular jet with a central recirculation zone. This annulus gradually coalesced until, by the region of slot 4, about 5 burner diameters downstream} the profile was fairly flat. The central circulation zone extended about $1\frac{1}{2}$ diameters downstream and this was followed by a region with a W-shaped profile. Although the fringe of the jet spread quite considerably, the radius of the maximum velocity peak remained pretty well the same as the outlet diameter of the burner quarl, until this peak merged into a single flat profile. Particular difficulties were experienced at certain points, particularly at slot 2, where blocking of the pitot with a tarry residue from the fuel was a problem. This prevented measurements being made in certain sections of the jet.

### 5.3.3.2 Temperature Measurements.

Temperature traverses were necessary mainly to determine the gas densities across the combustion chamber. These are shown in Figs. 47-48. These show that for both full power and low power, the temperature has a fairly constant value outside the flame of about 1200 and 900° C respectively. The temperature rises somewhat inside the main forward flow region to maxima of 1600 and 1400° C. There is also a sharp drop of temperature at slot 1. on the line of the annular jet from the burner, showing that at that point, recirculated gases have not mixed completely with the incoming air stream.

FIGS. 40-43.

Isoconcentration Plots for Full Scale Work.

FIGS. 44-46.

Velocity Profiles from Full Scale Work.

FIGS. 47-48.

Temperature Profiles from Full Scale Work.

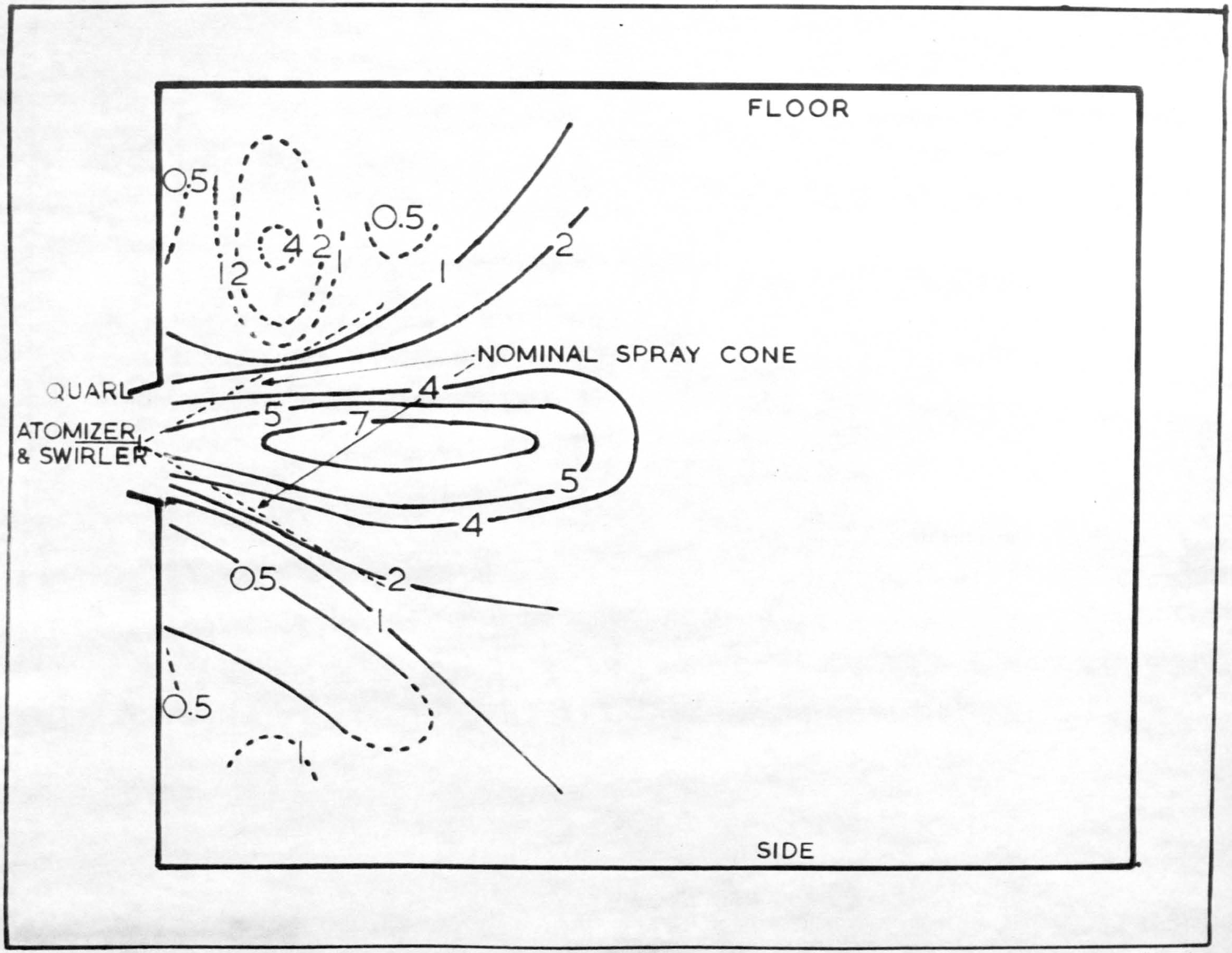



FIG. 40 a % CARBON MONOXIDE DISTRIBUTION - DUPLEX F.P. 60°
(INITIAL RESULTS - CURVED SECTION FLOOR TO SIDE)

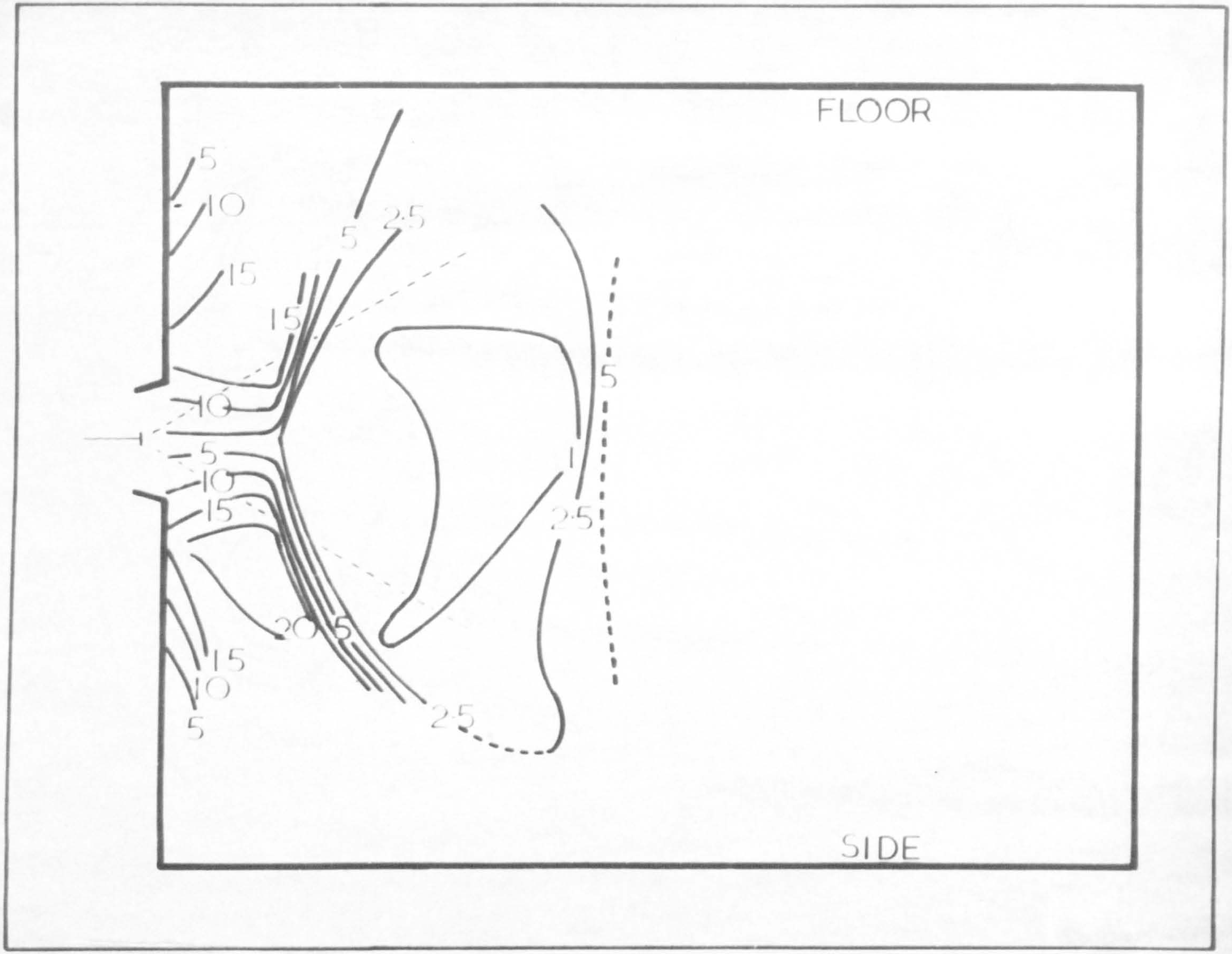



FIG 40b % OXYGEN DISTRIBUTION — DUPLEX F.P. 60°
(INITIAL RESULTS)

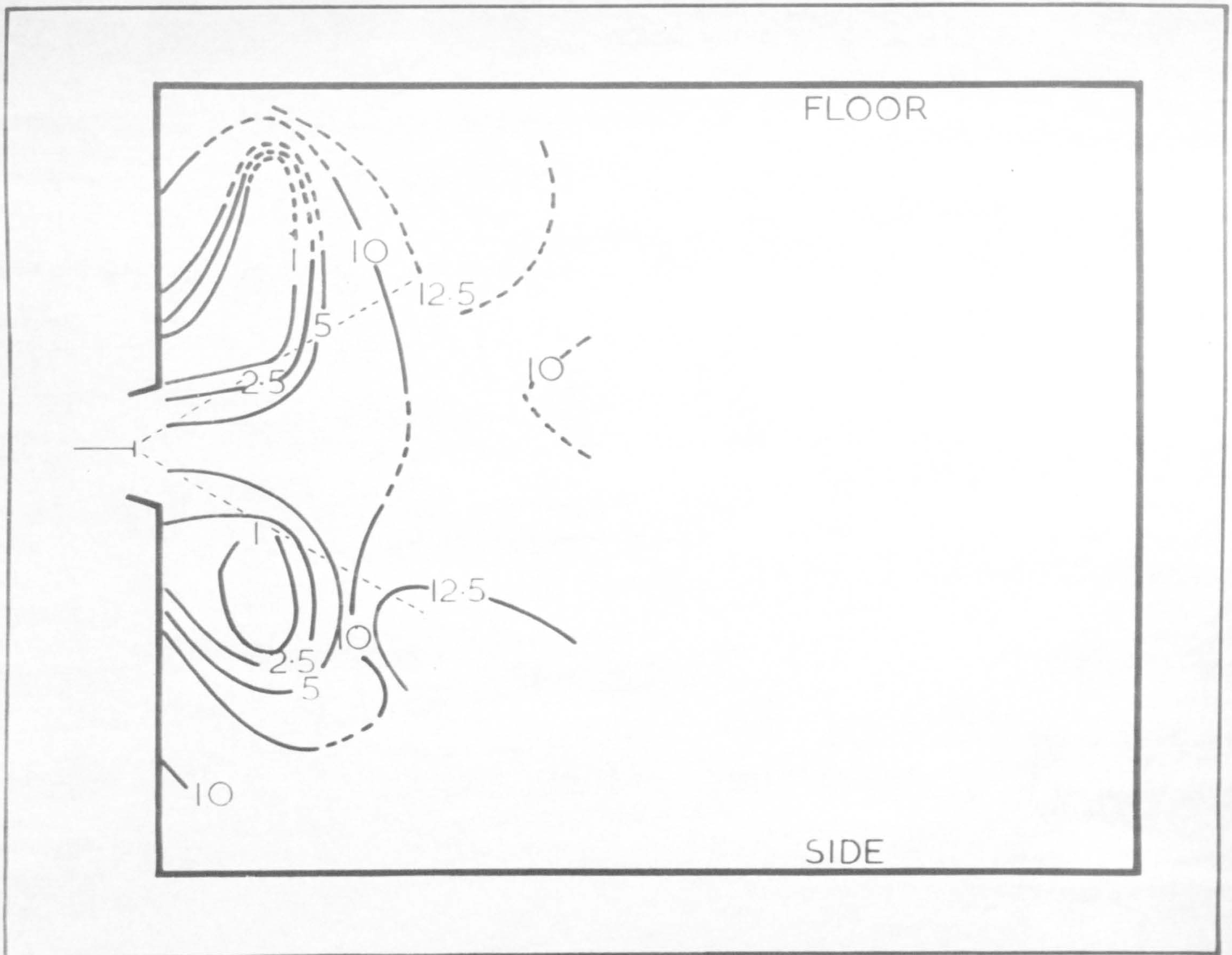



FIG. 40c. % CARBON DIOXIDE DISTRIBUTION - DUPLEX F.P. 60°.
(INITIAL RESULTS)

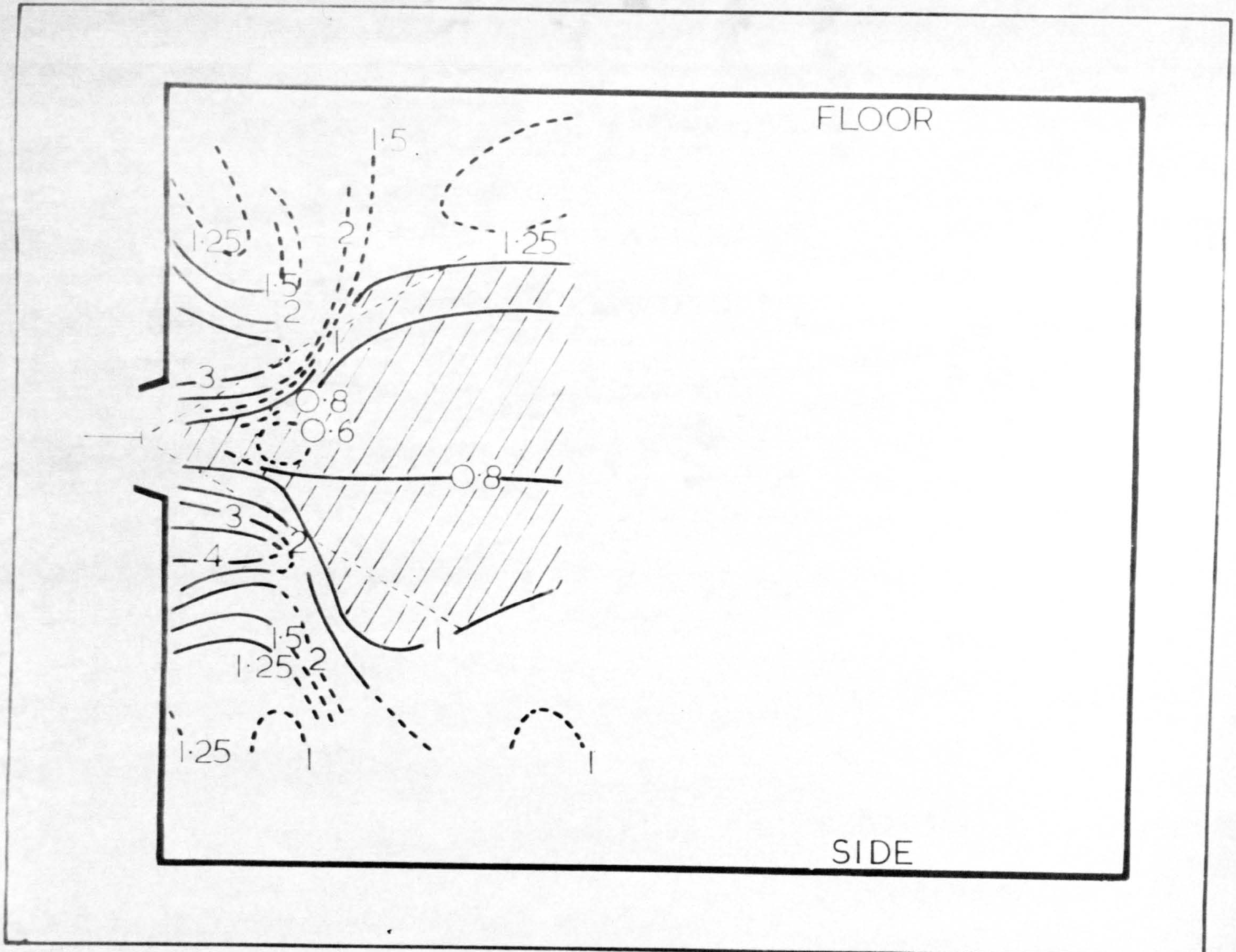



FIG. 40d. PLOT OF EQUIVALENCE RATIO – DUPLEX F.P. 60°
(INITIAL RESULTS)

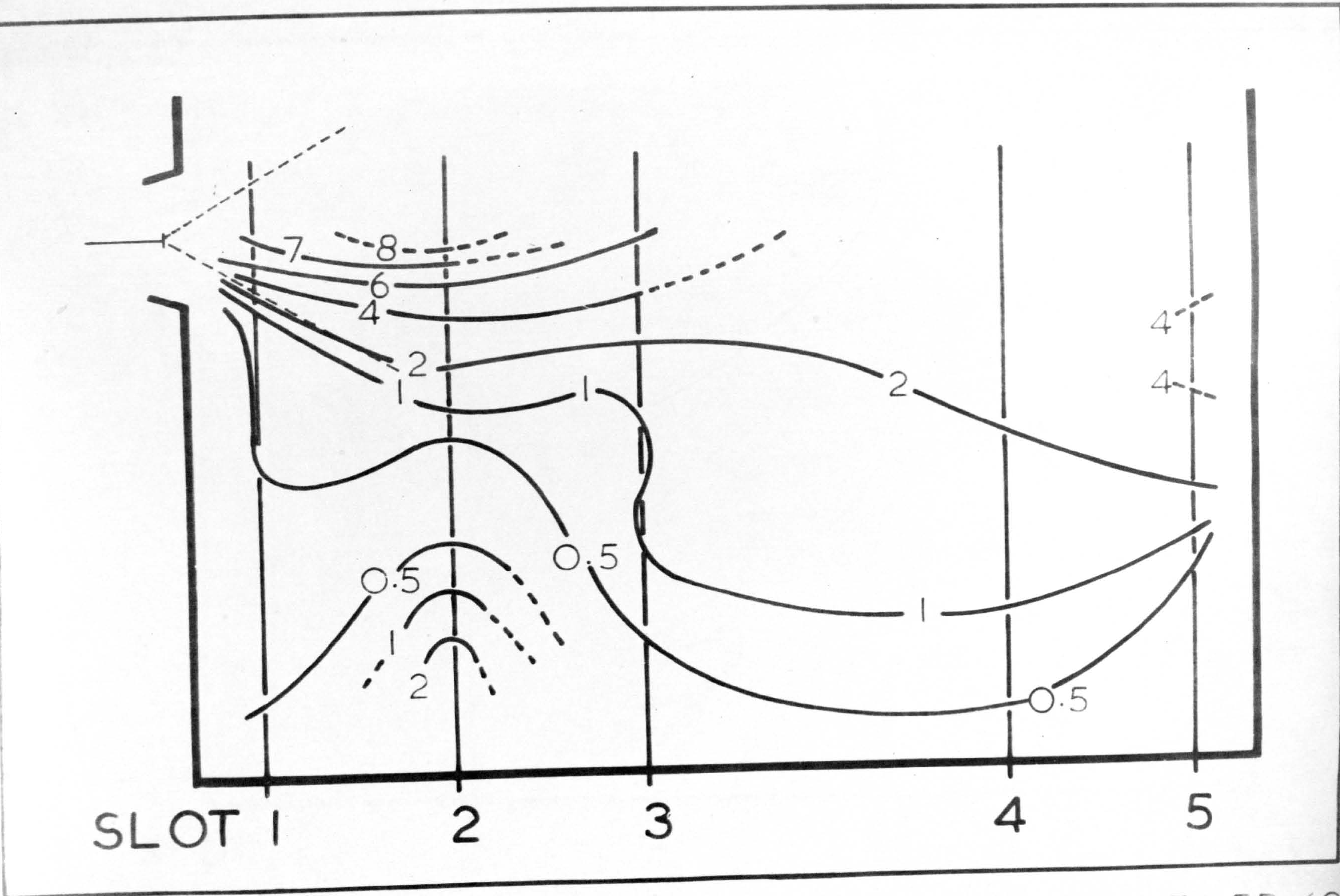



FIG. 41.a. % CARBON MONOXIDE DISTRIBUTION - DUPLEX F.P. 60°

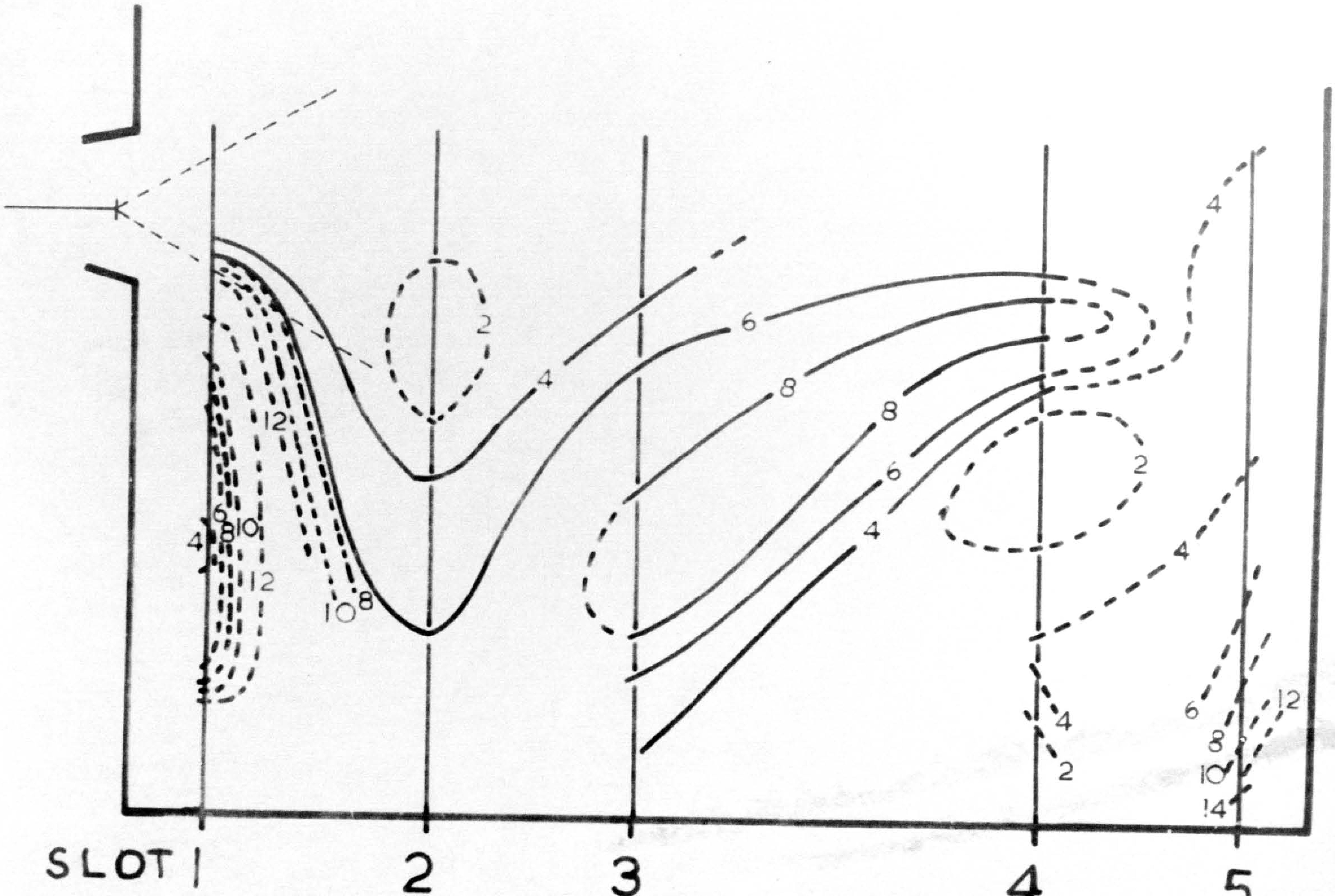



FIG. 41.b. OXYGEN DISTRIBUTION. - DUPLEX F. P.

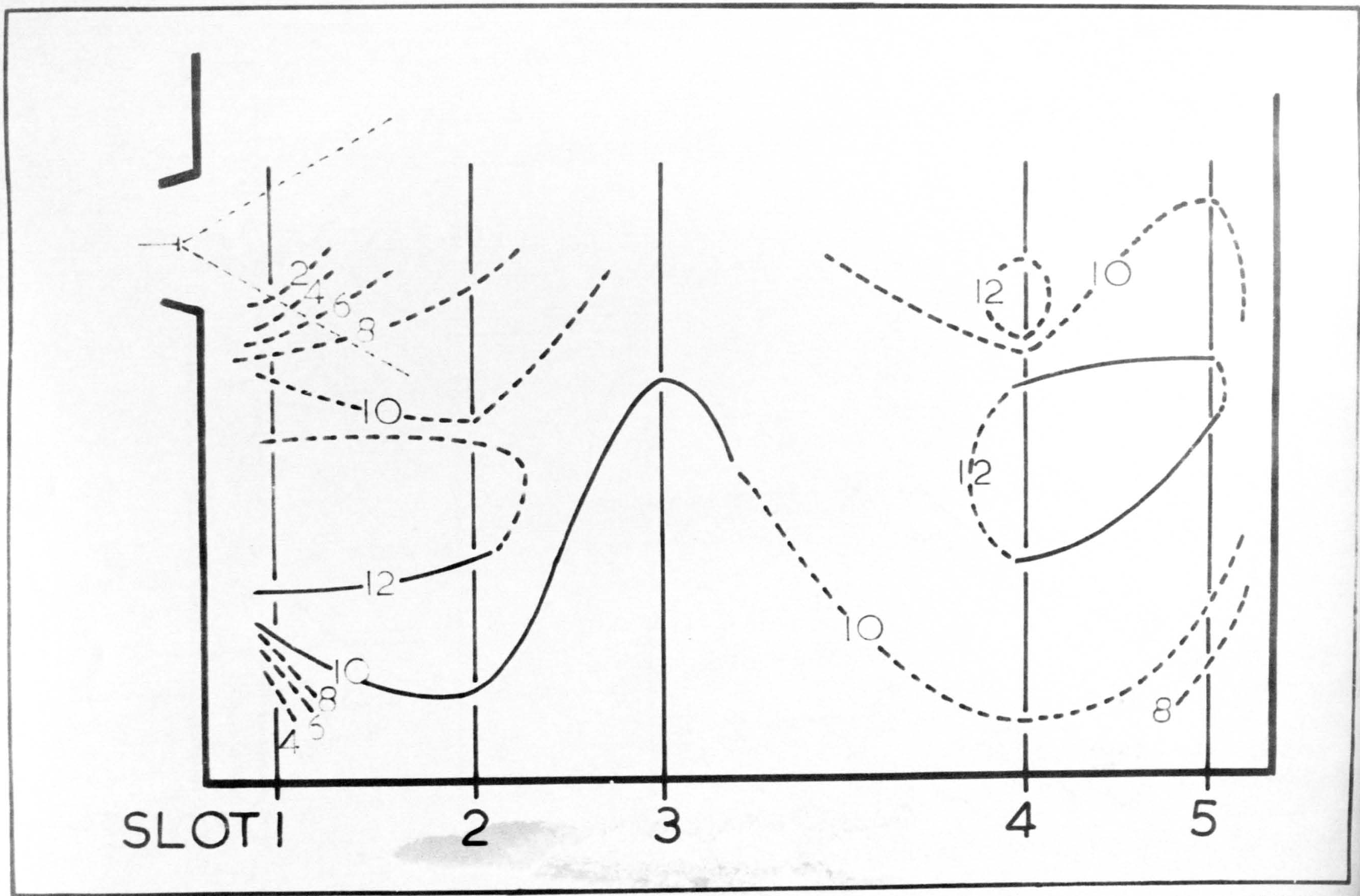



FIG. 41.c. % CARBON DIOXIDE DISTRIBUTION - DUPLEX F.P. 60°

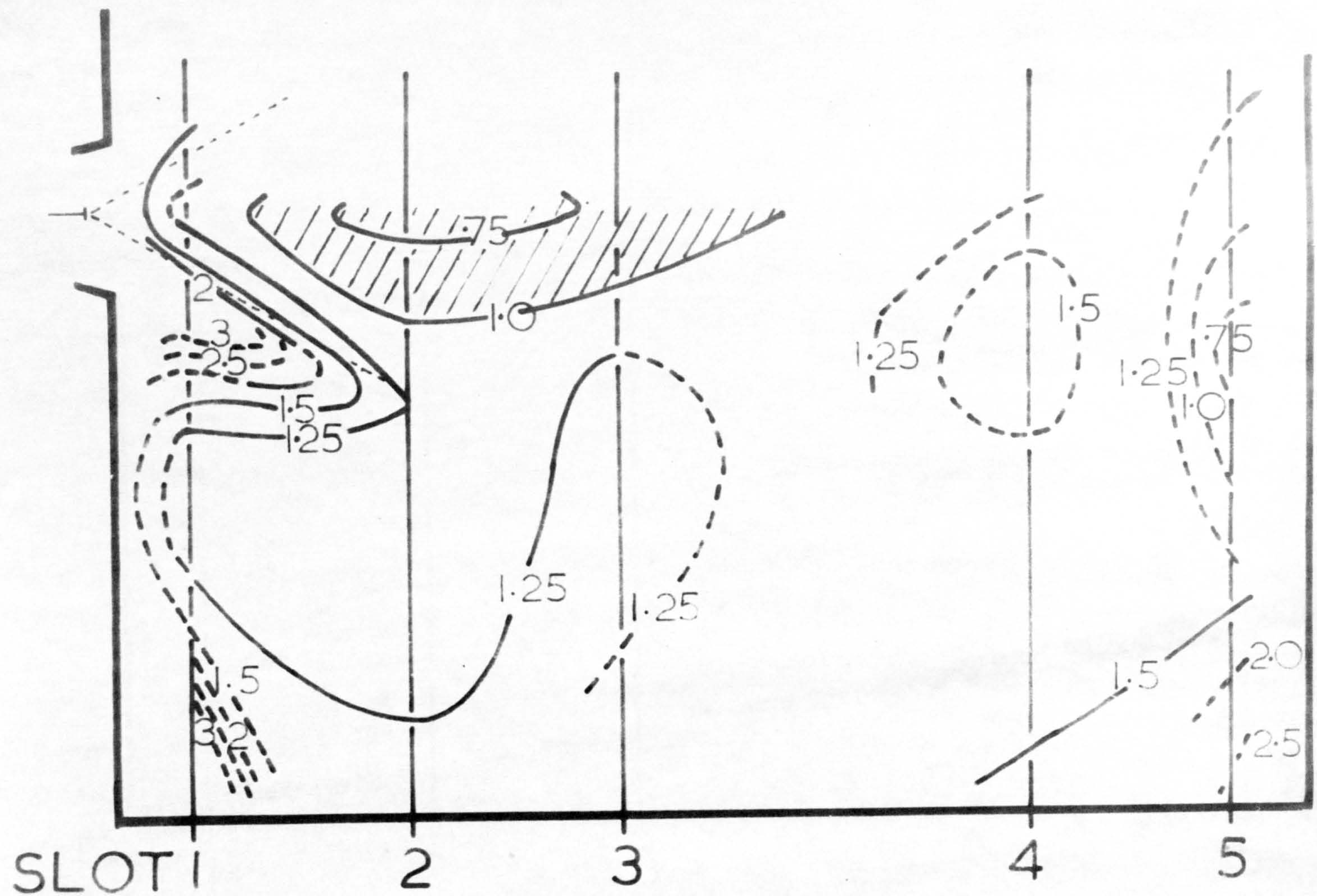



FIG. 41.d. EQUIVALENCE RATIO DISTRIBUTION - DUPLEX F.P. 60°

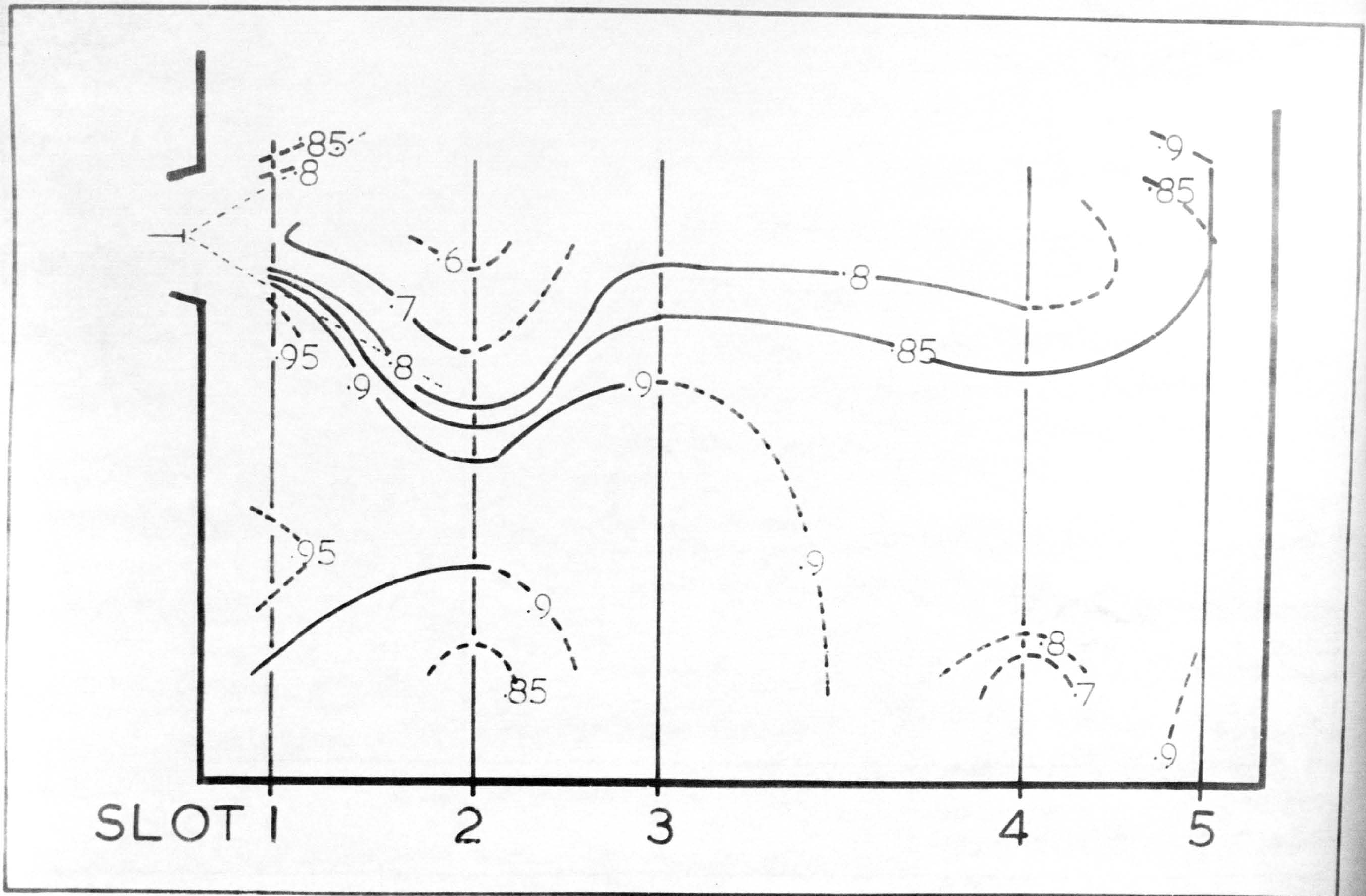



FIG. 41e. COMBUSTION EFFICIENCY DISTRIBUTION - DUPLEX F.P. 60°

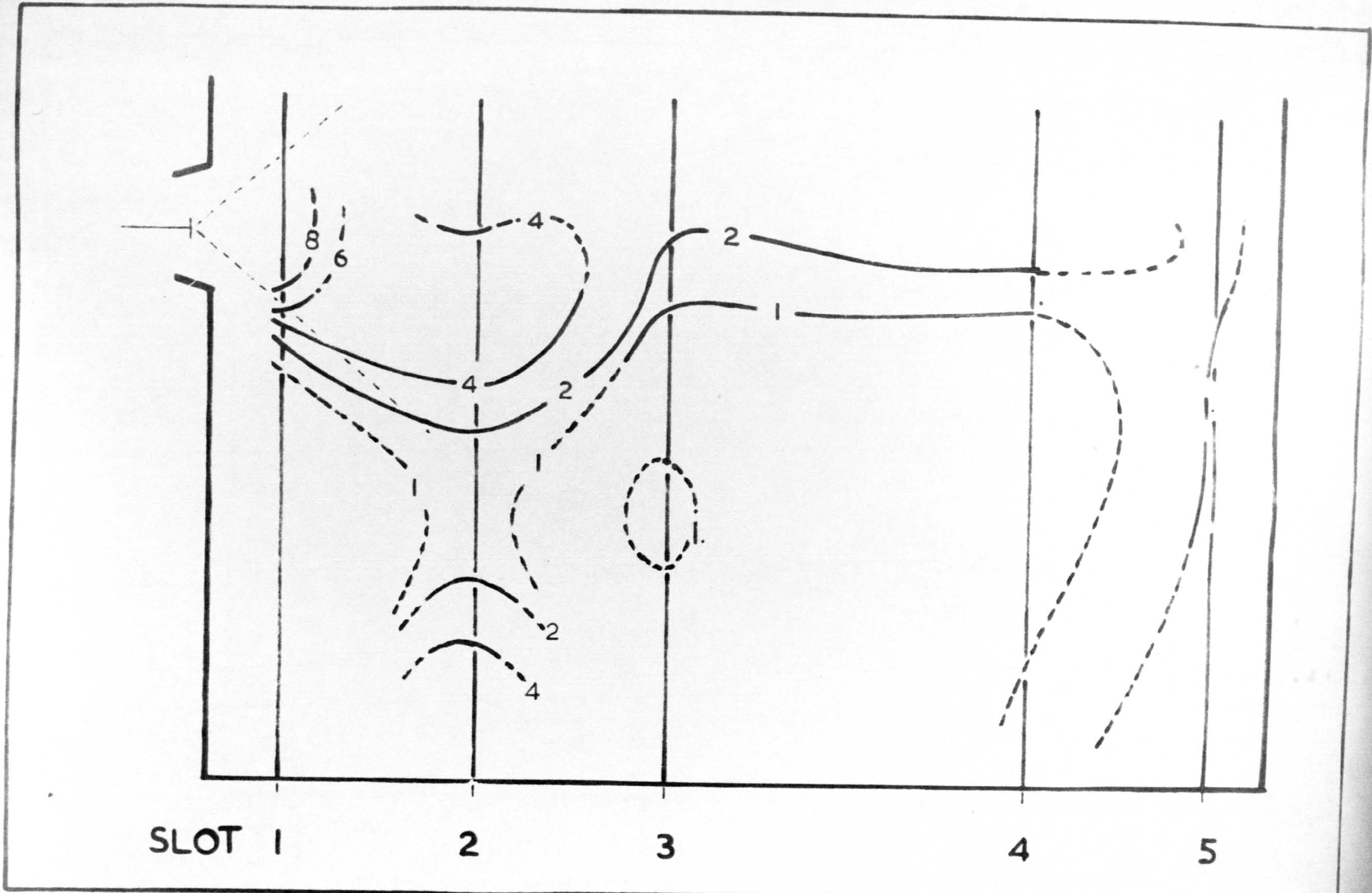



FIG. 42.a. % CARBON MONOXIDE DISTRIBUTION – SPILL F.P. 80°

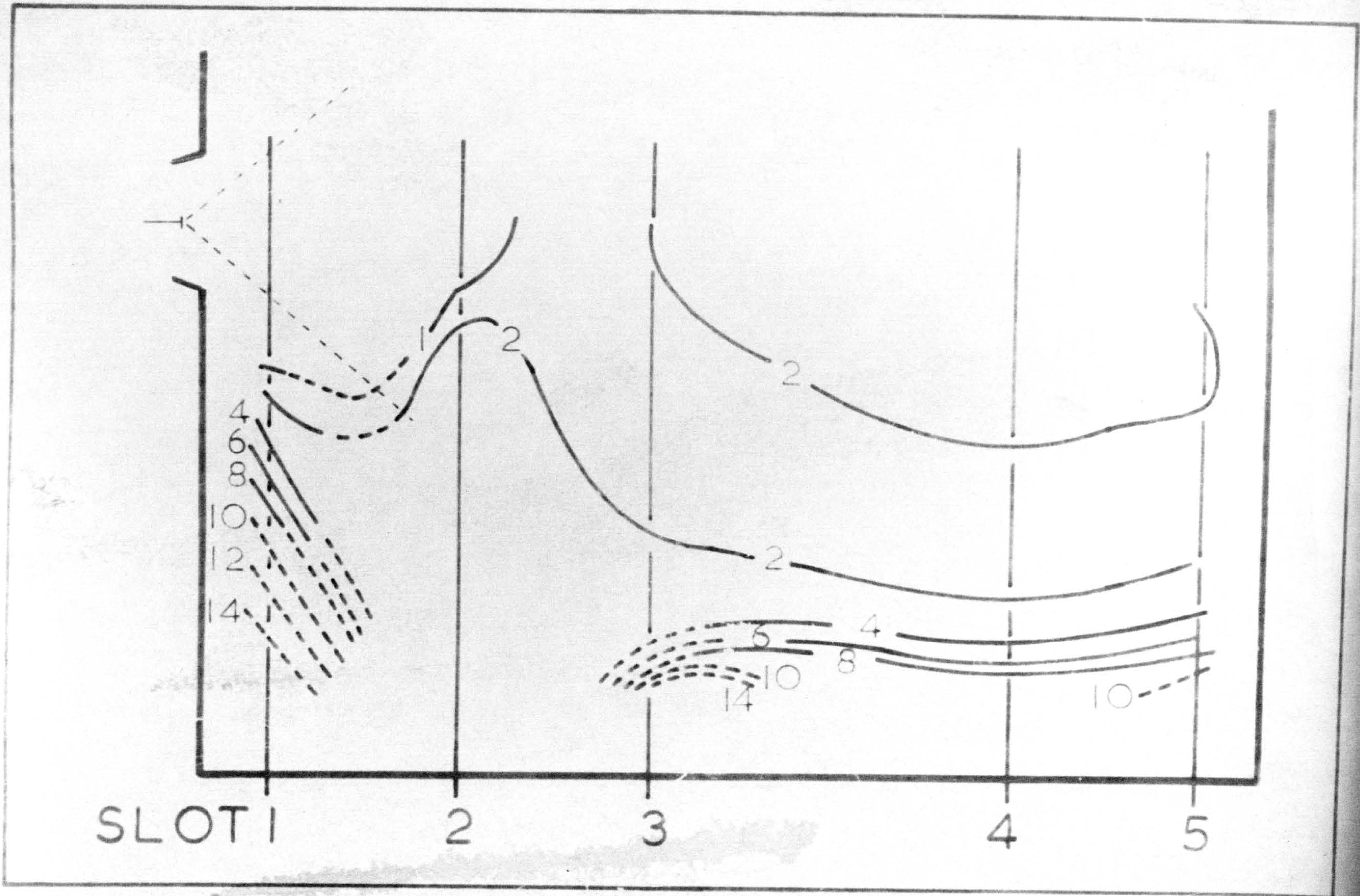



FIG. 42.b. OXYGEN DISTRIBUTION - SPILL FP 80°

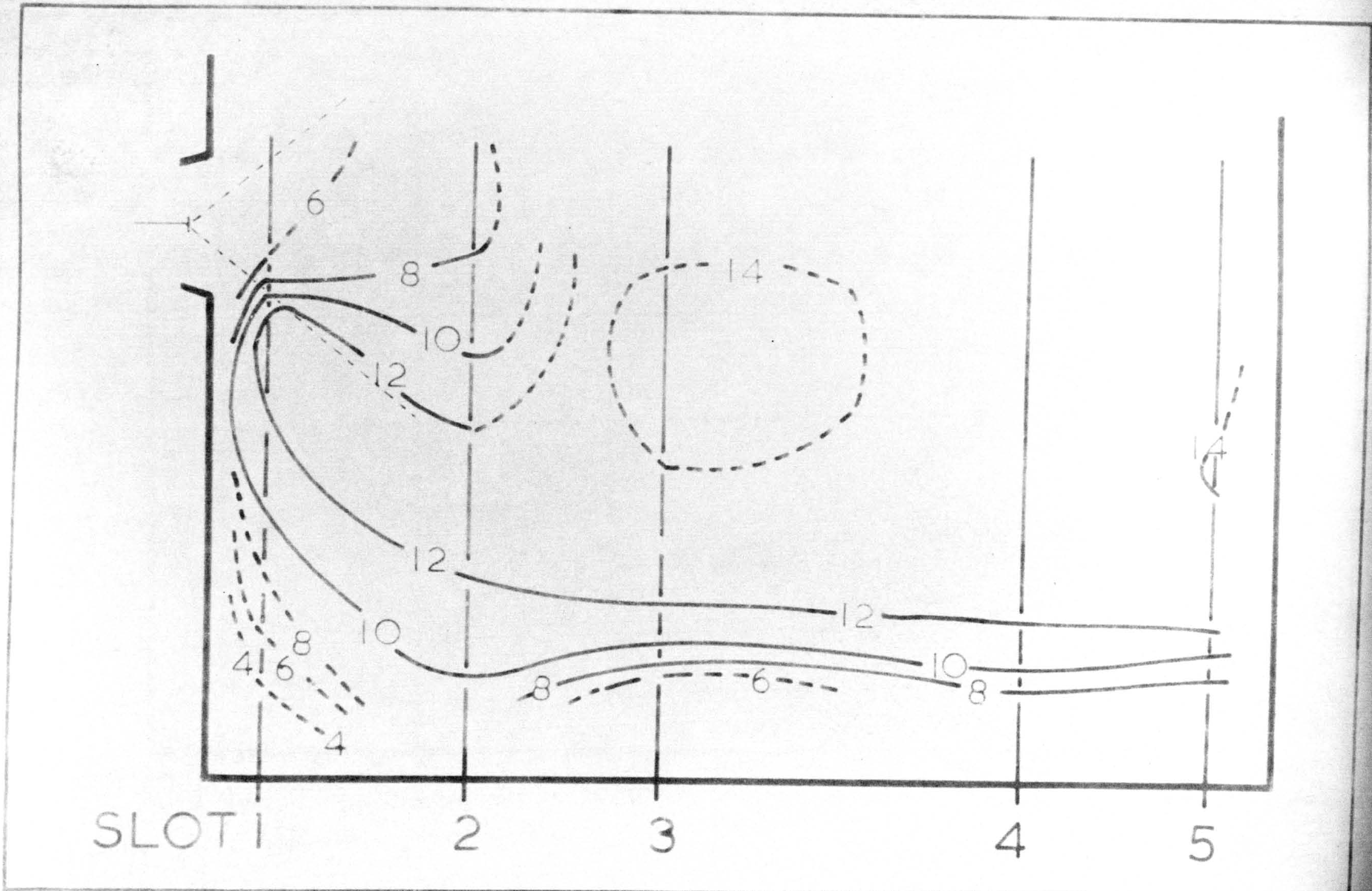



FIG 42.c. CARBON DIOXIDE DISTRIBUTION - SPILL F.P. 80°

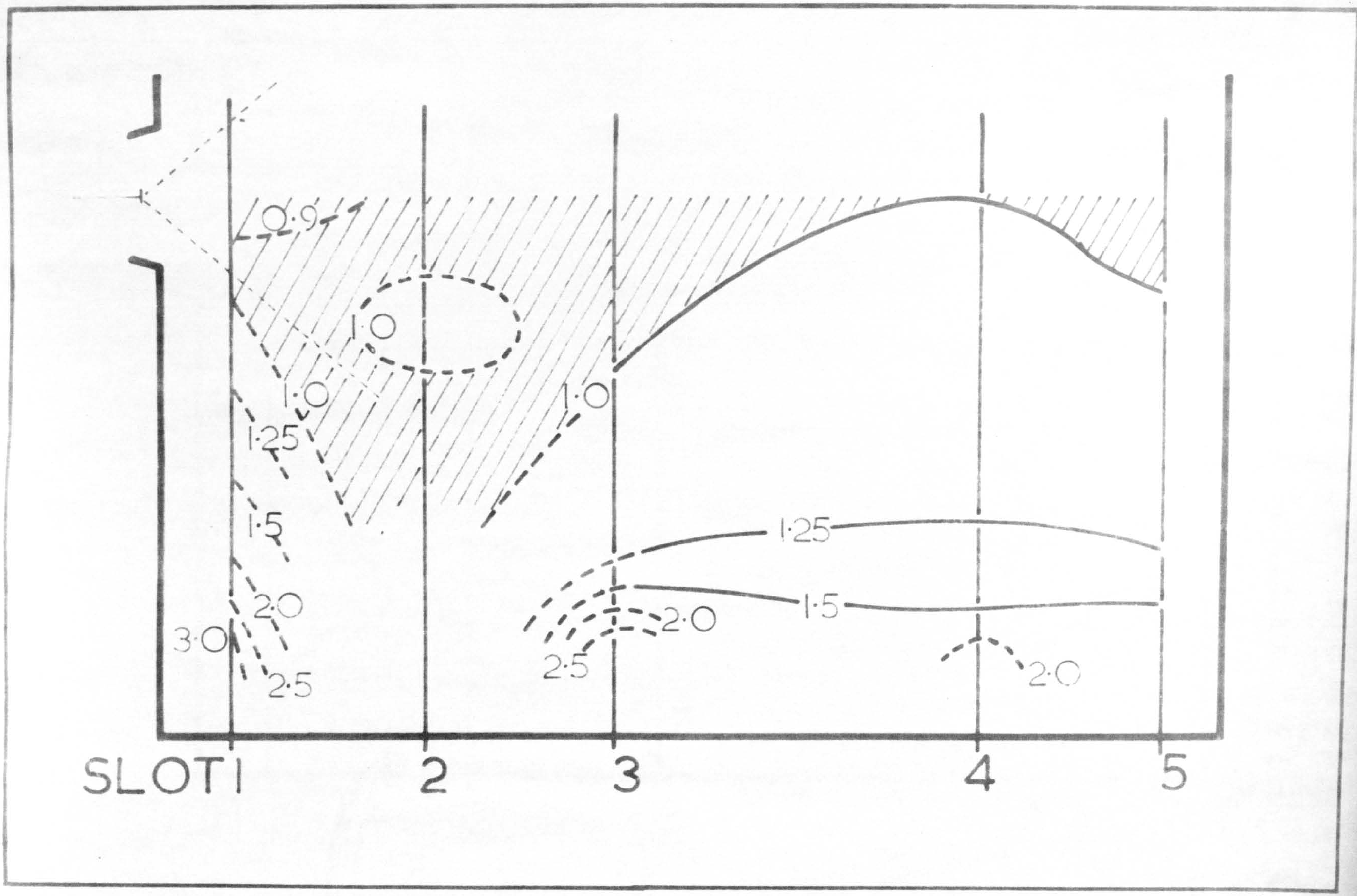



FIG. 42.d EQUIVALENCE RATIO DISTRIBUTION - SPILL F.P. 80°

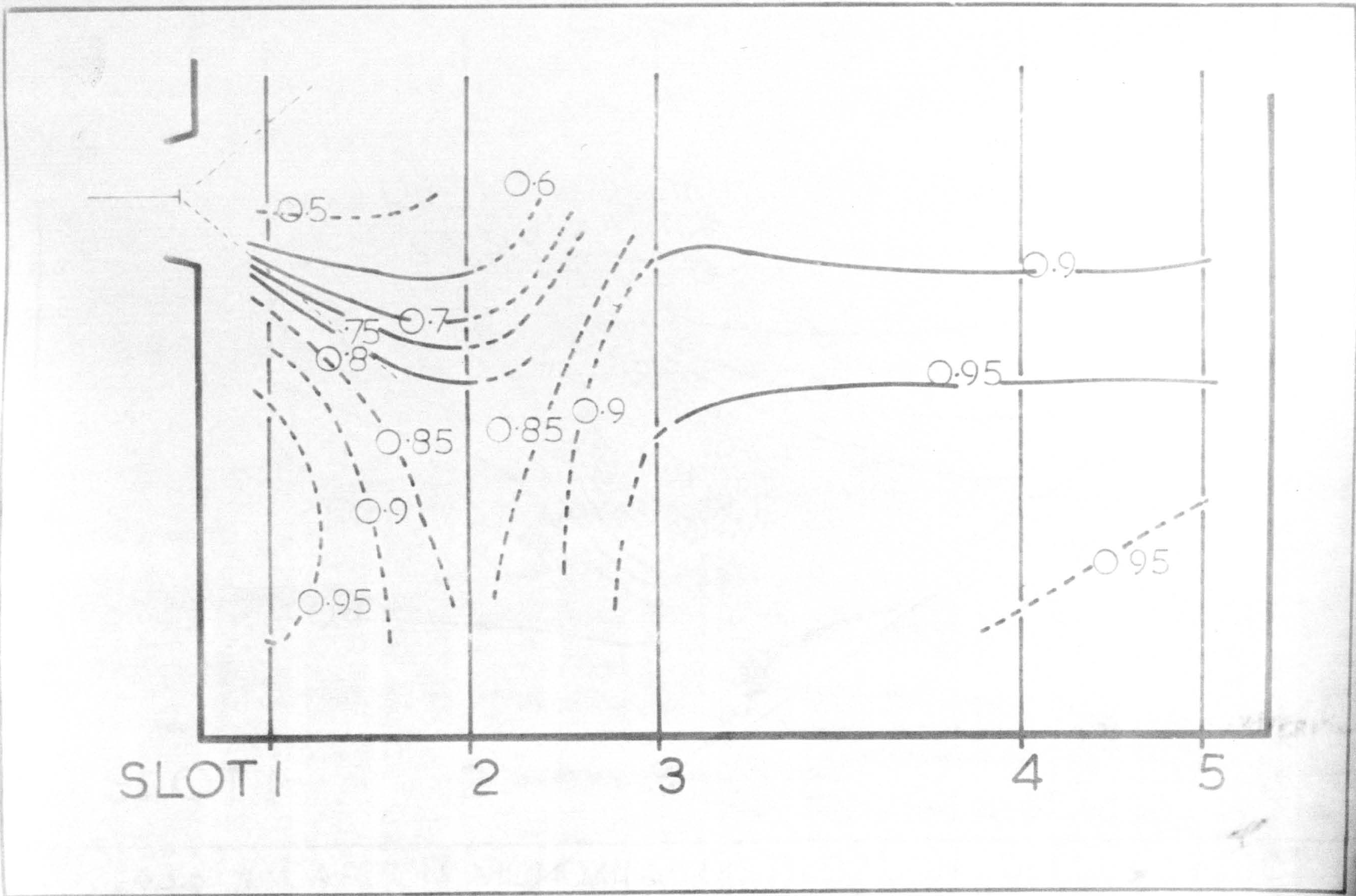



FIG. 42.e. COMBUSTION EFFICIENCY DISTRIBUTION - SPILL F.P. 80°.

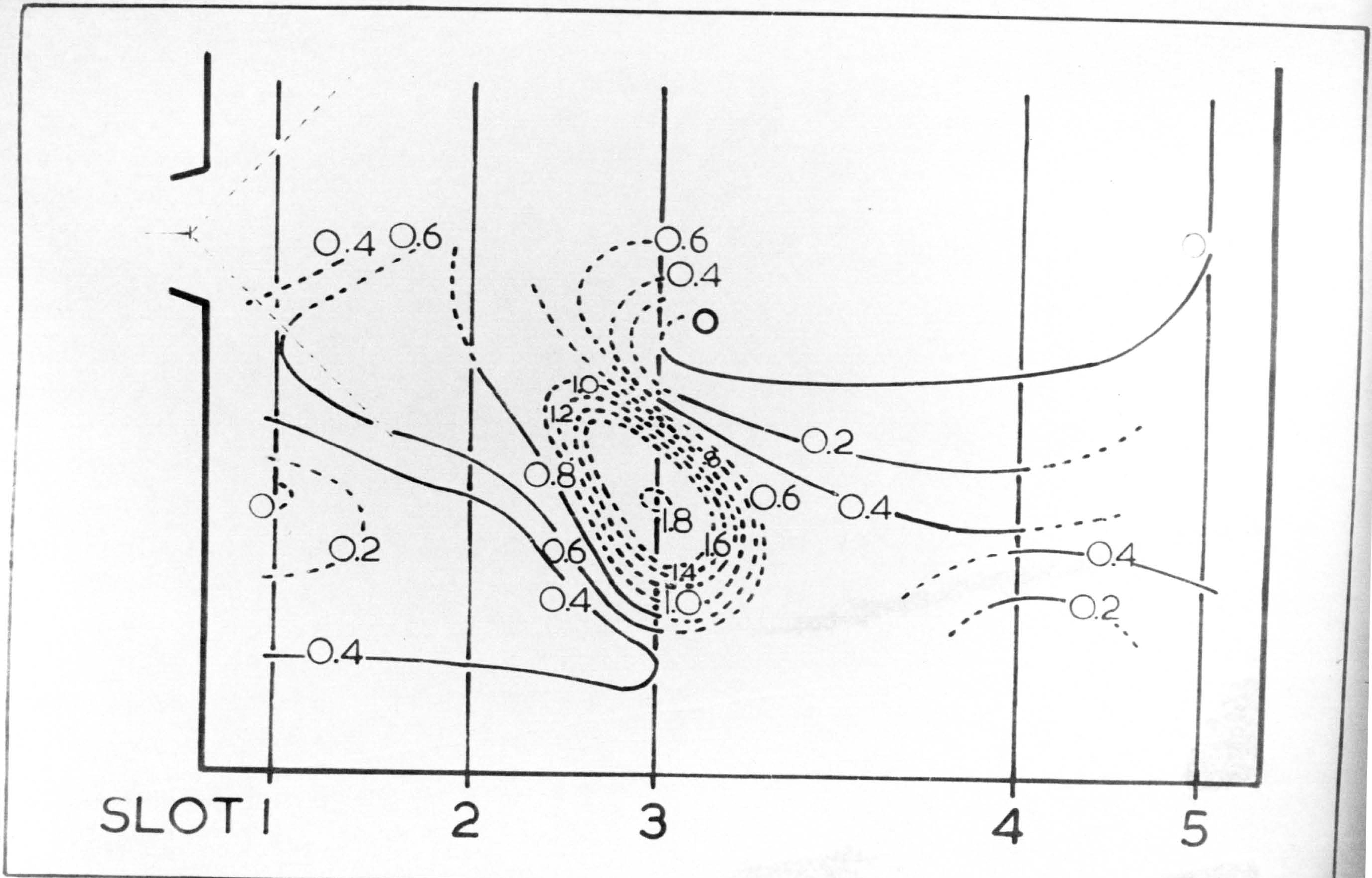



FIG. 43.a % CARBON MONOXIDE DISTRIBUTION - DUPLEX L.P. 90°

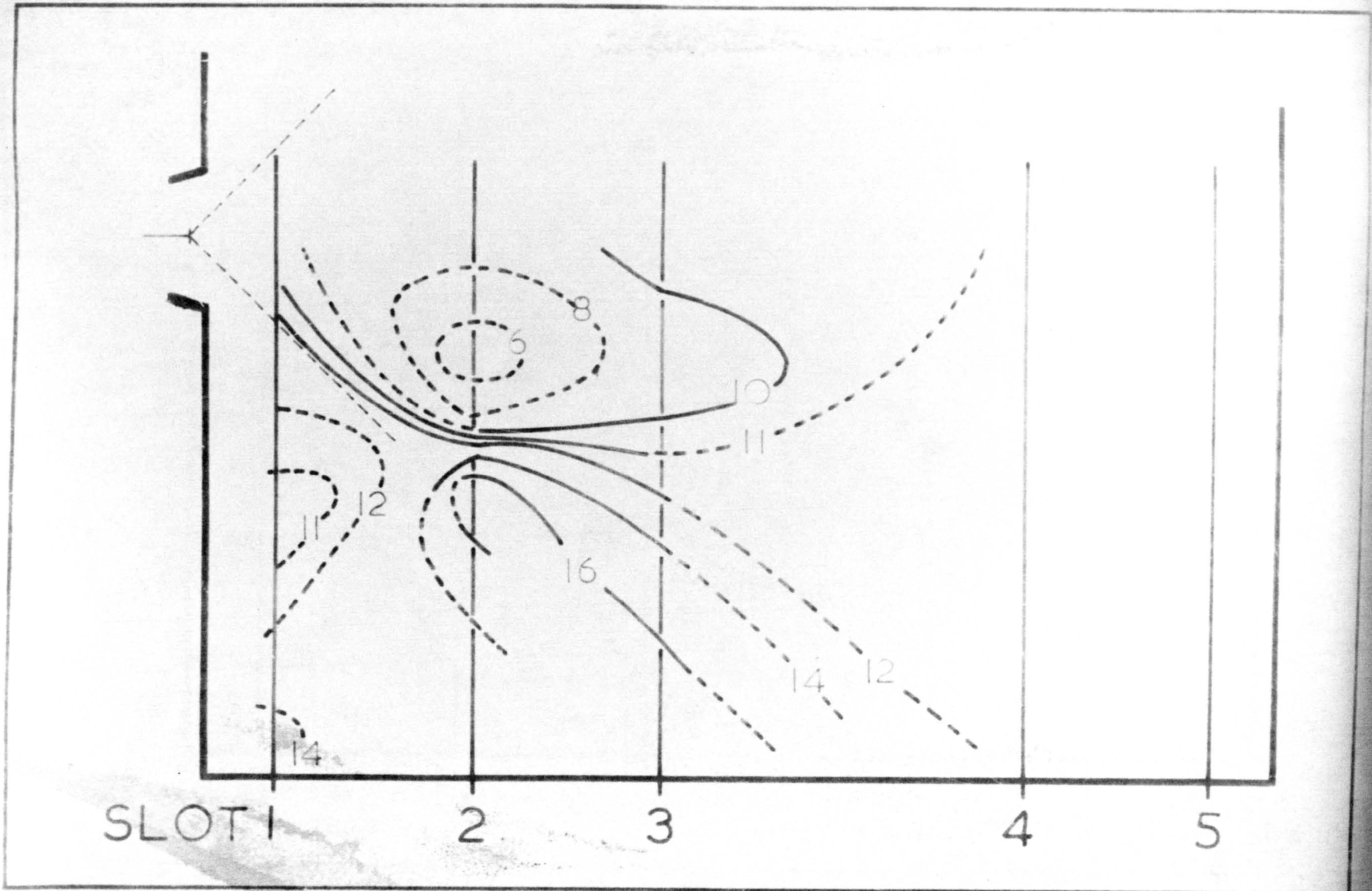



FIG. 43 b. OXYGEN DISTRIBUTION - DUPLEX L.P. 90°

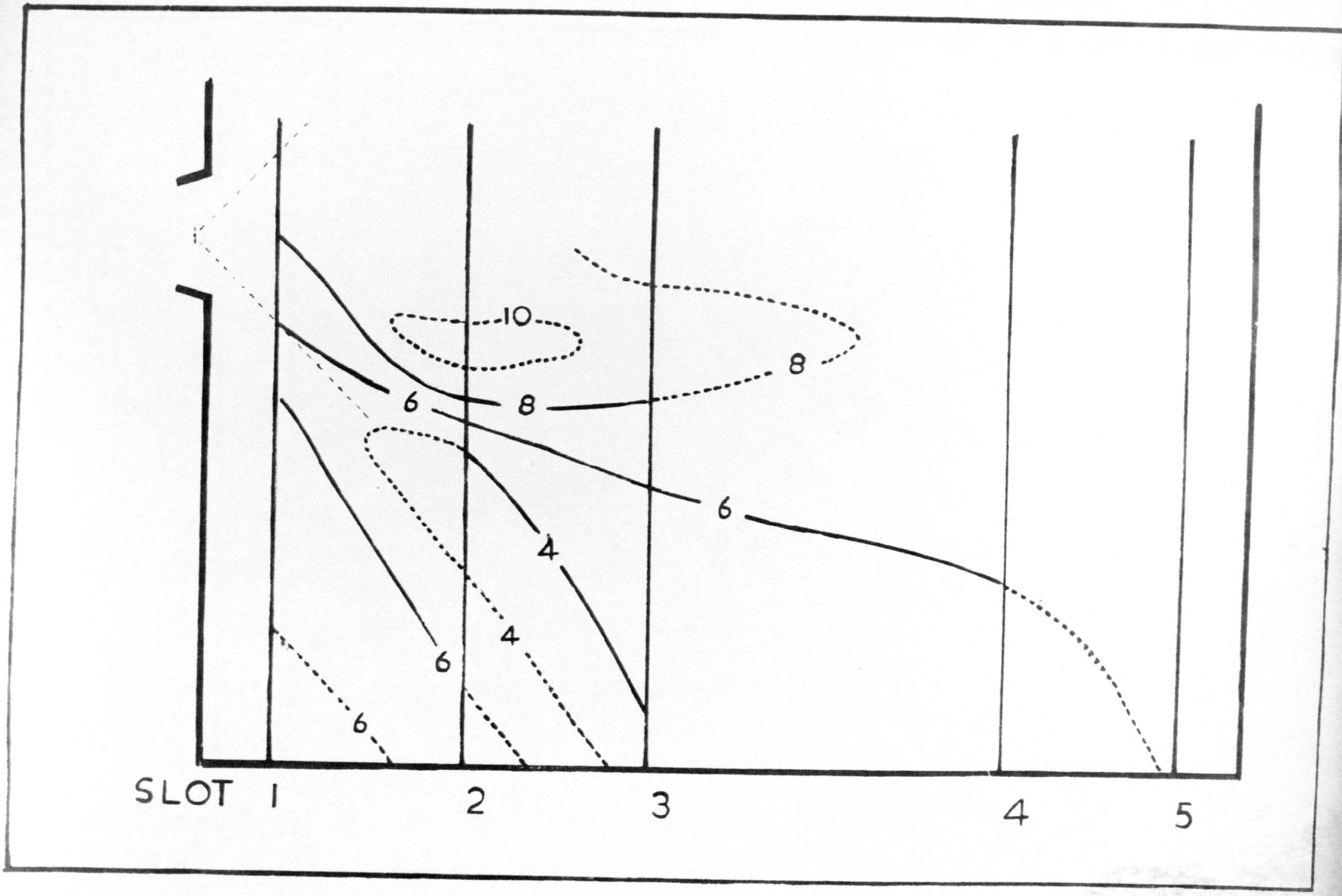



FIG.43.c. % CARBON DIOXIDE DISTRIBUTION - DUPLEX L.P.

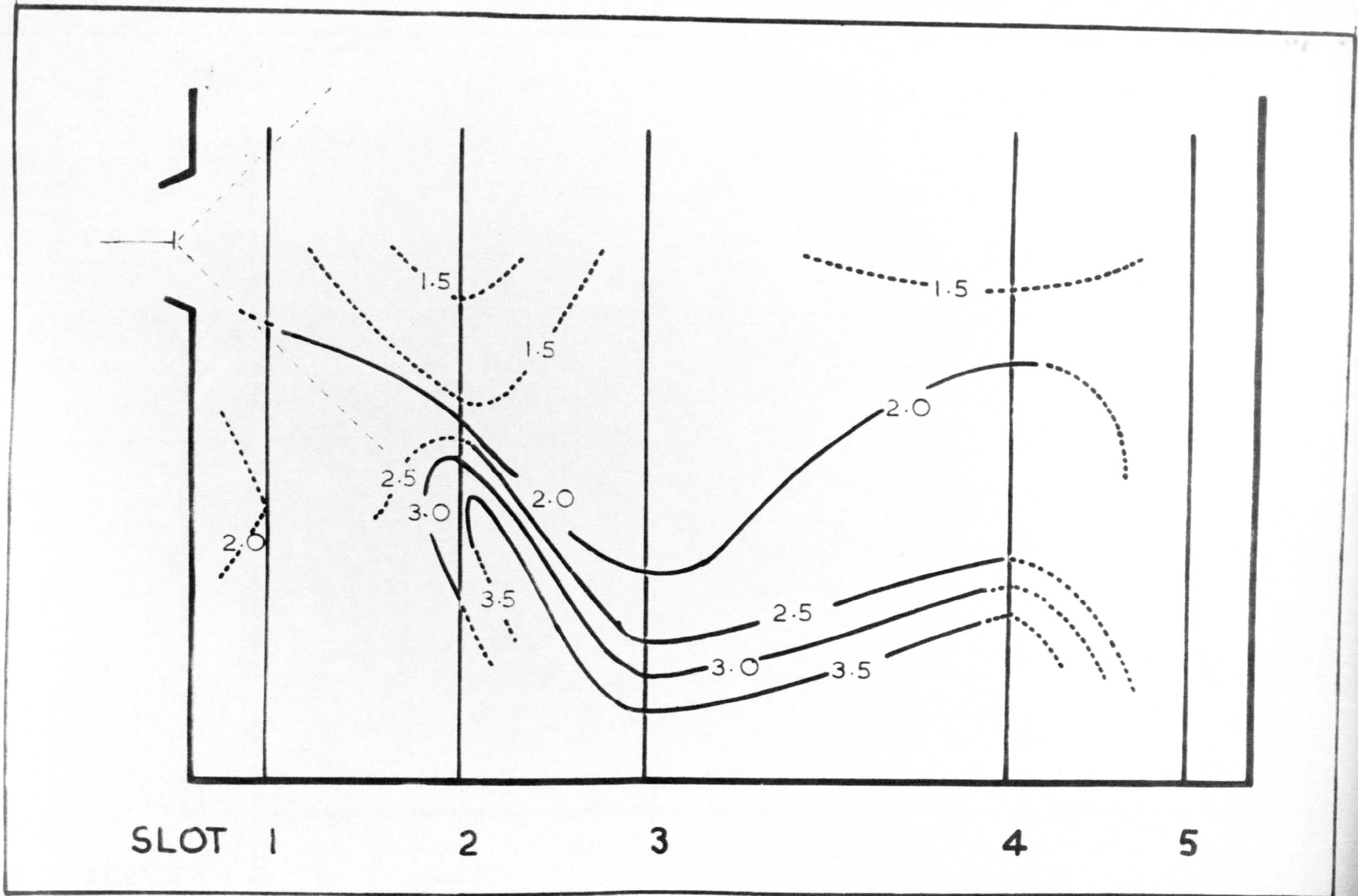



FIG. 43.d. EQUIVALENCE RATIO DISTRIBUTION - DUPLEX L.P. 90°.

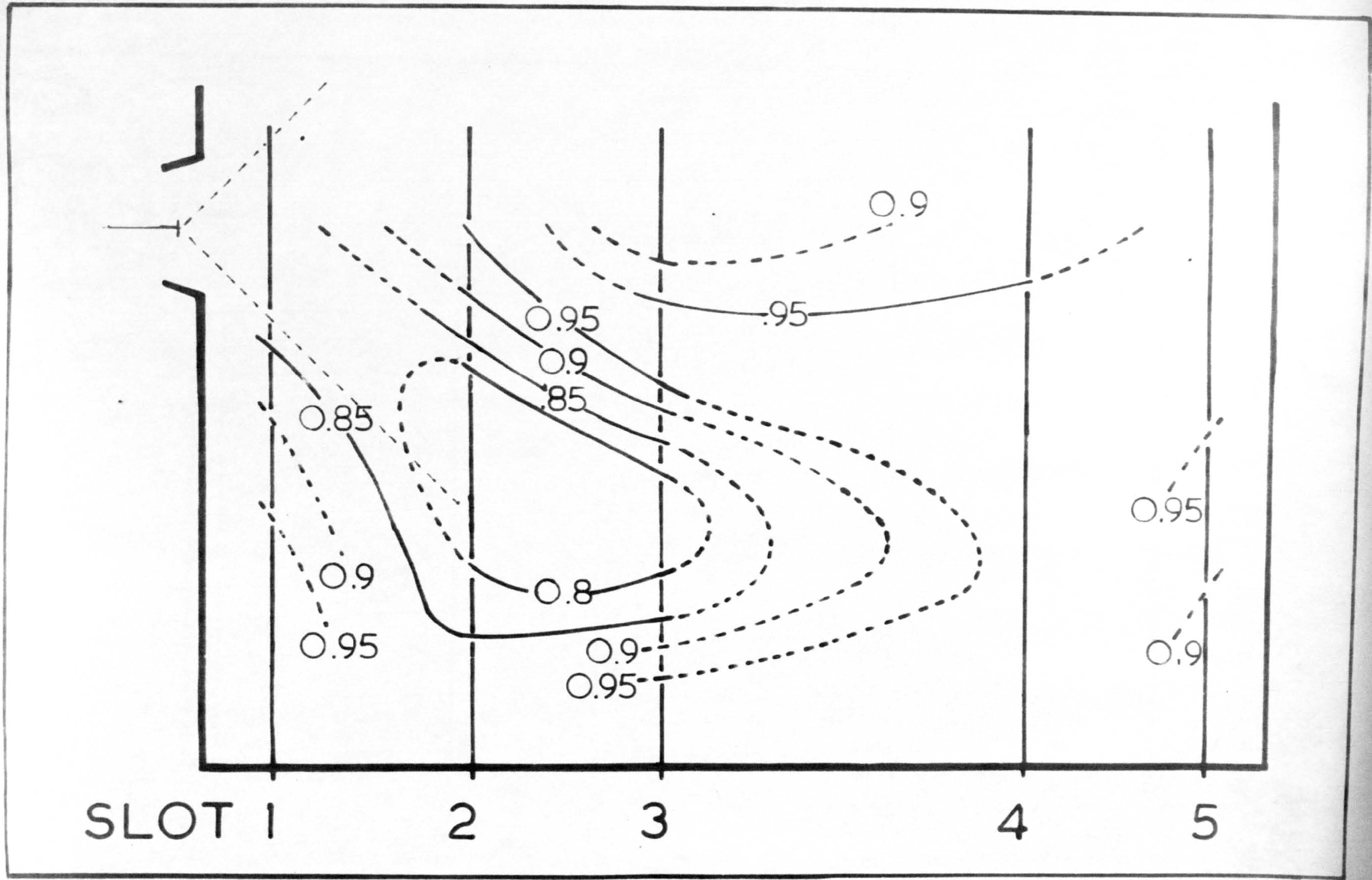



FIG. 43.e. COMBUSTION EFFICIENCY DISTRIBUTION - DUPLEX L.P. 90°

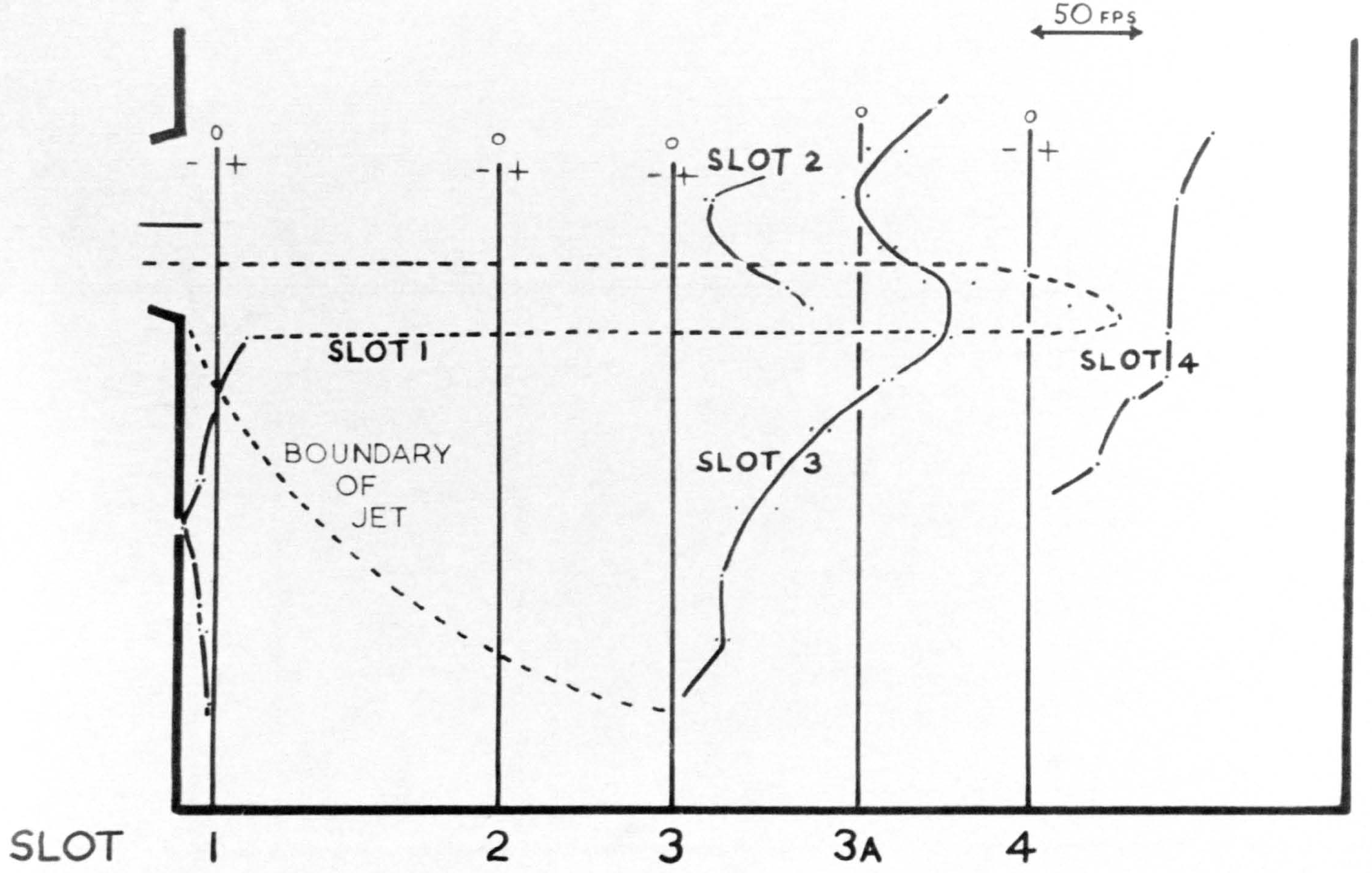



FIG. 44. VELOCITY PROFILES - 60° DUPLEX FULL POWER

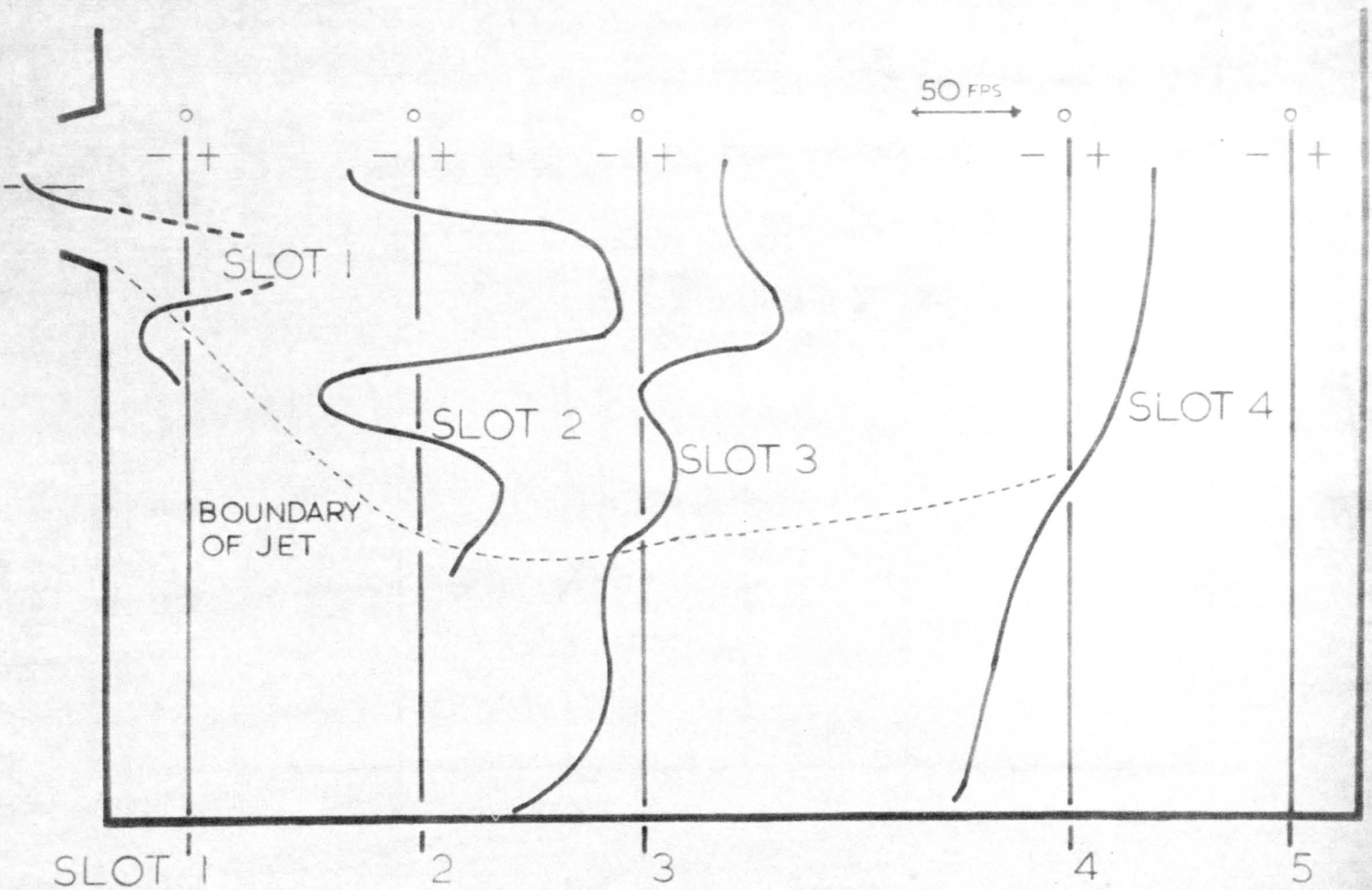



FIG. 45. VELOCITY PROFILES - 80° SPILL FULL POWER

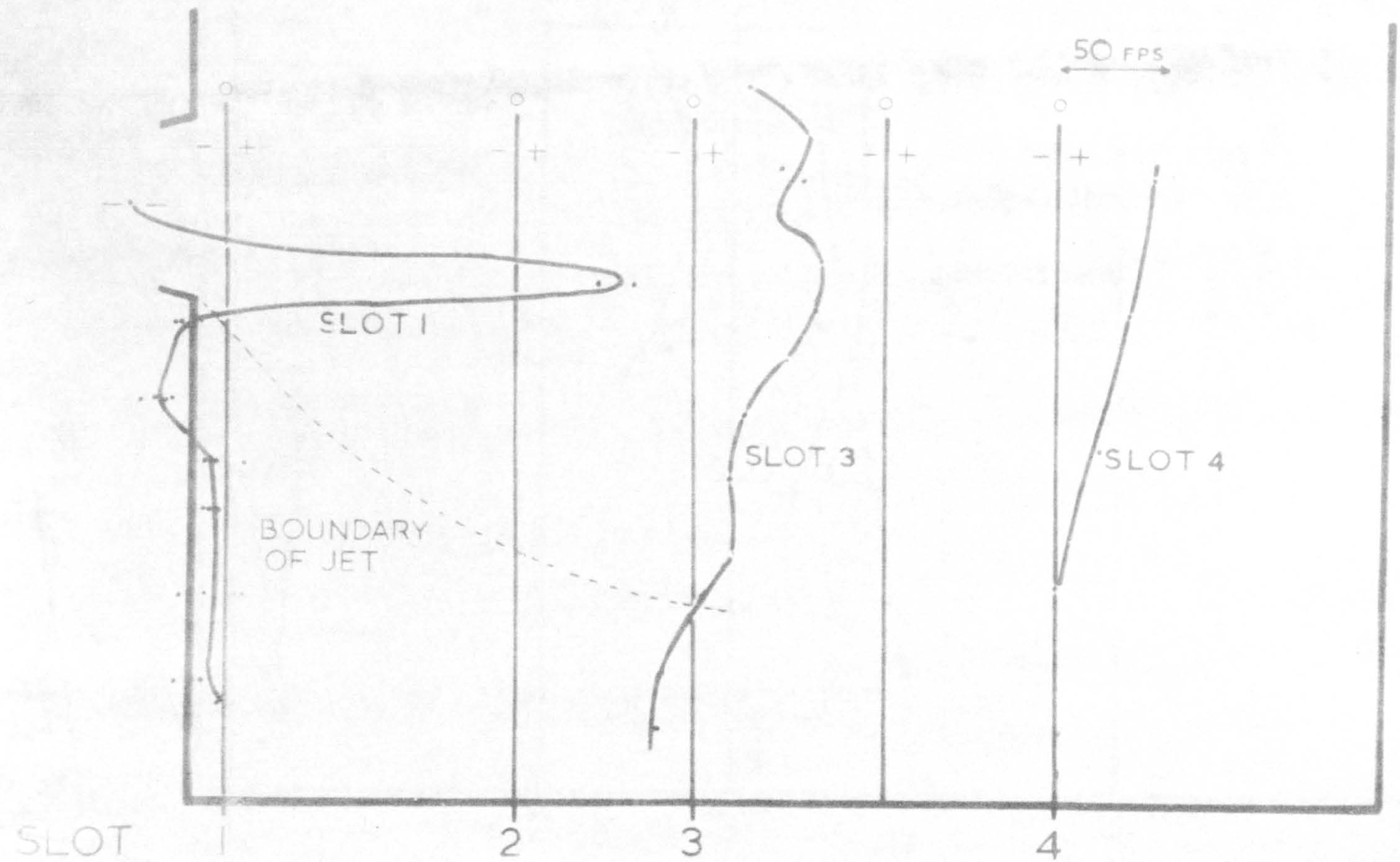



FIG. 46. VELOCITY PROFILES — 90° DUPLEX LOW POWER.

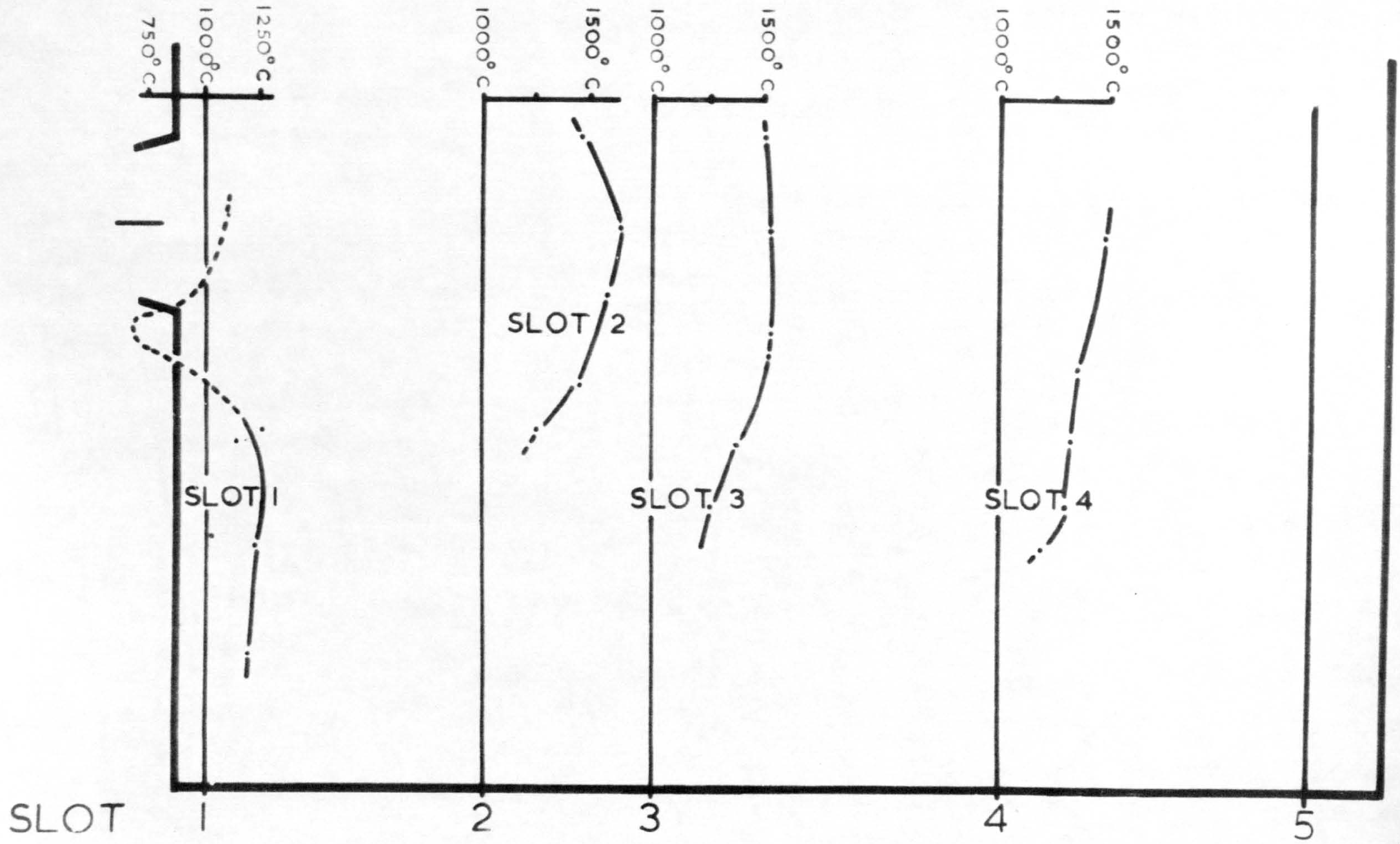



FIG. 47. TEMPERATURE PROFILES - 60° DUPLEX FULL POWER

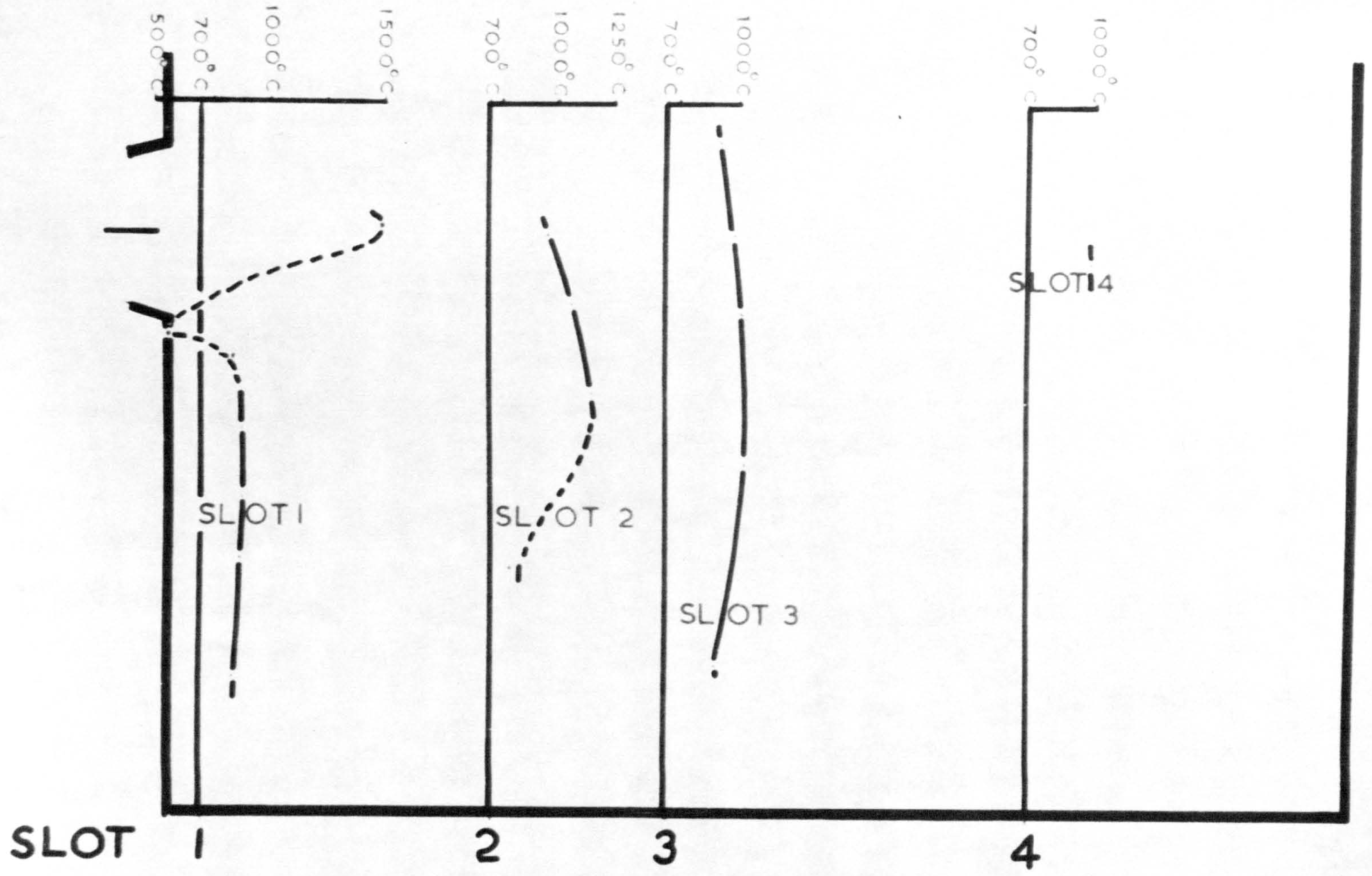



FIG. 48. TEMPERATURE PROFILES - DUPLEX LOW POWER

## 5.4 Experimental work with model burner.

### 5.4.1 Operating conditions chosen for study.

Details of the laboratory rig and the model burner design used have been given already in sections 4.2 and 3.4.2. The selection of the operating conditions was governed by the twin objectives of obtaining extra information to supplement the full size data and that of demonstrating the degree of modelling of full size flames that reasonably could be achieved.

As already discussed in section 3.3.4 the possible approaches to scaling pressure-jet flames have been reduced to two that seemed most likely to yield satisfactory results. These were cases 1. and 2. which are discussed in section 3.3.4. These provide for velocity, momentum ratio and air-fuel ratio similarity in the first case, and effective flame residence time, momentum similarity and air-fuel ratio in the second case.

As already indicated, case 1 effectively introduces a flow ratio of $L^2$, and case 2 a ratio of $L^3$. In the second case, strict momentum similarity as given by the group $P^{\frac{1}{2}}/v$ would reduce the pressure to 50.p.s.i.which is usually considered unsatisfactorily low for a pressure-jet atomizer. At very low atomizing pressures it is usually difficult to get a properly open spray cone.

Ideally, the next parameter, in order of importance, is the droplet size, as this effects the process of combustion and evaporation which is number 5. on Stewart's list (see Table II.). In practice, once the throughput rate and atomizing pressure are fixed, the droplet size distribution is fixed also, unless there is a very drastic change in fuel properties. Unfortunately, in the case of the atomizers used the different methods proposed for calculating the S.M.D. give such a wide scatter of results that it is difficult to be certain how far this alters between the atomizers used. Representative data are given in Table III.

For various practical reasons associated with the disposal of waste gases the model data given are divided between full-power and low power conditions, although in some ways this is not ideal, it does provide data under two widely differing air-fuel ratios. The lower air-fuel ratio is associated with the longer residence time so that it should not be unduly prejudicial to one set of results.

The two flames compared were the $60^{\circ}$ full power case and the low power case. These corresponded to the Duplex atomizer running at full and $\frac{1}{4}$ power. In the model, these conditions were represented by separate atomizers with characteristics as dictated by the scaling requirements and as far as possible similar spray angles.

TABLE IV.

CALCULATED OPERATING CONDITIONS FOR MODEL FLAMES ON DIFFERENT SCALING BASES.

| Prototype Flame | 60° Full Power | | Low Power | |
|---|---|---|---|---|
| **Prototype Conditions** | | | | |
| Fuel Rate lb./hr. | 2175 | | 570 | |
| Atomizing Pressure p.s.i. | 550 | | 500 | |
| Equivalence Ratio | 1.28 | | 1.98 | |
| Reference air velocity | 136 | | 51 | |
| Calculated S.M.D. (a) | 99 | | 74 | |
| Calculated S.M.D. (b) | 70 | | 58 | |
| Calculated Conditions for scaling:- | Case 1. | Case 2. | Case 1. | Case 2. |
| Fuel Rate | 196 | 58.7 | 51.2 | 15.4 |
| Atomizing Pressure | 550 | 49.5 | 500 | 45 |
| Reference velocity | 136 | 44 | 51 | 15.3 |
| Calculated S.M.D. | 54 | 105 | 42.2 | 75 |
| Conditions required to match actual atomizers using | Case 2. | | Case 1. | |
| Fuel Rate | 50 | | 46 | |
| Calculated Reference velocity. | 35.5 | | 54 | |
| Ratio $P^{\frac{1}{2}}/V$ | 1.5 | | .86 | |
| S.M.D. (a) | 68 | | 43 | |
| S.M.D. (b) | 72 | | | |

NOTE S.M.D. (a) Calculated using Radcliffe's formula.
(b) Calculated using Hudson & Clarke's formula.

See Appendix III.

The important full-size data are set out in Table III, together with corresponding calculated data using the scaling requirements selected under the headings of case 1. and case 2. in section 3.3.4 and the atomizer characteristics of the atomizers selected as being the nearest possible available.

The reference velocities quoted have been calculated on the basis of the minimum diameter of the burner throat (that is 12" in the full-size case and 3.6" on the model).

The full-power case was scaled to the requirements of case 2. This means effectively that the residence times in the flames should be the same for prototype and model. The result of the atomizer limitations has been mentioned already. As will be seen, the choice of 100 p.s.i.g. as an atomizing pressure increases the group $P^{\frac{1}{2}}/v$ by about 50%. On the other hand, according to Radcliffe's relation, the S.M.D. decreases by 31% while data due to Hudson and Clarke, representing the manufacturers of these two atomizers, gives a nominal increase in S.M.D. The degree of confusion arising over this matter can be seen by reference to the calculated values of S.M.D. from different relations given in Appendix III.

The low power case was designed to give, as far as possible, momentum similarity although this was not quite achieved. In view of the very conflicting requirements in all cases between different scaling groups this departure is unlikely to introduce differences which are too serious into the comparative experimental results.

It was assumed for modelling purposes that the flame temperature would be the same in prototype and model.

A distillate fuel, in this case Commercial gas oil, was used for each run. This had about the same atomizing viscosity as the heated light residual fuel oil used on the full size runs. The main effect of the distillate fuel is to alter the ignition delay time and have, of course, little residual material to have a long burn out time. This is an advantage in case 1. where it restores the ignition delay group (see section 3.3.3.8) to its correct value. In case 2, however, this factor is too small.

To avoid a leakage of air on the model rig, it was usual to run with a slight positive pressure in the combustion chamber.

### 5.4.2 Special points about the operation of the furnace and the presentation of the results.

In the early stages of operation difficulty was experienced with a very ragged flame which stabilized on the air swirler and had a poor combustion efficiency. It appeared to be due to flash vapourized fuel recirculating back to the swirler. The cause of this seemed to be due to the omission of the air directing cap which is mounted over the atomizer on the full size burner. This provides a small bleed of air across the atomizer face to keep this free of partially cracked fuel and probably helps to form a bluff body stabilizer. Once in place, the flame performance became quite normal. It appears from this that the local flow configuration very close to the atomizer can have a critical effect on the subsequent flame performance. As the laboratory furnace was designed before the full-scale work was completely planned, it was not possible to have all the slots coinciding. In particular, in the light of the initial full-size experiments reported in section 5.3, the opportunity was taken to place a slot near the front wall on the boiler. This was followed up later on the model by the insertion of two ports A. and B. near the burner on the furnace. In practice the slots in the model came at the following diameters:-

TABLE VA.

| SLOT | A | B | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|---|---|
| Distance from burner in. | $\frac{1}{2}$" | $2\frac{1}{2}$" | 6 | 15 | 24 | 33 | 42 |
| Burner diameters. | 0.1 | 0.52 | 1.25 | 3.1 | 5.0 | 6.9 | 8.75 |

This compares with the full-size slot positions:-

TABLE VB.

| SLOT | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Distance from burner in. | 6" | $26\frac{1}{2}$" | 49" | 83" | $102\frac{1}{2}$" |
| Burner diameters. | 0.39 | 1.65 | 2.9 | 5.2 | 6.4 |

It will be seen that the following can be regarded as corresponding:- Model slot 2, prototype slot 3; model slot 3, prototype slot 4.

The difficulty of adding extra holes in the laboratory furnace without weakening the brickwork too seriously, made it necessary to use an L shaped water cooled gas sampling probe which could be inserted in slot 2. of the model and the nose traversed at a position corresponding to slot 2. on the actual boiler.

5.4.3 Gas Composition patterns.

5.4.3.1 Using case 1. as the basis of similarity.

This represented the low-power prototype condition as indicated in Table III. The results are shown in Figs. 49, 51, 53. This requirement gave, effectively momentum similarity between the jets but not residence time similarity.

As in the full-size results the region where unburnt fuel occurred (see Fig.49.a.) spread out roughly along the line of the spray cone, leaving the central core with very little, if any, fuel. This suggests that the spray in this case crosses the peak air velocity region early on while the fuel is still largely in the form of a liquid spray. Such fuel as is entrained in this region is obviously very quickly burnt and appears only as $CO_2$. The highest unburnt fuel concentrations measured were the CO figures recorded at model slot 1., $1\frac{1}{4}$ burner diameters downstream, and by using an L-shaped gas sampling probe to traverse as near as possible to a position equivalent to slot 2. on the prototype, which was a little under two burner diameters downstream. These high fuel figures occur rather earlier in the model than on the prototype, but in about the same relative position on the fringe of the jet and were marked by a visible lazy tail to the flame. Despite the shorter residence times, away from these regions the CO figures were usually very low indeed, less than 0.1%. The oxygen and $CO_2$ plots (Figs. 49.b. and c.) both follow patterns that correspond to the CO characteristics already mentioned. High oxygen figures were noted along the axis in the later stages with some unburned fuel. It will be noted that conditions outside the boundary of the jet were very uniform, with no sign of the change in concentrations near the walls which occurred on the full size results. As steps were taken to operate the laboratory furnace under a slight positive pressure, this tends to support the view that this was due to leakage.

The equivalence ratio as in the full size case was almost everywhere a good deal greater than unity, while the combustion efficiency approached one except in the regions already mentioned where there were high CO figures.

### 5.4.3.2 Using Case 2. as the basis of similarity.

This set of scaling criteria were used with the scaled version of the full power flame under the conditions indicated in Table III. This gave residence time similarity, but some departure from momentum similarity.

The gas composition plots are given in Fig.50. These show the characteristic region of high unburnt fuel on the early part of the central core which has been noted already on the full-size counterpart. This extends to somewhere beyond slot 2., that is about 4 burner diameters downstream. The main difference from the full scale results is that the fuel rich zone outside the flame seems to occur a little nearer the front wall, roughly about one burner diameter downstream.

The oxygen figures, Fig.50.b. in general were the opposite of the CO figures being low where the latter were high. Near the burner, very high figures were recorded in the region before the incoming jet had mixed with the fuel and recirculated combustion products.

The equivalence ratio plot, Fig. 50. d. showed a similar flame envelope shape to the full size flame, and the combustion efficiency data followed a similar form to the CO data.

### 5.4.4 Velocity and temperature measurements.

The velocity and temperature data are very similar to those already reported for the full-size data. The same W-shaped profile was produced and the length of the recirculation zone and the other relative dimensions were very similar (see Figs. 51. & 52). For instance, the ratio of the minimum velocity on the axis to the peak velocity at slot 2., 3 burner diameters downstream, was 0.61 on the model, and at $2\frac{3}{4}$ diameters downstream was 0.58 on the prototype. In the model it has been possible to obtain much more data near the burner, and hence construct much better curves for the velocity decay, (see Fig. 57. b.), although even here some trouble was experienced with blockage near the burner.

The temperature data were fairly similar to the prototype. The high temperature in the jet dropped to a slightly lower and fairly uniform figure in the recirculated gases. The peak temperature measured in the full-power case for instance, was about 1500° C in the prototype and 1680° C in the model, while the temperatures of the recirculated gases were about 1200° C and 1400° C, respectively. This means that the requirement of temperature similarity will not have been quite

FIGS. 49-50.

Isoconcentration Plots from Model Results.

FIGS. 51-52.

Velocity Profiles from Model Results.

FIGS. 53-54.

Temperature Profiles from Model Results.

FIGS. 58-59.

Cold Flow Results.

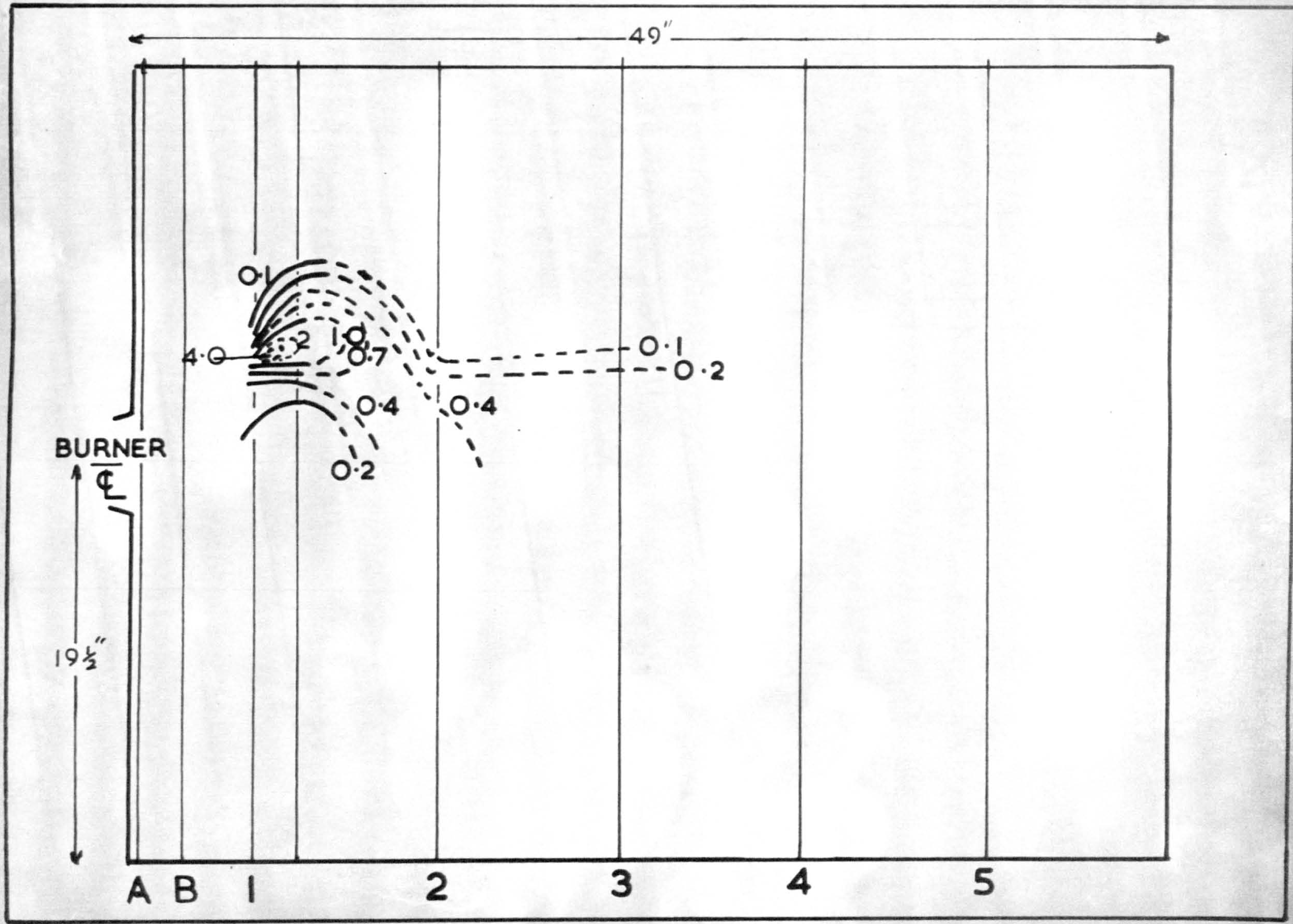



FIG. 49a. % CARBON MONOXIDE DISTRIBUTION - CASE I.

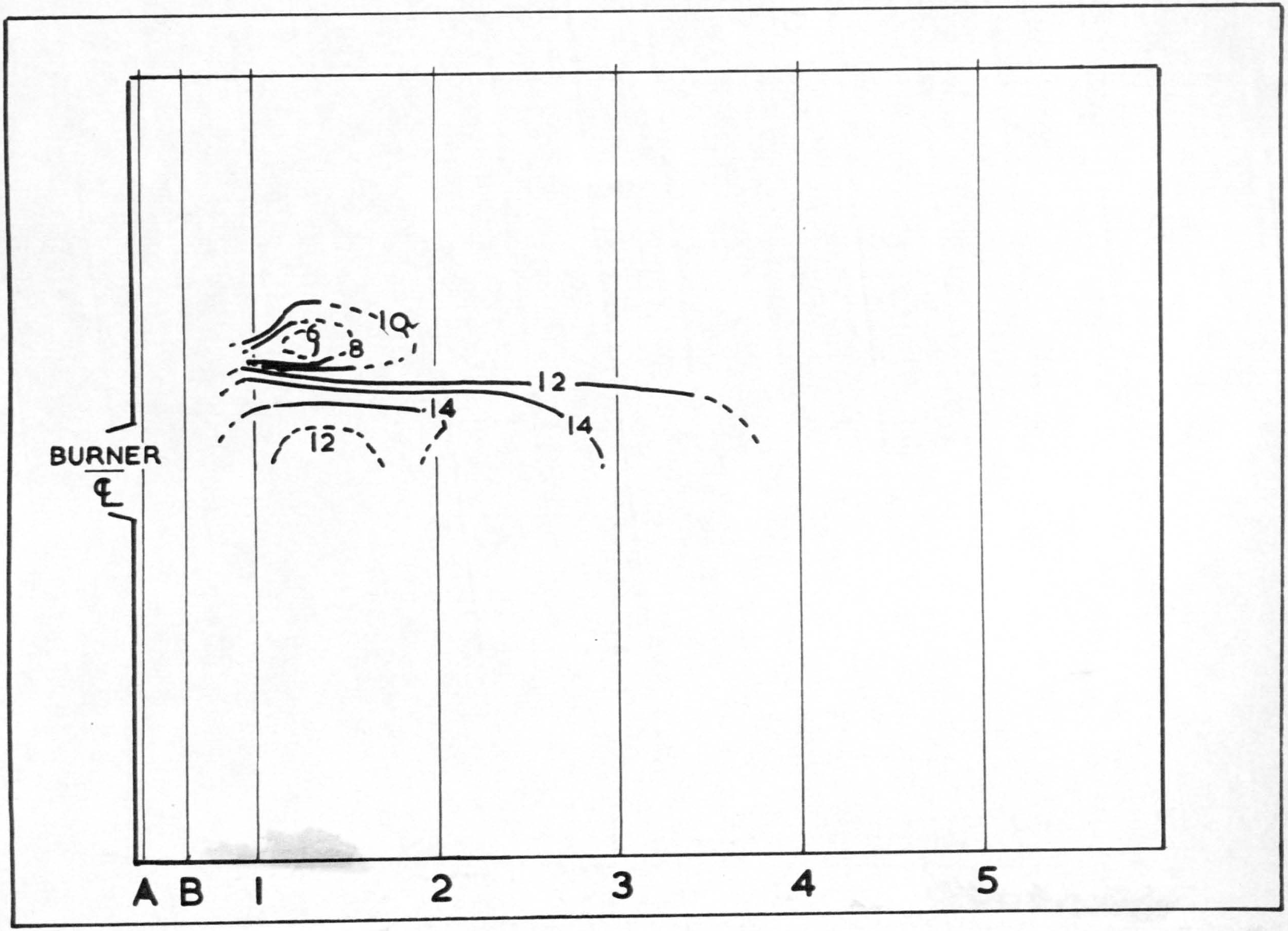



FIG. 49 b. % OXYGEN DISTRIBUTION - CASE I.

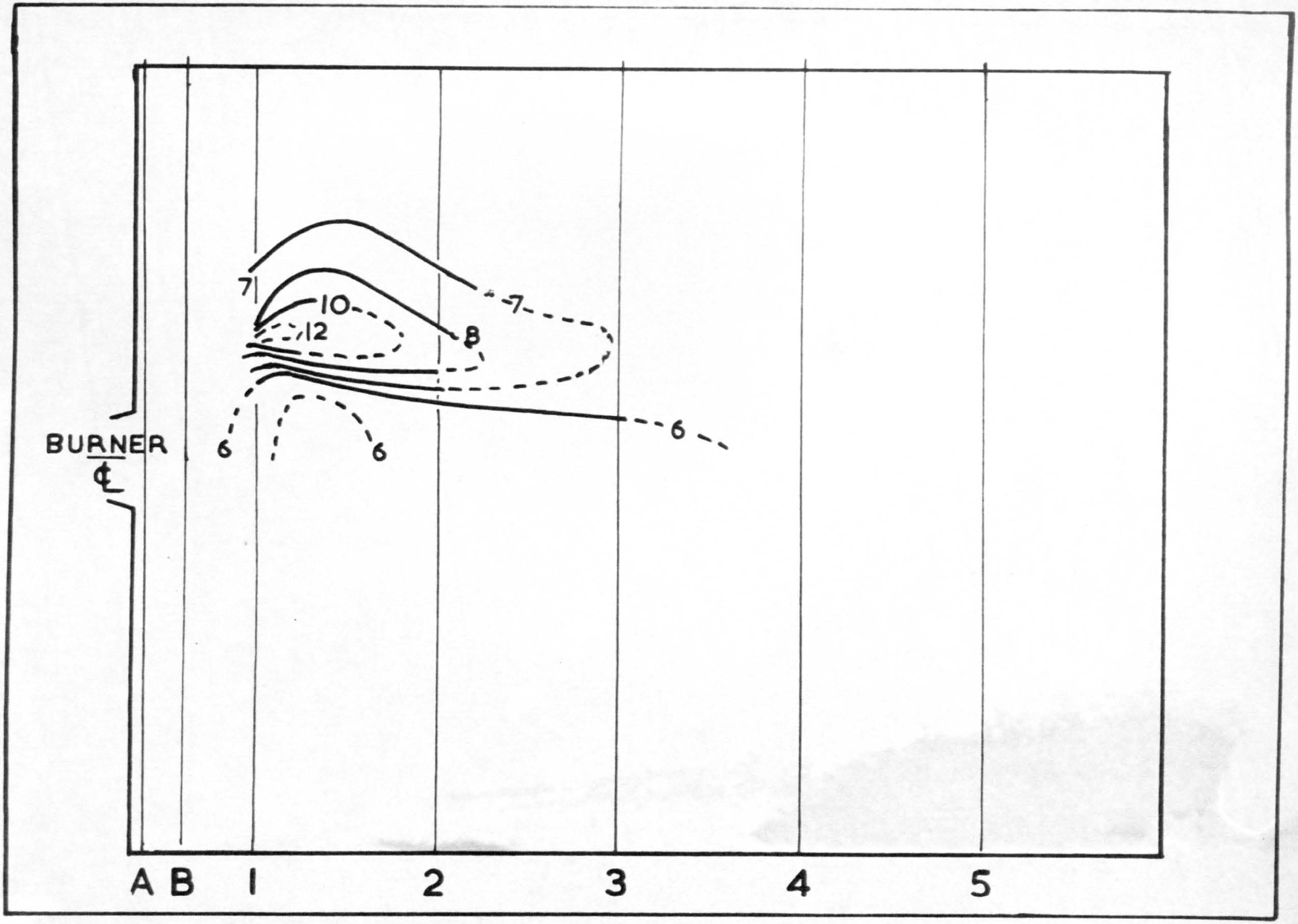



FIG. 49c. % CARBON DIOXIDE DISTRIBUTION - CASE 1.

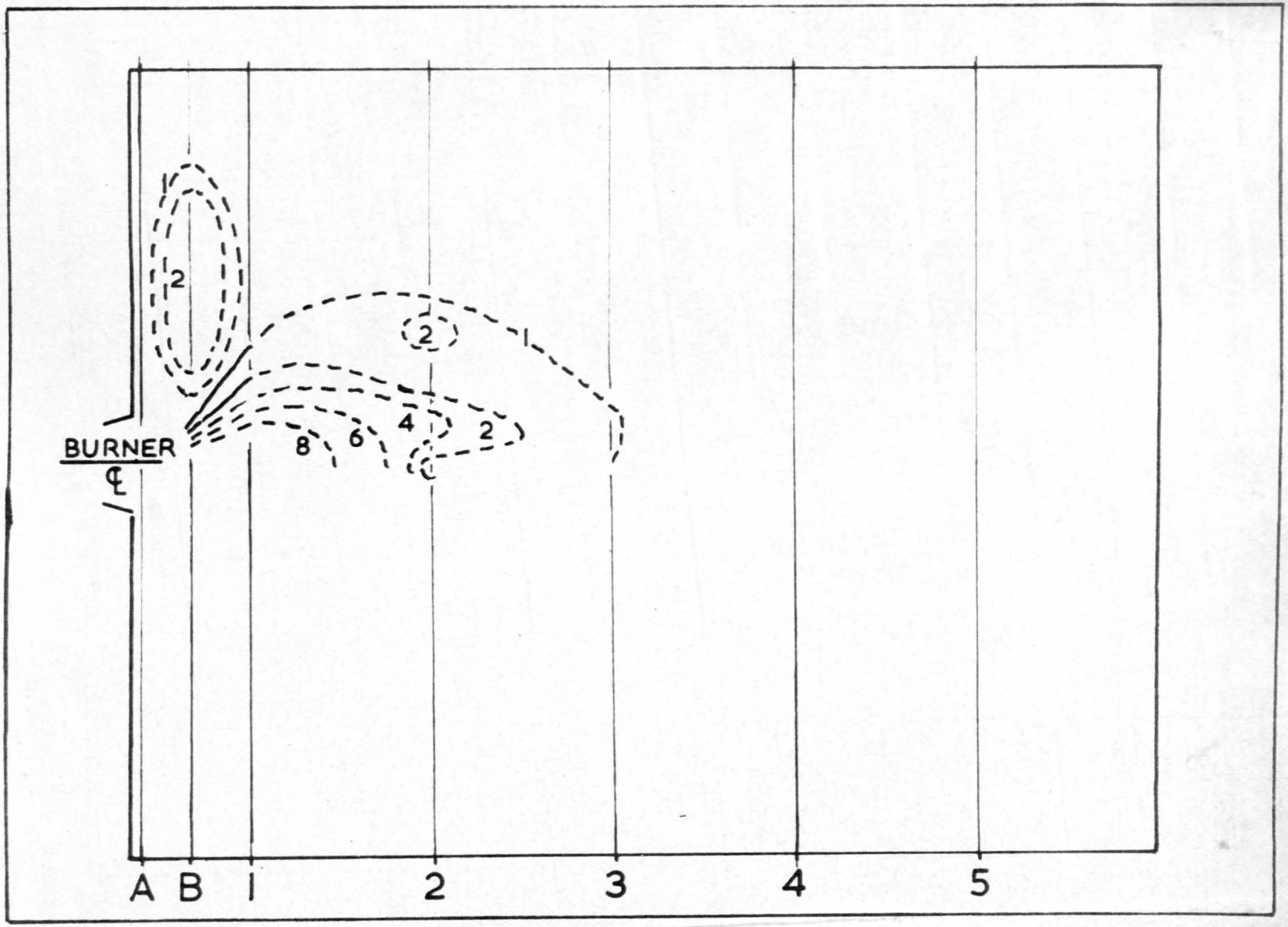



FIG. 50a. % CARBON MONOXIDE DISTRIBUTION - CASE 2

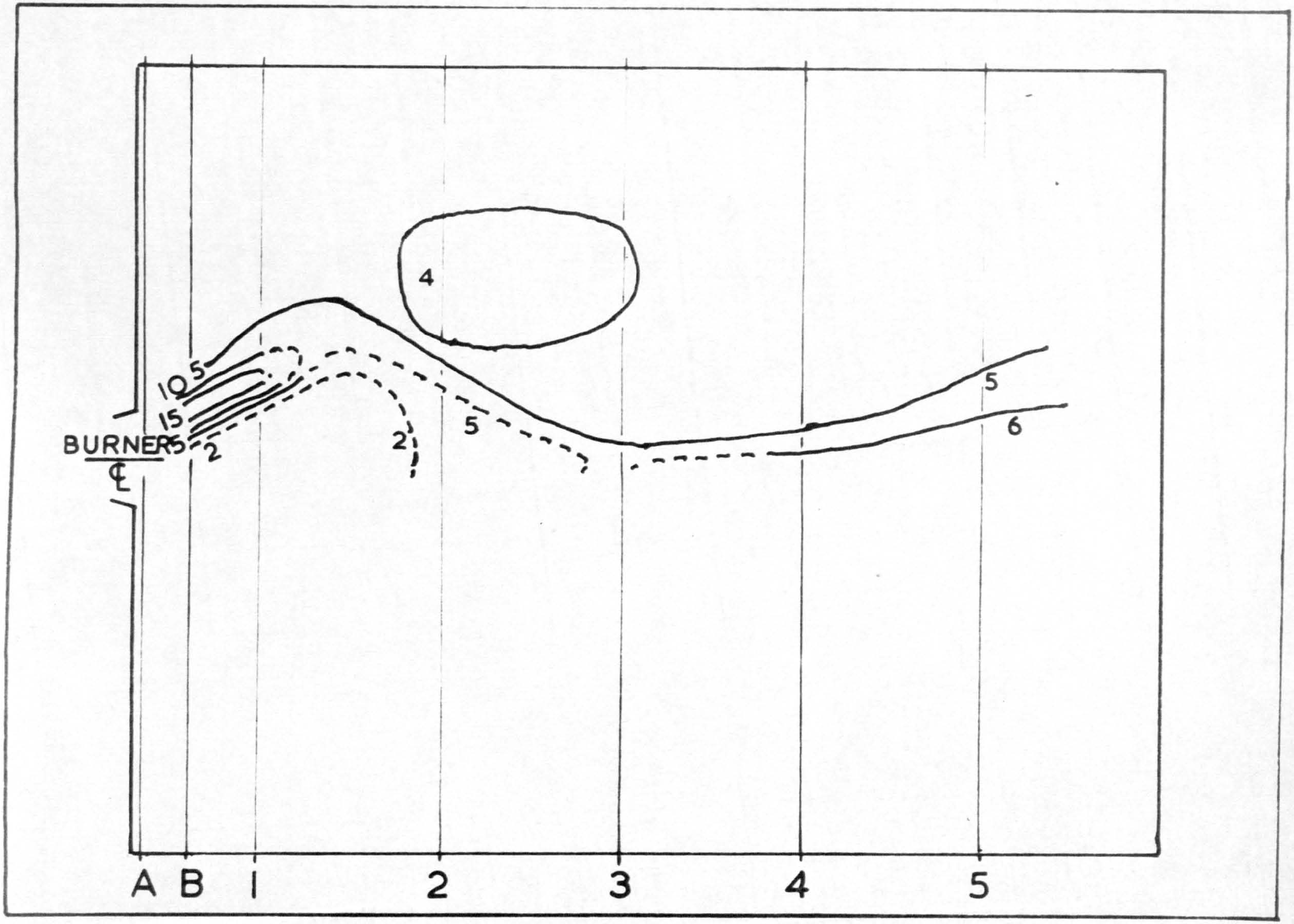



FIG. 50 b. % OXYGEN DISTRIBUTION - CASE 2

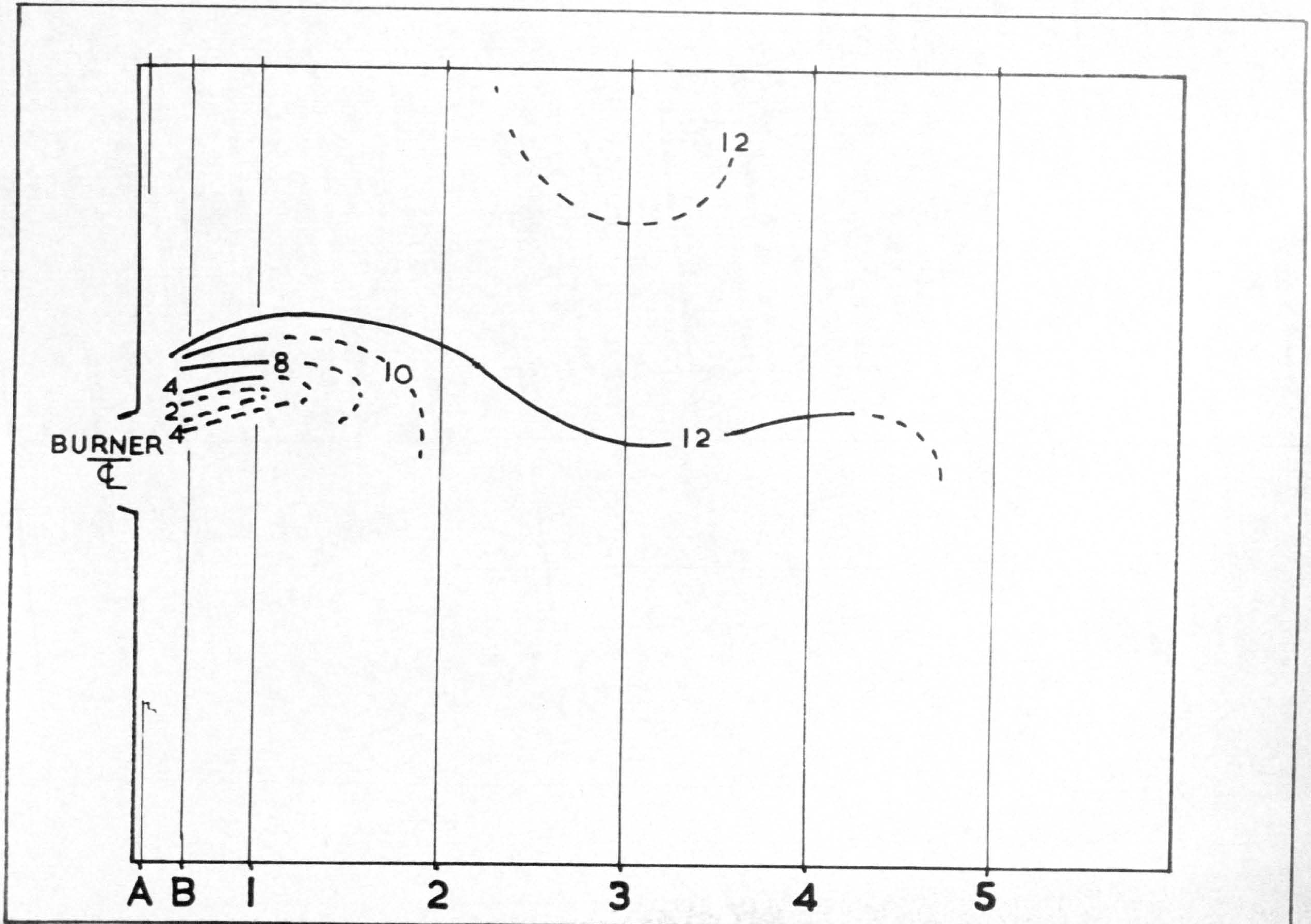



FIG. 50 c. % CARBON DIOXIDE DISTRIBUTION - CASE 2

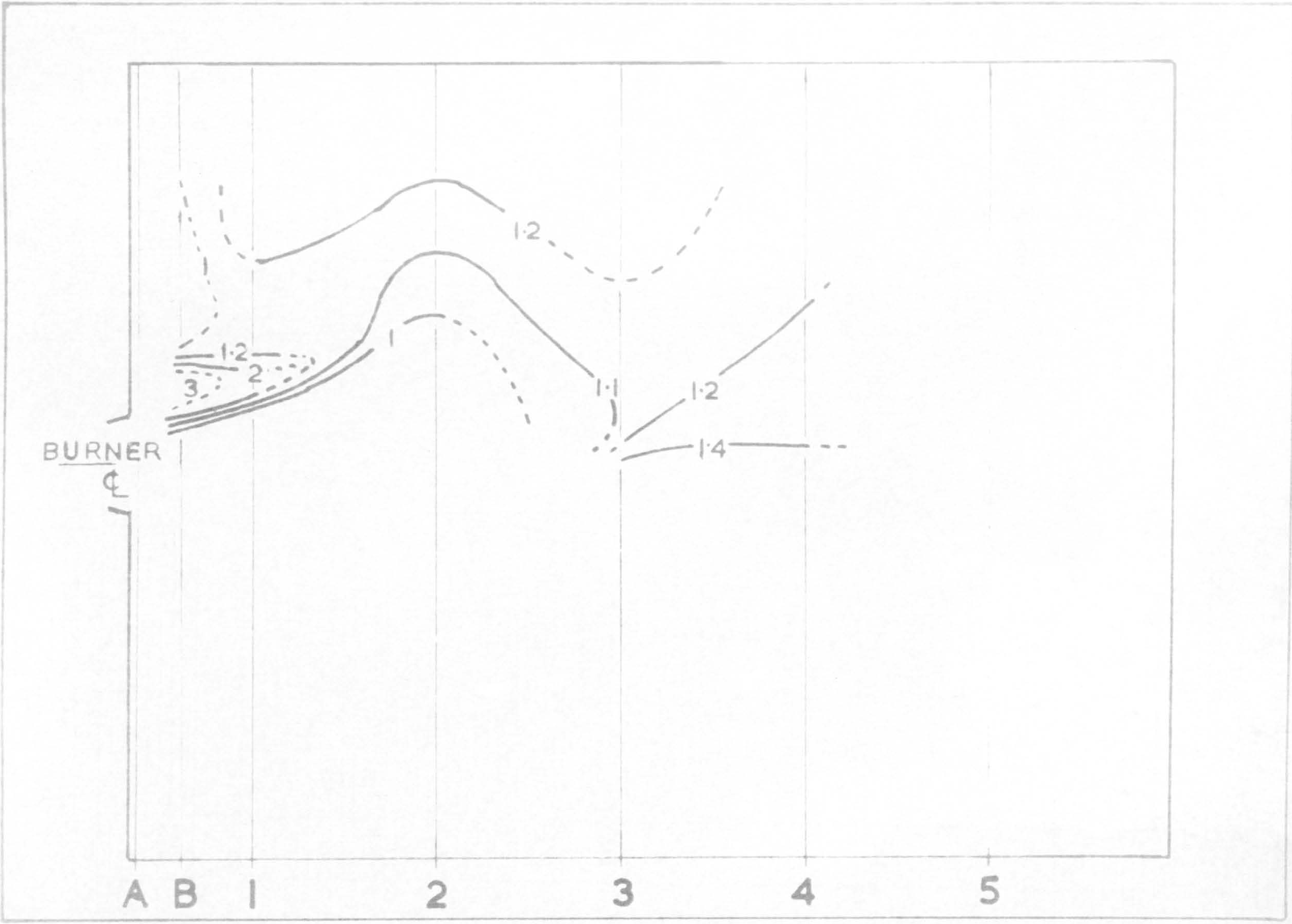



FIG. 50 d. PLOT OF EQUIVALENCE RATIO - CASE 2.

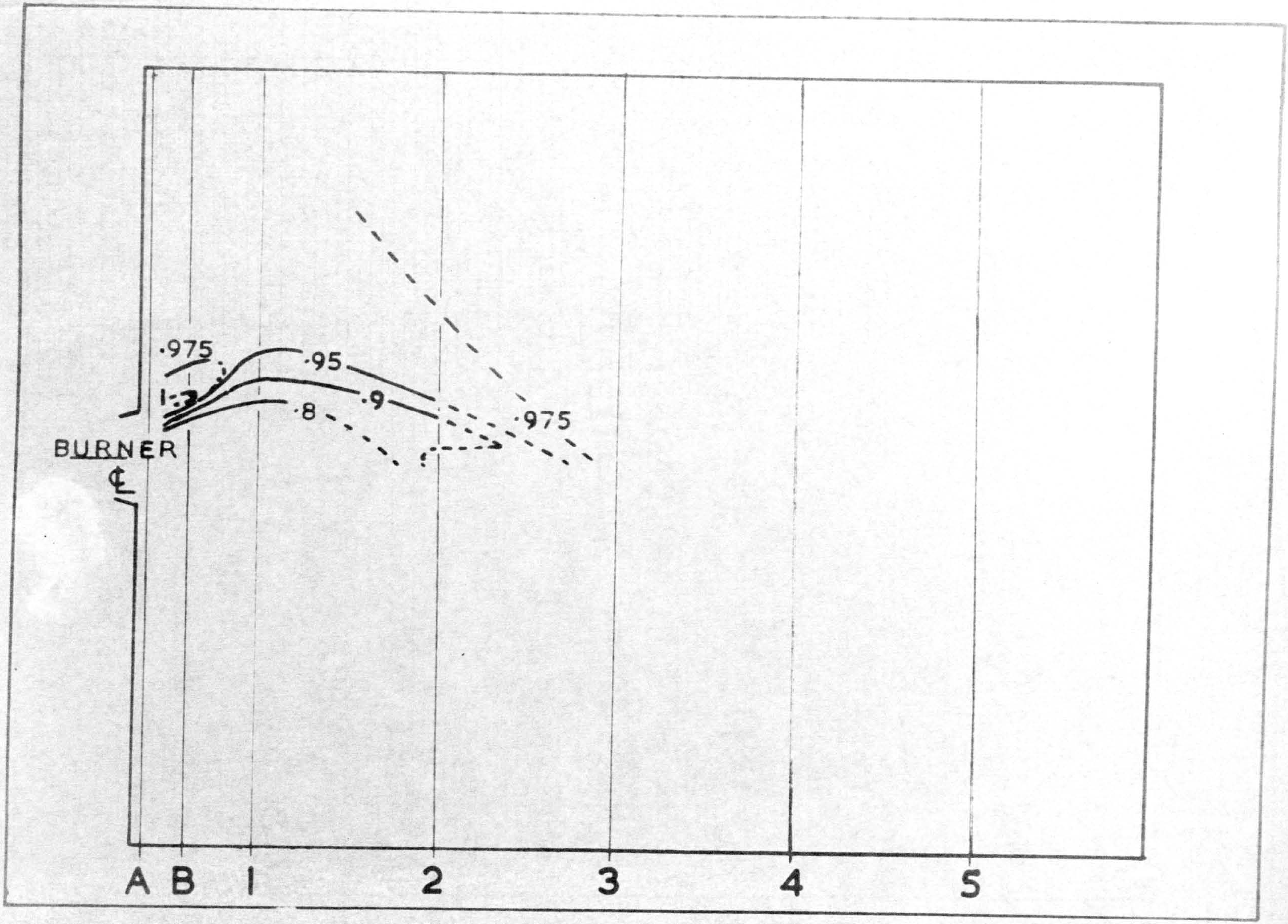



FIG. 50.e. PLOT OF COMBUSTION EFFICIENCY - CASE 2.

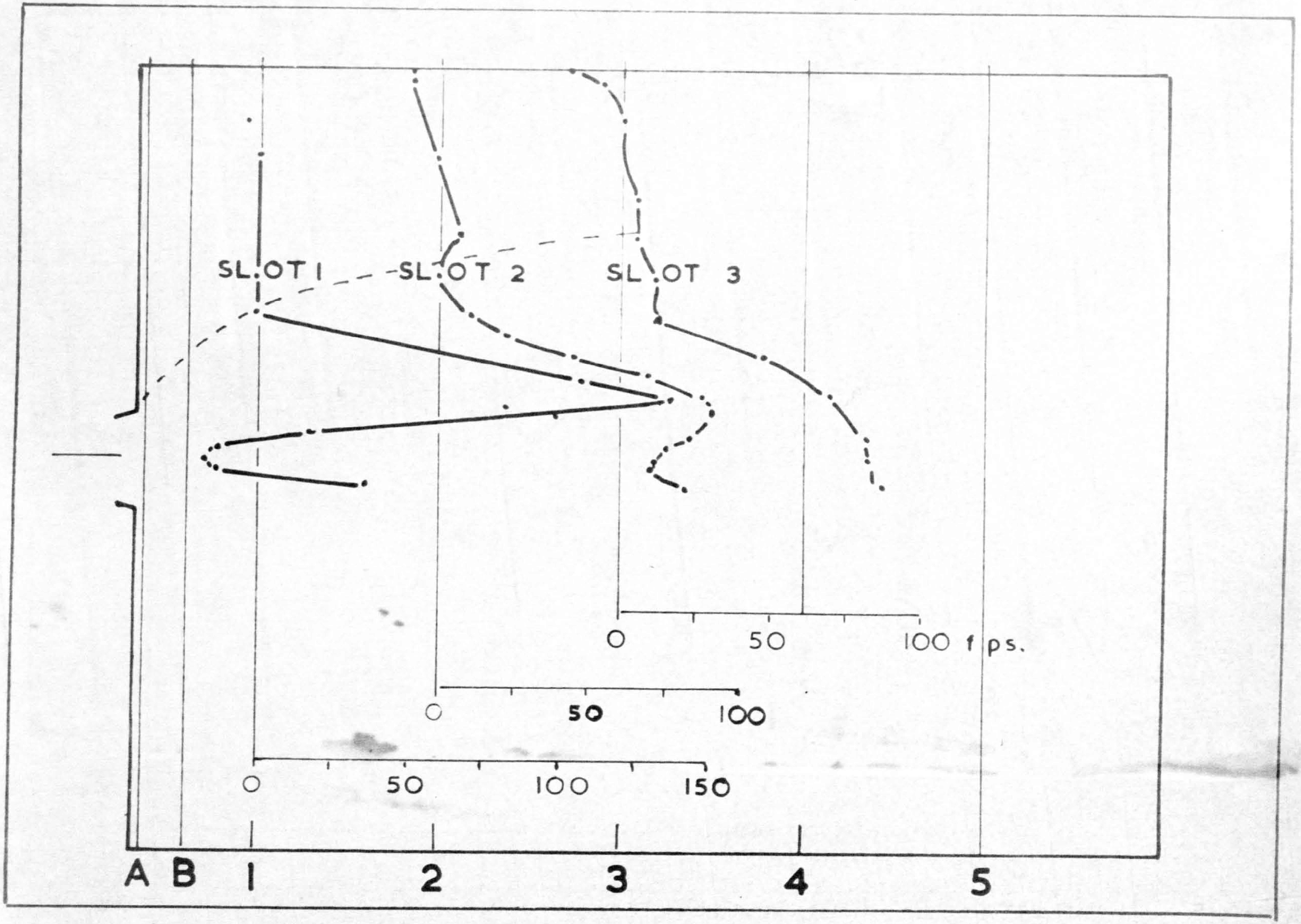



FIG. 51. VELOCITY PROFILES - CASE 1.

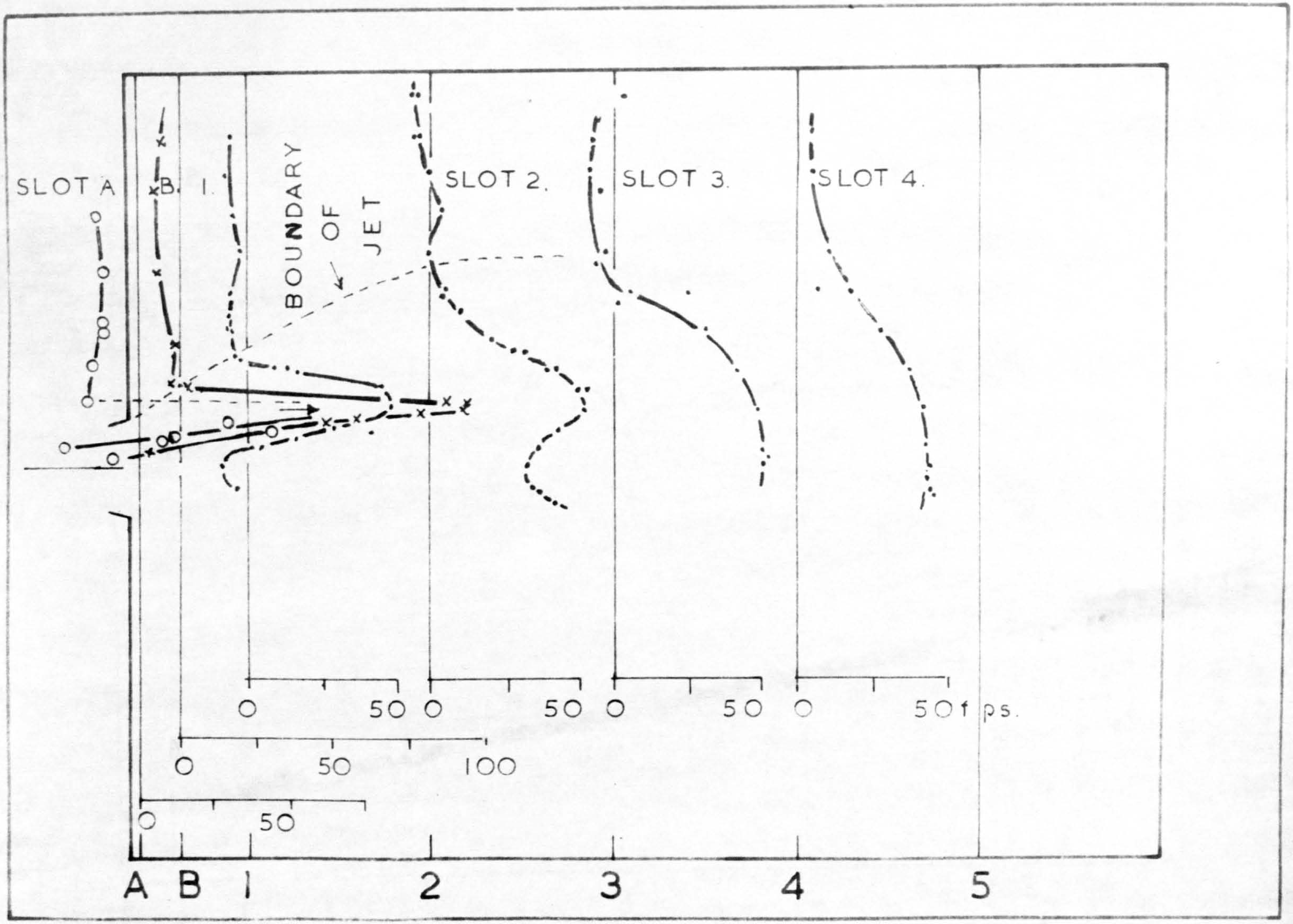



FIG. 52. VELOCITY PROFILES - CASE 2.

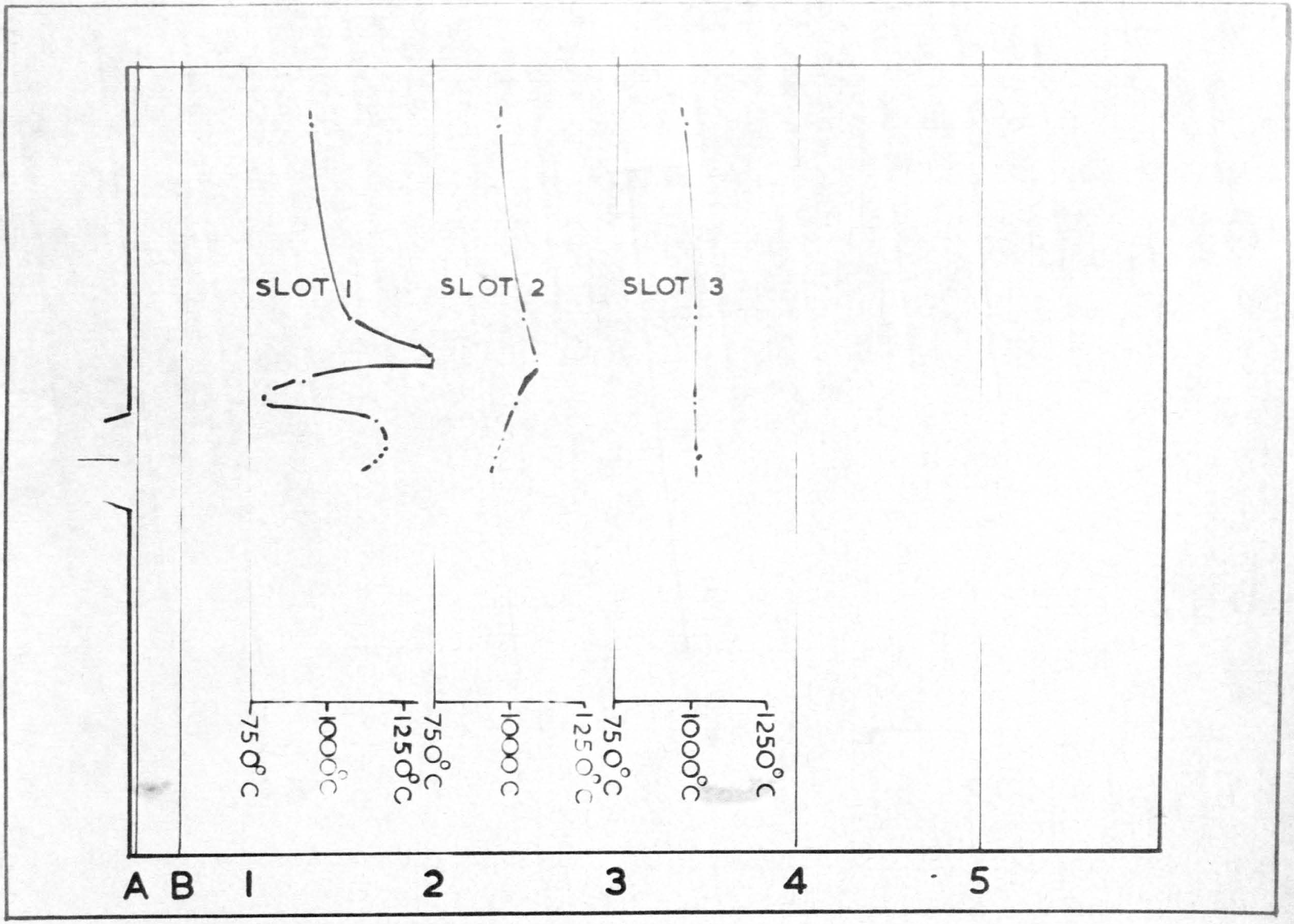



FIG. 53. TEMPERATURE PROFILES - CASE 1

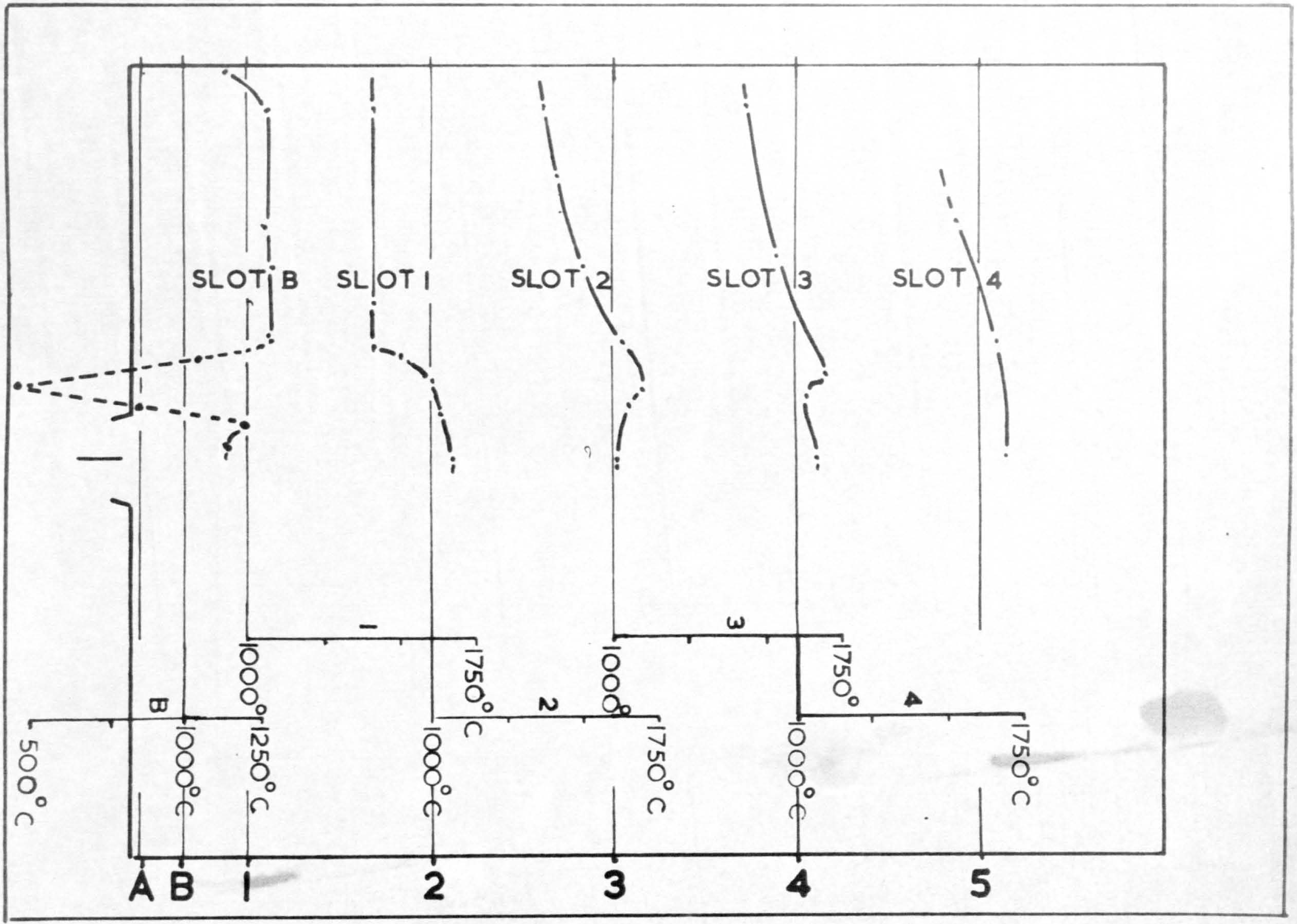



FIG. 54. TEMPERATURE PROFILES - CASE 2

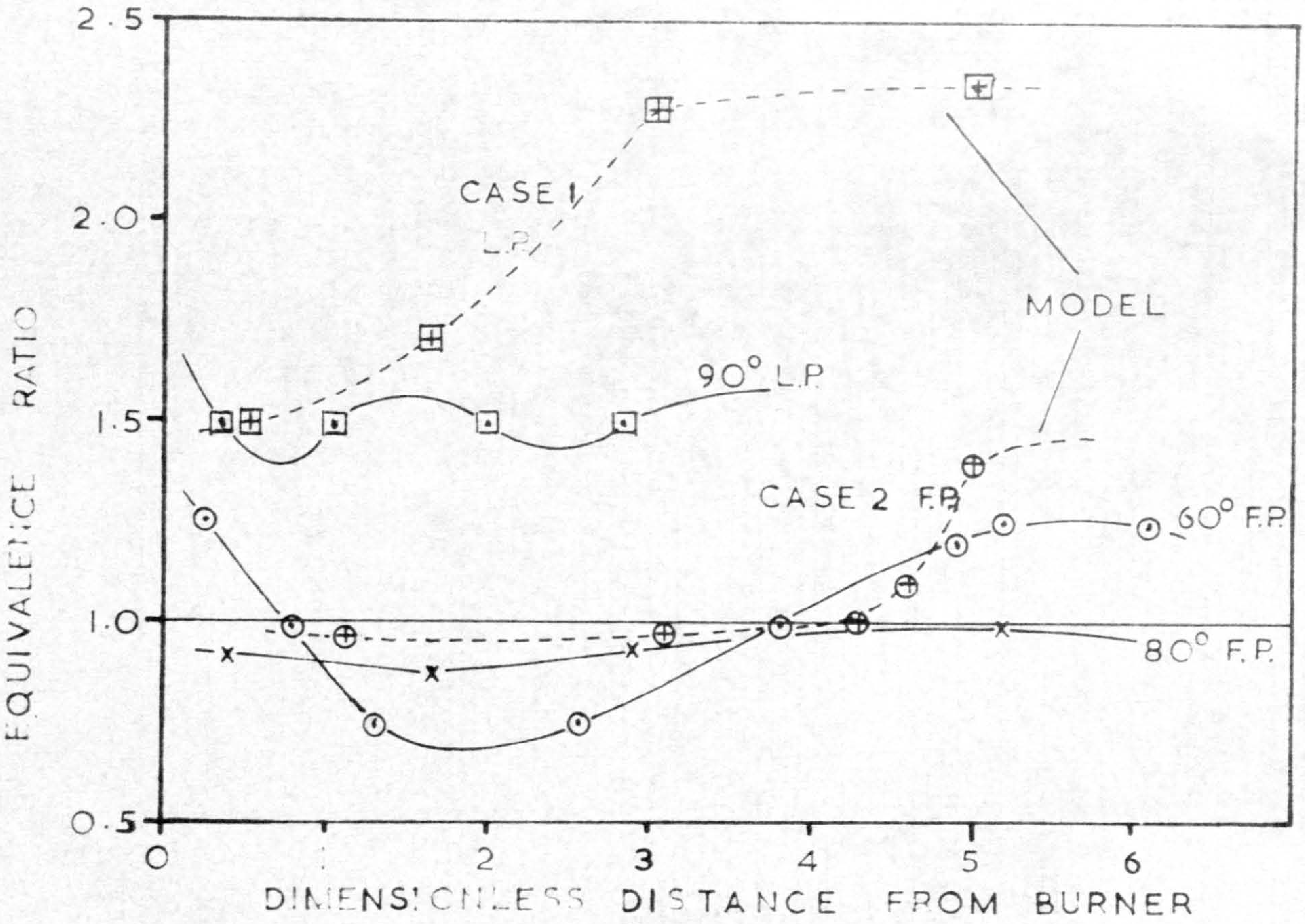



FIG. 55. VARIATION OF EQUIVALENCE RATIO ALONG THE AXIS.

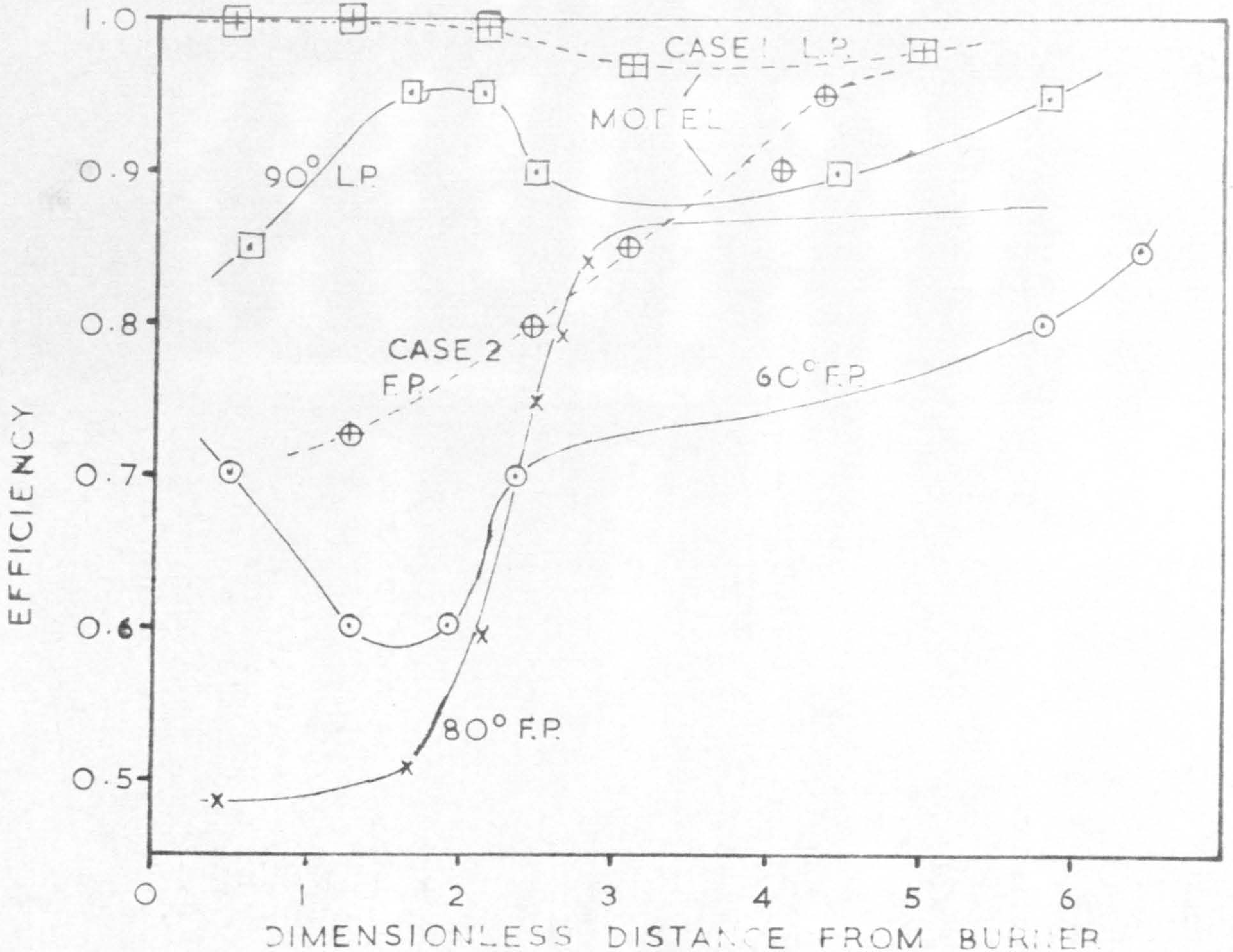



FIG. 56. VARIATION OF COMBUSTION EFFICIENCY ALONG THE AXIS.

FIGS. 57, 64, 66, & 67.

Legend

| | | |
|---|---|---|
| ⊙ | $60^{o}$ spray, full power | Full size results |
| ⊡ | $90^{o}$ spray, $\frac{1}{4}$ power | |
| ⊞ | $90^{o}$ spray using scaling criteria "case 1" | Model results |
| ⊕ | $60^{o}$ spray using scaling criteria "case 2" | |
| ┬ | High velocity | Cold model results |
| + | Low velocity | |
| x, ◊ | $G_{\phi}/G_{x}$ = 0 | IJmuiden results quoted for comparison. |
| o | $G_{\phi}/G_{x}$ = 0.06 | |
| □ | $G_{\phi}/G_{x}$ = 0.12 | |
| Δ | $G_{\phi}/G_{x}$ = 0.20 | |

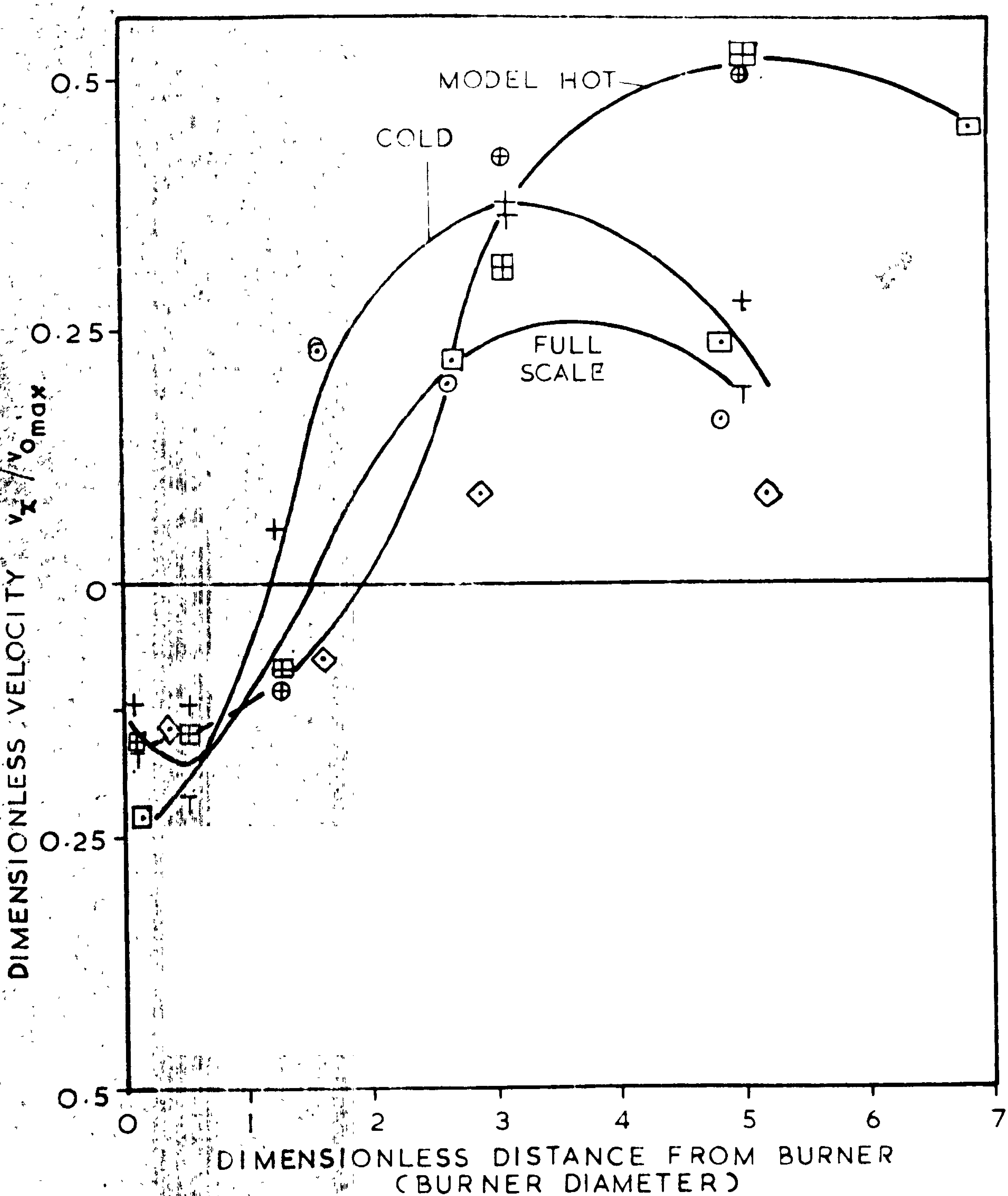



FIG.57(a) VARIATION OF VELOCITY ALONG THE AXIS FOR FULL SCALE AND MODEL RESULTS.

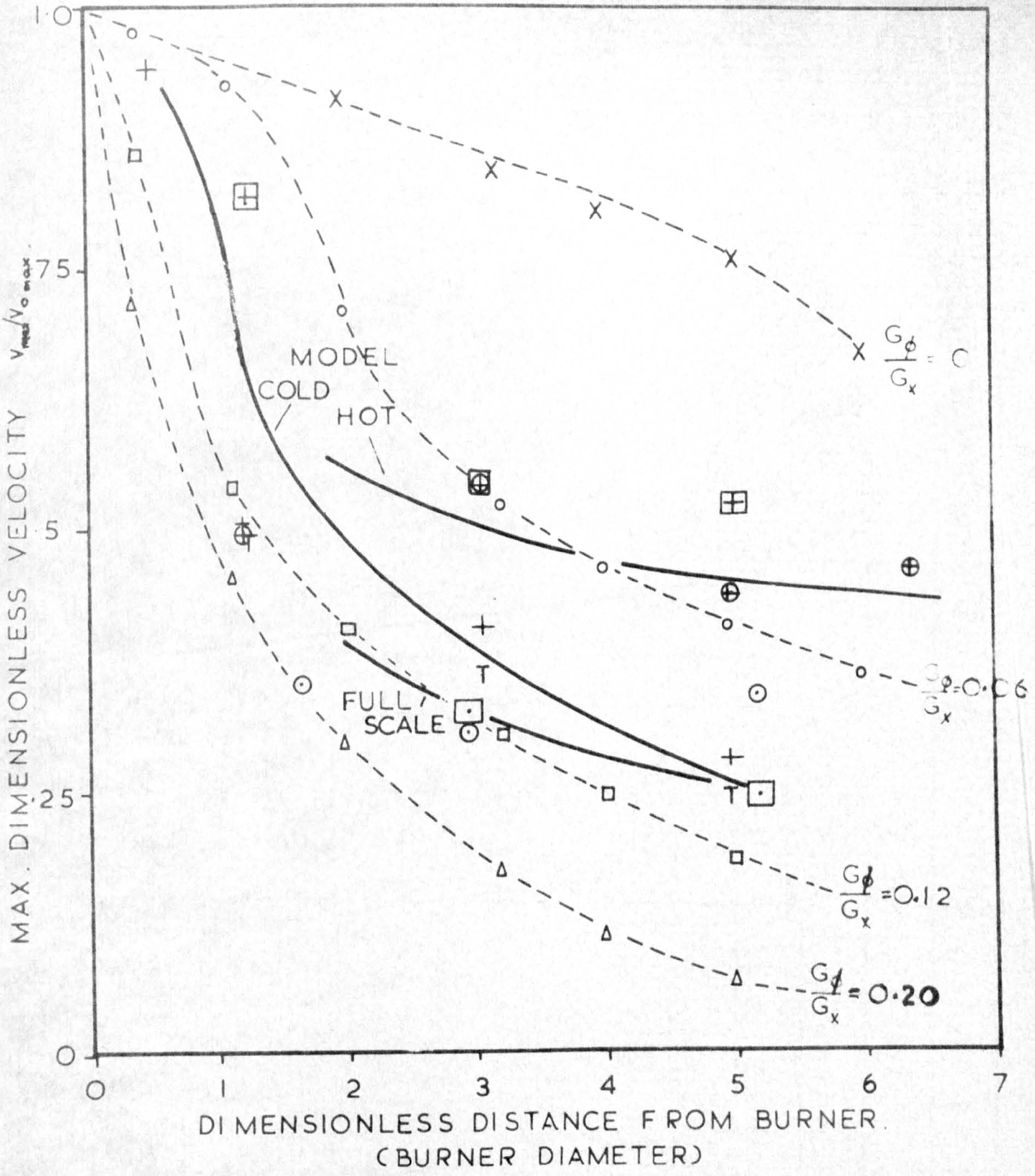



**FIG.57(b) PEAK VELOCITY DECAY FOR FULL SCALE AND MODEL RESULTS.**

(SOME IJMUIDEN RESULTS ALSO SHOWN FOR COMPARISON).

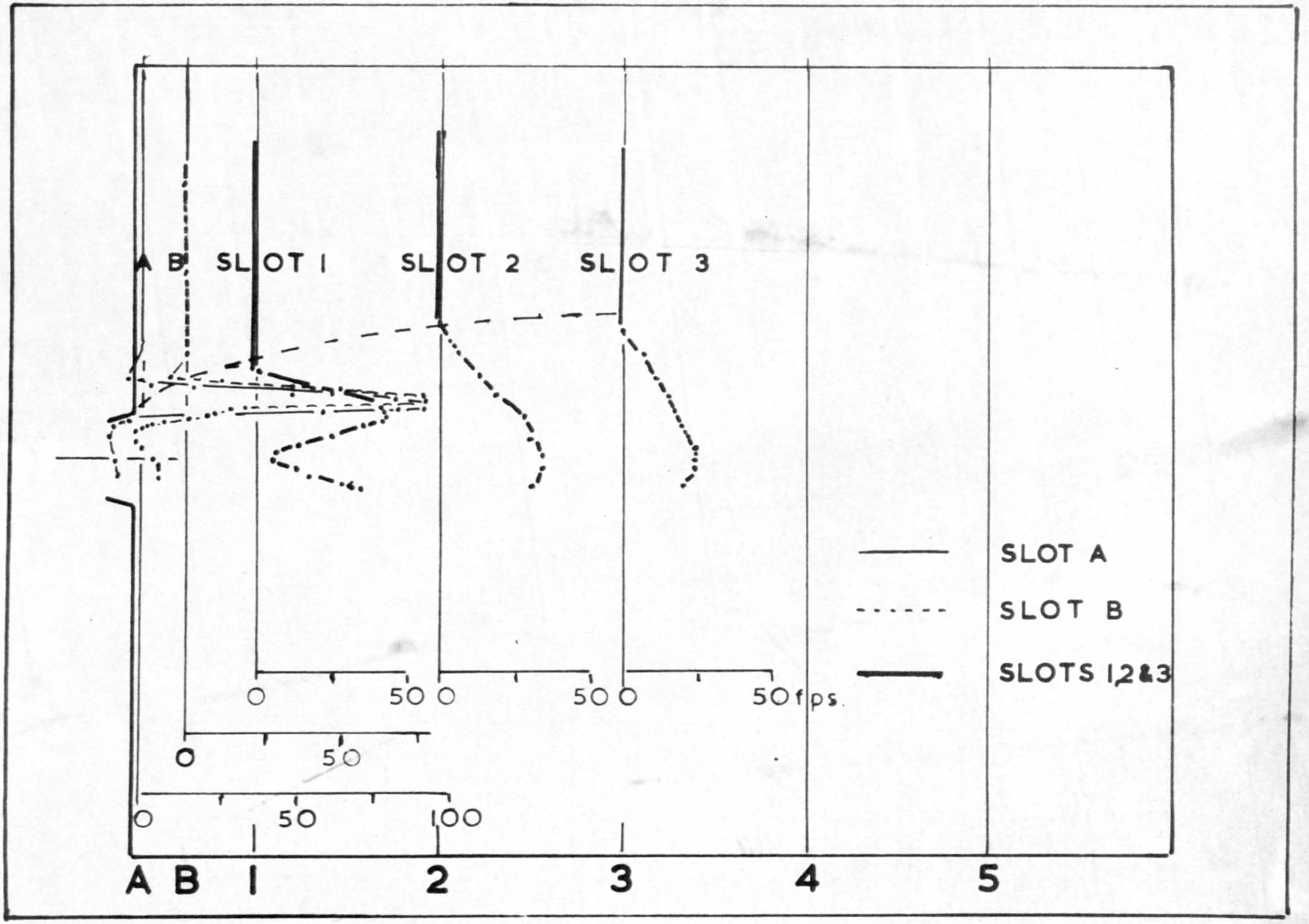



FIG. 58. COLD FLOW RESULTS - LOW VELOCITY.

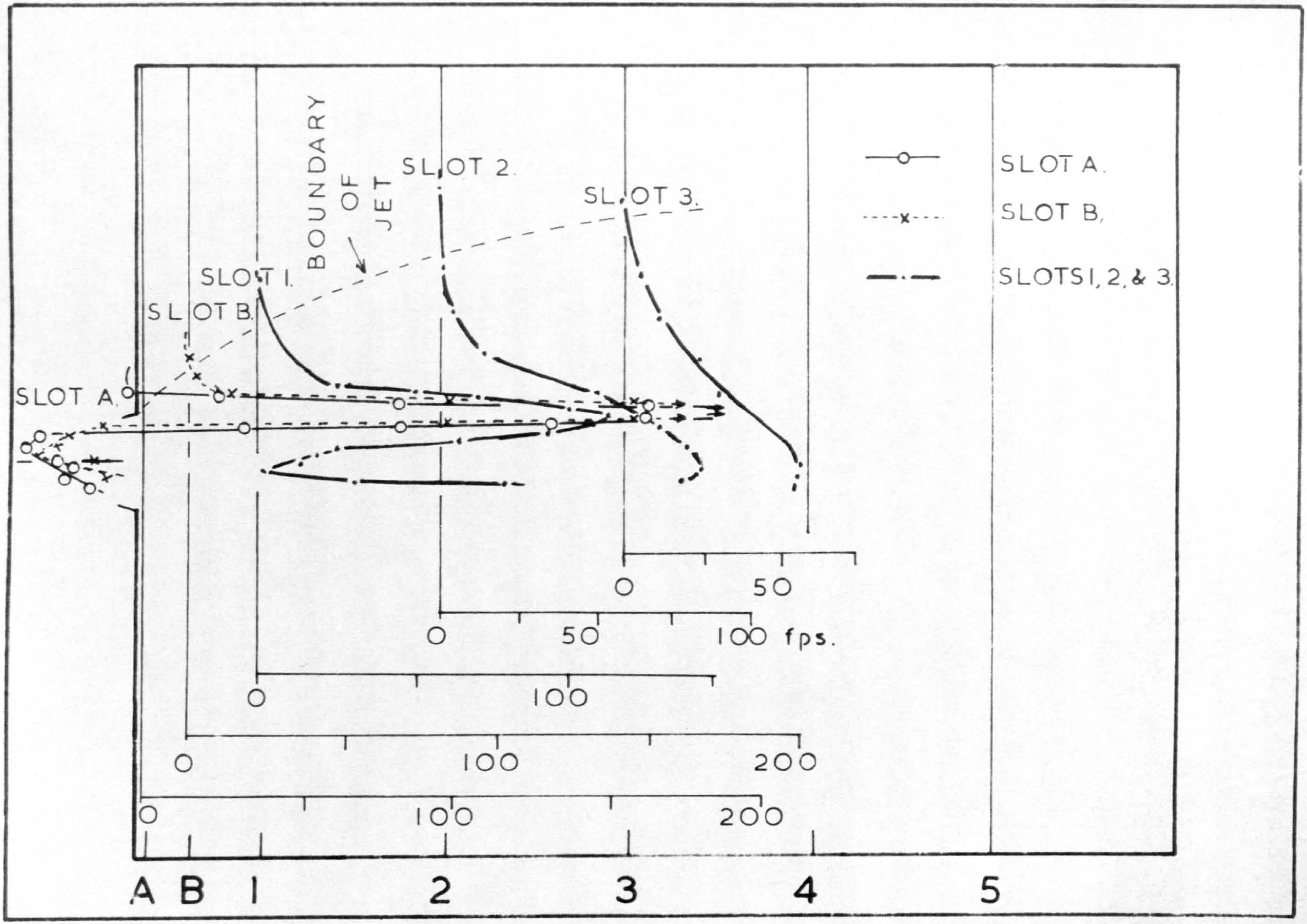



FIG. 59. COLD FLOW RESULTS - HIGH VELOCITY.

met. However, as these differences make only a 5% difference in the densities of the gases, this is not likely to be very serious.

### 5.4.5 Comparative cold flow work on model register.

Two cold runs were carried out on the model register in the combustion chamber to determine the differences between the hot and cold flow patterns. The velocity profiles are shown in Figs. 58 and 59. The first run was carried out at the same burner air velocity as the scaling case 2. run representing the full power condition. Hence burner Reynolds number was the same. The second run was at 2.35 times this velocity to give a reasonable change in velocity between the two runs.

The results showed that the central recirculation zone was notably shorter in length. The annular form of the earlier part of the jet was very similar in form to the hot runs, the radii of the peak velocity being almost identical. The annulus closed up nearer the burner than in the hot case as might be expected from the shortened recirculation zone. The radial spread of the jet boundary was also somewhat smaller. There were no significant differences between the two different velocities.

### 5.4.6 Experiments towards improved combustion systems.

The development of oil burners is not,at a fundamental level,a question of making small improvements to existing equipment. The development of improved systems is a sufficiently wide subject to warrant an extensive study on its own.

There was one set of useful experiments which could be carried out as a preliminary trial. This was to gauge the effects of upstream air injection from the back of the furnace with a more or less conventional burner. This was one method that, it was hoped, would be attractive in achieving low excess air combustion.

In these preliminary experiments which were carried out on the model rig, the conditions were similar to the second set of model experiments already discussed (i.e., the full-power condition scaled as case 2.). Upstream air injection was effected by a plain flat ended tube, which could be moved upstream from the back of the furnace on the axis of the burner. At this stage observation of the visible effect on the flame was considered sufficient to show if this method of secondary air injection was likely to be advantageous.

About 15% of the combustion air was introduced in the upstream direction with nominally about a third of the velocity of the air entering through the main register. Above about 7 burner diameters downstream there was a marked effect on the flame in reducing its length and producing a shorter, tidier flame than those obtained by high excess air figures through the normal register.

Unfortunately much nearer to the burner the effect of upstream injection was to spoil the flame stabilization and produced a very wide angled ragged flame. Something similar is sometimes experienced with the model burner even without upstream injection so that this may have obscured what on a larger scale might have been increased improved performance with injection nearer the burner.

The most satisfactory air injection seems to have been well outside the region where there is a low forward velocity or reverse flow in the centre of the flame. Presumably, the additional air serves to supply additional oxygen in the circulation round the flame making for much more complete combustion of the pockets of unburnt fuel known to occur fairly well down the flame. Further upstream, the air injection seems to have altered the aerodynamic character of the flame.

## 5.5. Measurement of Combustion Oscillations.

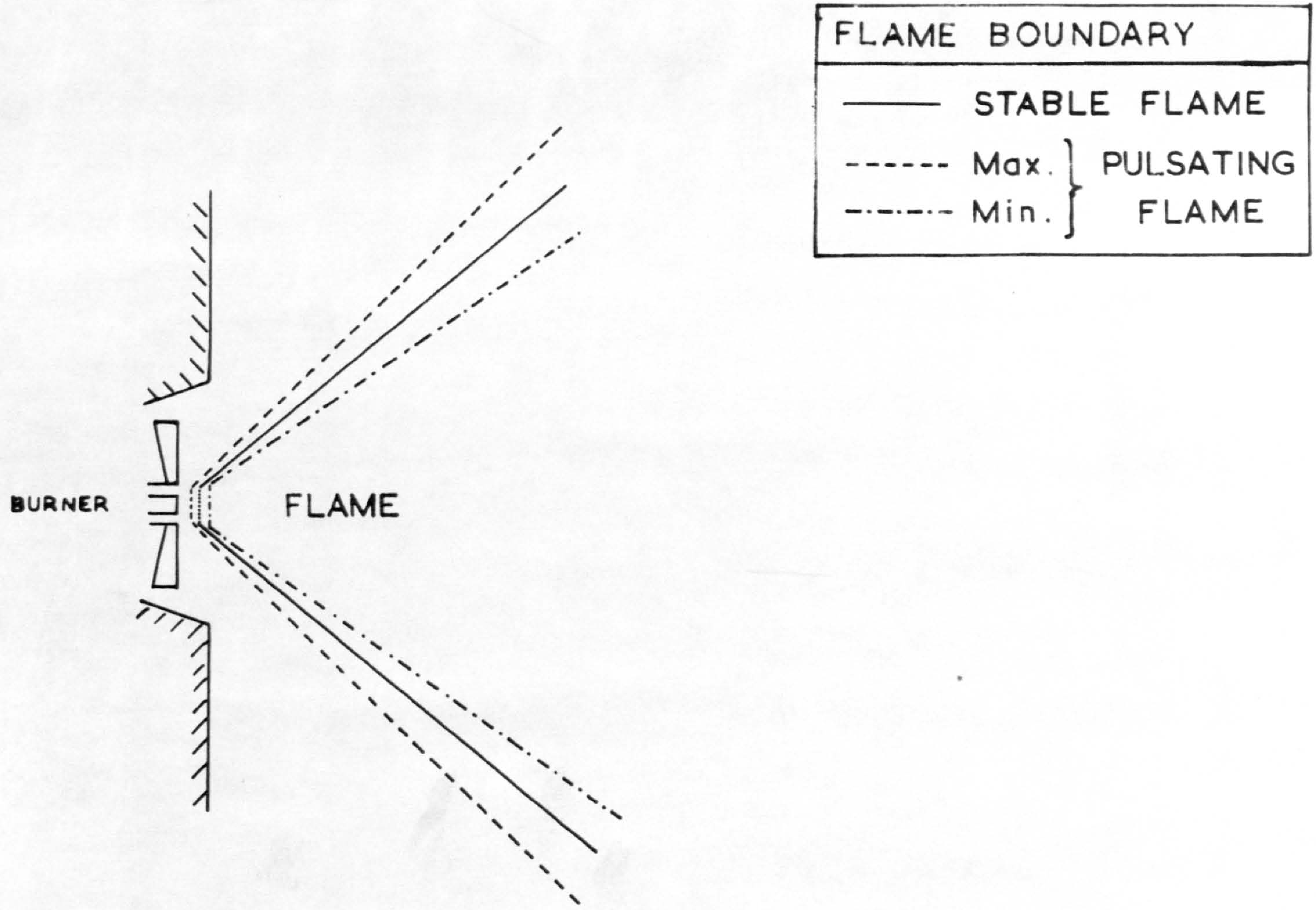
### 5.5.1 Preliminary Study.

The problems associated with unstable combustion have been discussed. At the time this study was commenced only somewhat uncertain descriptions were available about what actually happened when a boiler pulsated.

The preliminary work consisted in obtaining a 16 mm. cine record of the flame which was made to pulsate by reducing the air supply to about the black smoke point. For this work conditions were not otherwise controlled. Attempts were made to film the oscillating flame from the side through one of the sight holes and actually through the air register by filming the flame through the viewing hole provided in the register (a somewhat scaring operation for the camera operator!). To obtain a less obstructed view from the register, a polished metal mirror was placed in the air way and the register filmed from the side. The register used was the 12" type used for the experimental work. Several film records were obtained, mostly in black and white, but a few were obtained in colour, using Kodachrome stock. The use of a high speed black and white stock allowed the use of the higher speeds on the camera of 32 and 64 frames per second to obtain a somewhat slow motion record of the phenomena. These films were analyzed for frequency using a hand wound film editor which allowed examination of individual frames in the film and counting of the frames between each successive peak in visible flame (this was usually the point where the flame was blown back towards the register). The frequencies observed were found to be of the order of 10-20 c.p.s.

constant during the observed periods, but varying somewhat with conditions. The value of these measurements was in determining the order of the frequency of oscillation and in giving some visual record of how the pulsation cycle affected the flame.

It is not particularly easy to obtain a clear picture for reproduction of what is observed from the film. This has been reduced to a diagrammatic representation in Fig. 60. The results show that the flame has a fairly steady cone angle when burning stably which changes quite markedly during pulsation, being contracted from the steady position at one end of the cycle and considerably expanded at the other end. The approximate magnitudes of the changes measured from the film are given in the position of the cone as reproduced in the diagram, Fig. 60. This particular work was carried out with a single burner in a combustion chamber with a fairly low overall combustion intensity. As will be described later, this effect becomes even more pronounced in the case where several burners are firing together. The same effect was observed also in the shots taken through the register where the change in cone shape was equally visible. The effect was not sufficiently pronounced to blow any flame back through the register.




MOVEMENT OF FLAME BOUNDARY IN PULSATING PRESSURE JET FLAME.

FIG. 60.

It was observed that apart from low equivalence ratio, pulsation could be induced also by poorer mixing of the fuel and oil on an occasion when a boiler was fed with cool oil in error. This resulted in a consequent deterioration in the quality of atomization.

5.5.2 Quantitative Measurements.

In order to obtain data under conditions of the greatest current interest, quantitative measurements were made on a 3 drum Yarrow type boiler with two tube passes. This exhibited an extra phenomenon, apart from pulsation. This was a very low frequency oscillation which occurred under conditions of high excess air. In order to get a boiler of this type, it was necessary to carry out the work on one of H.M. ships. The instrumentation and boiler room arrangements have been described (see sections 4.3 & 4.5.4). The main problem was due to the fact that the boiler was an operating unit and it was necessary for the ship to be able to accept the steam at the level necessary for the measurements. This required co-operation with the navigating officer before changes in the firing rate of the boiler could be made. It had originally been intended to work these measurements in with black and white smoke limits measurements, but due to delay from burner faults, most of the measurements reported were made in the time left after the other work had been completed.

Combustion oscillations were induced by slowly closing the control valve for the air supply fan until the oscillations just started. The air level was then held at this point for as short a time as possible consistent with obtaining the measurements required, and the air rate then increased as quickly as possible. Communication between the boiler front and the instrument bay was maintained by portable telephone. Recordings of the oscilloscope trace were made during the steady part of the oscillation run.

The data obtained are plotted against fuel rate in Fig. 61. The frequency measured at the two different stations was the same, and as far as could be detected, these two records appeared to be in phase. Unfortunately, the characteristics of the microphone detectors measured the rate of change of pressure ($^{dp}/dt$) rather than actual values for differential pressure, and this last point was rather difficult to check except rather approximately.

The results show some very low values of frequency at low throughput rates, and a series of points with a slight positive gradient over the range of most of the measurements. Measurements were not taken right up to full power rating of the boiler in order to avoid the possibility of excessively high stresses on the boiler structure during pulsation.

Due to the burner fault, only five burners out of the possible six were in use during the period when measurements were made.

Gas sampling was used in an attempt to determine air/fuel ratios during oscillation, but difficulties were experienced in ensuring that the sampling procedure was consistent in the very limited time available to do this. More reliable relative data were obtained by measuring the apparent register draft loss (R.D.L.) just before oscillations commenced on the boiler room manometer. Values for R.D.L. could not be measured of course during oscillation due to the fluctuations of the gauge with the pulsations.* These measurements showed that except at very low powers the register draft loss at the onset of pulsation was independent of fuel throughput. The appropriate curve is superimposed on the black and white smoke limit curves for this particular boiler in Fig.62. Confirmation of this is obtained from the oxygen analysis on the stack gases shown in Fig.63. This shows an increasing oxygen figure at the limiting condition as the fuel rate was reduced.

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* Values for R.D.L. can only be regarded as relative,as the absolute value for flow depends on the flow characteristics of the register and the placing of the pressure tappings in the combustion chamber. It is consequently reported as R.D.L.

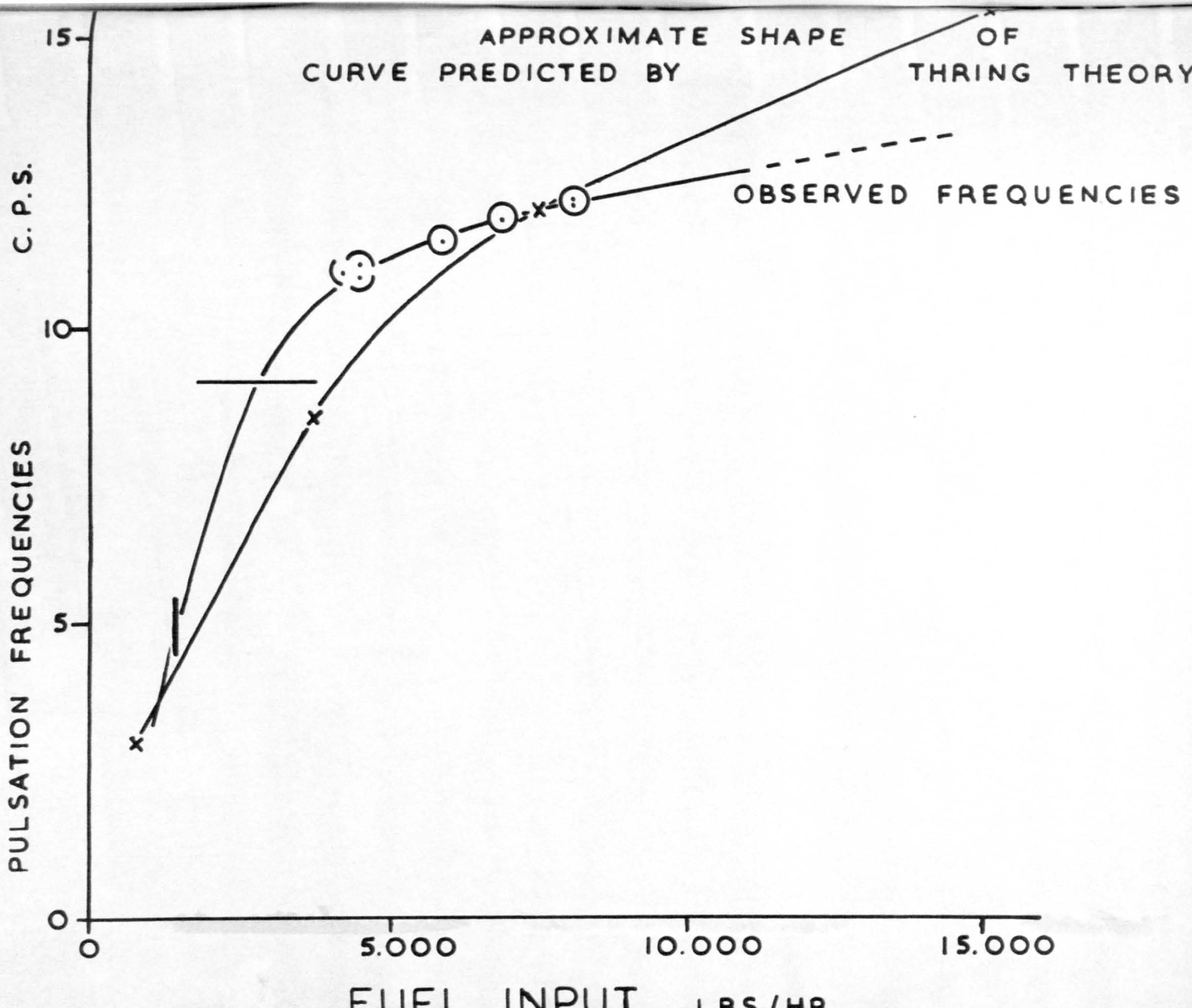
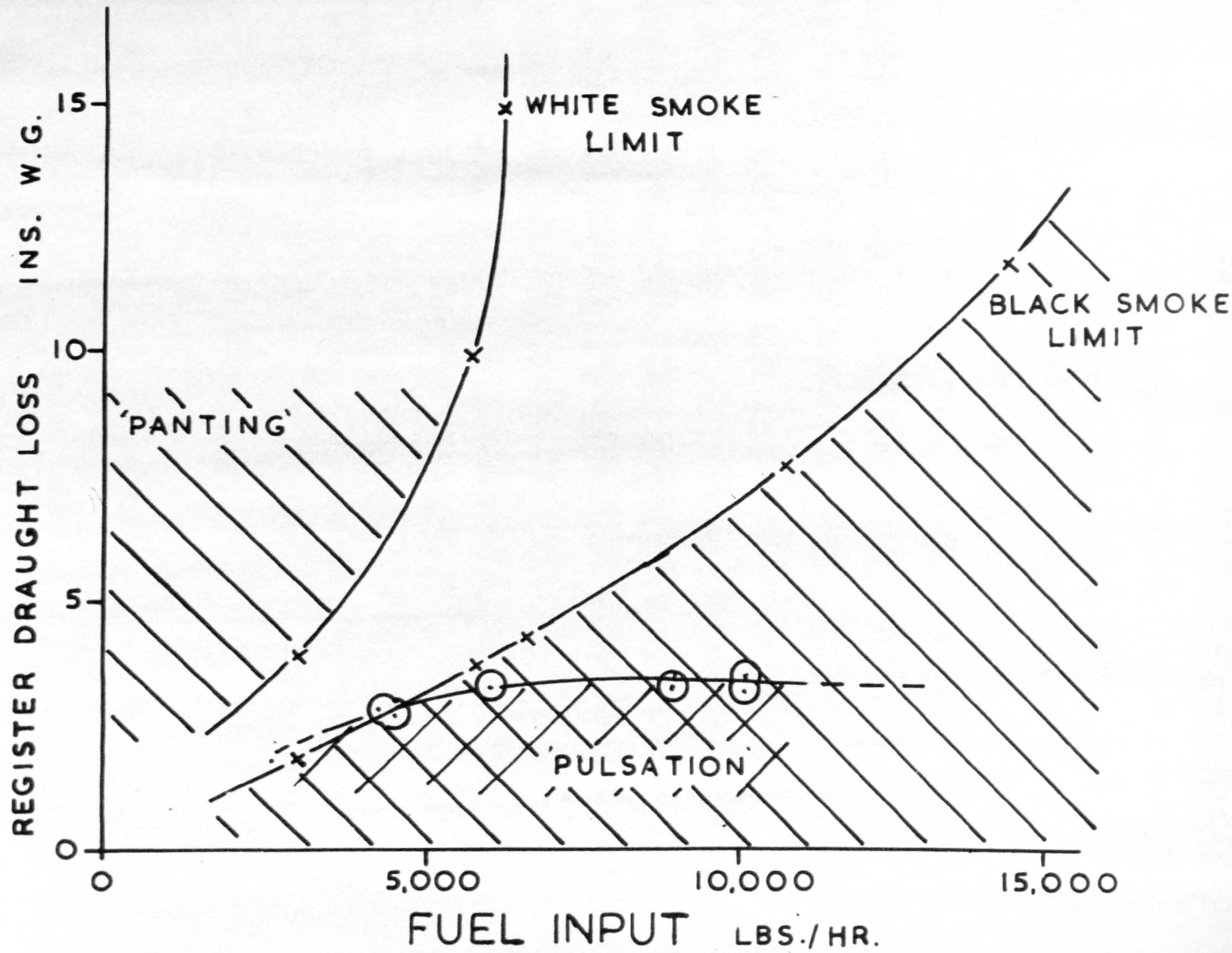



FIG. 61. PLOT OF OSCILLATION FREQUENCY AGAINST FUEL RATE.

FIG. 62.

Plot of Smoke and Pulsation Limits Against Fuel Rate.


WHITE SMOKE LIMIT
BLACK SMOKE LIMIT
'PANTING'
'PULSATION'
REGISTER DRAUGHT LOSS INS. W.G.
15
10
5
0
0
5,000
10,000
15,000
FUEL INPUT LBS./HR.

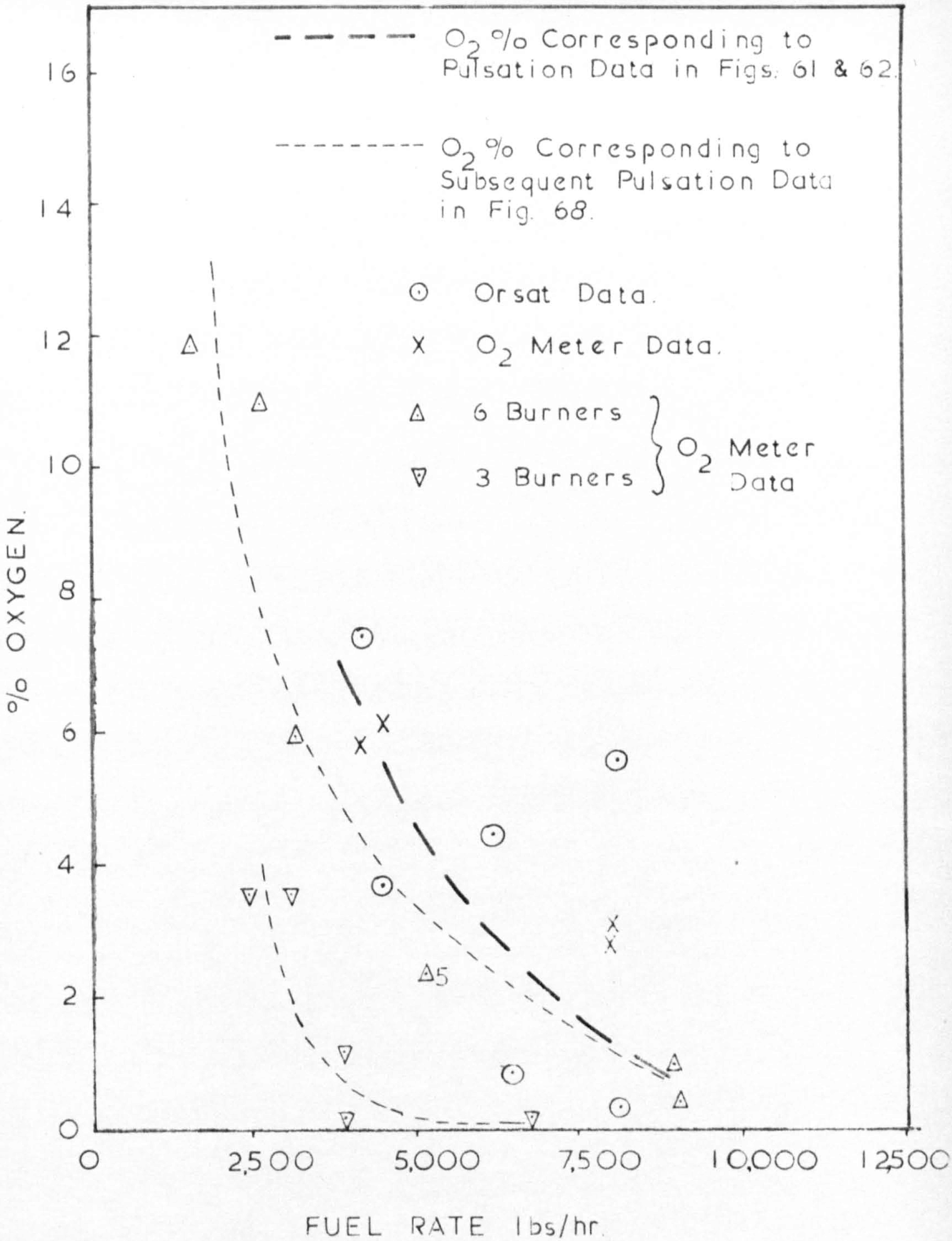



FIG. 63. PLOT OF VARIATION OF OXYGEN CONTENT IN FLUE GASES AT ONSET OF PULSATION.

A film record was made of a few of the pulsation runs using a water cooled periscope to which the cine camera could be attached. Like the single burner results described in section 5.5.1., the flame cone altered during the cycle. In this case, the change was much greater and the visible flame was pushed back practically against the wall in which the burners were mounted, and oscillated between this position, and approximately its normal shape during the oscillating cycle.

In addition to the measurements of regular pulsation obtained a few records were obtained under steady combustion conditions. These indicated that under these conditions that a certain amount of random noise existed. Although the frequency of this was not very well defined, it appeared to be about 5-10 times those of the oscillations measured, that is in the range 50-100 cps. There was no obvious direct connection between this noise and the regular pulsations measured.

# CHAPTER 6.

## Discussion of Results and Conclusions.

### 6.1 General features of the results.

#### 6.1.1 Relation of different plots.

The main question which remains, is the reliability and relation of the different results presented.

Allowing for the differences in sampling and burner locations, the two plots based on the $60^{\circ}$ full-power case show good agreement on the general features, where these overlap. Most of the differences occurred because of the different distribution of sampling sections. (See Figs. 40. and 41.) These show that results of this type are reasonably reproducible.

#### 6.1.2 Accuracy of results.

As far as possible, the results presented for gas composition and temperature are time mean values and for the velocity the r.m.s. value. The actual percentage error on individual readings will depend on the magnitude of the quantity in relation to the scale of the sensing device in use, the accuracy of calibration and so on.

Typical total fixed errors $w_f$ for each type of reading have been calculated from the sum of the squares of the individual errors, such as error of standard used, error in

measurement, error of calibration or correction used, etc.
Thus:-

$$w_f^2 = w_1^2 + w_2^2 + \ldots\ldots\ldots\ldots\ w_n^2 \ldots\ldots \quad (28)$$

where $w_{1,2 \ldots\ldots n}$ are the individual errors. For the gas composition results, the individual readings were reckoned to be a good deal better than $\pm$ 0.2% of the total volume. This would give an error in the equivalence ratio calculations of less than $\pm$ 2½%, except in a few awkward cases.

For temperature measurements, typical values for the results gave an error of $\pm$ 5% in the shape factor with a resultant temperature error in the range of temperatures measured of $\pm$ 20° C, and a consequent final error of about $\pm$ ½% in the density. For the velocity measurements, the error obviously will depend a good deal on the magnitude of the velocity being measured. In a good case, with a mid-scale reading on the recording instrument, the total errors in velocity should be better than $\pm$ 4%. In a poor case, at low velocities (say of the order of 10 f.p.s. under hot conditions), the figure might be as bad as $\pm$ 30%. This is taking into account errors due to fluctuations in readings as calculated from equation 27, and assuming an accuracy of $\pm$ 1% for the standard pitot used for calibration.

#### 6.1.3 Characterization of flames.

Ideally, it would be of considerably utility if the operating characteristics of the flame could be reduced to a single curve. Unfortunately, it is difficult to select data in such a way as to avoid ignoring some important feature. It seems necessary at the moment to make measurements across at least three of four different cross-sections in the flame, over the first four burner diameters, to ensure that no essential detail is lost.

### 6.2 Performance of flames investigated.

#### 6.2.1 The effect of fuel distribution.

The results from both the full-size and model experiments reported in the last chapter, showed that the fuel distribution changes markedly for different flames, even though only the fuel in the gaseous phase was measured.

The main conclusion was that the most critical features of the flame were affected strongly by the sprayer characteristics. In the system under investigation, the combustion pattern changed from 'burning inwards' with a high unburnt fuel region on the axis with the $60^{\circ}$ sprayer, to almost a 'burning outwards' system where a lot of the unburnt fuel was right on the fringe of the jet altogether for the low power condition. These results are in contrast to those of Beer[56], where the spray angle seemed to be relatively unimportant.

The difference in results is probably due to the burner aero-dynamics which in Beer's case should lead to a flame that was 'burning outwards' from a region of high oxygen content on the core, in all the instances investigated, so that the change in spray angle did not alter materially the relative dispositions of air and fuel.

It is obvious too from these results, that the main problem is not burning most of the fuel, but the last fraction. Thus, the combustion efficiency results show at least 50% of the fuel present was actually burnt at any point where measurements were made and more usually 80-90%. The real problem came from the combustion of the remaining percentage as this had usually by then reached a region of low oxygen concentration and regions of 'fuel escape' have been noted in these results on the fringes of the flame and downstream on the axis. It must be remembered that the measurements reported reflect the distribution of the gaseous component of the fuel only, although these are considered as representing fairly the distribution of combustible material. As significant amounts of fuel escape outside the jet, it follows that this must have been due to penetration by the fuel while still in the form of a liquid spray. It must be supposed, with the high air figures recorded, in the low power case that a considerable amount of the fuel spray must have crossed the main air stream.

Measurements made with a three-dimensional pitot suggest initially, that the fuel and air had approximately parallel cones. Once clear of the burner, the peak velocity region remained parallel to the axis, although the jet boundary still expanded for 2-3 diameters. This means that the fuel and air tended to "cross-over" fairly early on, probably about one burner diameter down stream from the quarl.

It is obvious that large changes in angle occurring over the operating range must be regarded as highly unsatisfactory. These results seem to show that for any given aerodynamic arrangement there is only likely to be one best fit spray angle. The use of air or steam assisted atomization would probably bring a big improvement in this direction.

### 6.2.2 Mixing.

The earlier stages of the mixing process have been touched on already. Here, the main body of the jet leaving the quarl is crossed by the oil spray. Much of the fuel must be entrained either as vapour or as fine droplets, leaving the larger droplets to penetrate beyond this region. In the case of the smallest spray angle, it is obvious that the jet entrains more than it can burn and large quantities of unburnt fuel appear in the central recirculation zone and on the axis further down stream. This is reduced for the $80^{\circ}$ spray and very little fuel appears at all in the central recirculation region.

for the $90^{\circ}$ spray. This is reflected in the plots of combustion efficiency on the axis in Fig.56.

Once clear of the region where the fuel distribution is dependent on the liquid spray, the mixing between the remaining fuel and the air will depend on turbulent mixing processes. This will result either in reentrainment from outside the flame into the early part of the jet or by the relatively slow process of turbulent diffusion, where the unburnt fuel region persists along the axis of the flame.

The equivalence ratio plots in Figs. 41.d, 42.d, 43.d, also demonstrate the regions where mixing of the available air and fuel is incomplete and shows the spreading effect of increasing the spray angle. Outside the flame this is particularly noticeable where the annular air jet leaves the burner. Here it takes quite a long distance for the air jet to entrain recirculated combustion products and this process is usually incomplete before the flame is encountered. This poor mixing, results in a considerable depression in furnace temperature in the early part of the air stream, see Figs. 47 and 48, which must have a very substantial retarding effect on the rate of combustion and evaporation of the fuel spray as it crosses this.

It was not possible to distinguish differences in mixing between the effect of high velocity, as opposed to high velocity gradient in the earlier part of the flame due to the "cross over" already described, resulting in the fuel crossing two regions of high velocity gradient. The pockets of unburnt fuel do, however, seem to occur away from regions of high velocity gradient but in the case of the axial zone where the velocities are quite high, this may be due only to the remoteness from the available oxygen.

### 6.2.3 Recirculation.

A plot of the bulk mass recirculation flow expressed as the ratio $m_r/m_o + m_a$, is given in Fig. 64. for the different cases investigated. This mass flow rate covers external recirculation round the flame. These data show that the maximum recirculated flow is about twice the initial mass flow from the burner. This is well short of the four times the initial flow quoted in the literature as the optimum (see Section 2.2.1.5). Furthermore, this maximum figure occurs about 3 burner diameters downstream. Thus, the contribution that the recirculated products make to the preheating of the air in the early stages of combustion is fairly small. As has been mentioned already, this is demonstrated by the low temperatures recorded in the air-jet near the burner.

The internal recirculated mass flow is very small by comparison, approximately a $\frac{1}{3}$ of the forward mass flow, but as

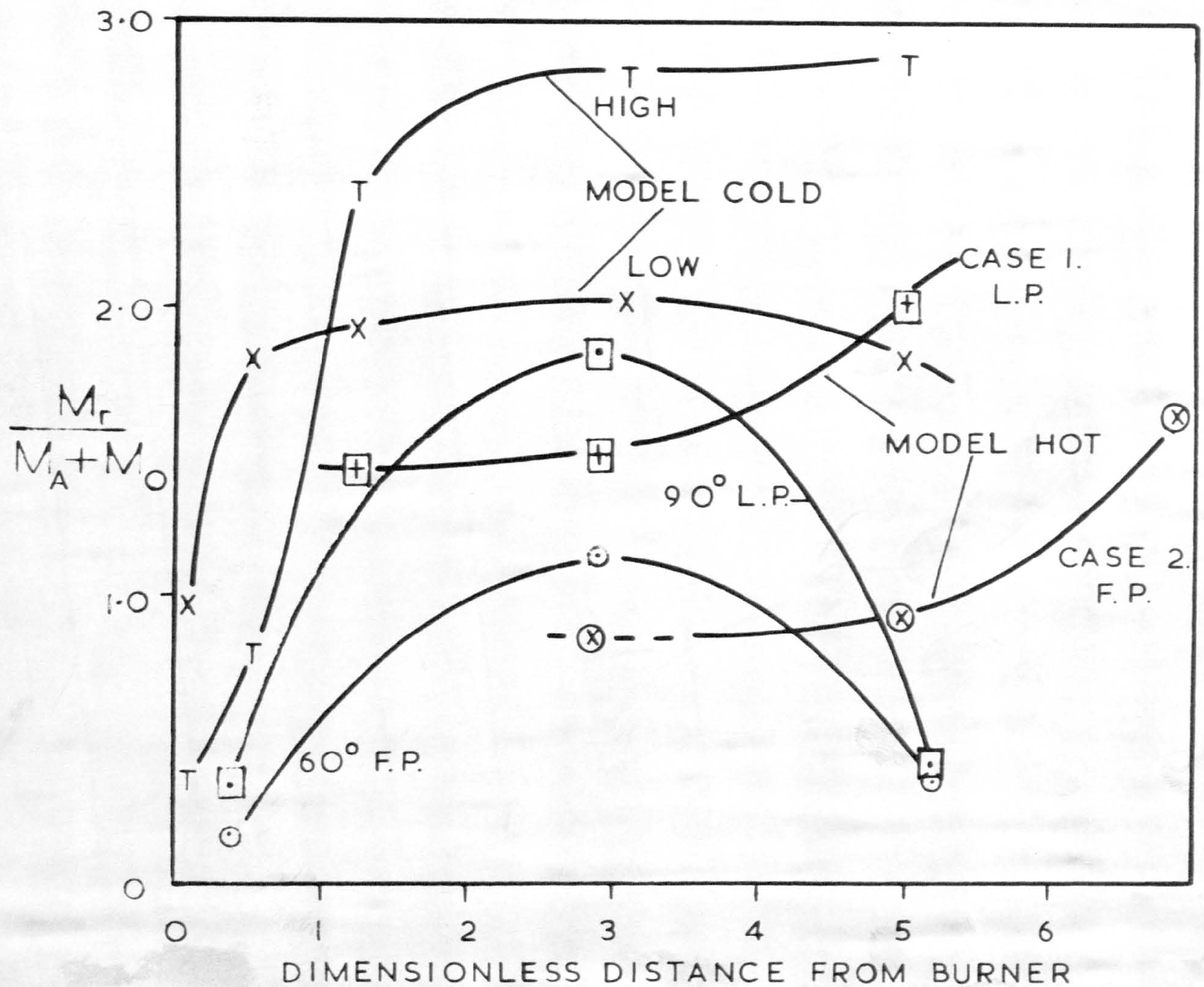



FIG.64. PLOT OF BULK RECIRCULATION FLOW.

this carries hot combustion products right back to the atomizer, it obviously plays a major part in providing sufficient heat for flame stabilization in the early stages of the spray.

Fortunately, as Thring and Masdin[20] point out, it is in fact more satisfactory in many instances to add air and combustion products in a stepwise fashion, sufficient being added only for the stage reached. The combustion products passing into the next stage provide the preheating of the combustibles. As the present system is by no means a perfectly stirred reactor, it is not feasible to make accurate sub-divisions of the flame in the same manner. Qualitatively, the following stages can be distinguished:-

(i) Very near the atomizer:- Probably sufficient combustion products recirculated axially to provide the necessary preheat for this zone. Air will come partly from the air bleed round the atomizer and in small quantities from the main annular jet. This corresponds approximately to zone 1. of Thring and Masdin's subdivided system shown in Fig. 6.

(ii) The region of peak air velocities:- Here, almost the whole of the combustion air and the fuel cross, but due to the entrainment of insufficient combustion products, this air is relatively cold. This corresponds roughly to the homogeneous reactor's primary zone in Fig. 5., but with a combustion products deficiency.

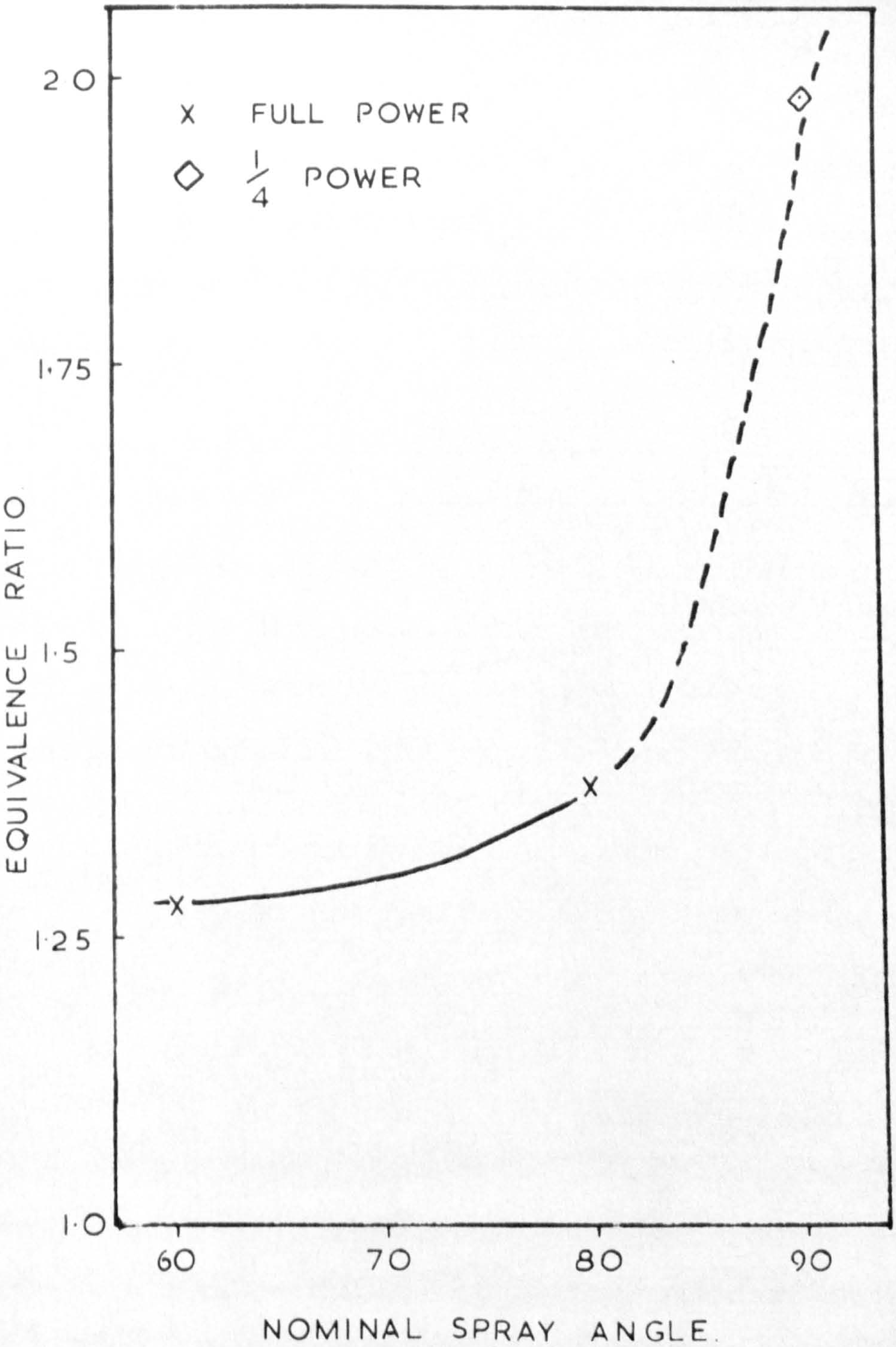



FIG. 65. PLOT OF VARIATION OF EQUIVALENCE RATIO AGAINST SPRAY ANGLE.

(iii) Outside the high velocity part of the air stream:- Here, the temperature is that of the recirculated stream, but both fuel and oxygen are in fairly low concentrations and the reactions are fairly slow, similar to the secondary zone in the homogeneous reactor.

### 6.2.4 Smoke limits.

It will be seen from Fig. 65., which is a plot of smoke limit expressed in terms of the equivalence ratio against spray angle, that the black smoke limit occurs at progressively higher equivalence ratios as the spray angle increases. The big jump comes with the change from full-power to low power as well as a change from 80° to 90° spray angle where the air-fuel ratio is twice the stoichiometric value. There are insufficient data to show which of these is the more significant. The formation of black smoke appears to be due to the slower burning constituents of the fuel not being burnt within the flame limits. This will obviously be a function of oxygen availability (oxygen concentration and mixing), residence time and gas temperature. It is interesting to consider which of these is most significant. If the air-fuel ratio is reduced towards the stoichiometric value, both residence time and temperature of the gases in the furnace should increase. This, in fact, takes the system into the region where smoke is formed which suggests that the smoke is governed primarily by transport of oxygen to the fuel.

It would have been interesting to record the effect of a $90^{\circ}$ spray angle under full power conditions as this would have shown if the limit was purely a function of the spray or whether the effects of increased turbulence levels, presumably occurring under full power conditions, were significant.

### 6.2.5 The aerodynamics of the system.

The aerodynamics of the system are of interest because of the central recirculation zone and its associated surrounding annular jet, which persists for about 3 diameters downstream. The peak velocities in this annulus maintained an almost constant radius which was very nearly the same as that of the end of the burner quarl. This is true, under both hot and cold conditions, although in the latter case the length of the annulus and the recirculating core are somewhat reduced. Data for the velocity changes along the axis and the peak velocities are plotted in Fig.57., while the recirculation data are given in Fig.64.

All the results give similar curves, although the model results are slightly higher, which is in line with the temperature data. When compared with Beer's data(65) for a swirling jet, the shape of the curves is somewhat flatter, probably because of the different outlet conditions. Beer's burner, for example, had a parallel end. These data suggest that the final air swirl produced by the register examined in this investigation had a ratio of angular to forward momentum $\frac{G_\phi}{G_x}$ of about 0.10. That is a little lower than one of Beer's sets of results which was for a ratio of 0.12.

Reference has been made already to Newby and Thring's concept of 'equivalent diameter'. As indicated, the mixing process in the earlier part of the flame is governed to a considerable extent by the characteristics of the oil spray, and in cases such as this one, where the initial jet is an annulus, the characteristics of the jet will vary in a rather more complicated way with temperature than a plain simple jet. However, there seems no reason why the entraining properties of the jet should be changed as this is assumed from equation 9. to be a function only of total mass flow. Thus, providing the burner diameter is replaced by an 'equivalent diameter' as defined in equation 13, it should be possible to correlate the recirculation data with existing data. Equivalent burner diameters are calculated for the full size and model burners in Appendix VIII. This is done in Fig. 66. where the result for $\frac{m_{r\,max}}{m_o + m_a}$ is plotted against $1/\theta''$, and in Fig. 67. where the dimensionless distance at which the maximum recirculation occurs is plotted against $\theta''$. Data from the initial pressure-jet trial at IJmuiden[108], and data from Cohen de Lara[50], (solid line), are plotted on the same graphs. The maximum quantity recirculated

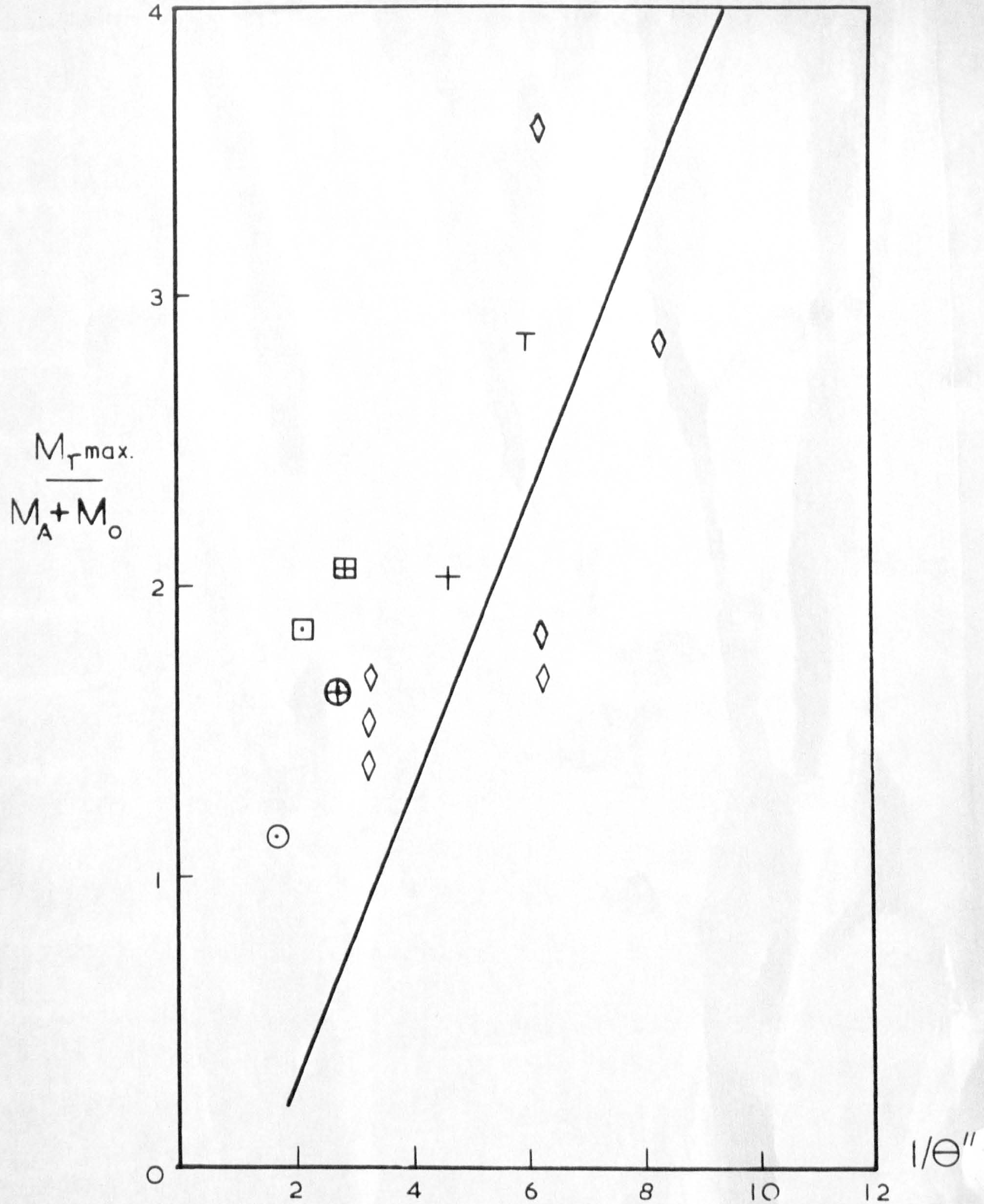



FIG. 66. VARIATION OF MAXIMUM RE-CIRCULATION FLOW RATE WITH $1/\Theta''$.

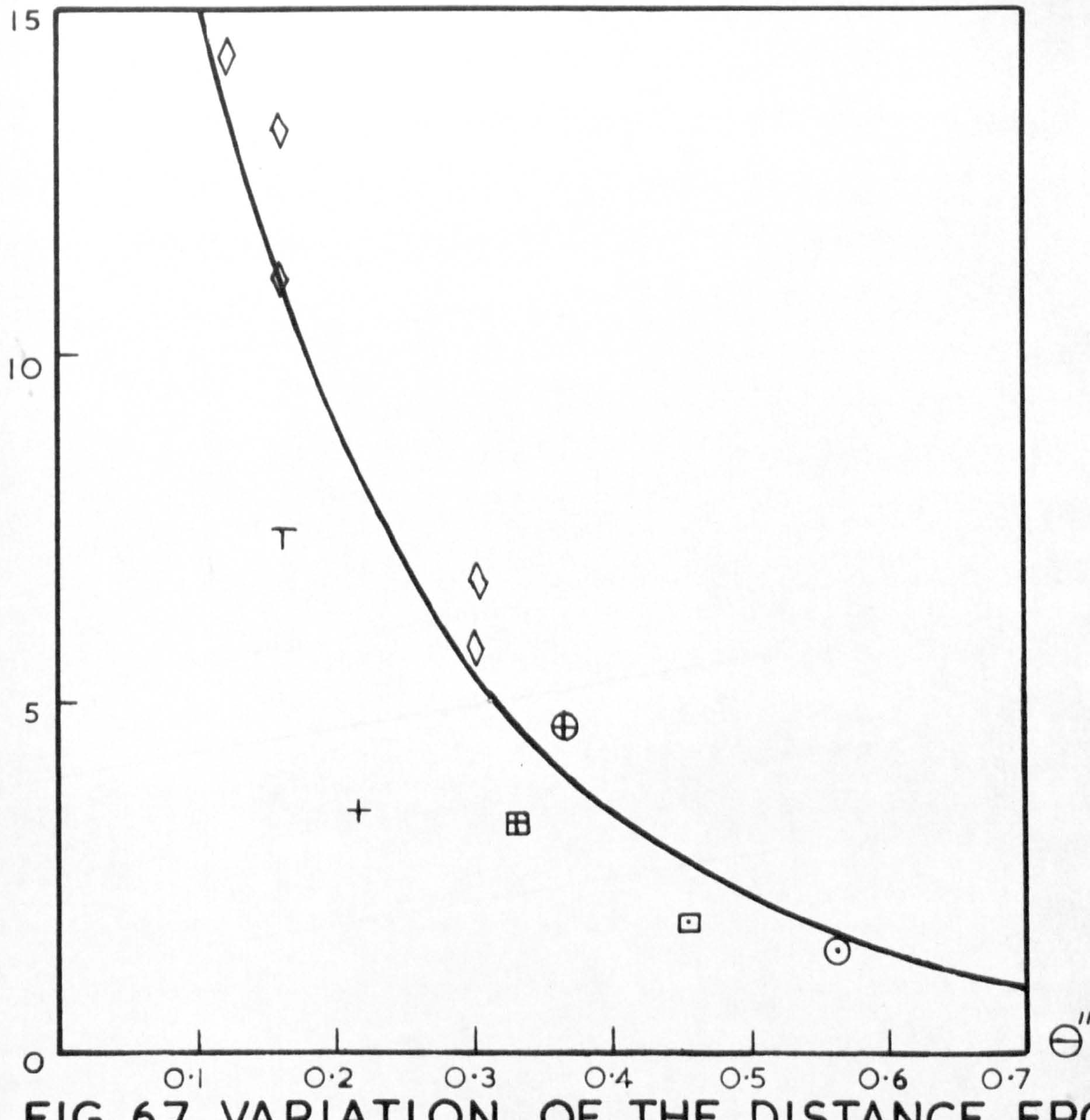



FIG. 67. VARIATION OF THE DISTANCE FROM THE NOZZLE WHERE MAXIMUM RECIRCULATION OCCURS AGAINST $\theta''$, COMPARED WITH THOSE OF A SINGLE JET IN A COLD MODEL.

in the present results can be seen to lie in a similar scatter to the IJmuiden results, although the distance for maximum recirculation is obviously shorter indicating that the jet, as predicted, was not fully developed before the end of the combustion space.

## 6.3 Discussion of scaling results.

The data obtained for both scaling cases show that the correct flow configurations can be represented without difficulty on the model. From these data alone the previous discussion in this chapter should allow anticipation of likely effects arising from variations in atomizer characteristic or flow configuration.

The gas composition studies are, of course, a good deal more complex as it is impossible to keep similarity correct for all the components of the combustion process. There are thus some differences between full scale and the two sets of scaling criteria tested. It is suspected from the fuel distribution data differences in the case of the low-power condition that the atomizer matching was not as close as had been hoped. The two sets of results show that it is possible to reproduce by either method the main characteristics of the full scale flames. (See Figs. 40-43 and Figs. 49 and 50.)

In both cases, the correct zones of high unburnt fuel were reproduced, although reference to the combustion efficiency plots and the CO plots show that in both cases combustion outside the regions of high unburnt fuel was almost complete as opposed to the values of 0.95 or slightly higher observed in the full size work. Prediction from the scaling conditions suggests that the case 2. criteria should be worse than case 1. in this respect when using a distillate fuel as in this experimental work. This is difficult to check as these were modelled on different conditions. Use of a residual fuel oil equivalent to that used in the prototype would be an interesting piece of future work in this case, and would probably help to keep the end combustion efficiencies more correct at the end of the process. Case 1, of course, presupposes the use of a distillate fuel to keep the ignition delay correct.

The recirculation data show a rather higher return mass flow in the model. This is accounted for by the fact that in the end it was not possible to keep $r_{o/L}$ identical in the model and full-size units, the model having a somewhat smaller $r_o/L$. thus allowing a high recirculation figure. This, however, is important only well down stream clear of the main part of the combustion processes.

The temperature data as mentioned already are slightly higher in the case of the model than the full-size, but only sufficient to make probably about 5% difference in the density figures. This difference is unlikely to have any significant effect. The increased temperature will, however, affect the reaction rate slightly. Additional water cooling could be added if necessary to correct this.

## 6.4 Combustion oscillations.

The main line of study in this thesis has set out to provide illumination on the combustion processes which may be relevant to the problem of oscillating combustion.

Direct investigation of combustion processes in pulsating flames would have necessitated very difficult experimental techniques, such as instantaneous sampling, and it was considered that more directly useful information would be obtained by the work, already discussed, on flames burning steadily.

At the start of the study, it was strongly suspected that the onset of combustion instability in the form of regular pulsations was affected by the performance of the flame in terms of mixing. This line of thinking was reinforced by the analysis of theoretical work discussed in section 2.9 and by subsequent measurements on pulsating boilers. From this, it was apparent that limiting air-fuel ratio increased as the fuel throughput decreased, which suggested that mixing deteriorated as power was reduced. Papadakis' results also lead to similar conclusions.(88)

Subsequent work carried out by Admiralty (see Fig. 68.) has amplified these data, and shown that there are probably several sets of limits, depending on the conditions, as was also noticed by Papadakis. Analysis of the possible theories advanced for combustion oscillation has shown that they fall into two classes, acoustic and non-acoustic theories. All the above mentioned results appear to have non-acoustic characteristics. None of the non-acoustic theories is as yet sufficiently defined to predict the limiting frequencies in these rather complex systems. The main difficulty is in determining the exact flame behaviour during the oscillation, and hence ignition delays and temperature variations during each cycle.

The theoretical ideas put forward by Thring[84] are used to calculate a 'trend' curve in Fig. 61. This is calculated on the assumption that this gives the correct value at half the full fuel rate. This avoids the difficulty of trying to calculate absolute values for the frequency. Although this curve follows the practical results in general form, the theory does not account for the jumps in limiting frequency observed.

From the practical point of view, each frequency band seems to have a limited energy output available to drive pulsation, and this might be avoided, from the present results, by choosing a system with a sufficiently high resistance across the burner. The alternative is to improve the mixing of the fuel and the air.

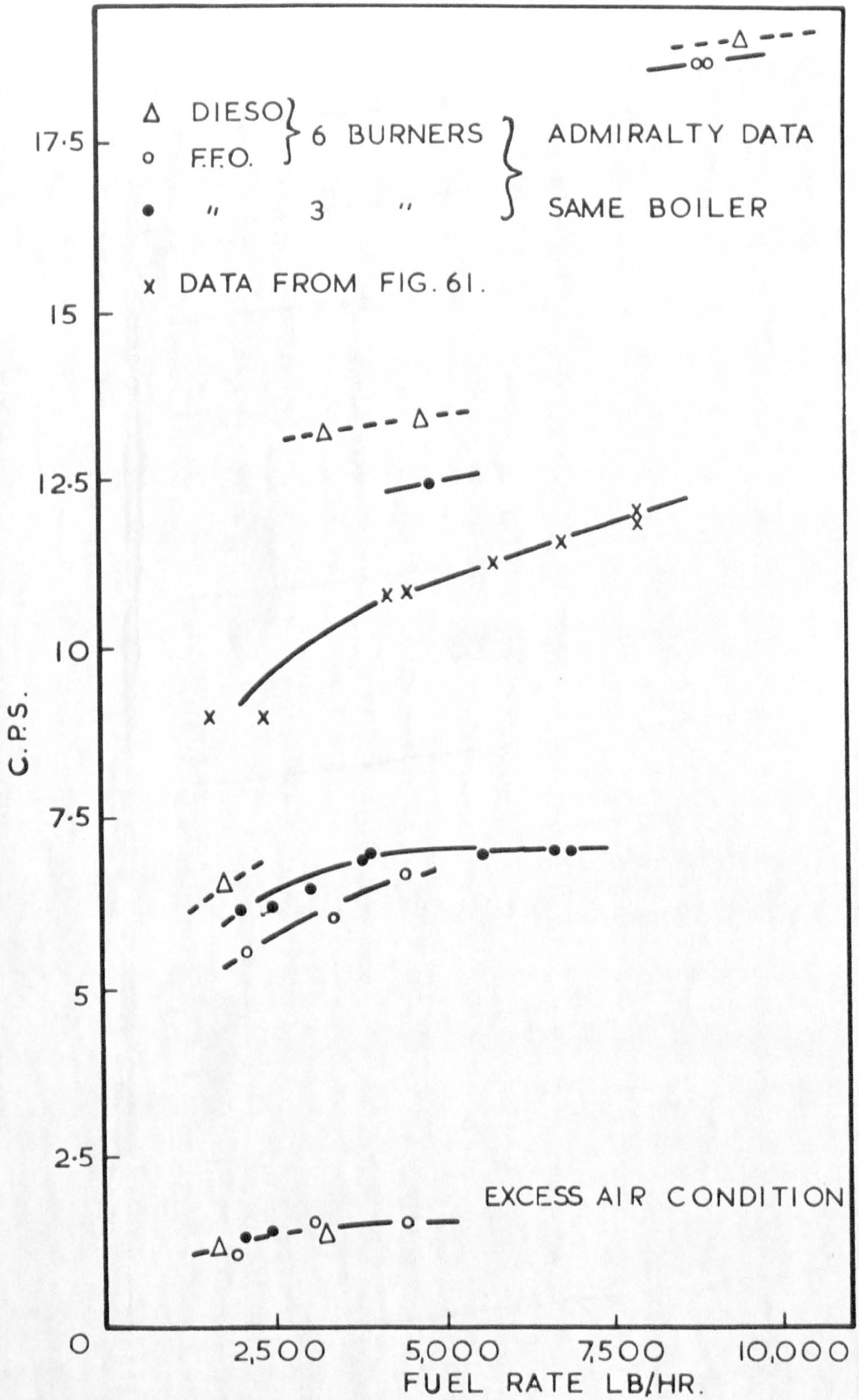



FIG. 68. PLOT OF OSCILLATION FREQUENCY AGAINST FUEL RATE FOR VARIOUS MARINE BOILERS.

It seems possible that pockets of unburnt fuel are a particular cause of trouble, since a perturbation in the system may result in a pocket being disturbed suddenly, with a consequent sudden change in reaction rate. It is noteworthy in this connection, that combustion oscillations are observed only when there is a wide distribution of unburnt fuel in the combustion chamber.

## 6.5. Overall Conclusions.

### 6.5.1. Full scale work.

At the commencement of the research, no data existed which showed the effect of burner variables on the mixing and combustion efficiency distributions or velocity and recirculation profiles for pressure-jet burners employing swirled air and designed for boiler use. Data of this type were obtained over a range of spray angles and at two power ratings. It was shown that:-

i) While mixing was adequate for most of the reactants, incompatibility between the hollow spray cone and the main part of the air jet which resulted in "cross-over" between fuel and air allowing fuel escape outside the flame. A second pocket of unburnt fuel was found to occur inside the flame and to extend someway downstream. This fuel loss outside the flame became more pronounced with wider spray angles and particularly at a ¼ power where the spray angle was widest.

It was concluded:-

a) that the marked increase in the air fuel ratio at which smoke first appeared was due to the increasing incompatibility between the oil spray and the air pattern as described above.

b) It is considered that the use of atomizers with a large angle change over their operating range is therefore unsatisfactory.

c) Changes in burner aerodynamics to supply air to these unburnt pockets are necessary. Simple experiments with an upstream secondary air supply showed that this appeared to improve the air distribution considerably.

ii) The general shape of the airflow pattern is shown to be unaffected by throughput rate, although some reductions in the relative mass recirculation flow were noted at high powers. The jet profile was annular in form near the burner, with a recirculation zone inside. Further downstream this became progressively W-shaped and then a more or less conventional single peak profile in form. Recirculation external to the flame appeared to be entrained too late to make any contribution to flame stabilization. It was concluded that the central recirculation zone was the main source of hot combustion products for flame stabilization despite a relatively small return mass flow. Control of this region should therefore affect flame stabilization substantially.

iii) In comparison with the theoretical optimum, these flames were very far removed from an ideal homogeneous reactor. The stepwise system of Thring and Masdin was deemed to be a more satisfactory target for this type of system. The recirculated flow needed in this case would be less, probably of the same order as that actually obtained.

6.5.2. Results from model furnace work.

i) It has been found possible to model flames satisfactorily, with respect to both flow configuration and the main features of the gas composition pattern.

ii) Dimensional analysis was used to determine suitable scaling parameters. It was found that it was not feasible to maintain similarity for all the possible groups at the same time. It was found satisfactory to use various combinations of the more important groups. Flames were modelled on two different selections of these groups, based in one case on maintaining velocity and relative momentum similarity and in the other residence time similarity. Geometric similarity of the burner was maintained in both cases.

iii) The results showed that it was possible to maintain good similarity for the velocity profiles between the model and the prototype under combustion conditions.

iv) It was concluded that the correct general features of the gas composition patterns could be reproduced by both the selections of scaling groups investigated. It is thought that there is little to chose between the results obtained by either method. It is possible that with additional experience, the first method may prove slightly better.

6.5.3. Comparison between Cold and Hot Results.

Data obtained from cold model runs showed that the general jet shape, particularly the radii at which maximum velocities occurred were similar to the hot case, but that the actual re-circulation zone was noticeably shorter under these conditions.

Cold measurements in the early part of the jet indicate that the initial angle of the air leaving the burner shows a maximum angle slightly greater than the oil spray.

6.5.4. Conclusions from the Investigation of Combustion Instability.

i) Data was obtained on combustion instability over a range of fuel input rates on a Naval three drum boiler. The pulsations observed at each fuel rate were shown to have a definite upper limit of air-fuel ratio, above which combustion was steady.

ii) Except at very low power levels, the limiting frequencies measured were very similar, increasing only slightly with power. At the limiting point, although the air-fuel ratio was different in each case, the actual R.D.L. was nearly constant.

iii) It proved impossible to make accurate measurements of amplitude for the oscillations observed, but this was thought to be of the same order as the R.D.L.

iv) Film records from pulsating flames showed that the flame cone angle oscillated with the pulsations through quite a wide angle.

v) Background flame noise was measured for steady combustion conditions, and although of a fairly random nature, had a frequency of 5-10 times those of the oscillations measured.

vi) The various theoretical treatments so far put forward are not yet sufficiently refined to predict the oscillation frequency at the onset of pulsation.

## 6.6 Recommendations for future work.

The following lines seem to be the most important next stages in research:-

i) Fuller investigation of the interaction of liquid sprays with air streams, particularly in practical burner configurations. (The current work of Priestley should go some way in this direction.)

ii) A much more detailed study of the mechanisms of flame stabilization in the sense of flame 'holding', in flames where there is a liquid spray and complicated aerodynamics due to swirl.

iii) The development of oil burners from first principles, as far as these are known. The real question is not that of finding an atomizer with characteristics to suit a particular situation, but rather in designing an air flow pattern to suit the oil spray. Useful development might take place along the following lines:-

i) To make the air distribution to the spray more uniform and so avoid external loss of fuel. Possible methods are by altering the aerodynamics of the burner, by the use of multiple oil sprayers to reduce the effective size of individual oil sprays, or by the use of secondary air injection.

ii) The other burner feature which might with advantage be altered, is the size and distribution of the recirculation zones. It is thought that advantages might accrue from concentrating most of the recirculation in the centre of the flame to reduce the tendency for pockets of unburnt fuel to occur on the axis. This could probably be achieved by the use of the "open" type of jet configuration, and a special quarl arrangement.

## APPENDIX I.

## Pressure Jet Burners for Boiler Use.

(Paper presented at the Institute of Fuel Conference "Major advances in Liquid Fuel Firing 1948-59").

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# APPENDIX II.

## Combustion Oscillations

(Paper given during 'Liquid Fuels Course',

Sheffield University, Summer, 1960).

*Reprinted from the* Proceedings of the Conference on " Major Developments in Liquid Fuel Firing, 1948-59 ".

# Pressure-jet Oil Burners for Boiler Use*

By A. M. BROWN, B.Sc. †

Recent advances in oil-burning technology have resulted in the solution of many of the problems formerly encountered with oil burners used on boilers which incorporate pressure-jet atomizers. Further advances, particularly in the direction of higher combustion intensities, will require much greater application of combustion theory. The present state of knowledge of high-intensity combustion, liquid sprays, furnace aerodynamics, and flame stabilization is reviewed, and attention is drawn to the need for much further work in many of these fields of study to make the results of real value to the engineer. The use of model techniques and the performance of some types of burner are discussed and an indication is given of the importance of aerodynamics in the boiler furnace as a whole.

## 1. INTRODUCTION

THE day is passing in which pressure-jet oil burners on boilers and in similar applications are operated with poor atomization of the oil and only a small pressure drop across the air register, often requiring large percentages of excess air for good combustion. Sometimes also little attempt was made to obtain a suitably shaped flame for the use required. In recent years efforts to obtain good atomization, attention to air register design and considerable increases in air velocities in the burner have resulted in much improved conditions of combustion. Today the range of burners in common use runs from the ½ gal/h domestic unit using distillate fuels up to 5,000 lb/h units firing preheated residual fuels with viscosities sometimes as high as 6,500 sec Redwood I at 100° F. The use of as little as 5 per cent excess air has been claimed for some well-designed burners and installations. Combustion intensities of 100,000 B.t.u./cu. ft/h/atm and upwards are now common, and considerable improvements in performance have been obtained by increasing the draught losses employed across the air register at maximum load from the range ½ in. to 4 in. w.g. to anything from 3 in. to 10 in. w.g. or higher in the bigger units. Pressure drops as high as 20 in. w.g. have been used in some high-intensity burners, and much higher still in jet-can combustors. Much more care is also being given to " fitting " flames to the more tricky applications such as shell boilers.

In the future the increasing demand for compact boilers, particularly for marine purposes, will necessitate higher combustion intensities and more stringent combustion conditions (see later, Fig. 3). In addition, economy and automatic operation will demand much greater knowledge and control of conditions in the flame at all loads. Although satisfactory results for many purposes are currently being obtained, there remains a very wide sphere for increased understanding of the combustion process, and for improved burner performance. This applies not only to the development of individual burners by themselves, but also to such matters as the interaction of adjacent flames and the aerodynamics of the combustion system and boiler as a whole. In the past, most methods of design have been largely empirical and the application of fundamental principles often neglected, although examples can be given of attempts to apply theoretical results to the parallel field of gas-turbine combustion chambers.[1,2,3] Despite the large amount of development work which must have been done by individual manufacturers, it is much to be regretted that with few exceptions [4,5,6,7] very little on the subject has appeared in print.

In this paper the present state of knowledge of aerodynamic requirements and mixing processes in pressure-jet burners is reviewed, rather than the handling and atomization of the oil. It is presented in the hope that this will stimulate discussion, and assist in the development of improved combustion equipment.

## 2. MAIN FEATURES REQUIRED FOR PRESSURE-JET OIL BURNERS

The main requirements for liquid-fuel pressure-jet systems are:—

(*a*) A high combustion intensity relative to the draught loss.

(*b*) A combustion efficiency (i.e. heat release compared with calorific value of fuel) of better than 99 per cent at maximum firing rate.

(*c*) Satisfactory flame stabilization, and optimum mixing of fuel, air and recirculation products.

(*d*) Satisfactory combustion over the specified turn down range (anything from 3 : 1 to 20 : 1). This is particularly important for automatic control systems.

(*e*) Avoidance of pulsations for any loading in the desired range of operation.

(*f*) Good heat transfer and combustion-chamber design.

(*g*) Tailoring of the flame to avoid damage to refractories or metal parts, especially the oil sprayer and air register, impingement on tubes or superheater elements, and avoidance of deposition of carbon or soot at any point.

## 3. OVERALL THEORETICAL COMBUSTION REQUIREMENTS

### 3.1. Chemical and Physical Processes

The first requirement is adequate heat transfer to raise the temperature of the fresh material to ignition point. There are several possible mechanisms, preheat, conduction or radiation back from the flame, or recirculation, but in the pressure-jet system only recirculation is really significant. Assuming small heat losses, a ratio of recirculated products to fresh combustion air of 0·75 is about the minimum for this purpose, and it has been shown experimentally that the maximum combustion

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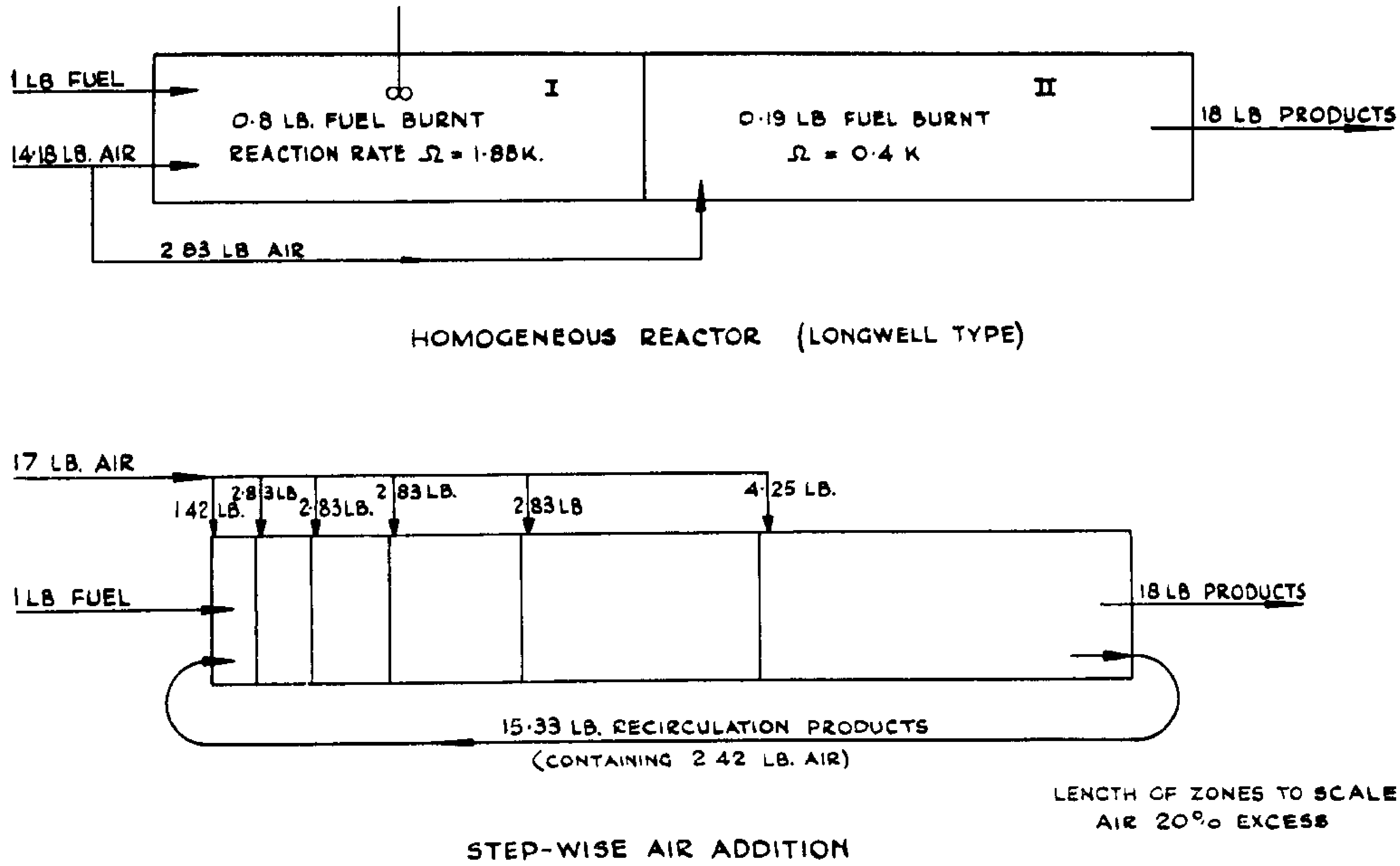



FIG. 1.—*Schematic arrangement of ideal high-intensity combustion systems.*

rate per unit volume is obtained at a ratio of about 4·0. A further increase in combustion products dilutes the reacting mixture, and slows down the combustion rate. With the exception of units such as small domestic boilers, heat transfer to the walls is unlikely to be sufficient to quench the flame in the combustion space. Thus, provided the percentage of excess air is fairly low, incomplete combustion (i.e. low efficiency) can only result from insufficient time to complete the chemical and physical processes of combustion before the reacting mixture is cooled by the tube bank.

From investigation of ignition delay, Mullins [8] has predicted that the maximum combustion intensity attainable in a liquid-fuel spray should be as high as $3 \cdot 9 \times 10^9$ B.t.u./cu. ft/h/atm. This figure is apparently only slightly affected by vaporization, droplet size, turbulence or air/fuel ratio, and is almost the same for all hydrocarbon fuels. Longwell's famous perfectly stirred reactor [9] has indicated that the maximum combustion intensity achievable by instantaneously mixing fuel, air and combustion products is $4 \times 10^8$ B.t.u./cu. ft/h/(atm)$^{1 \cdot 8}$ with 80 per cent completion of combustion. Both these results indicate that chemical processes are extremely rapid, and that the physical processes of mixing and evaporation are mainly responsible for incomplete combustion in any ordinary oil-burning system.

### 3.2. Ideal Combustion Systems

Applying the results derived by Longwell to cases of more immediate interest, Bragg [10] has shown that for higher combustion efficiencies in non-premixed systems the greatest intensity is obtained when 85 per cent of the stoichiometric air is brought into the homogeneous reaction chamber, giving 20 per cent of the fuel unburnt at the exit. A second chamber is attached where further air, up to 16 per cent excess, is added. For 99·9 per cent efficiency this secondary chamber must be the same size as the primary one.

Thring and Masdin [11] have recently suggested that by mixing air stepwise down the chamber each time the reaction rate begins to fall off, even higher intensities could be obtained than in the homogeneous reactor. The effect of this system is compared with a Longwell type reactor in Fig. 1, the increase in combustion intensity being about 6 per cent. A good gas-turbine combustion chamber might approximate to something like this.

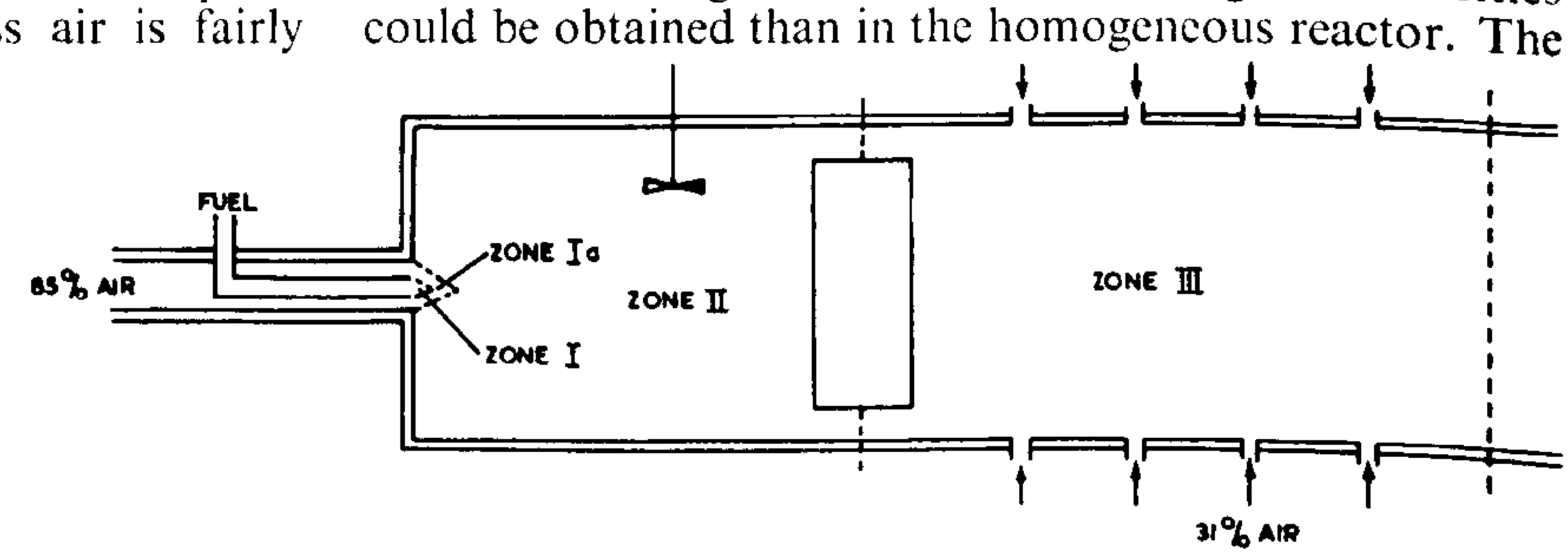



FIG. 2.—*An ideal non-premixed combustor for intensity combustion.*

*Zone I.—Fuel mixing with primary air.*
*Zone Ia.—Fuel-air mixing with four masses of combustion products.*
*Zone II.—Homogeneous reaction zone.*
*Zone III.—Remainder of air mixed in a reaction proceeding from* 80 *per cent completion to* 99·5 *per cent completion.*

These results must represent the target for practical combustion systems. Bragg's result above, if translated into practical effect, would look something like the combustion chamber shown in Fig. 2. The Longwell results have been applied to gas-turbine combustion chambers by other workers [3] with some success. The conventional pressure-jet burner system is a long way from this ideal because the flame by no means fills the combustion chamber. The form of the oil spray used is also such that it is nowhere likely to approach a homogeneous mixture with the air, nor is it usually possible to have secondary air jets downstream of the burner as in the gas-turbine chamber. One advantage of a non-premixed system is that, with the correct aerodynamic arrangement, it should be possible to vary the local air fuel ratio from distinctly rich at the entry to a weak final mixture, as in the Thring and Masdin suggestion, even without recourse to secondary jets. The exact understanding of the optimum course for mixing of fuel, air and combustion products still awaits experimental investigation, and this work is being undertaken by some of the author's colleagues. The theoretical work demonstrates nevertheless the great importance of obtaining the correct mixing pattern in the flame ; this depends principally on the aerodynamics and is discussed in a later section.

### 3.3. Actual Burner Systems Expressed as an Approximation to the Ideal

By assuming that the combustion air velocity, and hence pressure drop at the burner, is the principal factor governing mixing, a reasonable indication of how well practical systems approximate to the ideal can be obtained by plotting optimum combustion intensity per unit volume of the combustion chamber against absolute pressure loss.[7] The homogeneous reactor lies at one extreme, and all systems lying on this curve represent equally good attempts to satisfy a different compromise. As shown in Fig. 3, real systems are often far from this optimum, which means that either they are wasting pressure drop for other purposes than mixing and recirculation, or else the combustion chamber volume is not being fully utilized. The line $A$ represents the ideal, calculated on the assumption that the pressure drop arises entirely from the loss of dynamic head of the gases entering the chamber. Combustion intensity from this relation is proportional to $\sqrt{\frac{\Delta P}{P}}$ (where $P$ is the total pressure and $\Delta P$ the pressure drop across the register). The curve $B$ corresponds to Lubbock's empirical rule[12] where combustion intensity is proportional to $\left(\frac{\Delta P}{P}\right)^{\frac{2}{3}}$. The actual points represent typical values from different applications and, as will be seen, the mean line $C$ for these points is nearer to the relationship combustion intensity $\propto \frac{\Delta P}{P}$.

## 4. FUEL SPRAY

### 4.1. Liquid Droplets

Turning from overall considerations to small-scale effects, the picture is far less complete. Fluid dynamic results for the behaviour of moving bodies, the effects of drag, vortex formation, gas flow round obstacles under isothermal conditions and similar matters are discussed at length in the standard textbooks.[13,14,15]

However, as a useful review by the Battelle Institute points out,[16] our knowledge of hydrodynamics, evaporation of droplets and turbulence in combustion, for practical purposes, is extremely inadequate, and is further complicated by the differences in behaviour between single particles and the dispersed systems of interest to the engineer. Any problem concerning droplets in turbulent combustion systems is bound to involve a large number of variables. Many of these are very difficult to determine experimentally, in view of the problems encountered in establishing the precise conditions at any given point.

The behaviour of single droplets has been established with some success. There is little agreement, on the

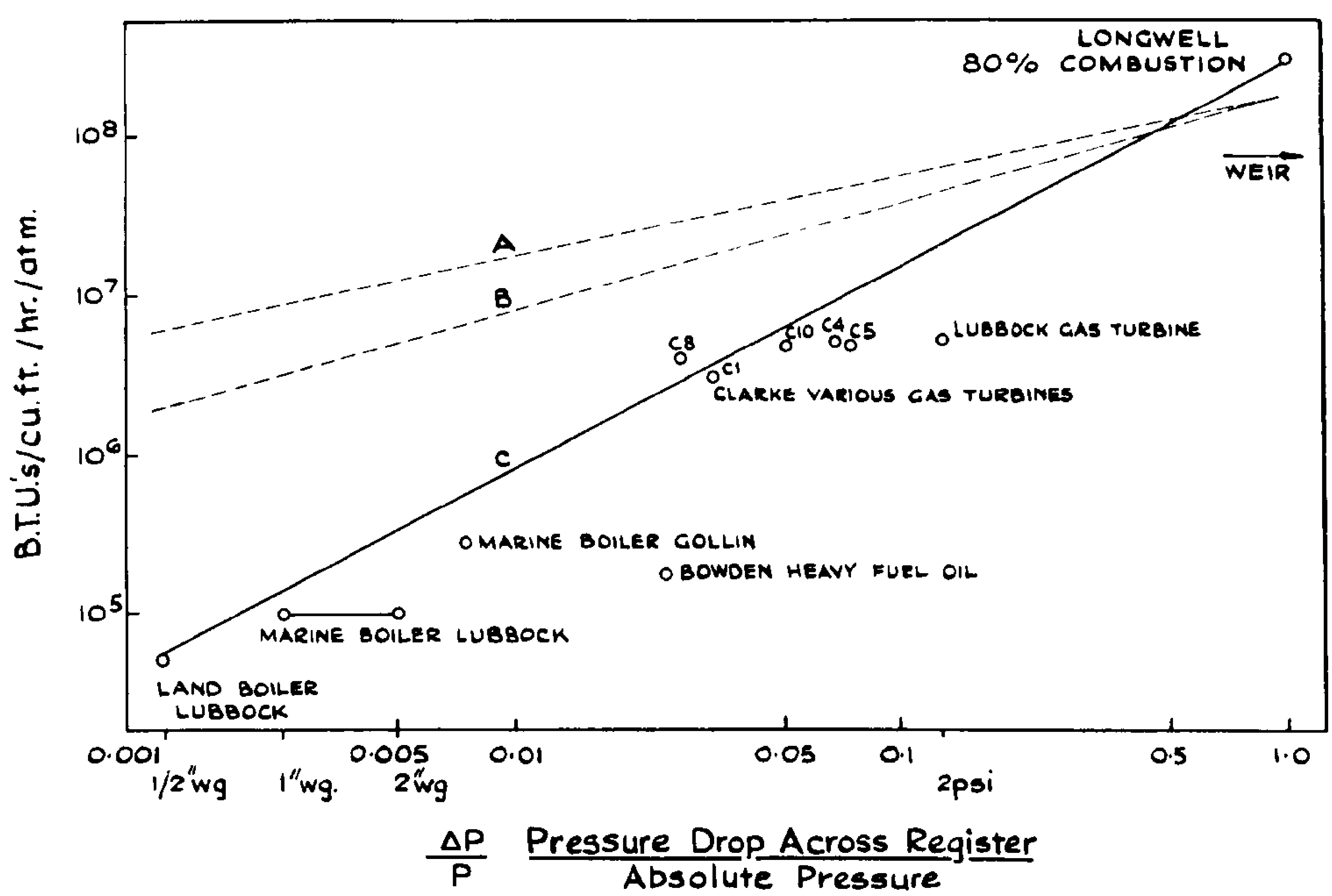



FIG. 3.—*Plot of values of combustion intensity against relative pressure loss for some real systems ; also two suggested optimum curves for comparison.*

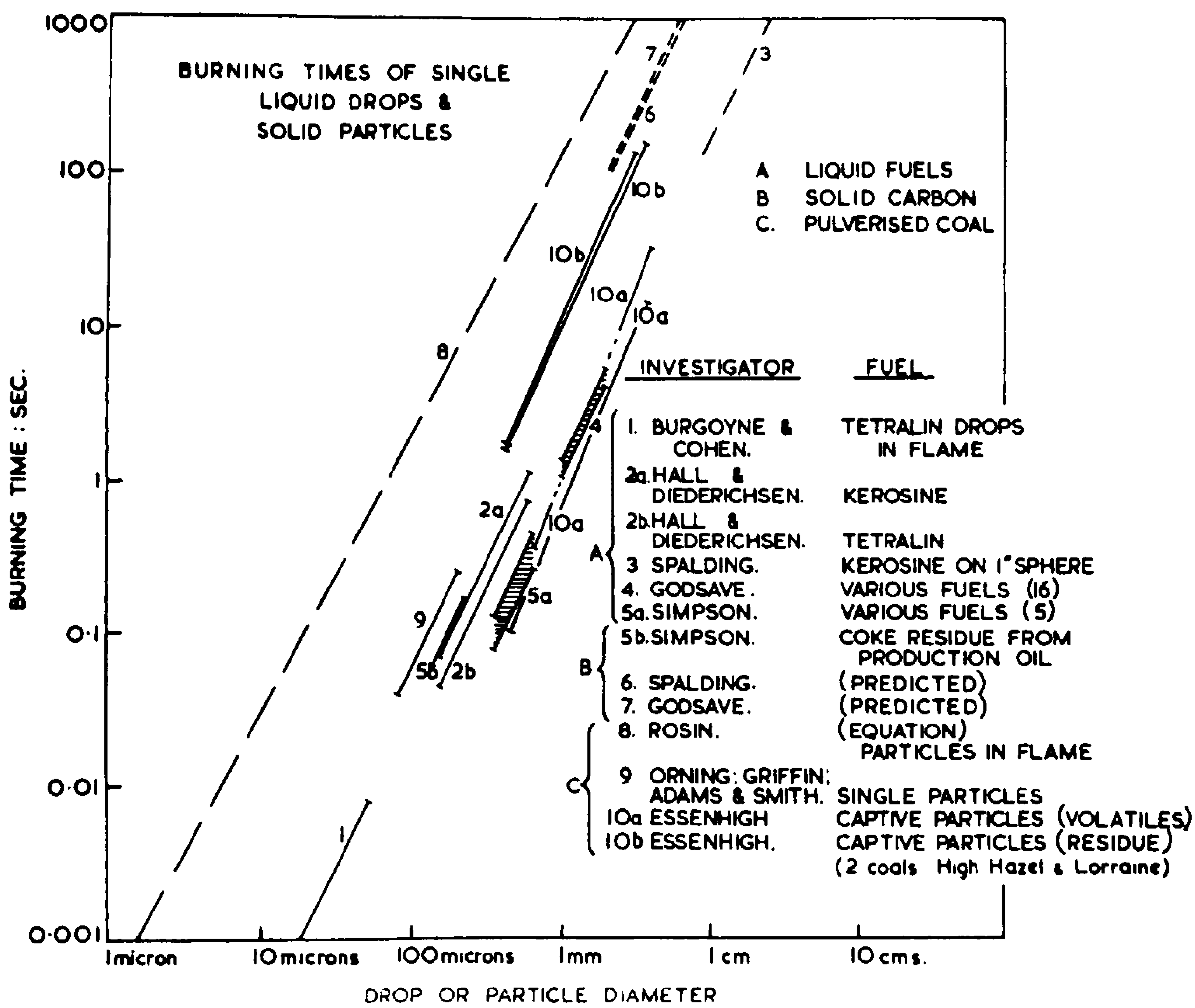



FIG. 4.—*Burning times of some single liquid drops and solid particles.*

other hand, about the paths described by freely moving particles, although satisfactory corrections have been produced for Stokes' law, notably by Langmuir.[17]

The process of droplet evaporation for liquids of fairly low volatility is comparatively well accounted for by Frössling,[18] provided that conditions are fairly uniform, although extension of this work to instances of high temperature and turbulence is badly needed. The relationship between heat and mass transfer, often treated as being virtually independent, also requires much more elucidation. For any practical calculations it is necessary to make rather drastic assumptions. Similarly for the calculation of droplet paths, it has been necessary to assume that the droplets behave as solid spheres, standard results for drag coefficients being then used.[19,20]

### 4.2. Atomization

Disintegration of liquids and droplet distribution have been studied by many workers. Although no theoretical treatment has yet proved really adequate, the Rosin-Rammler distribution is one of the least tedious to use and gives satisfactory results. The upper limit function proposed by Mugele and Evans [21] gives a slightly better fit to the observed data, and, since it takes into account the finite maximum droplet size, is probably more reliable for the calculation of mean droplet sizes.[16,22,23,24,25,26] It is not the purpose of this paper to discuss the different types of pressure-jet atomizer or the theory behind them, and good reviews are available elsewhere.[16,24,25,26]

### 4.3. Droplet Combustion

Droplet burning has been investigated both theoretically and experimentally by a number of workers,[27,28,29] and this has given some insight into the combustion process. The droplet size is of real importance to the combustion process in two ways. First, the difference in penetration and dispersion of different size droplets is one of the principal ways by which mixing of air and fuel is affected, and also has some part to play in stabilizing the flame and establishing its shape. Second, burning times of individual droplets are the other physical limitation of combustion efficiency, and some typical results are shown in Fig. 4.[7] The importance of burning times, and hence fineness of atomization, can be gauged when it is remembered that for combustion intensities of $10^5$ and $10^6$ B.t.u./cu. ft/h/atm the mean residence time for droplets in a boiler is 0·6 and 0·06 sec respectively. The long burn-out time for the carbon skeleton makes this fact particularly important for residual fuels. For boilers, a Sauter mean droplet diameter of 100 $\mu$ or less is regarded as essential. From the point of view of combustion intensity, it is also desirable that the aerodynamic arrangement should permit as few particles as possible to come out in less than the mean residence time.

## 5. AERODYNAMICS

### 5.1. Turbulence

Understanding of turbulence has always been hindered by the lack of a satisfactory physical picture, and by the failure to correlate statistical or empirical theories with observed phenomena.[16] The effect of both small-scale and large-scale turbulence in combustion is to increase the mass transport and surface area of the flame, and has been used to account for the greatly increased heat release in a turbulent flame as compared with a laminar flame.[30,31,3] In the combustion process the predominating effect is the combustion-produced large-scale turbulence, which is substantially greater than that due to the air flow alone.

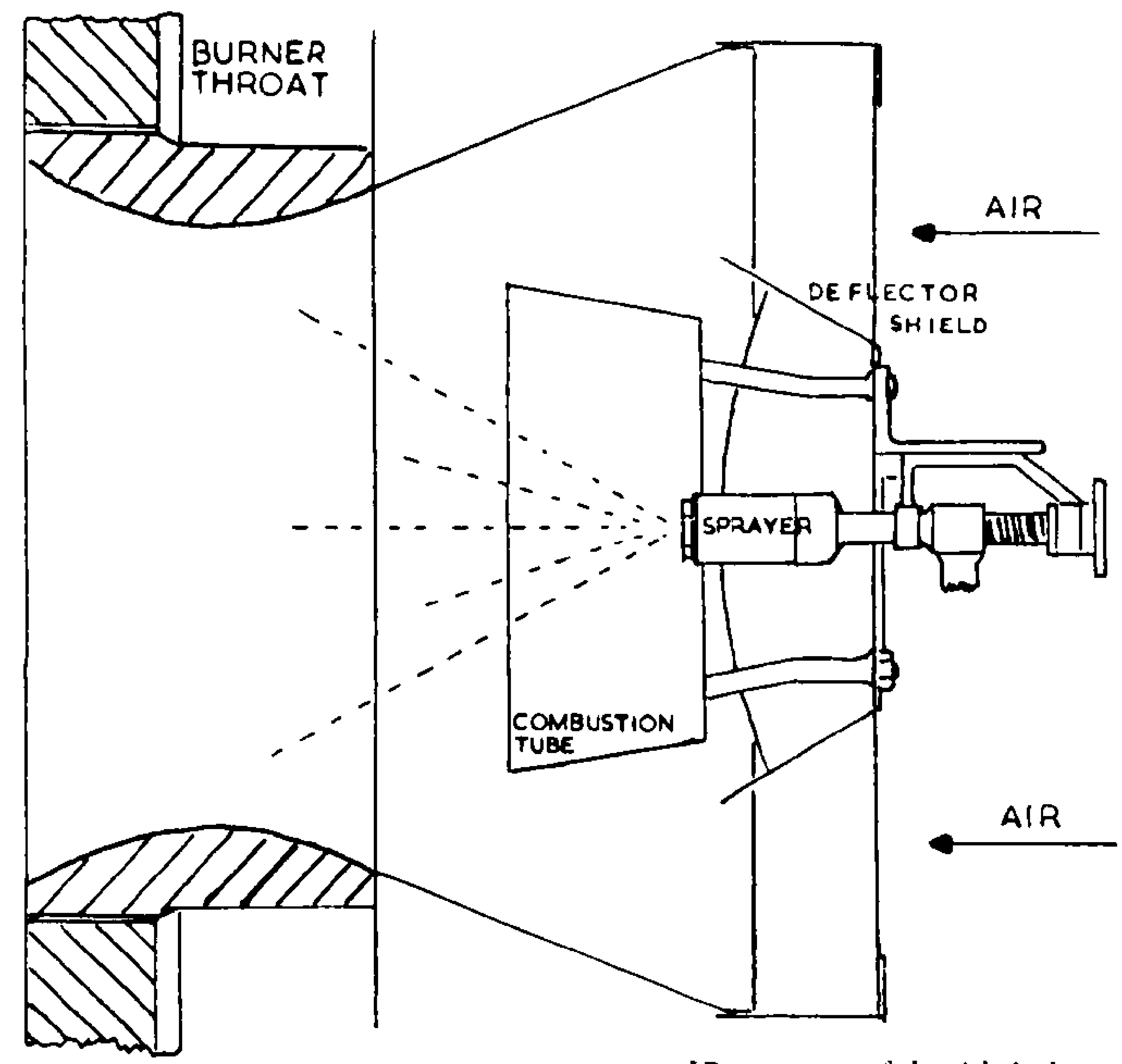
### 5.2. Mixing

All the theoretical methods for mixing in terms of mixing length give only mean results which are of little practical interest. The entrainment of air by free jets has been investigated in some detail,[32] but so far nobody has succeeded in estimating the course of entrainment of gases by liquid-fuel droplets in a pressure-jet spray. This result would be of some interest in determining the necessary aerodynamics of the air surrounding the spray.

Even in simple arrangements flow patterns can be quite complex, and at the moment it is impossible to predict with any accuracy the aerodynamics of any practical burner unit. Nor, consequently, is it possible to predict theoretically arrangements that will produce the mixing pattern demanded by the combustion considerations discussed in earlier sections. Here probably most of all a great deal of work is needed, if only to produce some empirical rules and methods whereby the efficiency of a given arrangement can be estimated. One partial solution, suggested for gas-turbine combustion chambers, is estimated in terms of the refreshment of a recirculating vortex.[19]

### 5.3. Flame Stabilization

Most of the classical work on flame stabilization has been carried out for laminar premixed systems, and the application of the results to turbulent diffusion flames must at best be distinctly uncertain. In general, flame stabilization is achieved by establishing an eddy system behind a bluff body. Predictions of the efficiency of this for simple systems in terms of blow-off velocity have been made by a number of workers.[33,34,35,36] An important practical consideration which must be kept in mind, particularly in the interpretation of cold flow results, is the very large alteration observed in flow pattern between isothermal flow over the stabilizer and when the mixture is burning (see Fig. 5).[37]




[By courtesy of the Admiralty.

FIG. 6 (*a*).—1941-*type Admiralty air register* (*impingement type*).

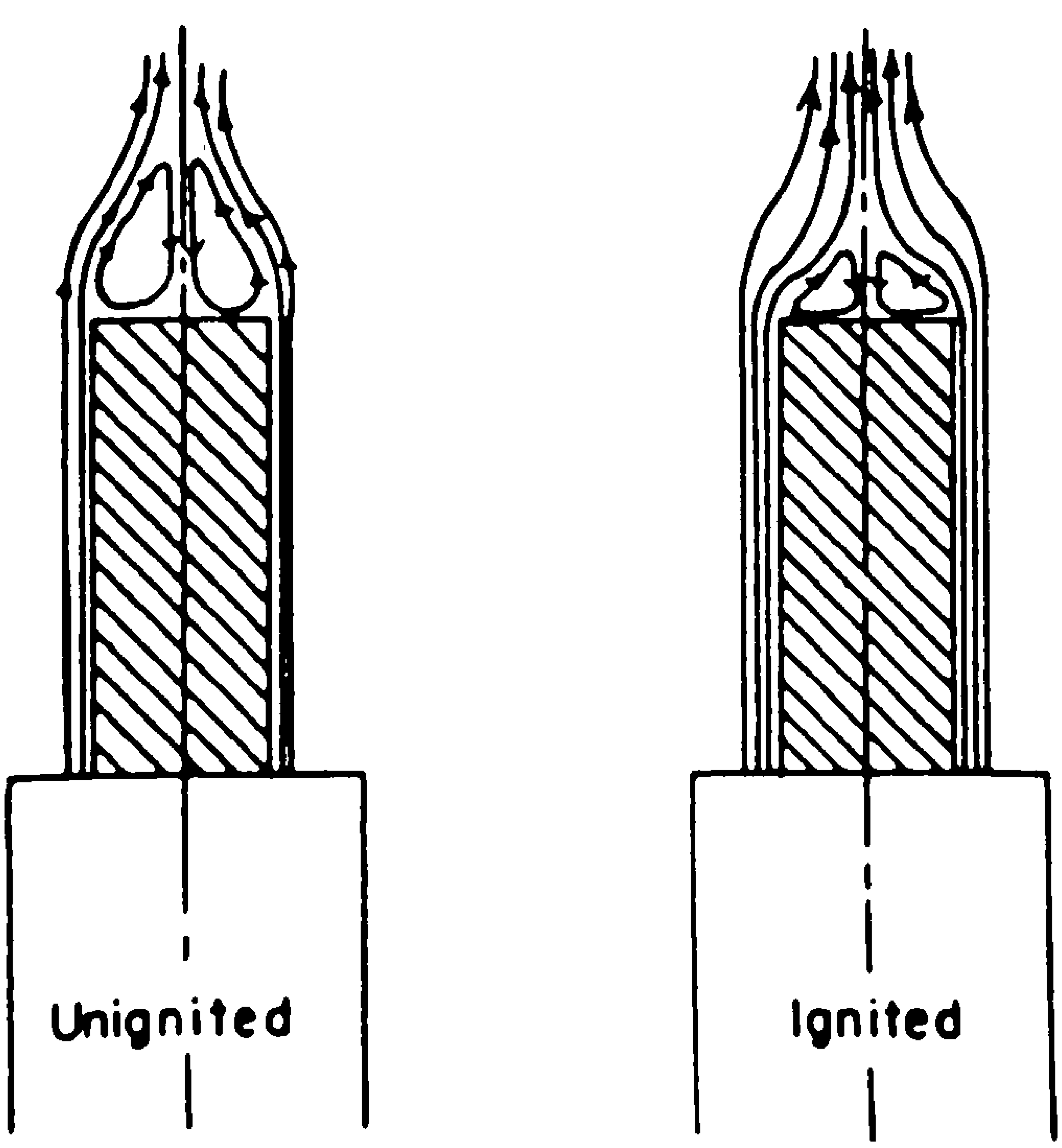



FIG. 5.—*Alteration of flow pattern round a flat-ended rod under unignited and ignited conditions.*

### 5.4. Combustion Instability

The development of low-frequency pulsation, usually of the range from 3 to 40 cps, has been a serious embarrassment in widely differing types of boiler installation. The amplitude of these pulsations can at times be as great as the draught loss across the air register, and can result in a distinct alteration of the flame cone angle during each pulse. Putnam,[38] working on small domestic boilers, has shown that this type of pulsation is acoustic, and has produced results indicating regions of instability without relating them to specific parameters of the combustion chamber or flame. The theory for " chugging " developed by Crocco and Cheng [39] has been successfully applied to rocket motors by Tischler and Bellman,[40] and adapted to the case of conventional oil burners by Thring [41] and Fritsch.[42] In essence, the theory depends on, as controlling factors, the restrictions in the system (e.g. at air register and stack), and postulates either a definite delay time between reaction rate variations and the consequent pressure variation upstream of the constriction, or an inertial effect of the fluid flow in some part of the system. For certain ranges of conditions this can result in the development of a stable feedback oscillation mechanism powered by the combustion process. There is some disagreement in detailed findings, but it is safe to say that for many installations there is unstable combustion at some range of loading, and that

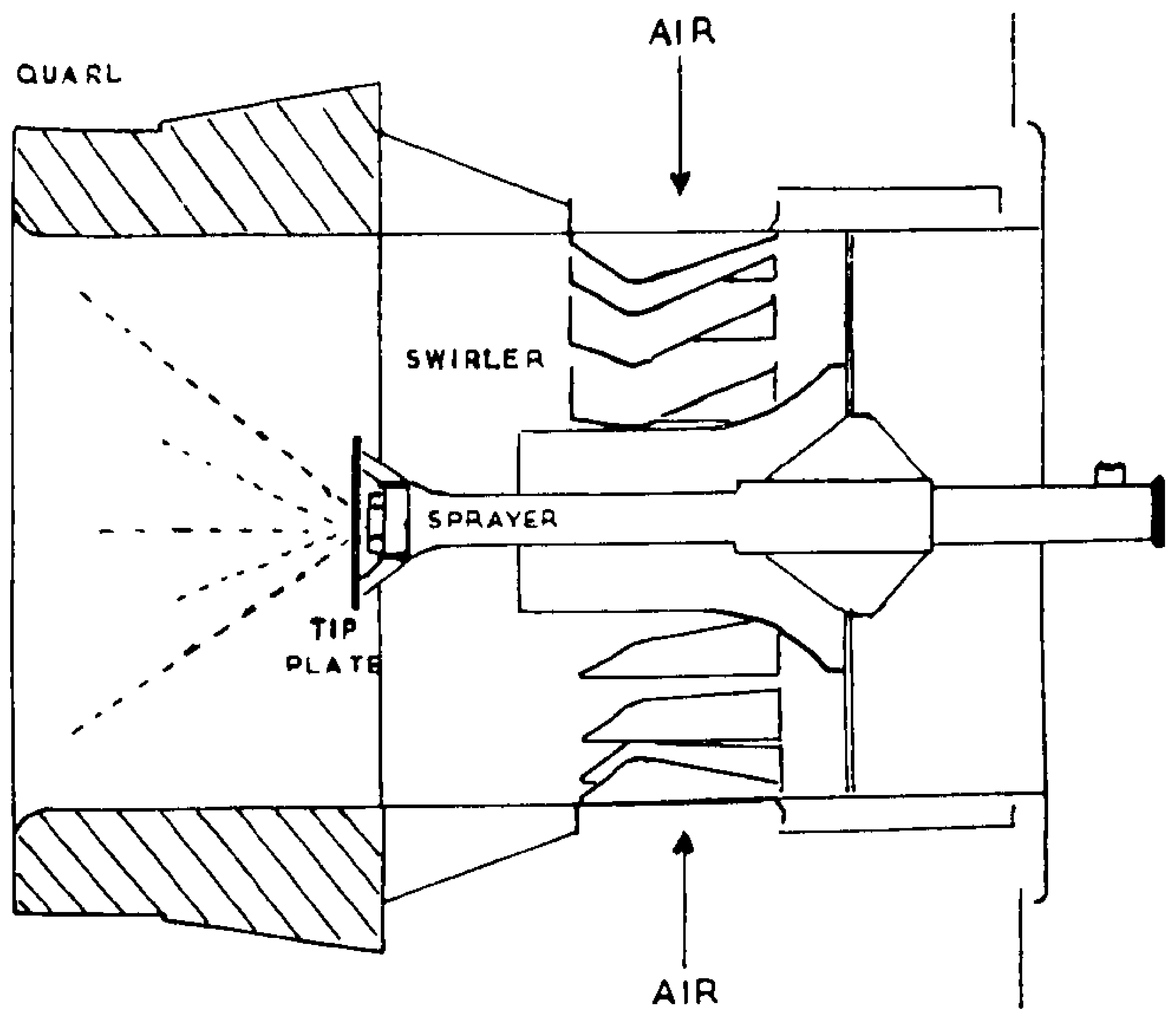



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FIG. 6 (*b*).—*1943-type Admiralty air register with swirler and tip plate stabilizer.*

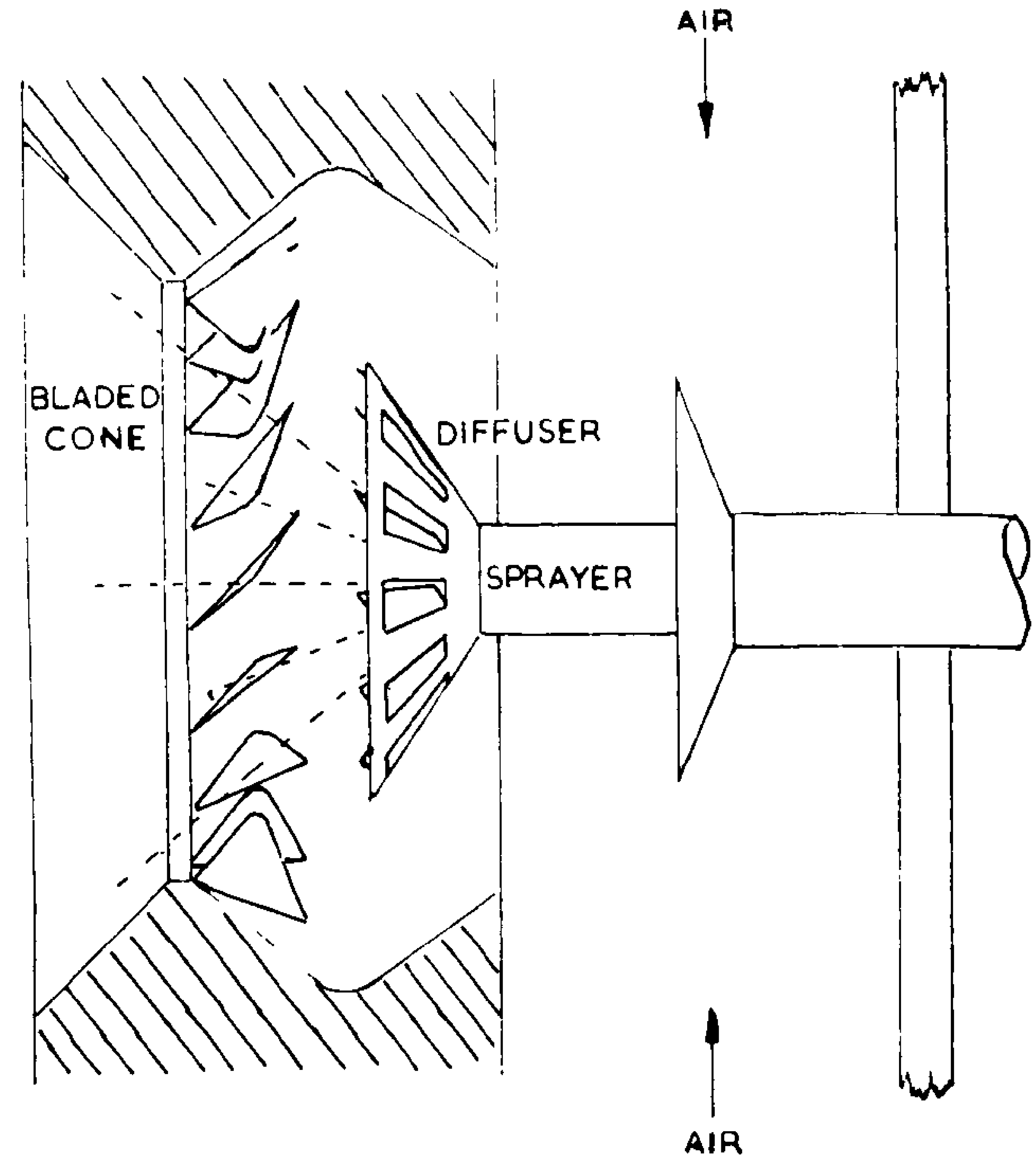


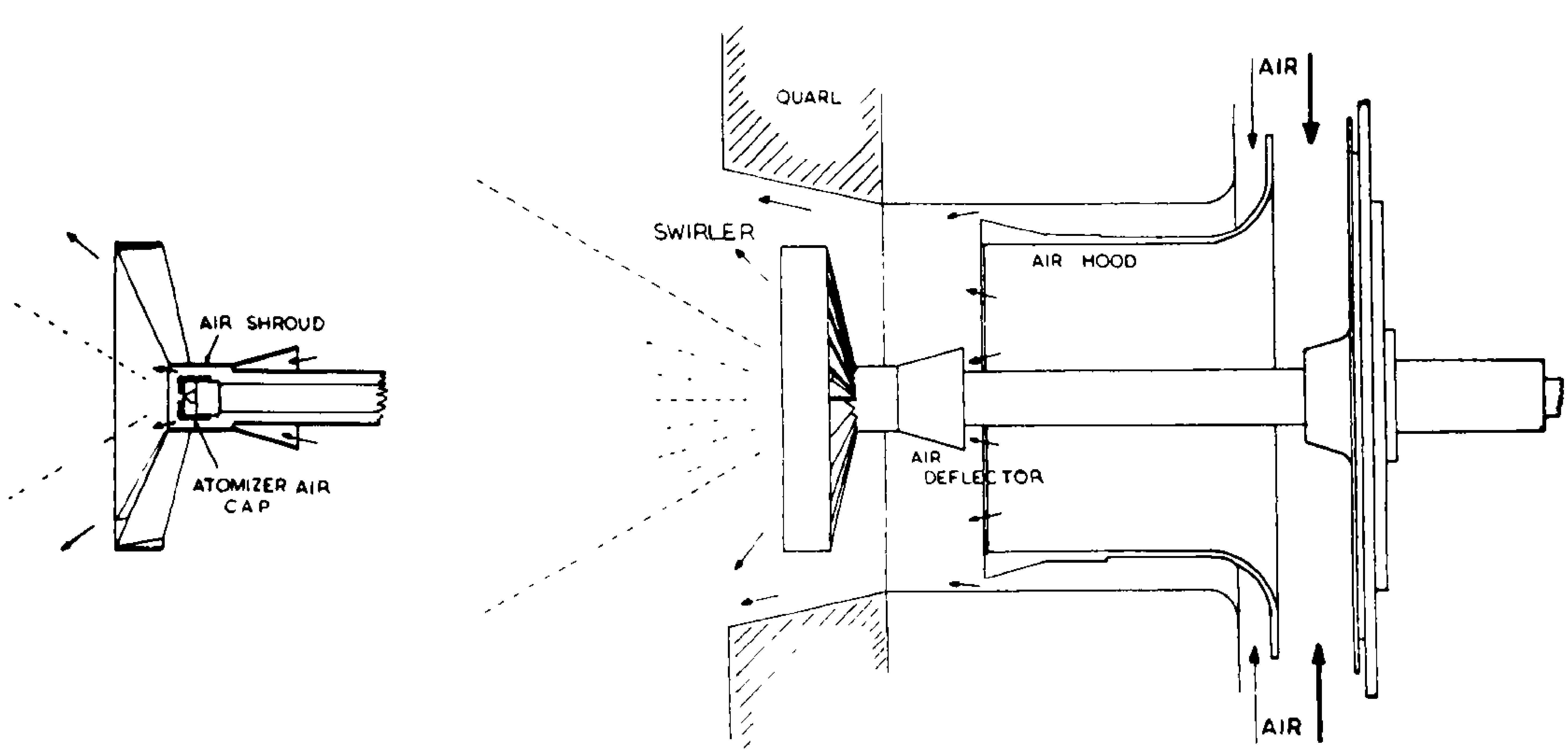

FIG. 6 (*c*).—*Babcock and Wilcox Carolina air register with conical diffuser and swirl vanes (some detail omitted).*

this is related, amongst other things, to the efficiency of mixing air with the burner fluid and the resistance to flow at the inlet and outlet. These factors are often easily modified, and this accounts for the fact that random alterations to the air flow through the register have often in practice cured the trouble. When this theory has been further developed, it should be possible to use it in designing boiler installations so that this trouble is avoided in the operating range.

## 6. PRACTICAL BURNER SYSTEMS

### 6.1. Individual Burners

Although the results of work on flame stabilization reviewed in section 5.3 cannot be quantitatively applied to pressure-jet systems, it illustrates the essential part played by eddies and recirculation in stabilizing the flame. All flames of this type are maintained in the desired position by the establishment of an equilibrium between the rate of flame propagation and the motion of fuel, air and combustion products. It must be remembered, too, that there will be quite large instantaneous variations in the flame, although it appears regular to the observer. Four typical burners and air registers are illustrated in Figs. 6 (*a*), 6 (*b*), 6 (*c*) and 6 (*d*). Each is stabilized by an eddy which is produced in Fig. 6 (*a*) by the downstream constriction of the burner quarl, in Fig. 6 (*b*) by a flat baffle, in Fig. 6 (*c*) by a conical baffle with slots and swirl vanes, presumably to produce smaller eddies, and in Fig. 6 (*d*), which is a modern high-intensity suspended flame register, by a large swirler producing stabilizing vortices in the actual combustion chamber.




[By courtesy of the Admiralty.

FIG. 6 (*d*).—*Modern high combustion intensity suspended flame air register.*

The effect of the stabilizers used in Fig. 6 (*b*) and Fig. 6 (*d*) can be judged from Figs. 7 (*a*) and 7 (*b*) respectively, which are somewhat similar configurations obtained for gas turbines.[19]

Mixing of the combustion air in the systems illustrated is achieved in several different ways, varying from directing the air into the oil spray at the quarl as in Fig. 6 (*a*), which is very sensitive to burner setting, to establishing a swirled-air cone round the spray in the suspended flame register illustrated in Fig. 6 (*d*). In this last instance, analogy with results obtained from a similar gas-turbine arrangement [20] shows how very sensitive the system is to shroud air flow round the atomizer and swirler angle. It appears to be easy to get a large vortex developing which carries the combustion air right away from the flame in the early stages, with consequent trouble from a secondary flame stabilized on the air register. Up to the present a study of the effect of swirl on mixing and flame configuration has only been undertaken for very limited fields,[43,41] and a much more general survey is badly needed.

Methods of design employed for air registers have in the past been as much art as science. There seems to be a wide field not only for the use of the criteria discussed earlier in this paper in the design of more efficient types of burner, but, in addition, for much greater understanding of the flow patterns required to obtain stabilization and good mixing. The obvious techniques for experimental work are scaled-down units and cold air and water analogues.[45,46,47] The scaling of such widely differing parameters as combustion intensity, turbulence, droplet penetration and so on, especially in systems with complicated aerodynamics, presents formidable problems. This work is the special concern of the author's research. Many workers have regarded Reynolds' similarity as being sufficient. However, the suggestion made by Thring [45,46] of allowing for the change of density on combustion by altering the scale of the inlet relative to the combustion chamber, is obviously a more correct approximation. This means that the model used is a model of a system with all gas flow at the flame density, but with the correct inlet mass flow and momentum. Any method of scaling is bound to involve more or less drastic simplifications, and the important results need to be checked on the prototype. Nevertheless, model techniques can result in big savings in time and work by providing the most likely solutions to try on full-scale work, and in clarifying the processes involved.

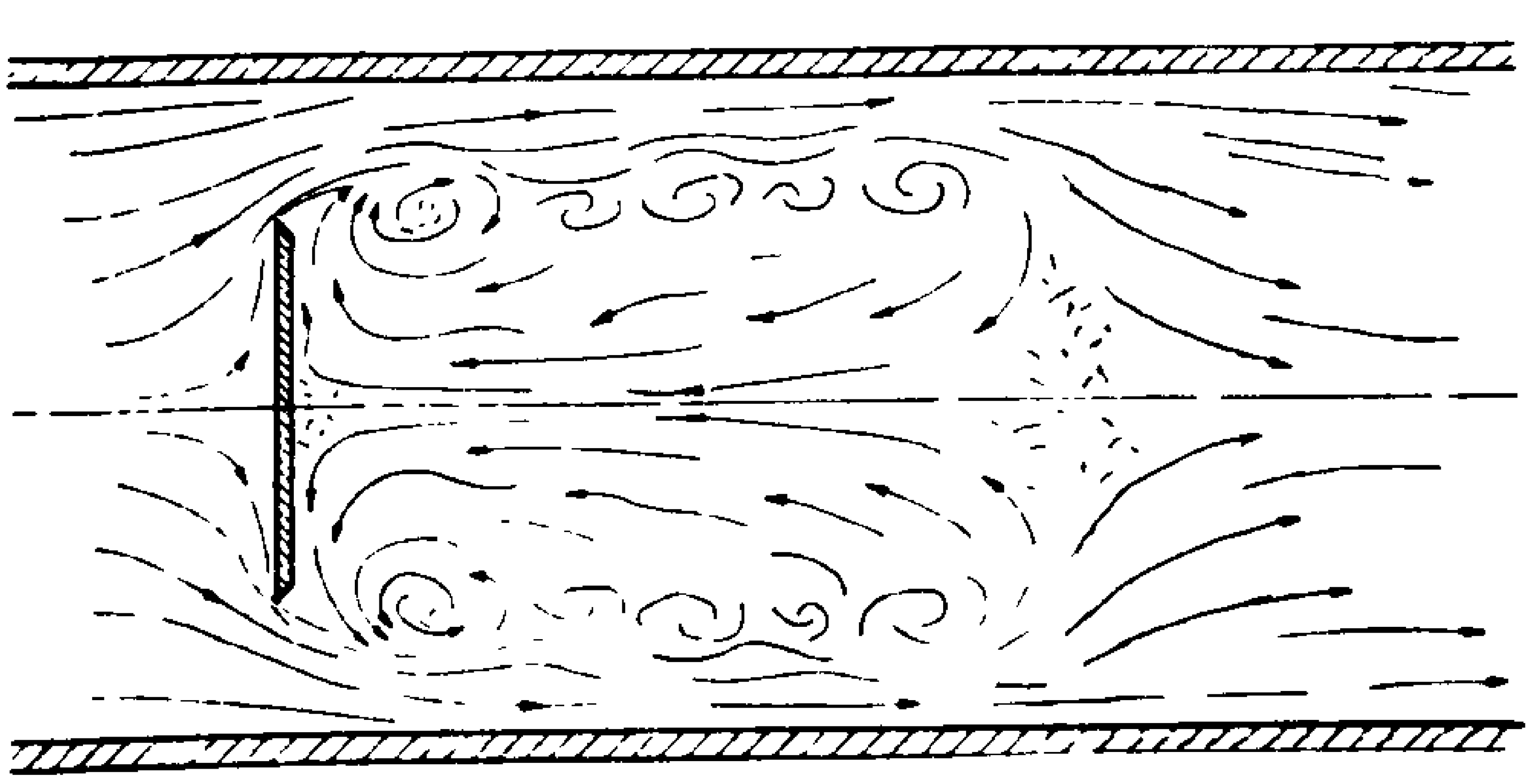


FIG. 7 (*a*) (*above*).—*Flow patterns in gas-turbine combustion chambers: eddies and flow reversal behind flat plate stabilizer.*

### 6.2. Combustion Systems in Boilers as a Whole

Very little consideration seems to have been given so far to the matter of gas flow through the boiler as a whole. The big differences in temperature distribution reported for both power-station boilers [48,49] and marine boilers [50] indicate that the subject of aerodynamic design of boilers also requires study. This has been further emphasized by the study of flow patterns in a high-combustion intensity marine boiler by one of the author's colleagues.[51]

The interaction of several burners firing together, and the increase in pressure drop usually required when doing this, and such matters as the spacing and relative angle of groups of burners and the effect of opposed directions of swirl, are all problems the answers to which would be of considerable interest. To mention a few specific examples, the heat-transfer differences of up to 6 per cent for different groups of burners running at the same load,[50] recently reported, have emphasized the importance of correct burner arrangement. Gollin [4] has recently drawn attention also to the need for evaluating different arrangements of burners in power-station boilers; some possible arrangements are shown in Fig. 8. This last problem is one in which cold model techniques would be particularly applicable.

## 7. CONCLUSIONS

In general, modern trends are towards higher combustion intensities, larger turn-down ratios in conjunction with automatic control and more efficient systems. The achievement of good mixing of fuel, air and recirculation products at all rates of firing requires the application of fundamental research to a much greater extent than has been done in the past, for the improvement of existing burners and the development of new types. The develop-

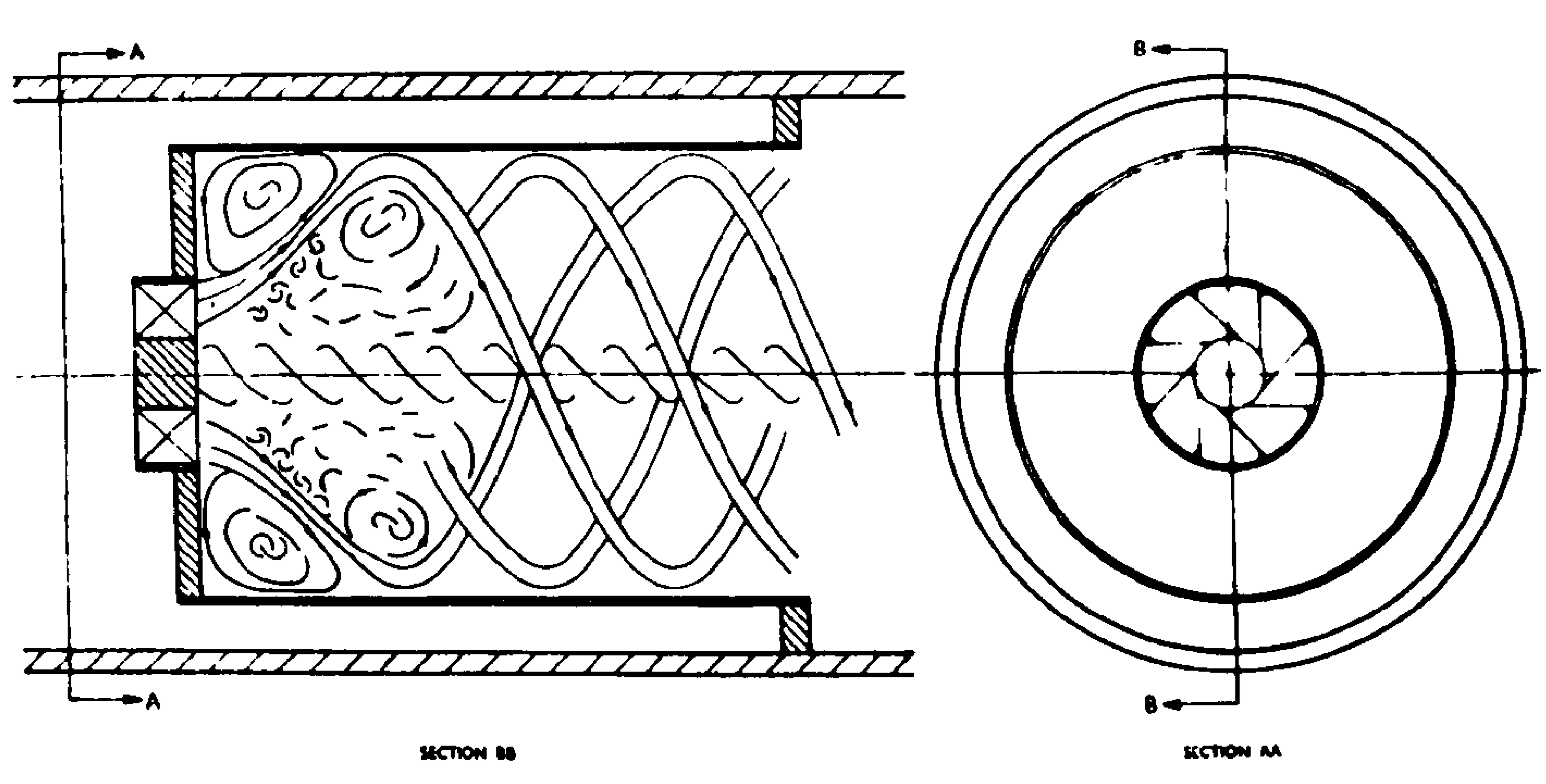



FIG. 7 (*b*) (*right*).—*Flow patterns in gas-turbine combustion chambers: recirculation vortices produced by an air swirler.*

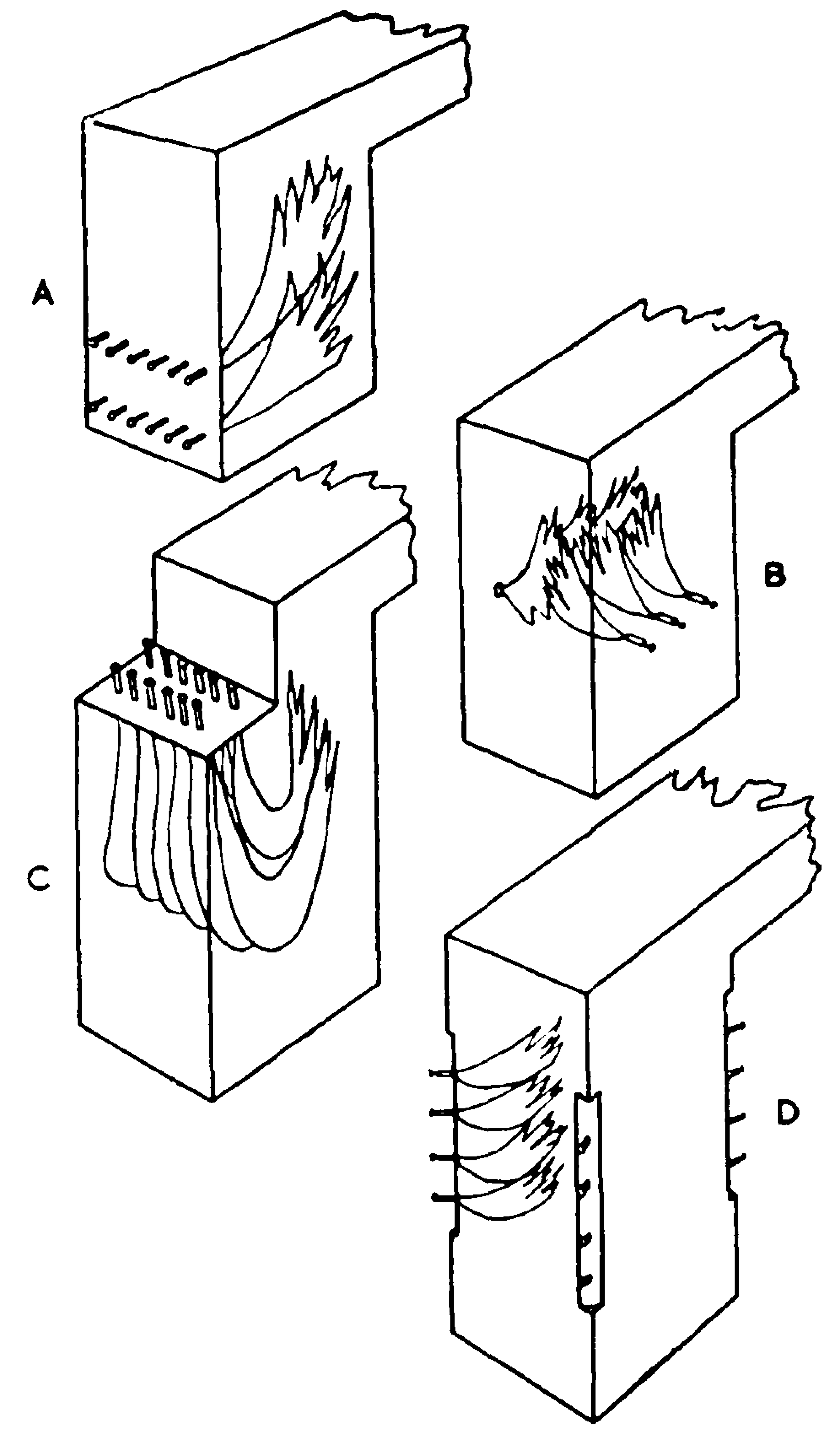



FIG. 8.—*Four possible ways of installing oil burners for a large power-station boiler.*

*A, on the front wall. B, on side walls firing opposed to each other. C, on the roof firing downwards. D, at the corners.*

ment of satisfactory model techniques to speed up this work is also important, and the application of analogues similar to the one-twelfth scale water model shown in Figs. 9 (*a*) and 9 (*b*) should provide a valuable tool at the drawing-office stage of the design of boilers and similar installations.

## 8. ACKNOWLEDGMENTS

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FIG. 9 (*a*) (*on right*).—*One-twelfth scale water model analogue of a high combustion intensity marine boiler.*

*A, burners and swirlers. B, combustion chamber. C, wire mesh simulating constriction caused by tube bank. D, uptake.*

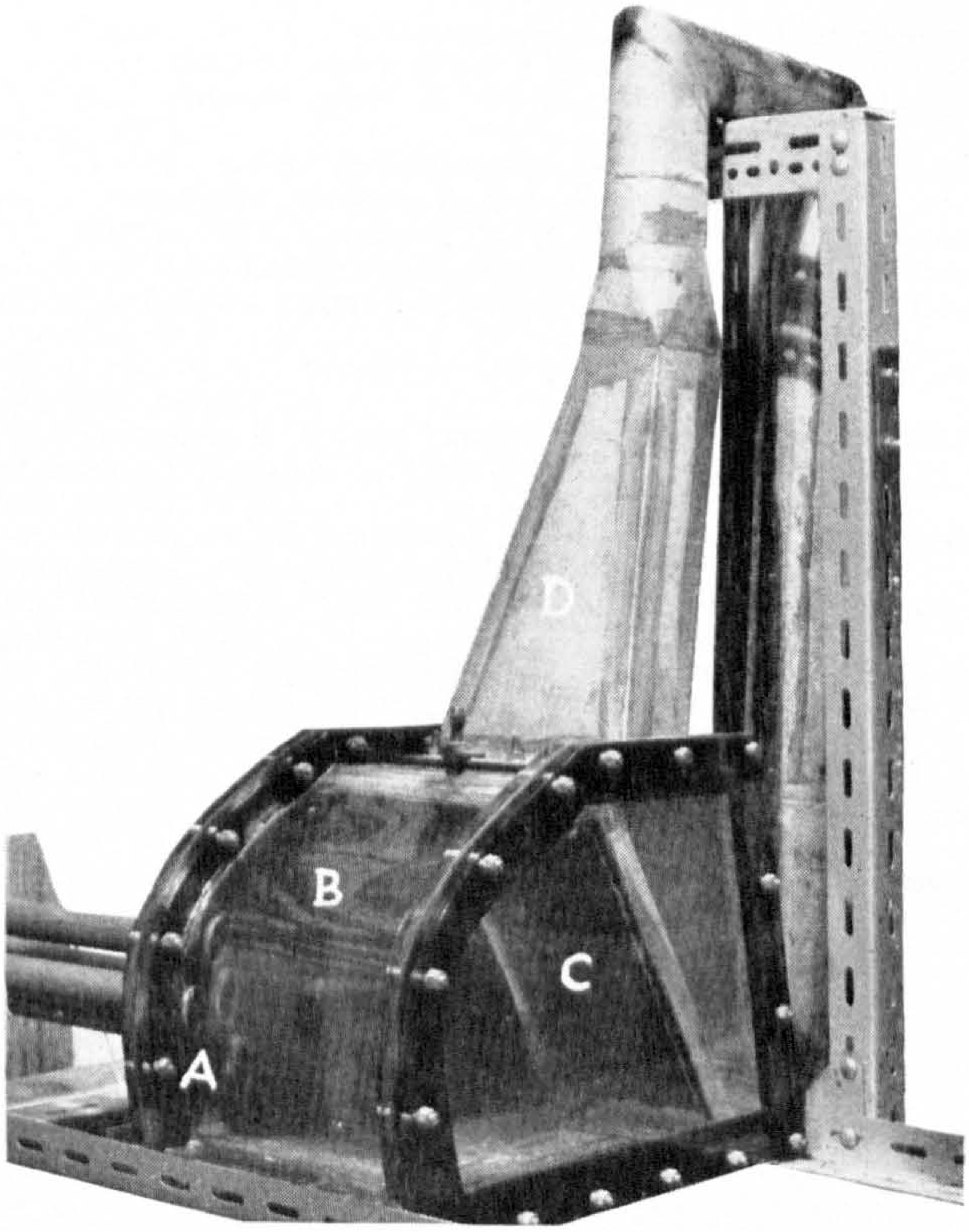
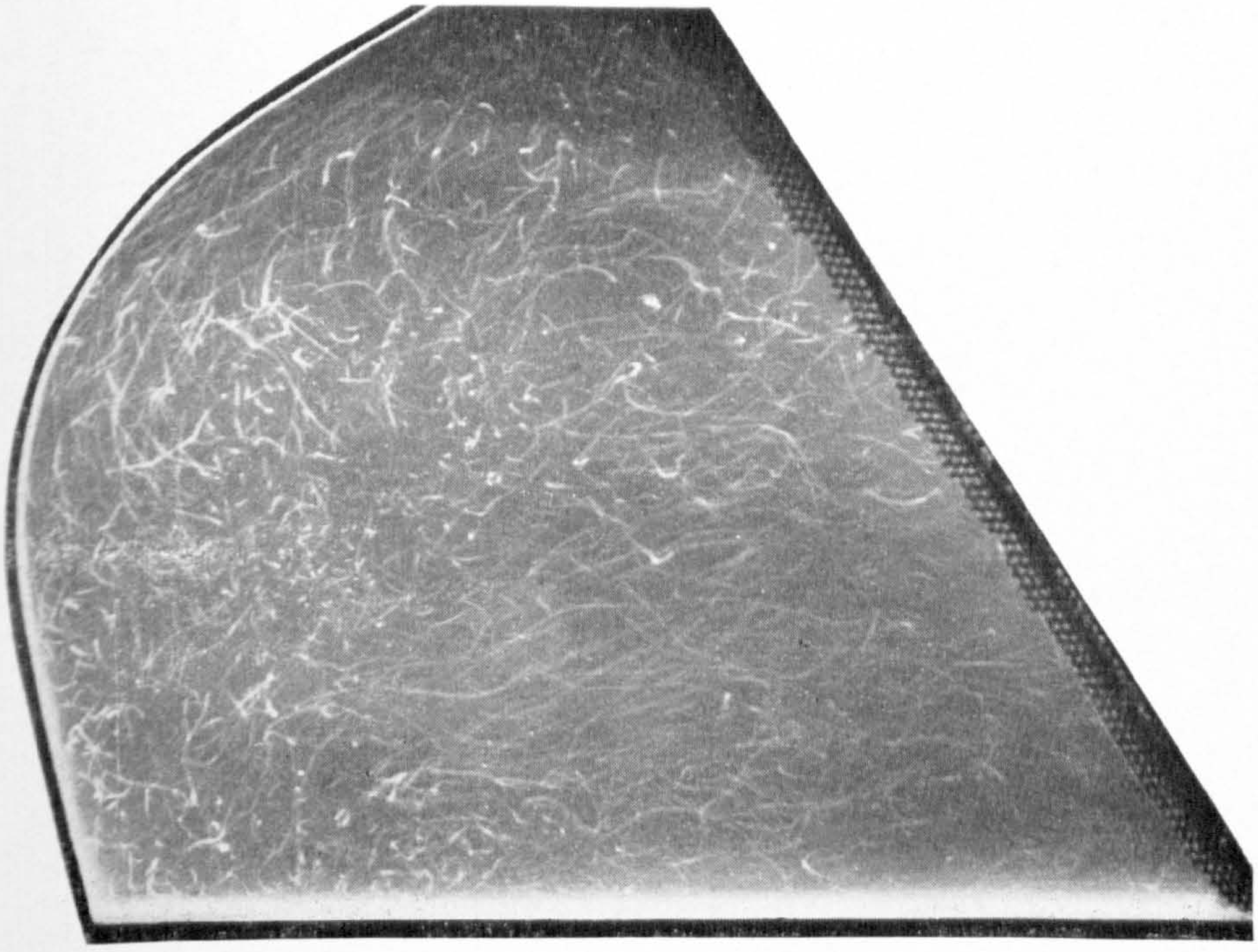



FIG. 9 (*b*) (*below*).—*Typical flow pattern obtained in this type of model, using polystyrene particles as flow indicator. In this instance the illustration shows a narrow plane one scale ft from the back wall of the analogue, with one burner in use.*

*Note*.—(1) *Flow round side of combustion chamber on left-hand side.* (2) *Flow towards tube bank on right-hand side.* (3) *Some flow still along axis of burner* (*left centre*).



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# COMBUSTION OSCILLATIONS

A. M. Brown B. Sc..*

## 1. INTRODUCTION

The phenomena of oscillatory combustion is one which has appeared in sufficient number of guises to merit a considerable amount of experimental research and theoretical study and has given rise to well over four hundred papers in the last twenty years.(1)

In a course on liquid fuel firing any paper of this sort must be concerned mainly with unstable combustion phenomena which occur with conventional liquid fuel firing of which boilers and gas turbine combustion chambers would be typical examples. The type of phenomena observed are limited to pulsations of low frequency in the first case and low and sometimes high frequency pulsation in the second case as well as flame noise. Before looking at these combustion problems in particular, it may perhaps be useful to make some general distinctions between the various unstable combustion phenomena which are known. Apart from effects like 'sensitive flames', which are a consequence of an externally applied force, usually a sound frequency, and oscillation effects which are purely aerodynamic, combustion oscillations can be divided into three classes:-

(1) Random frequency flame noise.

(2) High frequency pulsation with steady frequency often called 'screaming', 'screeching' or 'howling'.

(3) Low frequency pulsation or 'chugging'.

## 2. COMBUSTION OSCILLATION PHENOMENA.

### 2.1. Random frequency flame noise.

This can be important as far as liquid fuel firing is concerned since nearly all flames of any practical interest are non-premixed turbulent diffusion flames. Apart from any sustained oscillations which may occur in these flames, turbulent flames are characterised amongst other things by the flame noise and the diffuse outline of the flame as compared with a laminar flame. Something of the reason for this can be deduced by looking closely at what is happening in the flame. A slow motion ciné film for example shows that what may appear to be a regular shape when normally viewed is in fact a series of rapidly changing transient conditions. This can be seen by the irregular edges of the flame which is made up of ever changing random micro effects, the whole being in a dynamic equilibrium. The presence of turbulence, while not giving rise to very much noise in an unignited flow, may be surmised to give rise to highly irregular local mixing rates and the sudden very rapid ignition of pockets of fuel and air in the flame. Strangely enough very little seems to be known about flame noise and the unpleasantness of noisy flames is a spur to further study. Flame noise of course seems to have a quite random frequency but under some circumstances it may act as a trigger to some of the steady frequency combustion oscillations described in the following sections.

## 2.2. High frequency instability.

The term high frequency instability or 'screaming' in rocket motors and other high intensity combustion equipment covers a quite extensive range of different phenomena ranging in frequency from about 100 c.p.s. upwards, something line 20 kcs being the highest frequency so far observed. This type of instability is characterised by greatly increased heat transfer, sometimes as much as an order higher than during steady combustion and very large pressure changes in the combustion chamber resulting in many cases in the damage or even distruction of the combustion chamber. The oscillations can have several modes of vibration either on their own or superimposed on each other, these can be longitudinal, transverse or tangential in form. There also seem to be three different wave forms possible, their characteristics being set out in Table 1 and Fig. 1.

TABLE 1.

| Wave Form | Amplitude as % of combustion chamber pressure. | Type of wave |
| --- | --- | --- |
| Sine wave | Only moderate amplitude 20 - 50 % | Acoustic |
| 'Saw-tooth' wave | Up to about 100% | Acoustic in form but involving a relaxation effect |
| 'Saw-tooth' wave with high velocity | About 5 times combustion chamber pressure, but increasing rapidly with time. | Shock or detonation wave, much faster than sound, Mach 2 - 3, very non-linear. |

Theoretical analysis of high frequency instability has in the main attempted to relate the frequency and amplitude with some function of the combustion chamber dimensions. The causes of damping are not very well understood although it has been noted amongst other things that the viscosity of the gases makes very little difference, that spray damping may be an important effect and that a large initial triggering energy is required to start the oscillation off. The question that still largely remains to be answered is the real nature of the driving mechanism in the process. Until a good deal more is known about this, chances of really effective control and avoidance of 'screaming' are somewhat limited. In the analysis of longitudinal oscillations, which is the only mode so far really attempted, Crocco(2) draws attention to what appears to be one of the most important variables and that is the time delay in the system due to the combustion process the time delay is accounted for by the fact that a finite time must elapse after the injection of the fuel and oxidant for mixing and ignition before they can effect the pressure and temperature of the system. Crocco regards this delay time as made up of a short (0.1 - 0.5 m.sec.) duration 'sensitive' component of the time delay which is affected by the variation in the rate of the processes in the combustion chamber and a rather longer (1 - 5 m.sec.) 'insensitive' time delay due to some of the dimensions of the system, for example the configuration of the injector. This concept of delay time seems to be a basic requirement for all combustion driven oscillations of both high and low frequency types and control of this means a big step in the control of oscillations.

## 2.3. Low frequency pulsation or 'chugging'.

The distinction between high and low frequency pulsation appears in many cases to be somewhat diffuse with a steady transition from one type to the other, but since a much wider range of combustion equipment is likely to exhibit 'chugging' involving in some cases factors which are of little importance in the previous section it is convenient to distinguish between the two. Low frequency oscillation is generally classified as that below about 200 c.p.s. and in fact in most reported cases in conventional oil fired equipment 60 c.p.s. and often well below are the usual case. Although 'chugging' is a much smaller amplitude effect than 'screaming' and can be tolerated without destruction of the combustion chamber it can be very inconvenient in a number of ways. It reduces the combustion efficiency very considerably, it disturbs control equipment that is dependant on combustion chamber pressure or metering the supplies of air or fuel. In rockets. chugging is also a possible source of the energy necessary to trigger off high frequency instability. In boilers it gives rise to a noise nuisance making smoke and occasionally flame blow out, while in some cases prolonged pulsation can result in damage to the boiler. It is possible too at high turn down ratios that the range of air/fuel ratios over which stable combustion can occur can be very small indeed. For a given set of conditions a steady frequency is usually maintained and seems to be developed from a progressive amplification of small oscillations to a limiting amplitude. Several forms of 'chugging' have been observed:

(1) Pulsation which is relatively insensitive to mixture ratio, and sensitive to changes in a characteristic dimension of the combustion chamber and total pressure, below a limiting value of $\Delta P/P$. This is only likely to be observed in rockets where pressure changes are large.

(2) Pulsation in which the combustion process provides a driver for a resonant frequency in the system. Here the mixture ratio affects the amplitude considerably, but does not affect to any great extent the frequency which will largely be a function of the chamber dimensions, temperature, and total pressure, if this varies very much. The rate of fuel input also appears to be fairly unimportant.

(3) Where conventional oil firing techniques are used, which means that there are large disparities between the fuel input pressure and that of the air, only the air side will be affected by the pressure fluctuations in the combustion chamber resulting in changes in the actual mixture ratio during the pulsation cycle. In many cases below a certain limiting pressure drop for the air supply sustained oscillations can be maintained under fuel rich conditions. The frequency of the oscillations seems to be very sensitive to mixture ratio while at the limiting condition, where pulsation just starts, is sensitive to the fuel throughput and to variations in inlet and outlet resistances. Important factors which affect the amplitude are the air/fuel ratio and mixing conditions. These observations are more fully described in a later section.

(4) On some large boilers with a trunked air supply to the burner, in addition to the pulsation described under 3, a very low frequency 'panting' is observed at low powers under certain conditions of considerable excess air. This has a frequency over the range observed which is unaffected by the firing rate. The frequencies observed were of therange 1 - 3 c.p.s.

Another rather curious form of very low frequency oscillation has been reported as occuring in a power station boiler fired on pulverized brown coal. This seems to have been insensitive to alterations in the combustion condition but apparently to have been a function of the moisture content of the coal.

It will be seen that observed pulsation phenomena of the 'chugging' type show wider variations than is generally realised. It is therefore, all the more to be regretted that on top of the relative sparsity of literature on 'chugging';especially as far as conventional oil firing is concerned, much of the published data is very inocmplete. This makes comparisons difficult. Nevertheless it is obvious that no one mechanism is likely to be applicable to all the above observations, especially as these can differ even in the same combustion chamber according to the actual conditions. It is possible to envisage three likely modes of vibration:-

1) <u>A mode of vibration dependant on total pressure.</u> Here the physical conditions of mixing, mixture ratio, injector characteristics etc., alter relatively little during the pulsation cycle and only the reaction rate has a large variation with a consequent change in entropy. All that is required for this mode is a variable which will have a substantial effect on the reaction rate, the most obvious one is the absolute pressure. In most rocket motors the mixture ratio during the cycle is unlikely to vary. The flame largely fills the combustion chamber and pressure fluctuations which are high are likely to have an entropy mode of vibration.

2) <u>A mode dependant on the mixing conditions in the system.</u> This might be described as a flow disturbance mode since in this case the pulsation can be regarded as having a large effect on the mixture ratio, mixing conditions recirculation or vortex pattern in the combustion chamber and must be regarded as a three-dimensional effect. This sort of mode is likely particularly to occur in a system like a

boiler where the flame only occupies a small fraction of the total chamber volume and flame shape, mixing conditions and air/fuel ratio are all likely to vary during pulsations.

3) An acoustic mode. This will require a driving force if oscillations are to be continously sustained. The driving force will presumably be provided by one of the other modes, the resonant frequency of the system being super-imposed on the natural frequency of the driving oscillation. The only requirement for this is a suitable resonant frequency and a power feedback from the combustion process sufficiently high to overcome the damping forces. Otherwise for low gas velocities at least it is independant of the combustion process.

## 3. ANALYSIS OF THEORIES PUT FORWARD TO EXPLAIN LOW FREQUENCY OSCILLATIONS.

### 3.1. Theories not involving acoustic quantities.

The clearest way of looking at the various theories is probably along the lines of the comparative table, suggested by the author and Professor Thring(3) shown in Tables II (a) and (b). Here the work of resprestative authors are compared but it is, of course, by no means exhaustive of the literature available. These theories apply to 'chugging' although not necessarily limited to it only. The division is made into 'non-acoustic theories', where the total pressure and mixing modes described in 2.3 have their place, and into theories, dependant on the 'Acoustic Mode'. For the 'Non-acoustic Theories', the 'total pressure' theories can be regarded as dealing with a constant mixture ratio for the fuel and oxidant throughout the pulsation cycle, while in the 'mixing mode' theories the mixture ratio is likely to alter by virtue of the reaction of the pressure fluctuations on the low pressure air supply, but not on the high pressure oil supply. (i.e., a pressure of less than 20 inches w.g. as opposed to several hundred p.s.i.).

Crocco & Cheng[2] have considered idealized rocket combustion chambers for 'chugging' for the cases where the rate of injection remains constant and only the chamber pressure varies (i.e. affecting only the 'sensitive' component of the time lag) and also cases where the rate of injection varies as well (i.e. affecting the 'insensitive' part of the time lag) for both mono and bi-propellant systems. They assume for their simplified model that:-

1) Gas pressure is practically uniform throughout the combustion chamber at every instant and oscillates about a mean value as a whole.

2) Temperature variations are small and practically constant and uniform irrespective of the pressure oscillations.

3) The mixture ratio remains constant during the pulsation cycle; in the case where small variations are considered as occurring, corrections are made to the mass balance in the system, which is then regarded as having a fixed mixture ratio.

The result of this is to make the reaction rate solely a function of pressure variation.

$$f \sim P^n$$

where 'n' is the pressure index of reaction between combustion processes and the oscillations in the combustion chamber. Unfortunately this index is in practice dependent on other factors than pressure and with complexities of the combustion process insufficient information is available to be able to calculate this directly. While the above assumptions are reasonable as far as rocket motors are concerned obviously under conditions of conventional oil burning, assumption 3 cannot be regarded as valid. Nor can the pressure variations, which are usually less than 1% of the absolute pressure in the combustion chamber, be assumed to have any significant effect on the reaction rate. This introduces the need for a mode of oscillation dependent on the physical conditions

TABLE II(a).

NON-ACOUSTIC THEORIES FOR LOW FREQUENCY COMBUSTION INSTABILITY.

| AUTHOR | MODE OF OSCILLATION | SYSTEM | ARRANGEMENT CONSIDERED |
|---|---|---|---|
| CROCCO & CHENG[2] | Total pressure mode. | Mono- & Bi-propellant rockets | Ratio of oxidant to propellant assumed sensibly constant during pulse. Reaction rate altered only by fluctuating chamber pressure. |
| TISCHLER & BELLMAN[4] | As above. | Bi-propellant rockets. | Similar to above. |
| THRING[5] | Mode dependent on mixing conditions. | Boilers. | In this theory the pulse is assumed to affect <u>only</u> incoming air rate. Hence mixtures are not constant during pulse and small fluctuations in pressure have a large effect on the reaction rate. It is thus mainly concerned with the inlet conditions. |
| FRITSCH[6] | As above. | Boilers (water tube, 3-pass, etc. | Similar to above, but here the main concern is the interaction of combustion chamber, tube passes, stack, etc., down stream of the combustion chamber. |

MECHANISMS SUGGESTED.

With all non-acoustic theories it is necessary to explain the 'inertia' and 'spring' effects as well as the feedback mechanism.

Crocco & Cheng and Tischler & Bellman assume that the feedback is the effect of absolute pressure on the reaction rate (increase P then the reaction rate increases and so on, +ve system). Thus large fluctuations in P are required to alter substantially the reaction rate.

The theories dependent on the mixing conditions assume, on the other hand, that the reaction rate and consequently the combustion chamber temperature are dependent on the effect of the pressure drop at the inlet on the air flow rate and mixing in the combustion chamber and are thus sensitive to small fluctuations in the pressure in the combustion chamber.

in the combustion chamber where mixing, mixture ratio and input flow of air may all be regarded as affected by the pulsation. A mode dependent on the mixing conditions in the system has been considered by Thring[5] and with some embellishments by Fritsch[6].

In Professor Thring's approach an idealized system is considered. This is assumed to consist of an air inlet pipe, oil supply system, combustion chamber and stack. Two things are considered necessary for stable oscillation. (a) A tuned circuit or a system with a natural stable frequency which can arise if there are two delay times in the system of the same order of magnitude which interact with each other, and (b) a power feedback mechanism which replaces the energy absorbed in the natural damping of the system. One of the delay times is regarded as being the dwell time of gases in the combustion chamber and the other the delay time between the fan and the combustion chamber or in the stack. The ignition time delays as opposed to residence or decay times in the system can be considered as represented by the inertial effects in the supply tube in responding to pressure changes in the combustion chamber and the delay time due to the time taken for mixing between the air and the oil. Several possible conditions could be postulated.

(1) The upstream pressure is kept constant and the mass flow rate across the register is regarded as responding immediately to a change in the combustion chamber pressure but only completes this exponentially after a certain decay time. See Fig. 2. (a).

(2) The pressure is regarded as constant at all points in the inlet pipe, but varies with time as a consequence of the flow rates in and out of the pipe. See Fig. 2. (b).

(3) The same thing but with the second delay volume on the exit side.

TABLE II(b).

ACOUSTIC THEORIES FOR LOW FREQUENCY COMBUSTION INSTABILITY.

| AUTHOR | SYSTEM | PHENOMENON CONSIDERED |
|---|---|---|
| RAYLEIGH[7] | Helmholtz resonator. | $\frac{1}{t} = \frac{c}{2\pi}\sqrt{\frac{A}{V.l}}$ |
| ELIAS & GORDON[8] | Rockets | Treated as modified Helmholtz resonator. The gas flow is ignored. |
| PUTNAM et al.[11] | Rijke tube, gas and oil fired domestic boilers etc.. | Treated as 'organ-pipe' or single or double necked Helmholtz resonator as applicable. Rayleigh criteria applied. Open end 'organ-pipe' $t = \frac{2L}{c}$ Closed end 'organ-pipe' $t = \frac{4L}{c}$ |
| MERK[10] | Rijke tube & idealized combustor. Premixed and linear conditions assumed. | Rayleigh criteria applied. In this case the total combustion chamber pressure affects the flame propogation velocity and thus the position of the flame front and the velocity of the hot gases in the combustion chamber. |
| SANDERS & LAWRIE[12] | Domestic oil fired boilers. | Low amplitude oscillations only and treated as Helmholtz resonator. |

MECHANISM.

With all acoustic theories the only problem seems to be to explain the feedback mechanism. The conditions for instability are always regarded as being those proposed by Rayleigh: 'The fluctuating component of the heat addition must be in phase with the pressure wave'. The necessary condition for this being an ignition time delay. Hence the rate of heat addition is a +ve. function of the absolute pressure. In the Helmholtz resonator the gases in the neck provide the 'inertia' term and those in the 'body' the'spring'term. In Putnam's case no theoretical estimate is made for damping and feedback and hence the mechanism is not clear, but basically it appears to be as above the effect of pressure on heat liberation.

The temperature can be assumed to vary proportionately with the mass flow rate or with some derivative of this. In fact the case where temperature varies with mass flow rate in conjunction with condition 1 above appears to be the combination that will in fact sustain oscillations of a likely frequency. In all non-acoustic treatments it is necessary to account for an 'inertia' term, a 'spring' term, 'damping' and feedback. Taking

$$T - \bar{T} = k(\dot{m}_a - \bar{\dot{m}}_a)$$

these requirements can be satisfied by writing a differential equation of the form:-

$$t_1 t_c \frac{d^2 \dot{m}_a}{dt^2} + \left\{\frac{\bar{p} t_1}{R_e \bar{\dot{m}}_e} + t_c \left(1 + \frac{k\bar{p}}{R_a \bar{T}}\right)\right\} \frac{d\dot{m}_a}{dt} + \dot{m}_a \cdot \left(1 + \frac{R_a}{R_e}\right) \frac{\bar{p}}{R_a \bar{\dot{m}}_a} = \text{Const.}$$

where the first term will be the inertia, the next the damping and feedback, and the last the spring term. This implies that stable oscillations will be possible if:

$$-k \geqslant + \frac{R_a \bar{T}}{\bar{p}} + \frac{t_1}{t_c} \cdot \frac{R_a}{R_e} \cdot \frac{\bar{T}}{\bar{\dot{m}}_e}$$

or in other words if the second term is zero or negative. Linear relationships have been assumed between mass flow and the pressure drops in the system have been assumed to be linear to simplify the mathematics but this does not obscure the processes involved. This means in effect that the flow rate into the combustion chamber will be out of phase with the temperature and pressure fluctuations.

This also means that oscillations will be inhibited if the inlet resistance is increased, if the ratio of combustion chamber volume to high velocity inlet volume is reduced, if the exit resistance is reduced or if the mass flow through the system is reduced, although these are not always possible or desirable. The most satisfactory thing to do, however, is to reduce the ignition delay time by reducing the fraction of the combustion chamber that is occupied by unmixed air by improving the mixing of air and oil at the burner.

The suggestions put forward by Fritsch are on generally similar lines but are particularly concerned with multi-pass boilers where there are several delay volumes in the system. He also introduces a term to account for the dynamic behaviour of the flame including the heat losses from the flame to the walls. This term is an added refinement, although it makes the theoretical treatment more complicated. Unfortunately some of his conclusions seem at variance with observed trends, at least for normal fuel rich pulsation.

### 3.2 Acoustic Theories.

Some contributions to the theory of acoustic oscillations are given in Table 11 (b). The mechanism of pulsation in this case is regarded as being a standing wave in an 'organ-pipe' or 'Helmholtz' resonator type of system, typical cases being illustrated in Fig. 3, and the only factor which requires to be added to this to completely describe the system is the feedback mechanism and the damping force. This means that the observed frequency is at or near a resonant frequency of the combustion chamber at the particular pressures and gas temperatures encountered. This frequency should be largely independant of the feed rates for fuel and air, which in the main will only affect the conditions for which stable oscillations can be maintained.

The basic requirement for sustained oscillation was put forward by Lord Rayleigh[7] that is:- 'a component of the fluctuating heat addition must be in phase with the pressure wave'. In other words the rate of heat addition must be a positive function of pressure if the oscillation is to be driven. As even in completely pre-mixed gas flames there is an ignition delay time suitable conditions can be produced by the combustion process when the velocity of air or mixture at the inlet will lag or lead the pressure and temperature fluctuations by a phase shift of the order $\pm\frac{1}{4}$; $\pm1\frac{1}{4}$; $\pm2\frac{1}{4}$; etc.

Most of the other papers under this heading are largely concerned with additions to the theory of resonating systems. Elias and Gordon(8) regard the combustion chamber as a straight forward resonating system and ignore the flow. Corrections for this have been made by other workers(9) and these are of the form

$$f_L = f_{L_0}(1 - M^2).$$

$f_L$ = longitudinal frequency.

$M$ = Mach No.

It can be seen that unless unusually high velocities occur in the system this correction will be very small, representing a frequency change of 3% for a 50ft/sec flow at combustion chamber temperatures.

An analysis of some of Putnam's results carried out by Merk(10) are particularly interesting, although for premixed gas flames, the analysis is an attempt to take into account the alteration of conditions arising from variation in the size and shape of the flame during combustion as a result of variations in the velocity of flame propagation.

On more conventional oil fired systems Putnam and others(11) (12) have done a considerable amount of work on small domestic oil fired unit. In all the cases listed under this head the evidence is considered as showing that standing waves are set up of the frequency range 20 to 40 c.p.s. see Fig. 4. Since their concern has largely been the noise nuisance of pulsating flames they have mainly been concerned with amplitude rather than frequency and a band of high amplitude over certain air/fuel ratios is reported, see Fig.5. Although the reason for this is not clear, the experimental results seem to show reasonable agreement with the suggested theory, except for a few important exceptions. There is, however, a conspicuous lack of explanation for the necessary feedback mechanism and for the nature of the damping forces.

Unfortunately despite this large range of mechanisms described in the last two sections the combustion process is a very complex matter and all of these are only an approximate attempt to indicate simply the important factors affecting flame instability, and it is not always easy to make certain comparisons between theory and practice. To show some of the complications involved some of the work recently undertaken by this Department is out-lined in the next section.

## 4. RECENT EXPERIMENTAL WORK ON OIL FIRED COMBUSTION UNITS.

### 4.1. Pulsation phenomena observed on large marine boilers.

Recently two investigations have been carried out on marine boilers on H.M. ships to try and determine the nature of the pulsation phenomena observed under actual service conditions. In both cases the boilers were of the 'Yarrow' type, fairly similar in design. They have a trunked air supply to the six burners and are designed to give fairly high combustion intensities with register draught loss figures going up to about 15" w.g. A block diagram of the type of boiler together with the various measuring points is shown in Fig.6. Measurements of pressure fluctuation were made in the first case with microphones and in the latter case with strain-gauge pressure transducers. Two quite distinct regions of pulsation were observed, one quite violent, one in the fuel rich region with a frequency of between 5 and

20 c.p.s., and another region at low powers at <u>high</u> air fuel ratios with frequencies of about 2 - 3 c.p.s. Except in a few cases the fuel was a heavy fuel oil.

For the fuel rich type of pulsation which occurred more or less in the 'black smoke' region of the boiler's operation the following principal observations were made:

(1) The onset of pulsation occurred quite sharply over the whole power range as the air and hence the air/fuel ratio was reduced. Except at the very low powers this onset occurred at a definite R.D.L. figure, perhaps representing the point where the damping exceeded the possible feedback power. Frequencies at or rather near the limiting value are plotted for one boiler in Fig.7. which shows the very restricted range of stable combustion at low powers and also the conditions beyond which black and white smoke occur.

(2) The frequency of pulsation varied rapidly up to about $\frac{1}{4}$ power (see Fig.8.) and thereafter linearly with fuel input rate. It appears that for the two boilers there are significant differences in the limi6ing R.D.L. and hence air input rate and also frequencies for comparable fuel input rates, suggesting perhaps that alterations in resistances and delay times in the system have substantial effects. It was also noted that a switch over can occur for higher fuel rates to a pulsation with a higher limiting R.D.L. and corresponding frequency. At the start of the pulsation it was also noticed that the frequency was a good deal higher than the steady value and that this decreased as the amplitude built up, suggesting the influence of inertial forces increases as the pulsations get properly established.

(3) The change from a heavy fuel to a distillate one also seems to have quite a marked effect on the frequency, presumably because of alterations in the mixing conditions and ignition delay times between the two fuels.

(4) Pulsations initiated in the combustion chamber quickly lead to the establishment of coupled pulsation of similar amplitude in the air trunking, air box and uptakes. This was usually in phase although occasionally, one section would be out of phase with the others.

(5) The present pulsation increased the mean pressure drops in the system.

(6) Alterations to the flame cone were also considerable during the pulsation cycle, oscillations about the steady state position of the flame cone are shown for fairly mild pulsation in Fig.10. and in the film extract. In many cases this distortion of the flame front results in the flame blowing right back on to the front wall presumably with considerable interference with the normal mixing and recirculation pattern in the flame.

In the case of the low frequency oscillations the main points observed are:

1) That the frequency is almost constant over the fuel input range measured and for different burner combinations.

2) That they occurred between definite air/fuel ratio limits, it being possible to increase the air to the point where combustion was again stable.

## 4.2. Pulsation phenomena observed on a small scale experimental unit.

Work is being carried out in parallel with the large boiler work on two small experimental units in the

Department in which it is much easier to alter the various variables involved and these are being shown in the demonstrations. As far as oil firing is concerned, Capt. Papadakis has been able fairly easily to get the fuel rich type of pulsation, but so far the panting low frequency type of pulsation has not been reproduced. In general the results show similar trends to those observed on the large boilers with a few additional facts:-

1) The frequency of pulsation seems to show big changes with the mixture ratio as the mixture becomes more fuel rich, becoming lower as the air decreases until the flame is extinguished.

2) The frequency does not show any close correlation with the mean gas temperatures in the combustion chamber as would be required for an acoustic mode of oscillation. The temperature appears to fall slightly at the highest frequencies.

3) The presence of a secondary source of ignition, in this case a spark igniter near the burner when turned on, stops or greatly attenuates the amplitude of the pulsations. This was also noted in the case of the oscillation with the wet brown coal fuel mentioned earlier, and is presumably due to reductions in the ignition delay time for the flame.

4) Coupled pressure fluctuations occur throughout the system and there is no evidence of amplitude modes as observed by Putnam.

### 4.3. Pulsation involving a resonant frequency of part of the boiler structure.

There is no reason why the oscillations should not drive some resonant frequency in the structure of the system instead of an acoustic mode. In one experimental boiler where combustion gas velocities were unusually high, violent oscillations of the boiler tubes occurred, and in this particular case it was found by 'cold blowing' that the oscillations were entirely aerodynamic in origin and were independent of the flame. However, tube vibration or similar effects might well impose a cyclic variation on the outlet resistance and hence allow flame driven oscillation. This can be shown on a bench demonstration where the outlet has a damper mounted on a spring.

### 4.4. Relation of experimental results to theory.

As has already been indicated, combustion oscillation theory is still a long way from being sufficiently refined to explain all the observed results. It is possible, however, to use these results to determine the type of oscillation mode which applies for a given system and to compare the trend predicted by theory with the observed fact. In the case of the fuel rich oscillations observed, these are almost certainly of the non-acoustic type dependant on the physical conditions of mixing etc., in the system. In this connection it is interesting to compare some observed results with the shape of the curve predicted by the Thring theory, assuming this to give the correct frequency at 1/2 power. Fig.9. The results from experimental work also help to show which variables are actually most important in the oscillating systems and even this can be a considerable assistance in controlling or avoiding pulsation.

## 5. CONTROL AND PREVENTION OF PULSATION.

With such a large number of possible phenomena there is no easy way out. As our knowledge of the processes involved increases it should in many cases be possible to 'tune out' the pulsations so that the right conditions for oscillations do not occur over the working range of the equipment, and some ways of achieving this have already been indicated. There are two basic things which can be done. The damping in the system can be increased or the ignition delay time can be decreased (this automatically increases the oscillation frequency and hence the damping forces). The first of these can be achieved in ordinary oil fired systems by increasing the inlet resistance at the air register and hence the register draught loss or by increasing the combustion chamber size. Neither of these are always possible and in the second case is distincly undesirable. In oil fired systems the factor which will most affect the ignition delay is the mixing of the air, recirculated combustion products, and the oil, and this is a field that would seem to be a very fruitful one for study. In pressure jet flames, particularly, penetration of the air to the centre of the spray cone always seems to be poor, this is very well demonstrated, in a gas composition survey of what may be regarded as a good pressure jet flame with plenty of turbulence, which has been carried out by the author. The plot of CO concentrations shown in Fig. 11. shows how high the concentration of unburnt fuel is at the centre of the flame which means that any alteration in the mixing conditions is likely to produce big alterations in the heat release rate, a factor not conducive to avoiding pulsation.

## 6. USES OF OSCILLATORY COMBUSTION.

The final question is can oscillatory combustion be used for anything? At least higher frequency combustion oscillation can certainly be used. This is based on the greatly increased heat transfer rates due to the alteration in diffusivity and hence burning velocities during the process. This offers the very interesting possibility, as has been suggested in a number of patents and is already in operation in Germany, of constructing combustion chambers with very high combustion intensities indeed and hence very much smaller than a combustion unit fired by any conventional method. One of the chief problems in this is likely to be in avoiding objectionable noise at least on larger units.

The other possible use is the very ingenious one suggested by Reynst[13] in which the oscillating combustion is made to draw fuel and air into the combustion chamber in the low pressure part of the cycle where it is ignited by residual pockets of burning material left over from the previous pulse. As such an arrangement avoids pumps and compressors it is claimed that such a process could be made more efficient than for example a gas turbine. One possible use of this process would be to drive the striated gas systems necessary for the direct conversion of heat to electricity using electro magnetic gas braking.

## ACKNOWLEDGEMENTS.

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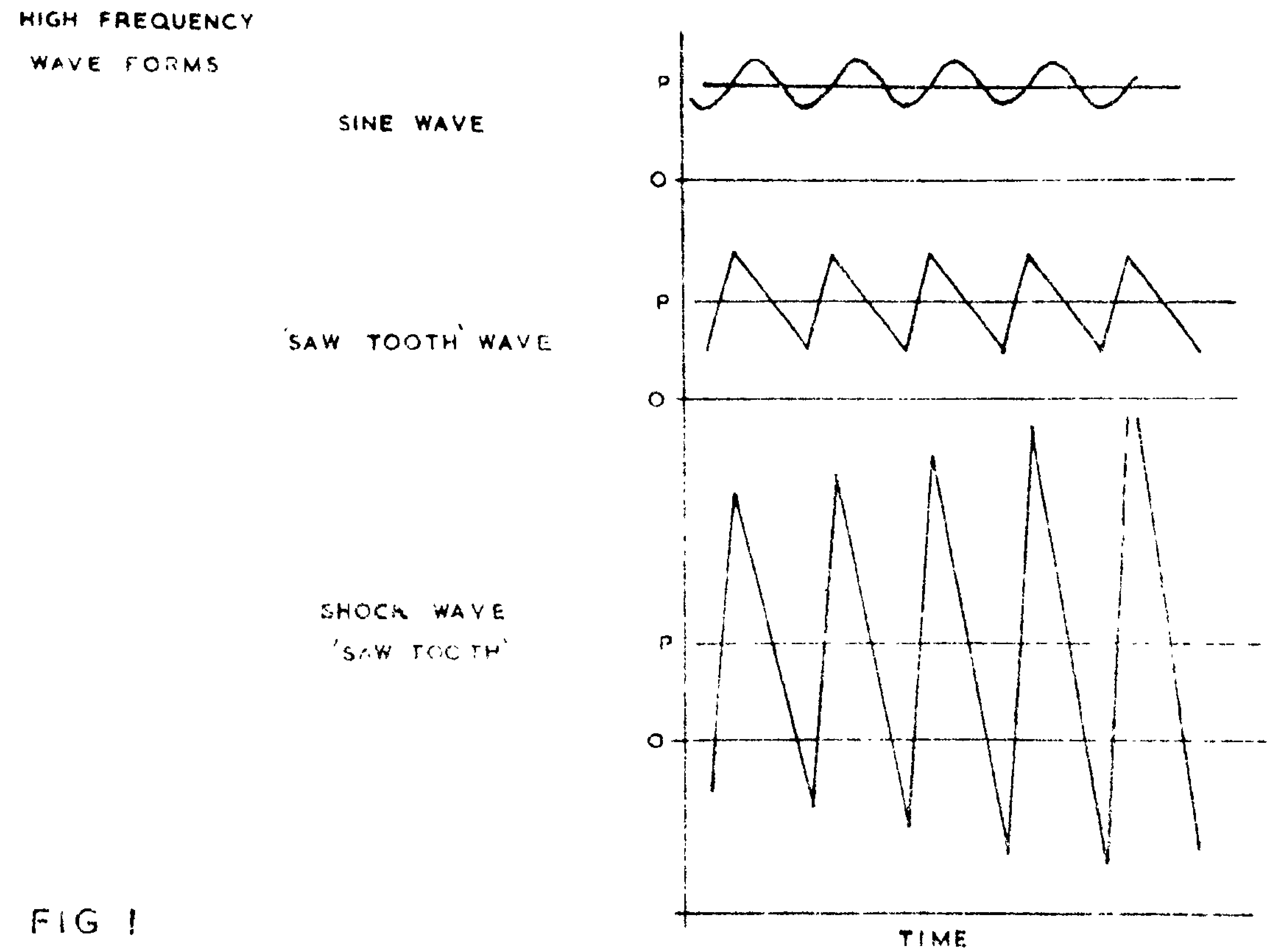



FIG 1

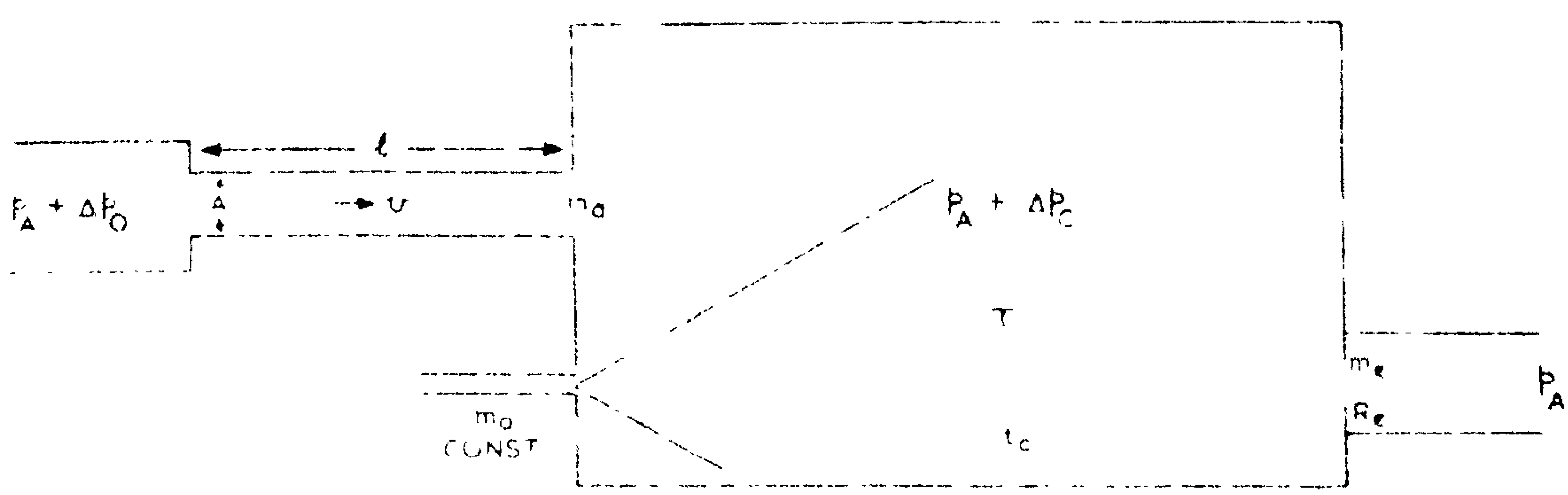



FIG. 2a

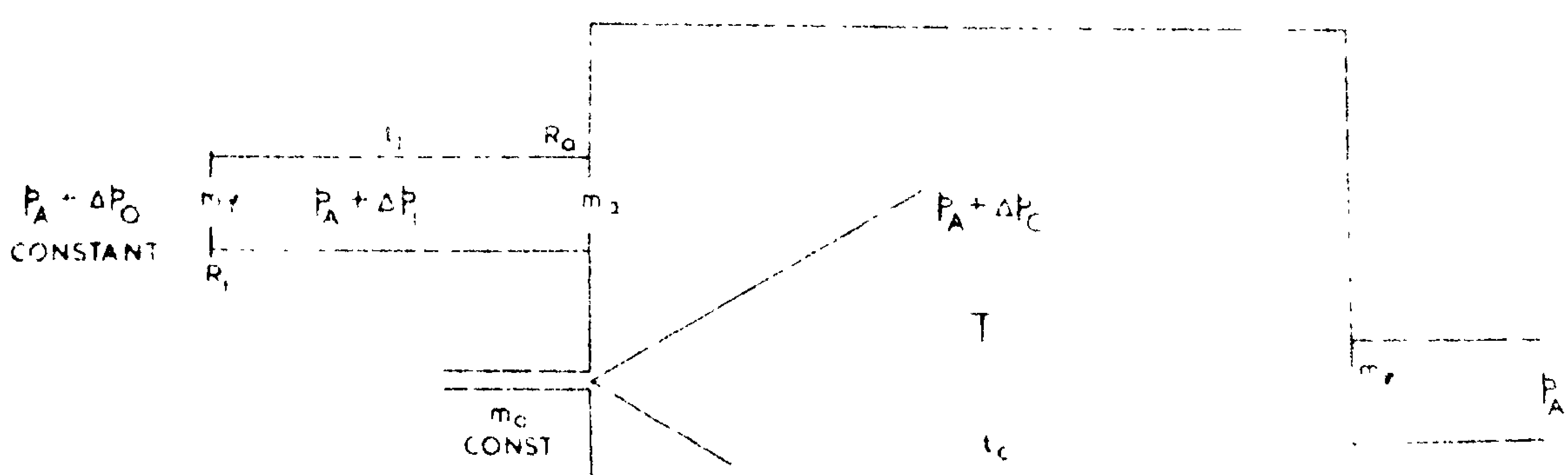
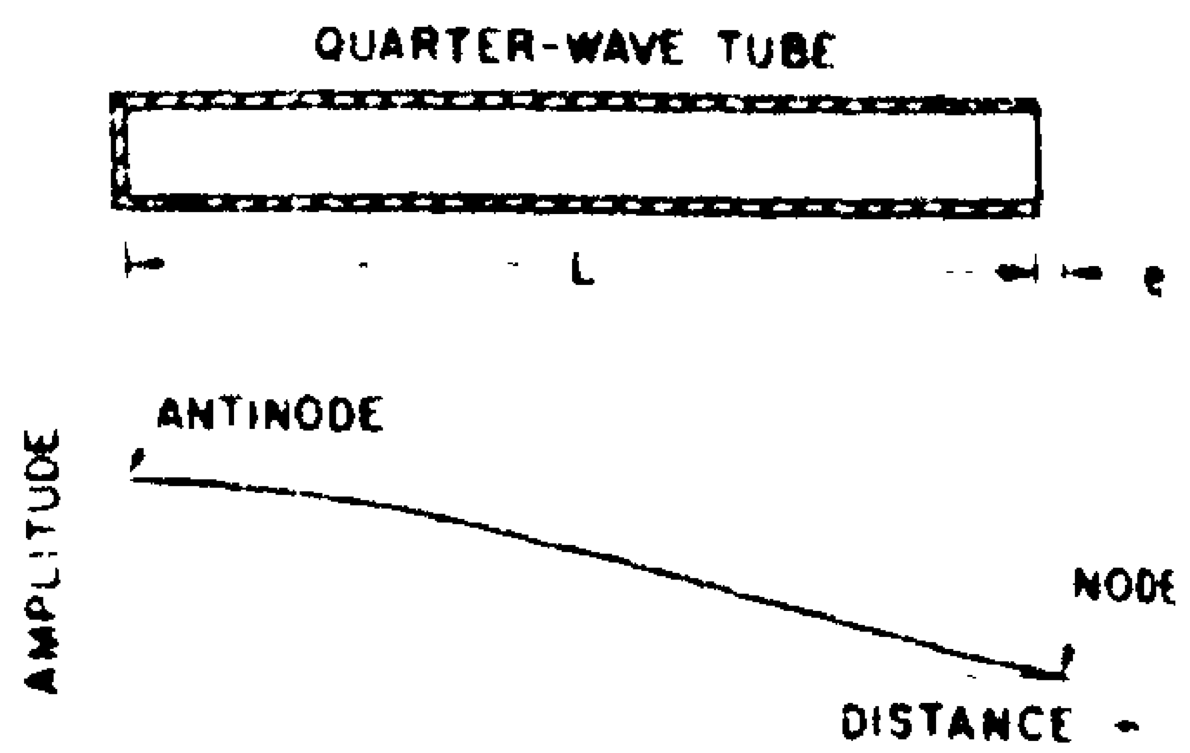
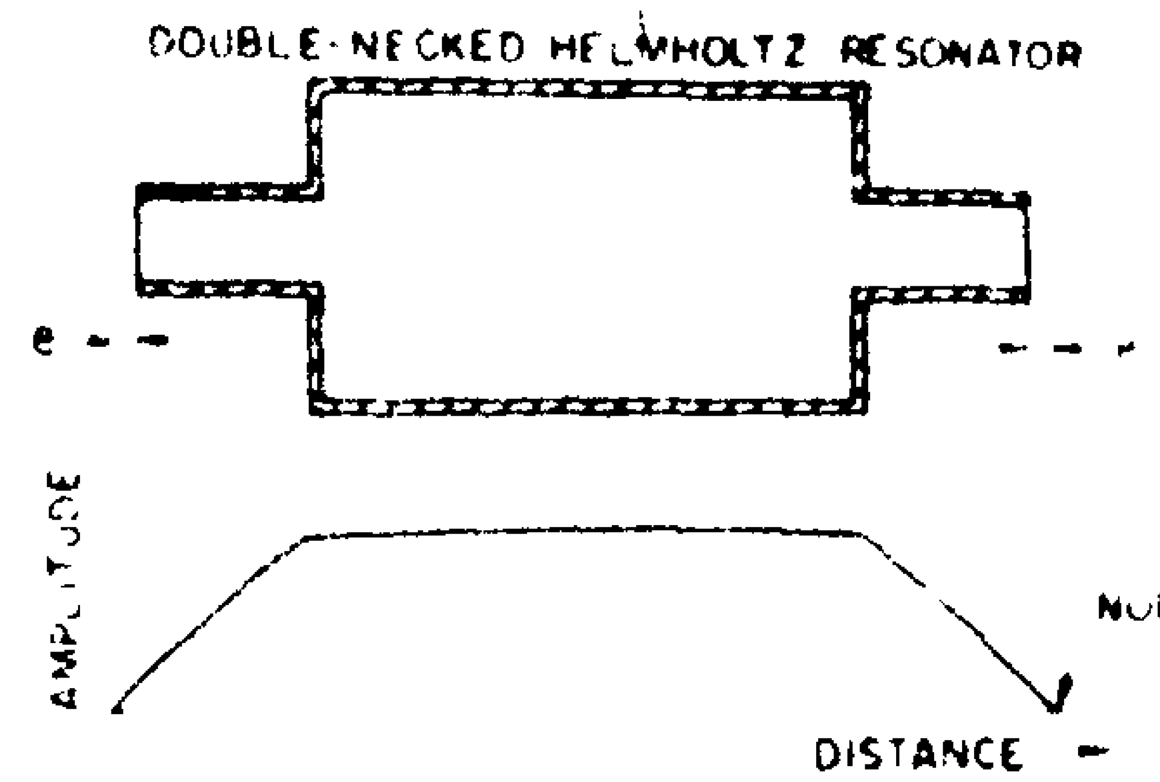



FIG. 2b.



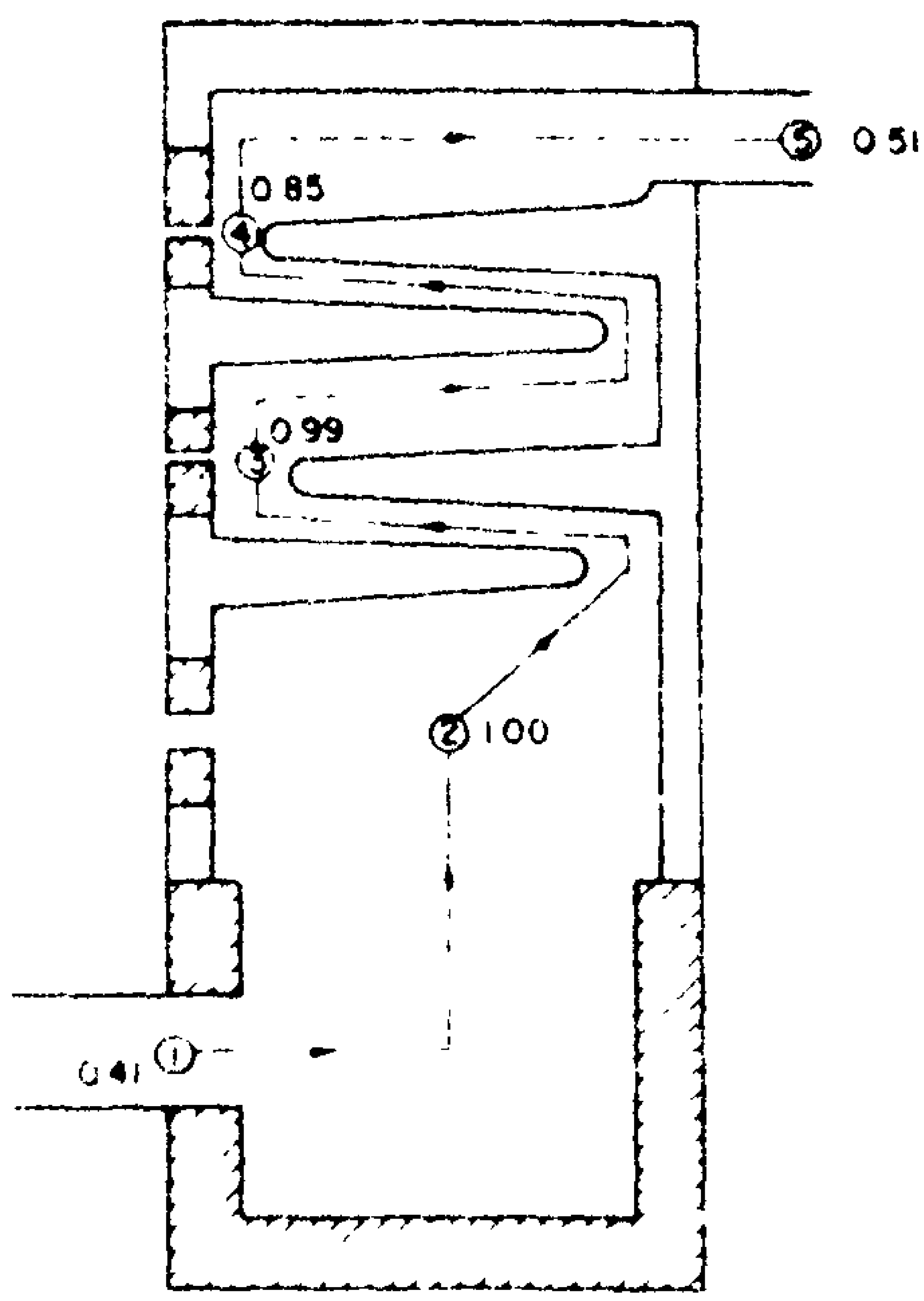

**Standing wave in a closed organ pipe (quarter-wave tube)**

FIG. 3a.



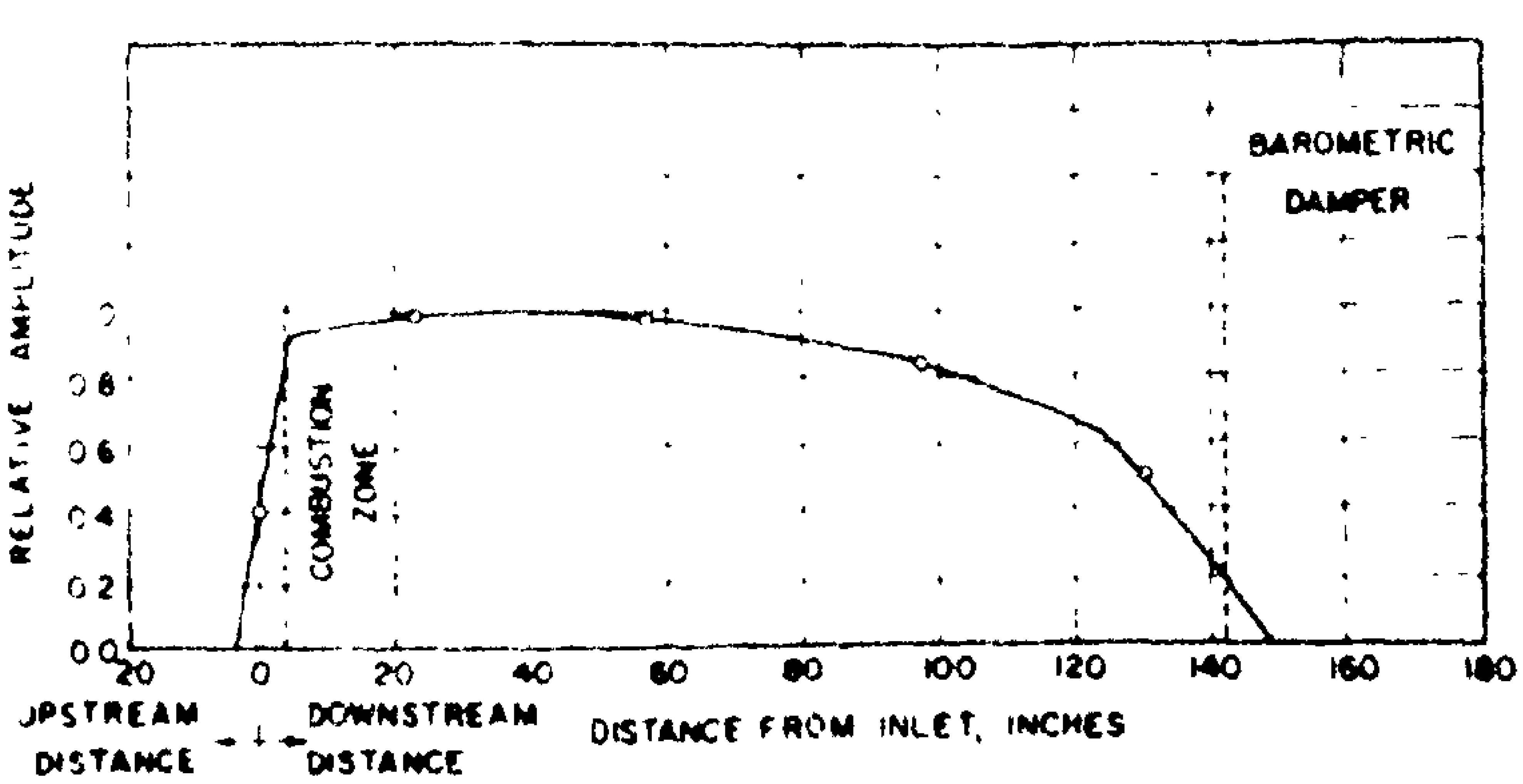

**Standing wave in a double-necked Helmholtz resonator**

FIG. 3b.

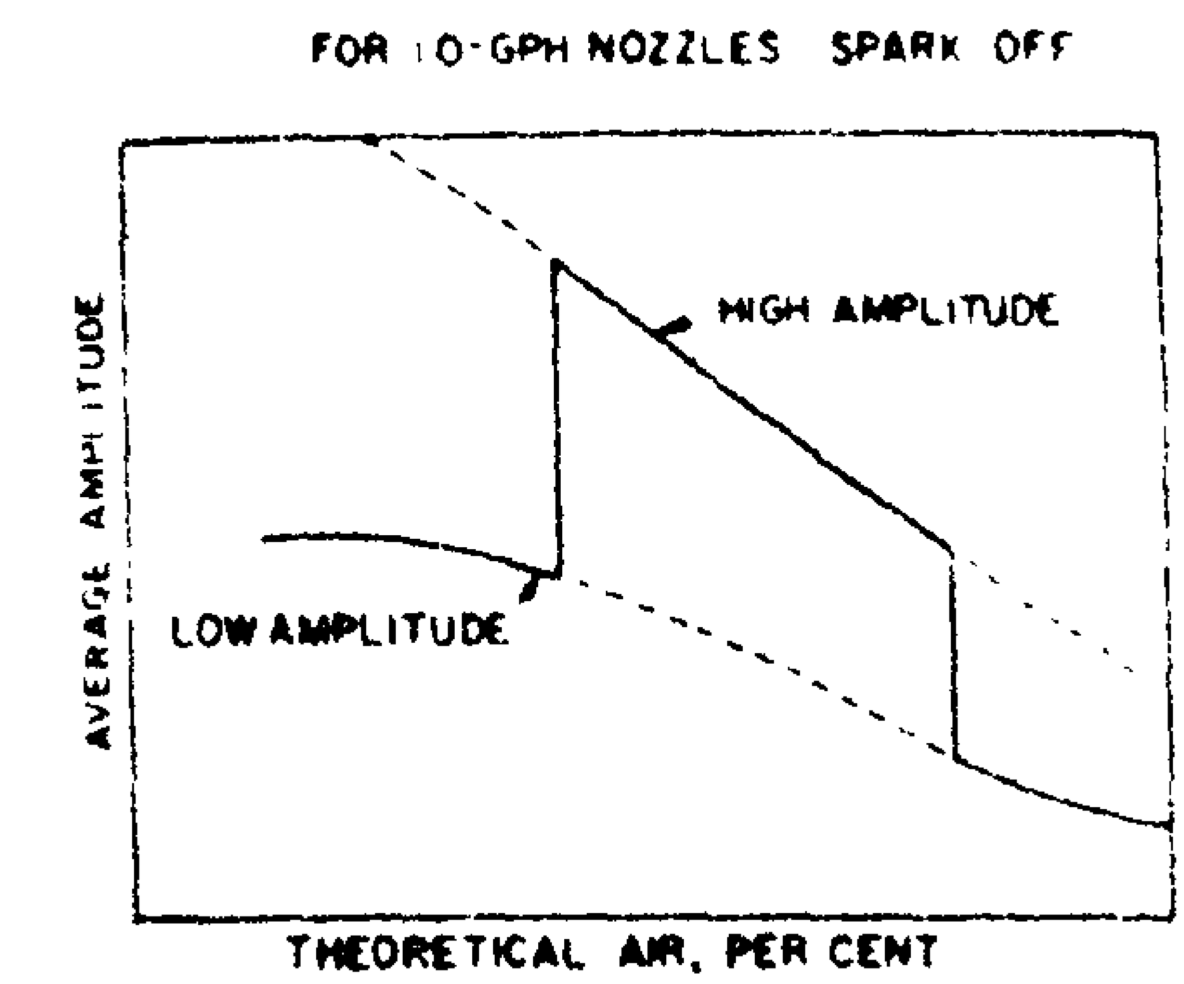



***(left)*—Sketch of tubeless boiler, showing path of gas travel (numbers denote amplitude relative to amplitude at point 2, taken as 1.00)**

FIG. 4.

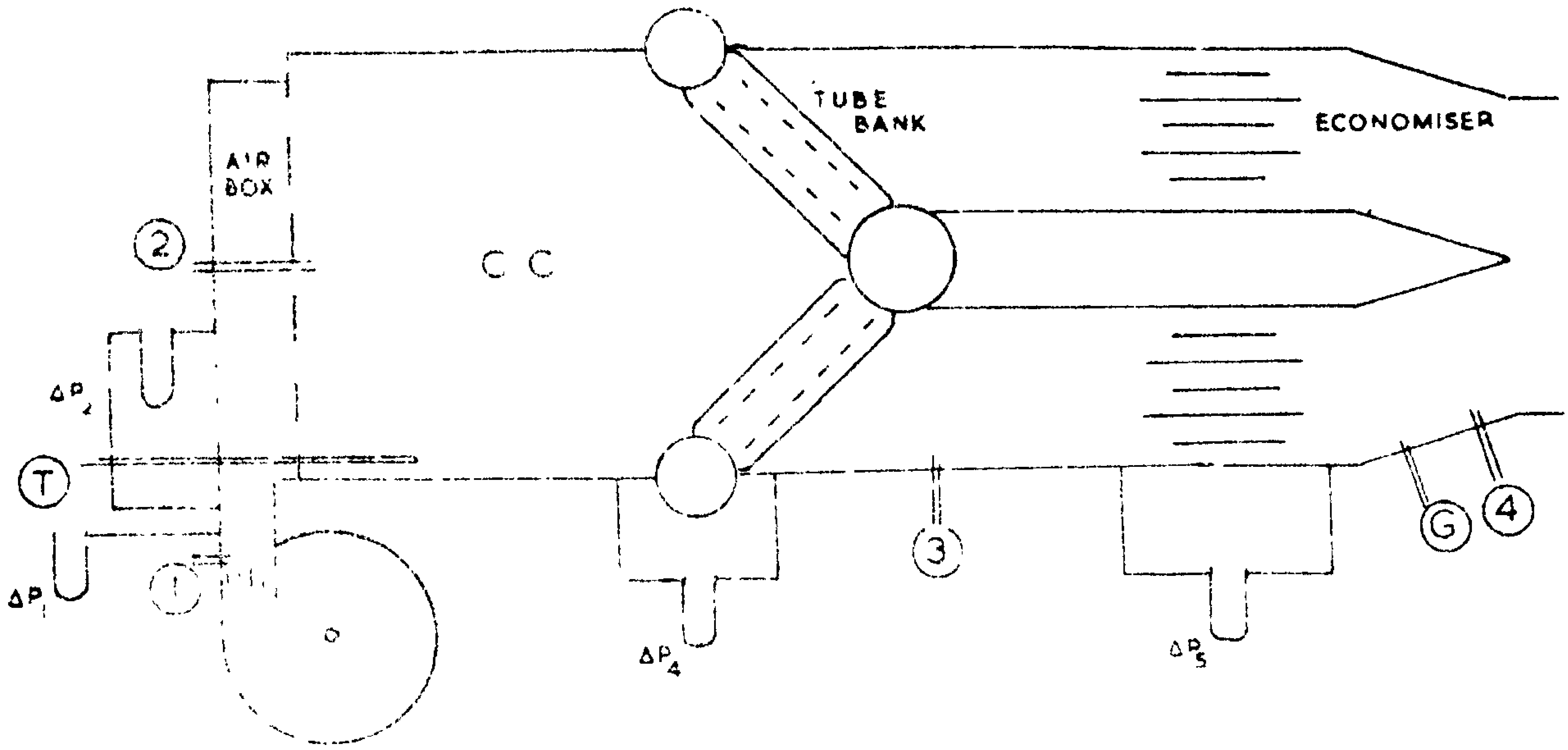



**Oscillating-pressure distribution in tubeless boiler**




**Sketch of low- and high-amplitude ranges of oscillation**

FIG. 5.




BLOCK DIAGRAM SHOWING MEASURING POINTS
FIG. 6.

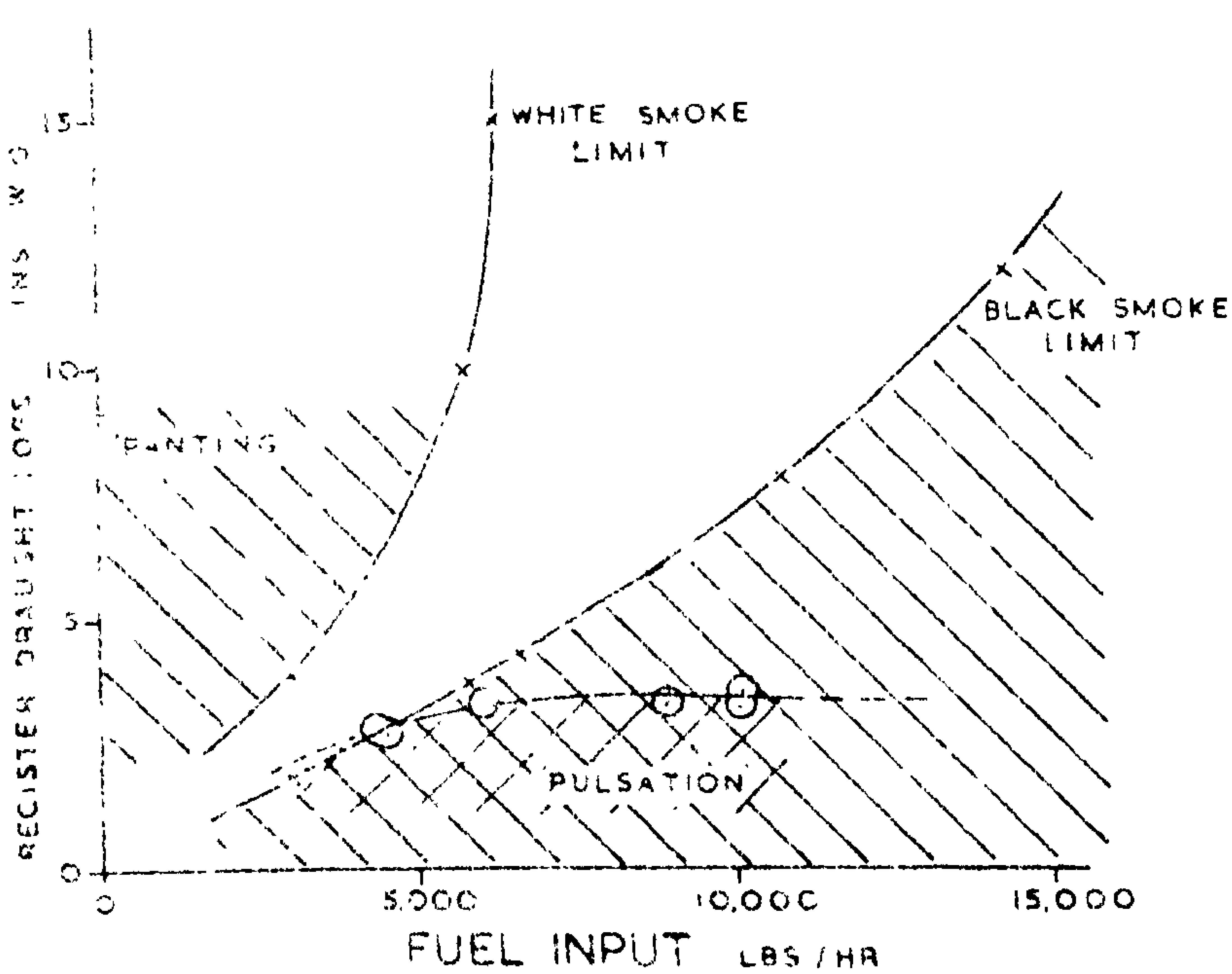
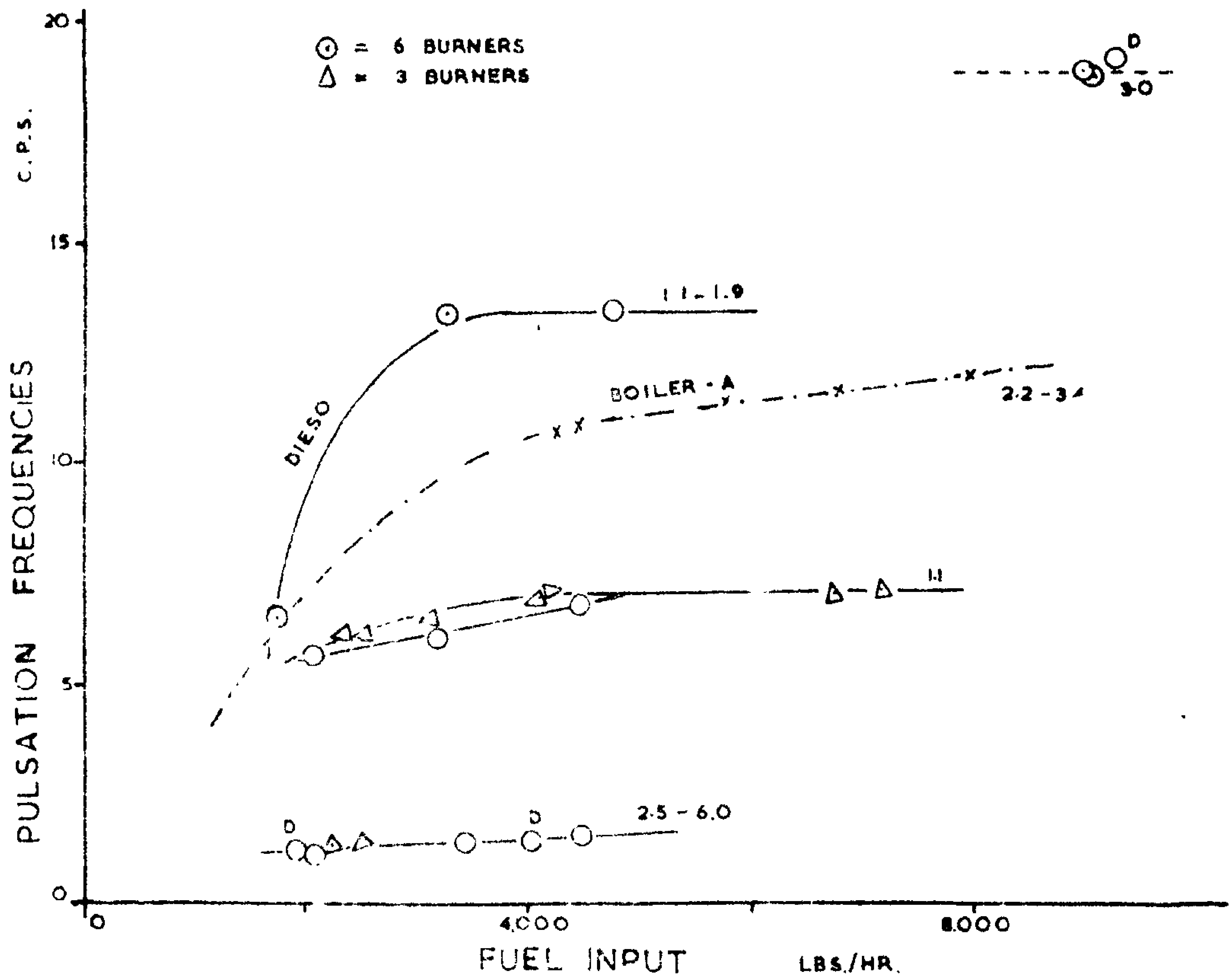
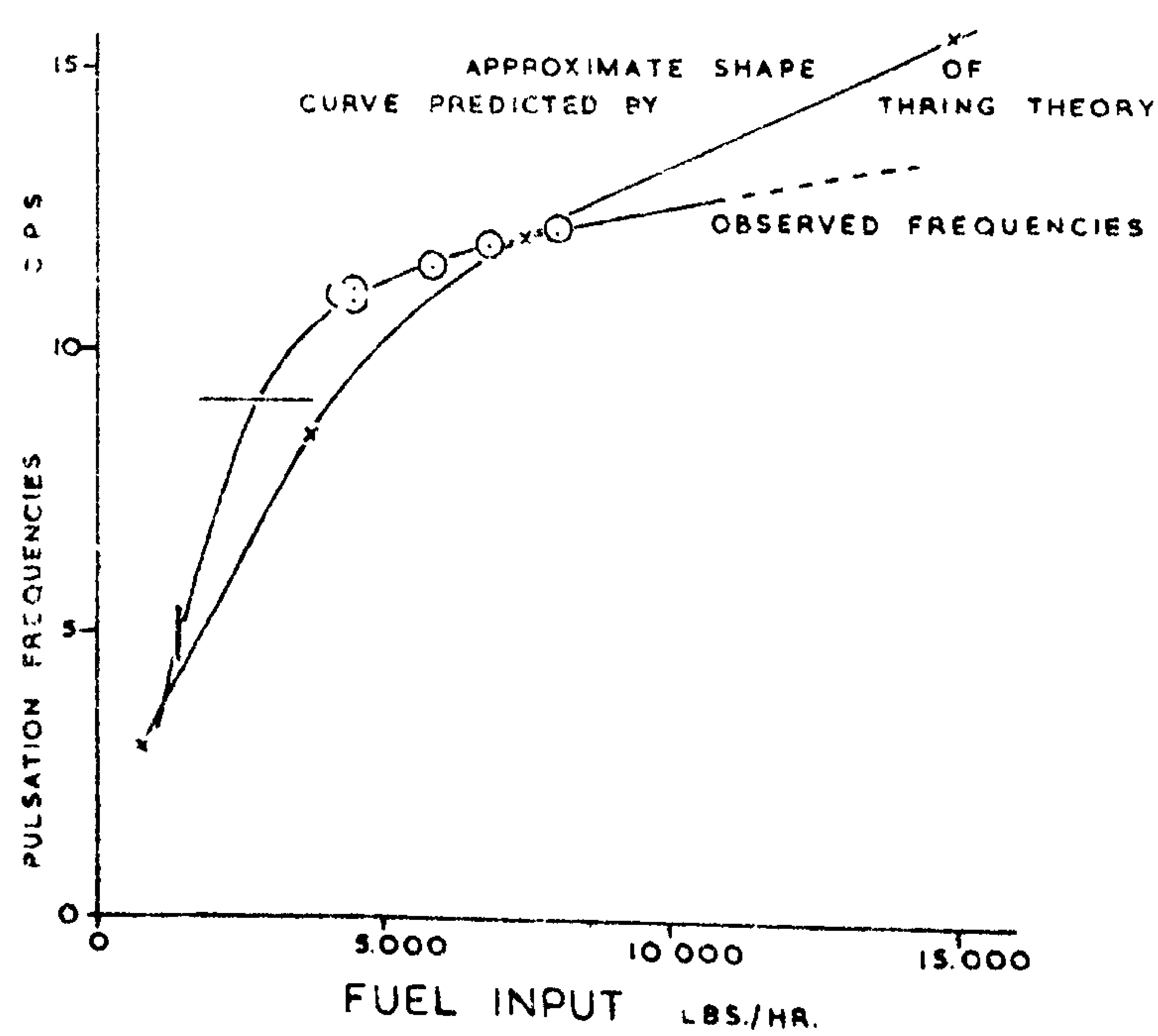



FIG. 7.



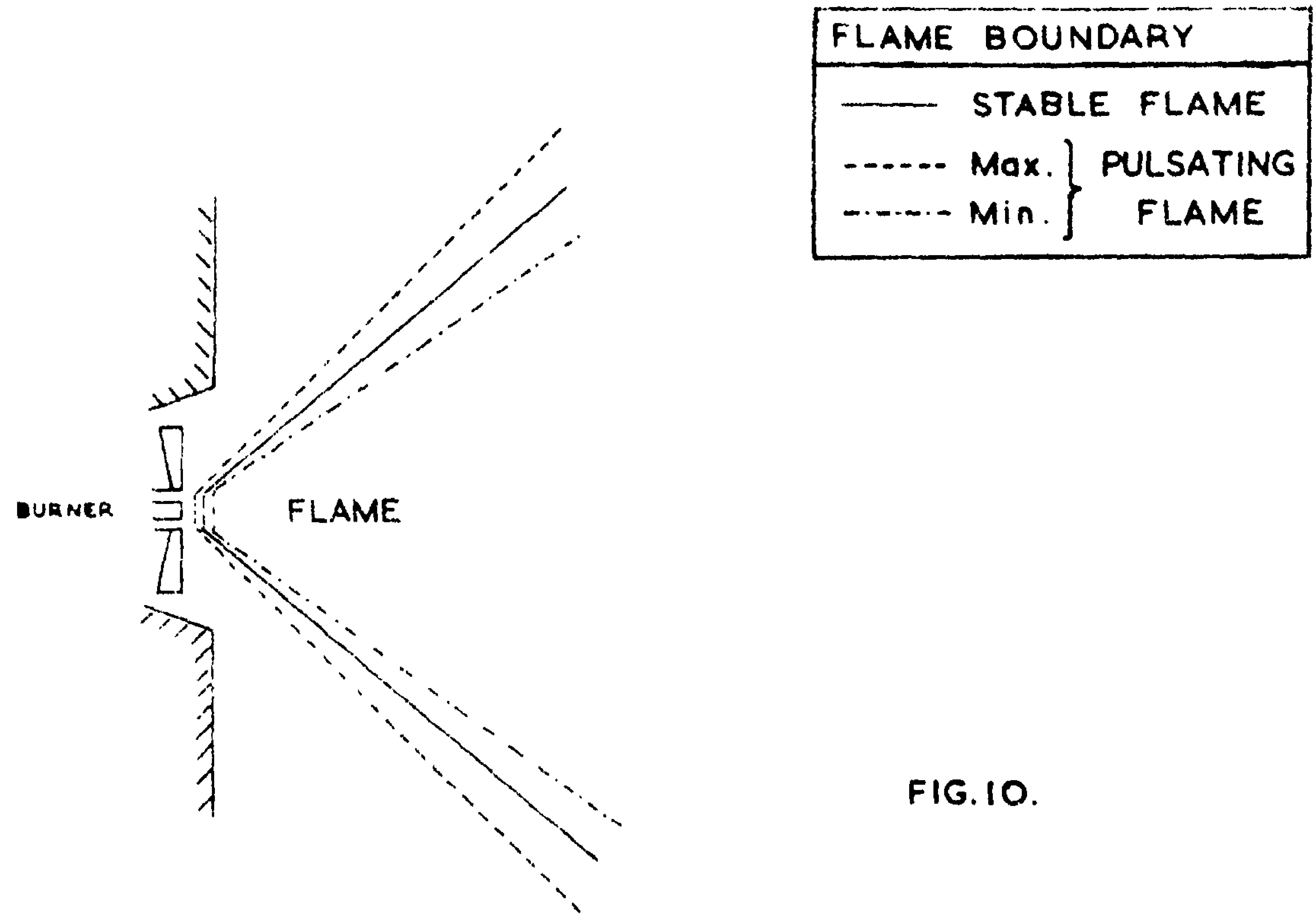
⊙ = 6 BURNERS
Δ = 3 BURNERS
C.P.S.
PULSATION FREQUENCIES
1.1 - 1.9
DIESO
BOILER - A
2.2 - 3.4
1.1
2.5 - 6.0
3.0
D
FUEL INPUT LBS./HR.


FIG. 8.



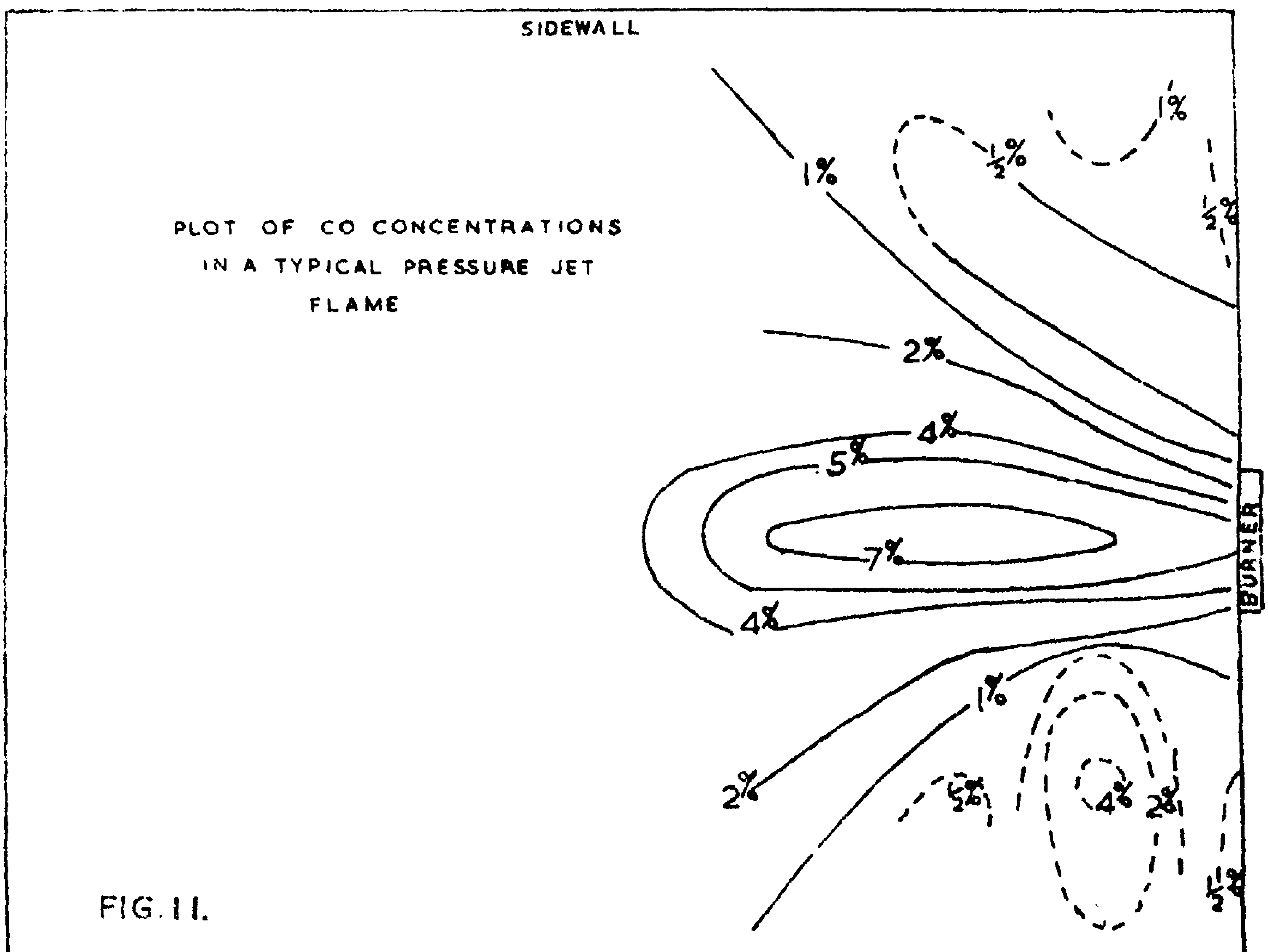
APPROXIMATE SHAPE OF CURVE PREDICTED BY THRING THEORY
OBSERVED FREQUENCIES
PULSATION FREQUENCIES C P S
FUEL INPUT LBS./HR.


FIG. 9.



FLAME BOUNDARY
STABLE FLAME
Max.
Min.
PULSATING FLAME
BURNER
FLAME


FIG. 10.



SIDEWALL
PLOT OF CO CONCENTRATIONS IN A TYPICAL PRESSURE JET FLAME
1%
½%
1%
½%
2%
4%
5%
7%
4%
BURNER
1%
2%
½%
4%
2%
½%
FLOOR


FIG. 11.

# APPENDIX III.

## Calculation of Sauter Mean Diameter for Sprays.

As already indicated in Chapter 2. on theoretical grounds the general relationship for the mean droplet diameter of sprays in pressure atomizers is:-

$$d \propto \left\{ \frac{FN \cdot s}{\theta \cdot P} \right\}^{\frac{1}{3}} \left( {}^{\rho_f}/_{\rho_a} \right)^{\frac{1}{6}} \qquad (5)$$

The viscosity does not appear directly in the above relationship although it is reflected in the flow number.

A considerable number of empirical relationships have been reported(28)(8)(26), in some cases based on the same data. The actual range of experimental data appears to have been limited to a very few Simplex atomizers, some N.G.T.E. experimental atomizers, some small Lucas atomizers and Power Jet spill atomizers, and were for flow numbers less than 4 and small flow rates for distillate fuels. This means that extrapolation is only really justified over these ranges and for strictly similar atomizers.

In the case of the experimental work reported here, the Lucas atomizers used in the laboratory were almost certainly of the same general type as those for which there are reported relationships and what appeared to be the best of these is due to Radcliffe[28] who gave the relation:-

$$SMD = 182 \frac{Q^{0.25}}{P^{0.4}}$$

Fraser[26], however, includes a density term

$$SMD = K \frac{(FN)^{0.25}}{P^{0.275}}$$

Extrapolation of these results to large atomizers and heavy fuel oils is rather questionable. The makers[8] and Humby[109] quote other expressions:-

$$SMD = 335\ P^{-0.348}\ FN^{0.209}\ \nu^{0.215}$$

$$SMD = 120\ Q^{0.282}\ P^{-0.397}\ \nu^{0.204}$$

although these still appear to be based on similar operating ranges to those given above.

The atomizers used in the full size experiments were a so called Duplex atomizer and a spill atomizer (See Fig.9.). For the first of these a rather complicated form of simplex atomizer has had to be assumed. Most of the very limited data on spill atomizers assumes that they are run with the spill shut or fails to take it into account at all.

Radcliffe, however, gives a curve for the effect of

$$\frac{\text{SMD Spill Open}}{\text{SMD Spill Closed}} \quad \text{against} \quad \frac{\text{Q Spill Open}}{\text{Q Spill Closed}}$$

This indicates that an SMD calculated as if the atomizer is behaving as a Simplex atomizer at the same line pressure as the Spill atomizer, is unlikely to be more than 5% high, providing the flow rates do not differ by more than 30%. The results for the Spill atomizer therefore has been corrected accordingly.

The SMD figure calculated from each formula are tabulated below.

TABLE A.

Calculated S.M.D. Values

| Atomizer | Relationship used for S.M.D. | | | |
|---|---|---|---|---|
| Due to:- | Radcliffe | Fraser | Clarke & Hudson | Humby |
| Ref:- | 28 | 26 | 8 | 109 |
| Duplex | | | | |
| 60° | 99 | 131 | 70 | 110 |
| 90° | 74 | 76 | 58 | 80 |
| Spill | | | | |
| 80° | 99 | 116 | 62 | 95 |
| Model | | | | |
| 60° | 68 | | 72 | 80 |
| 90° | 43 | | | |

## APPENDIX IV.

### Determination of suction pyrometer "efficiencies"

The determination of fuel temperature readings from those actually measured involves a number of problems and uncertainties. The importance of the temperature measurement is chiefly in the calculation of actual velocities, and exceptionally accurate measurement is likely to be lost in the variations and inaccuracies in other measurements. It is desirable to make a better estimate than accepting the 'with suction' temperature as the true value, which with the shield arrangement used, might well introduce errors of up to 100° C in the final result.

The "efficiency" of a suction pyrometer is defined as, the ratio of the measured temperature difference to the actual temperature difference between the case with a constant rate of suction through the shield and the case with no suction. Three methods for determining this have been suggested by Land & Barber[99] who also give theoretical justification for these methods.

1. The first method depends on the fact that this efficiency is a function of the gas velocity through the shield. Temperature increase and hence curve shape is a function of

the characteristics of the individual shield system. Any index can be used to specify the shape of this curve, but $\dfrac{T_{max} - T_o}{T_{max} - T_{\frac{1}{4}}}$ has been found to be a convenient reference factor and allows direct use of Land & Barbers curves for "shape factor" against "efficiency". Figs. lV.1 and IV.2.

2. The second method depends on the fact that the time lag is different when the suction is turned on and when it is turned off. For a simple shield this ratio $\dfrac{\tau_o}{\tau_1} = \dfrac{h}{2\sigma T^3} - 1$

where $h$ = convective heat transfer coefficient between gas and tubes.

$\sigma$ = Stefan's radiation constant.

This ratio is also an index of the efficiency and a similar procedure therefore can be carried out to that used under Method 1. to determine actual temperature. Fig.lV.3.

3. The third method is by direct calculation. This is a fairly laborious calculation which can be simplified by relating the performance of the refractory shield used to an equivalent number of blackened metal shields, and using standard results for the latter calculated by Land & Barber. The equivalent number of blackened metal shields is given by

$$\sqrt{(1 - \frac{2}{E}) + 4\sigma T^3 \frac{W}{k}}$$

where $E$ = the total emissivity of the surfaces.

$W$ = the tube thickness.

$k$ = the thermal conductivity of the refractory.

The relative merits of the three methods may be summarized as follows. 1 and 2 both have the advantage that they depend on an actual measurement for the efficiency, as this probably varies slightly between different suction pyrometers constructed of the same material but with different shield lengths. Method 3, while being an absolute calculation, has the disadvantage that values for E and k are either not known, or are suspect, for the temperature ranges of interest. The result is that calculated efficiencies are badly at variance with the other two methods which were found to give fairly good agreement. This variation in method 3 seems probably due to the actual values E being much lower than those quoted in the literature. At attempt is being made to have this verified using an arc image furnace[110].

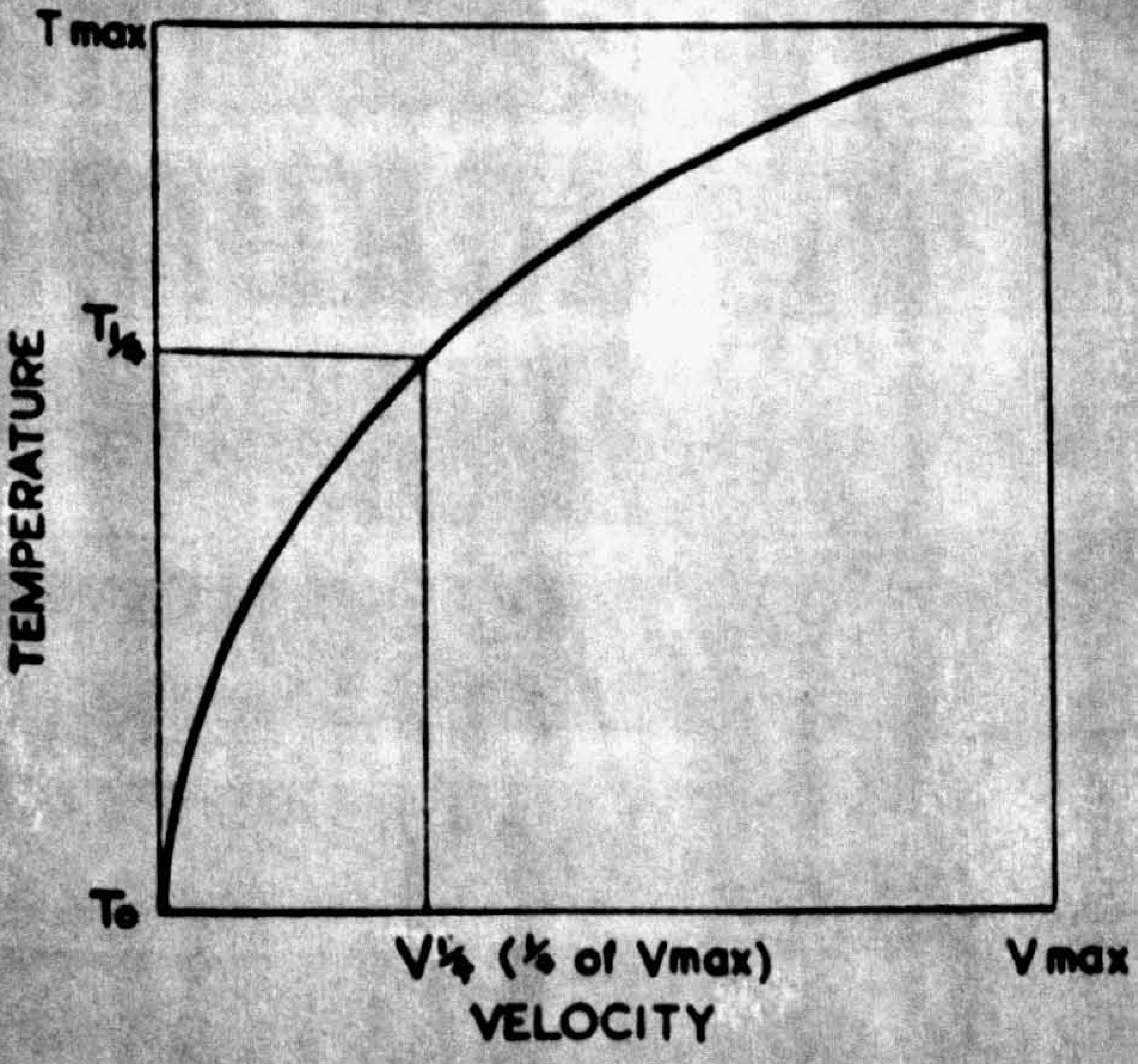



Fig.IV.1 Velocity – Temperature curve

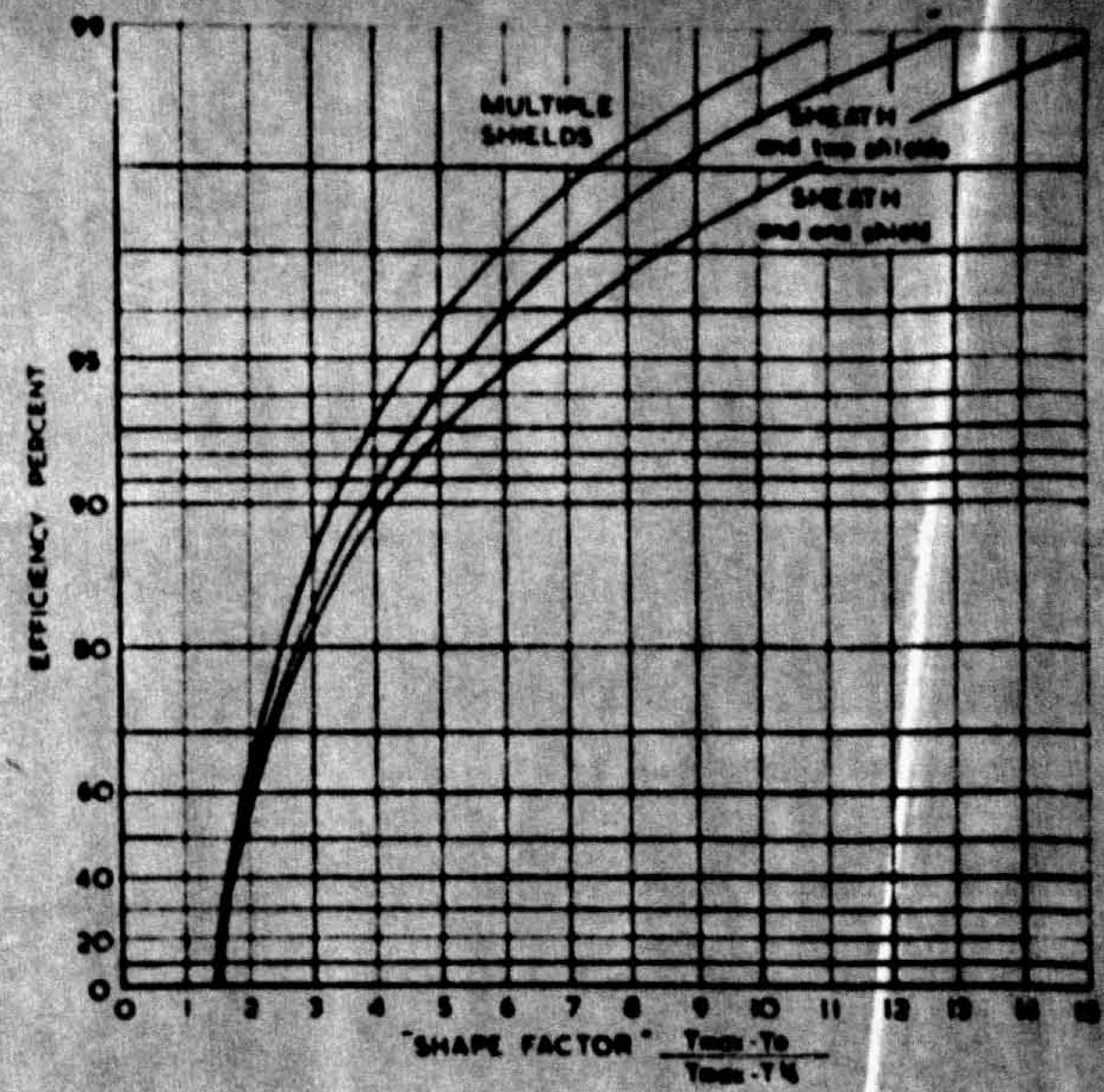



Fig IV.2 Efficiency of Sheathed Couple Pyrometers related to "Shape Factor"

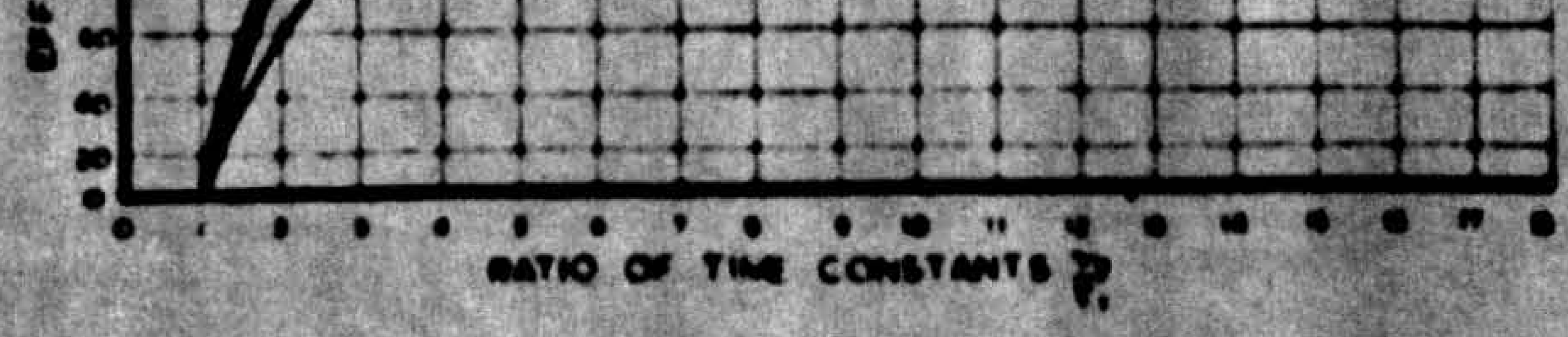



Fig IV.3 Efficiency of Sheathed Couple Pyrometers related to $\frac{T_0}{T_1}$

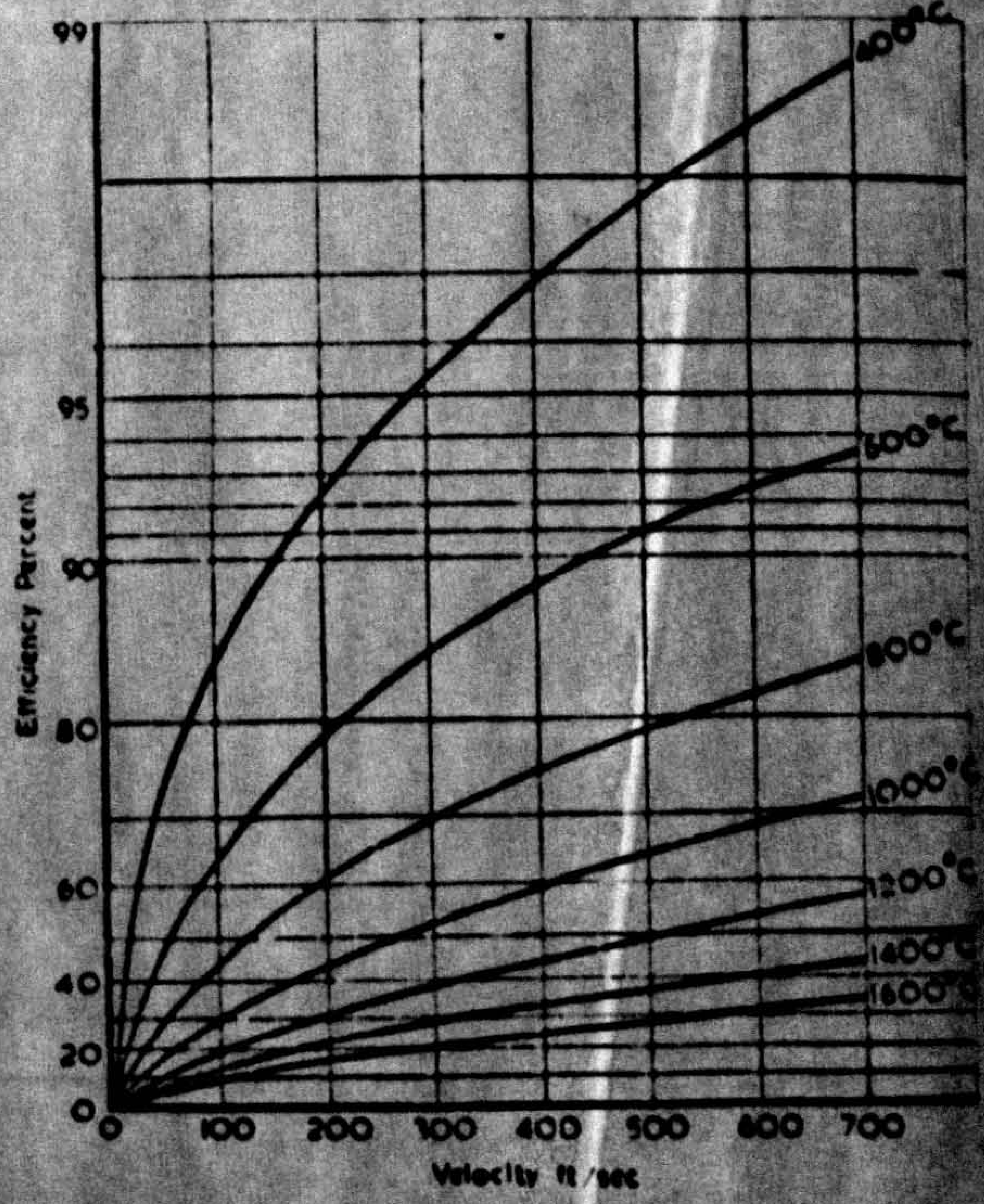



Fig IV.4 Efficiency of Pyrometer with Blackened Metal Sheath and one Shield

## APPENDIX V.

## Calculation of Air Fuel Ratios and Combustion Efficiencies.

### 1. Air Fuel Ratio

The value of the air/fuel ratio at any point was calculated on the basis of a series of material balances at that point.

Carbon Balance at point P.

$$\frac{Z}{359}(CO_2 + CO + CH_4 + 2C_2H_4) + \frac{U}{12} = \frac{C}{12} \quad \dots\dots\dots\dots \quad (1)$$

Oxygen balance at P.

$$\frac{\nu . 0.23}{32} = \frac{Z}{359}(CO_2 + O_2 + \tfrac{1}{2}CO) + \frac{S}{36} \quad \dots\dots\dots\dots \quad (2)$$

Hydrogen balance at P.

$$\frac{H}{2} = \frac{Z}{359}(H_2 + 2CH_4 + 2C_2H_4) + \frac{S}{18} \quad \dots\dots\dots\dots \quad (3)$$

Nitrogen balance at P.

$$\frac{\nu . 0.77.}{28} = \frac{Z.N_2}{359} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \quad (4)$$

### 1.1 Air Fuel Ratios (Hydrogen balance)

Eliminating S and Z from equations 2,3 and 4.

$$\nu_N = \frac{H}{0.0292 - \frac{0.055}{N_2}(2CO_2 + 2O_2 + CO - H_2 - 2CH_4 - 2C_2H_4)} \quad \dots\dots\dots\dots \quad (5)$$

1.2 Air Fuel Ratio (Carbon balance).

Eliminating Z between equations 1 and 4

$$\nu_C = \frac{C - u}{\frac{0.330}{N_2}(CO_2 + CO + CH_4 + 2C_2H_4)} \quad \text{..........(6)}$$

Normally it is assumed that u the quantity of unburnt carbon = 0. However, the value of $\nu_H$ and $\nu_C$ rarely coincides exactly and it is possible to calculate solid carbon in the flame (assuming no other losses) from (6).

$$C - \nu_H . \frac{0.330}{N_2}(CO_2 + CO + CH_4 + 2C_2H_4) = u \quad \text{.......... (7)}$$

1.3 Combustion Efficiency.

The combustion efficiency $\eta$ will be

$$\frac{\text{Calorific Value of Fuel Burnt}}{\text{Total Calorific value of Fuel.}}$$

Calorific Value of Fuel Burnt per lb. of fuel reaching P:-

$= \quad Z\,(188CO + 191H_2 + 593CH_4 + 930C_2H_4)$ C.H.U. per lb.

Thus Combustion Efficiency $\eta$

$$= \quad 1 - \frac{\nu .\ 9.87}{K.\ N_2}(188CO + 191H_2 + 593CH_4 + 930C_2H_4)$$

Where:-

C = lb. of Carbon per lb. of Fuel.

H = lb. of Hydrogen per lb. of Fuel

u = lb. of soot formed per lb. of fuel.

$CO_2$, $CO$, $O_2$, $H_2$, $N_2$, $CH_4$, $C_2H_4$ refer to fractional volumes of these gases analysed in the dry soot-free sample.

$Z$ = cu.ft. of dry soot-free combustion products at the point $\rho$ per lb. of fuel reaching P.

$\nu$ = air-fuel ratio.

$S$ = lb. of steam formed per lb. of fuel reaching P.

$K$ = Calorific Value of the fuel (CH U/lb.)

## APPENDIX VI.

### The Corrections made for errors in the Oxygen Analyzer.

As mentioned in the text, section 4.5.2, the detector stage of the Kent Oxygen Analyzer is sensitive to a number of sources of error.

These are:-

1. The effect of carrier gas composition.

(a) Calibration: The range error is approximately proportional to the $\frac{C_p}{\nu}$ for the whole gas mixture. Normally, the analyzer is calibrated against $O_2$ in $N_2$, and a calibration factor introduced based on the mean likely gas composition. Serious errors will result only from the presence of large percentages of hydrogen or some hydrocarbons. In this case, it would probably be necessary to calculate a new calibration factor from $\frac{C_p}{\nu}$ data, or carry out special calibration.

(b) Zero Shift: Errors can be introduced also in the zero by a big change in gas composition. The only remedy is special calibration. However, for the gas compositions normally measured with these instruments, errors are unlikely to exceed $\pm$ 0.2% $O_2$ on the 0-10% range.

2. The effect of temperature and conductivity.

(a) Temperature: Temperature changes in the cell affect, fairly markedly, the output of the bridge circuit. Normally, keeping the instrument heater switched on is sufficient under normal circumstances to maintain a constant cell temperature. Again, in the event of large percentages of hydrogen appearing, the sample this balance would upset the temperature. This, and thermal conductivity changes would make special calibration necessary again. Under normal conditions the gas flow was regulated to the same volume as that used in the calibration. The throughput is so low that minor changes in gas temperature are eliminated by leaking or cooling to instrument temperature before the gas reaches the cell.

(b) <u>Gas density changes.</u>

The instrument is very sensitive to gas density changes which makes it necessary to compensate for atmospheric pressure and any substantial pressure in the instrument. On the gas sampling rig a mercury manometer was connected to the outlet side of the instrument, so that the atmospheric pressure could be adjusted to give the true instrument pressure. On calibration, the instrument was set to give the correct scale reading at 29.5 Hg. and the correction factor then necessary was the ratio $\left(\frac{29.5}{p}\right)^2$.

# APPENDIX VII.

## Measurement of three dimensional flow.

### 1. Principles of design.

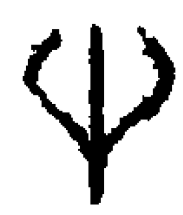
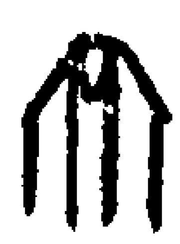
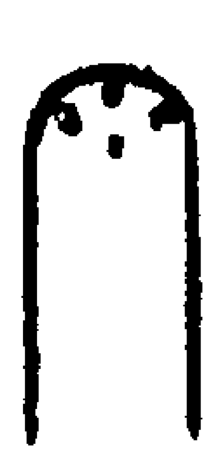
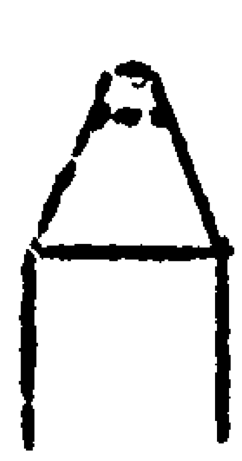
Pitots for 3 dimensional flow measurement have yet to become standardized, and the individual design being largely dictated by the application and the conditions of turbulence and velocity gradient expected. A short review is therefore given of previous work in this field and the reasons for the design of particular types. The most widely used forms are tabulated in Table B. which also provides an explanation of the special properties of each type.

### 2. Methods of use in measuring yaw and pitch angles.

Two general methods of use are possible for the measurement of yaw and pitch angles. The ideal method of measurement is by rotating the instrument in both directions until the two pairs of diametrically opposite tappings both have a null reading. This gives directly the flow direction from the two angles of rotation of the pitot, (in steady flow regions to about $0.2^{\circ}$ angle). The dynamic head is then obtained by the differential pressure between the curve hole and one of the side holes. To obtain the true dynamic head it is usually necessary to calibrate the instrument, although with careful design it should be possible to get the factor K in equation

## TABLE B

**Possible Probes for the Measurement of Three-dimensional Flow**

| | Head | Characteristics | Ref. |
|---|---|---|---|
| | | Uncooled Types | |
| 1 |  | Claw Pitot Fragile and possibility of vortex formation from arms. Not suitable for water cooling. Good characteristics. | 116 |
| 2 |  | 5-way Tapered Head. Good characteristics. Tappings very close. | 115<br>117 |
| | | Types Suitable for Water Cooling. | |
| 3 |  | Wedge. Very sensitive, but only suitable for 2-dimensional flow. Difficult to obtain representative dynamic reading. | 115. |
| 4 |  | Hemispherical Head. Used at right angles to flow. Reasonable characteristics, not suitable for fields of high turbulence or velocity gradient. Must be yawed in use. | 115<br>111 |
| 5 |  | Ricardo Grooved Head. Also used at right angles to flow and requiring to be yawed in use. Much closer tappings and greatly improved characteristics compared to No.4. Approach flow characteristics possibily suspect. | 111<br>34 |
| 6 |  | Spherical Head. Reasonable characteristics, Head necessarily large. Tappings widely spaced. (Can be designed to operate at right angles to the flow, or facing into it. | 114<br>117<br>112<br>113 |
| 7 |  | Hemispherical Head. Similar to No.6. which does not seem to have any advantages over this form. Tappings fairly well separated. | 56<br>118<br>119 |
| 8 |  | Conical Head. Really an enclosed form of No.2. Probably better characteristics than Nos.6&7. Tappings very close and very sensitive to yaw and pitch. Can be provided with extra ring of static tappings. Yaw and pitch angles can be measured by rotation to null point or from differential pressures Tappings can be located accurately. | 117<br>120 |

No. (26) near or equal to one. For a cylindrical or spherical surface, a tapping normal to the surface at an angle of slightly less than $40^{\circ}$ to the dynamic tapping should give a true static pressure when the dynamic head tapping is lined up with the direction of flow.

This method of operation is obviously very complicated in anything except very simple measurements where yawing and pitching devices can be freely manoeuvred. For practical purposes pitch angle and sometimes both pitch and yaw angles are usually determined from the differential pressure measurement between opposite tappings after calibration. Each instrument must be calibrated individually as minor differences in manufacture make a common calibration for a particular type unreliable.

One of these latter methods of measurement are usually used in practical cases to keep the movement of the pitot to a minimum. It is the preferred method in regions of turbulent or unsteady flow where null points are difficult to determine. In general, however, rotation to determine the null point is more reliable and increases the angles that can be measured. The differential pressure method is limited to angles of less than $\pm 60^{\circ}$ in either plane from the direction of alignment of the pitot.

FIG. VII.1.

Five Hole 3-dimensional Pitot for Cold Work.


DEGREES

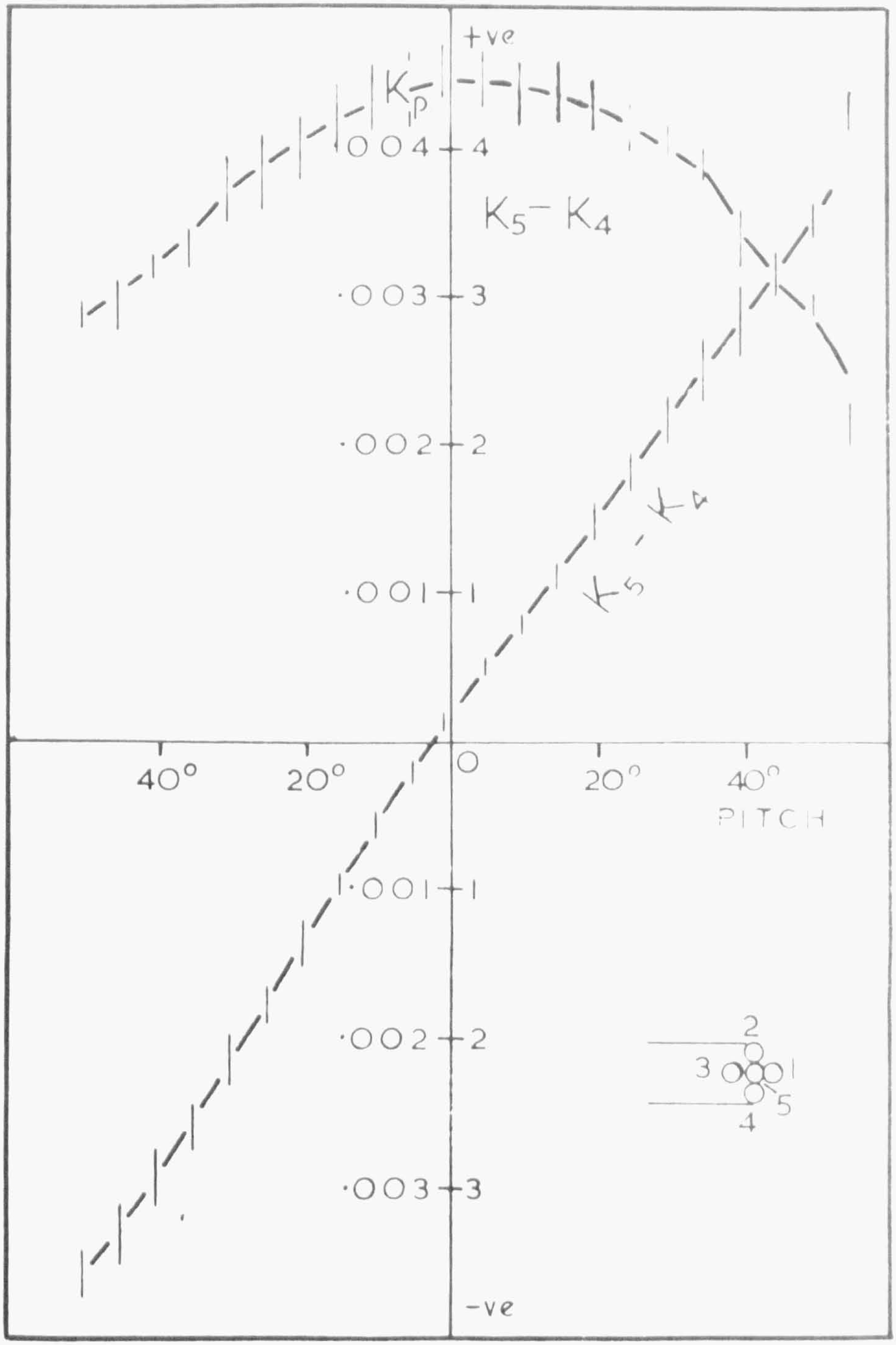



FIG. VII.2. CALIBRATION CURVES FOR 5-HOLE PITOT FOR COLD WORK.

(FOR VALUES OF $1/2\,\rho v^2$ IN THE RANGE 28 TO 85) through the half angle of the yaw tappings to use these same two tappings for a dynamic and static measurement. Although this instrument is reported as behaving well, the tappings were still quite a long way apart and some possible objections could be raised on the grounds that the dynamic head was read at 45° to the axis of the instrument and might be expected to suffer from interference from other parts of the instrument.

No. 6. and No. 7. are basically very similar. Peploe[112] amongst others, developed a spherical head water cooled pitot and Pengelly[113] has succeeded in producing heads down to about 3/4" diameter. A good deal of work has been done also on their calibration. Apart possibly from more uniform flow round the head, it is difficult to see any advantage for the spherical over the hemispherical head and in the sizes necessary for water cooled instruments the tappings are again too far apart for work in other than regions of little or no velocity gradient.

To overcome the difficulty of the wide spacing of holes a ½" diameter spherical head was developed which was mounted on a water cooled probe up to a distance of about ¾" from the sphere. The final section was made solid and was turned and bored from a single piece of stainless steel. Fig.III.3. Three advantages were expected from this, firstly, reduction in size of head and corresponding reduction in the distance between

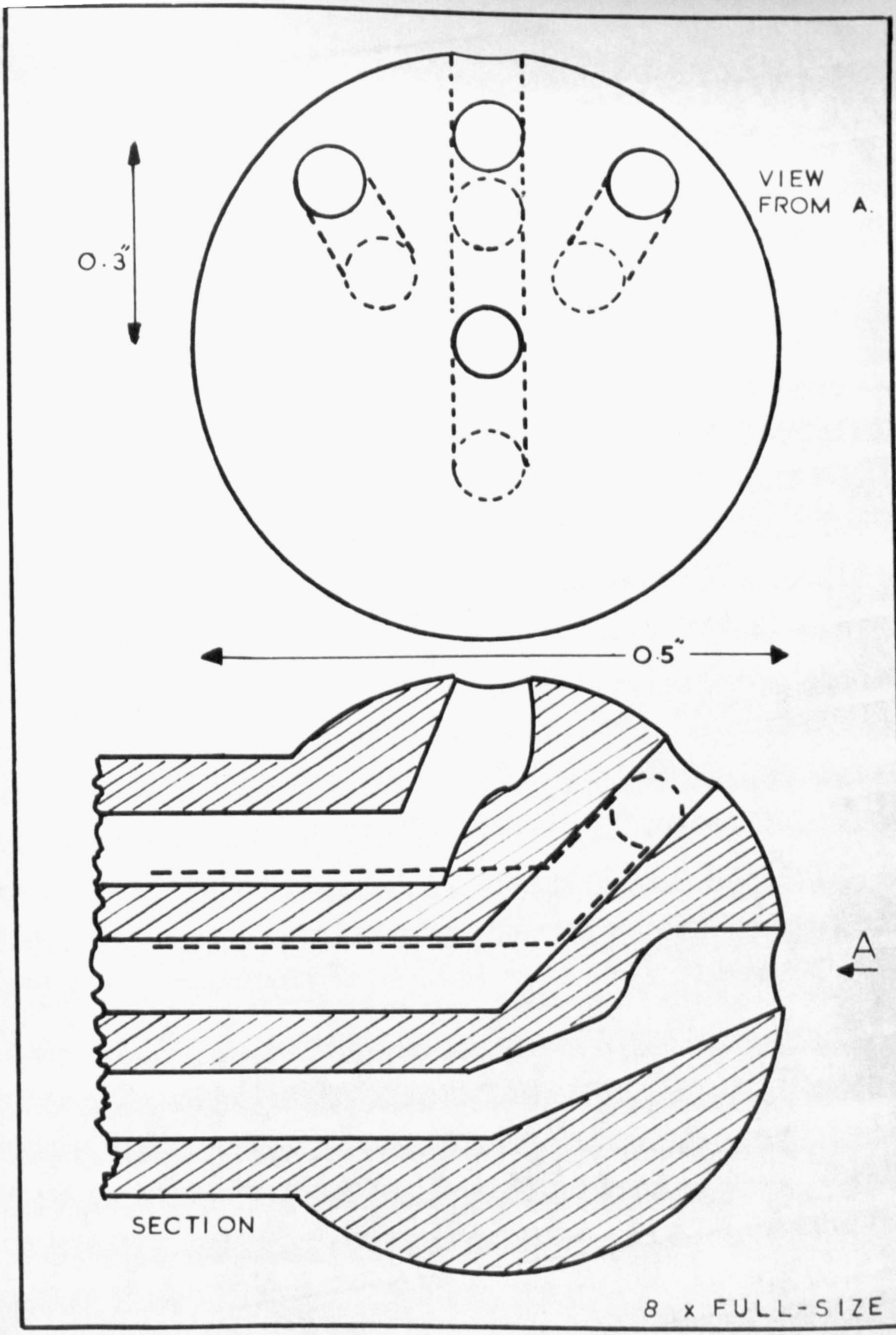



FIG. VII.3. SPHERICAL HEAD FIVE-HOLE PITOT.

SOLID STAINLESS STEEL HEAD.

tappings, secondly, by running the probe tip hot it was hoped to avoid the excessive deposition of the tarry fuel residues that tend to block water cooled instruments, thirdly, considerable easing of the constructional problems. In this particular case, the placing of the taps was about an axis 45° to that of the probe in an attempt to cover flow directions from right angles to the probe to in line with the probe axis. For this reason the tappings were set at 45° to the central tapping instead of the customary 40°. The range of operation desired is approaching the limit for this kind of instrument. How far this target was achieved can be seen from the calibration curve, Fig. V11.4. reasonable results being obtained for the range ± 40°.

The use of the hot end was found to reduce but not eliminate, blocking problems, but unfortunately the head suffered very badly when used in reducing regions of the flame, which severely curtailed the useful work which could be obtained from this instrument.

No. 8. is the design eventually adopted for use in the laboratory furnace. It is illustrated in Fig. V11.5. This overcomes all the previous disadvantages of spacing of tappings as the outside tappings are only 0.11" centre to centre, although the diameter of the cone at its base is 0.45". The steeply angled cone makes it particularly sensitive to yaw and pitch measurement.

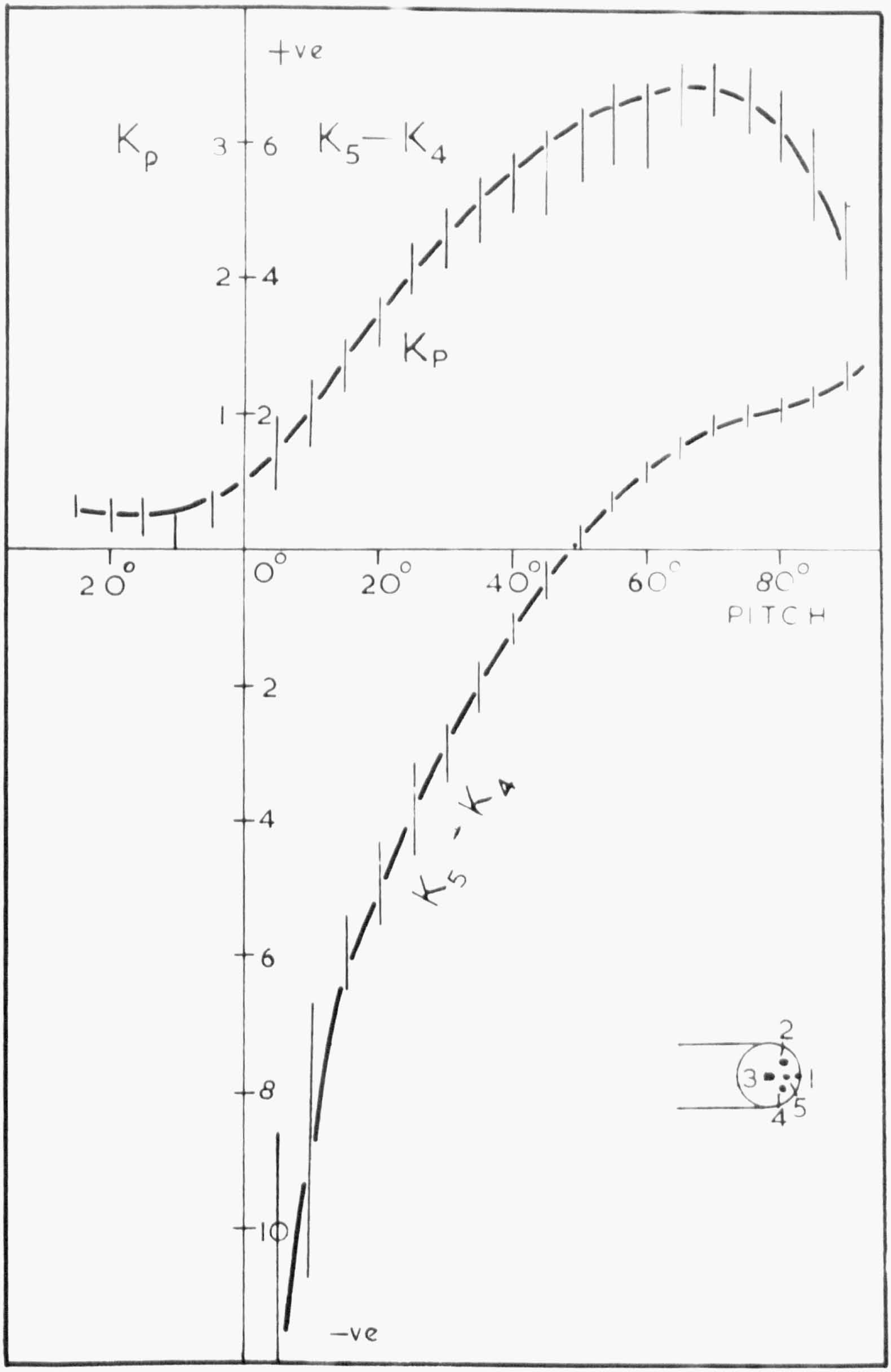



FIG. VII.4. CALIBRATION CURVES FOR FIVE HOLE SPHERICAL HEAD PITOT.

(FOR VALUES OF $1/2\ \rho v^2$ IN THE RANGE 2·6 TO 55·2)

As constructed, a special 3/8" diameter 9-way stainless steel tube was used for the stem and by making the stem enter at the base of the cone, the crooked arrangement was avoided which has been used by some other workers, and a very difficult constructional problem eliminated. The stem was slightly bent at its outer end so that the tip of the instrument was in line with the axis of rotation. The 9-way tube provides for 5 pressure connections to the probe head in the middle, and 4 divisions outside these to provide the out and return paths for water cooling. The arrangement of the probe head is shown in Fig. V11.5. The rear end of the probe was fitted in a mounting bearing and provided with a $360^{\circ}$ scale and pointer (See fig. V11.5.c.).

4. Calibration.

The calibration curves for the three dimensional pitots are given in Figs. V11.2, 4, & 6. These are plots of $\frac{\Delta p_{\alpha,\beta}}{\Delta p_o}$ the ratio of the differential pressure reading at a given angle to the reading with flow normal to the probe, against angle.

The calibration curves for each pitot are given in a single chart for each instrument. Following Van der Hegge-Zijnen[114], these are represented as follows, the convention being shown in Fig. V11.2. The following orifice coefficients are defined assuming that the instrument is rotated to the null point for a yaw measurement.

FIG. VII.5. (a).

Conical Head 5-Hole Pitot.

FIG. VII.5 (b).

Photograph of Head.

FIG. VII.5. (c).

Photograph Showing Scale and Pressure Outlets.

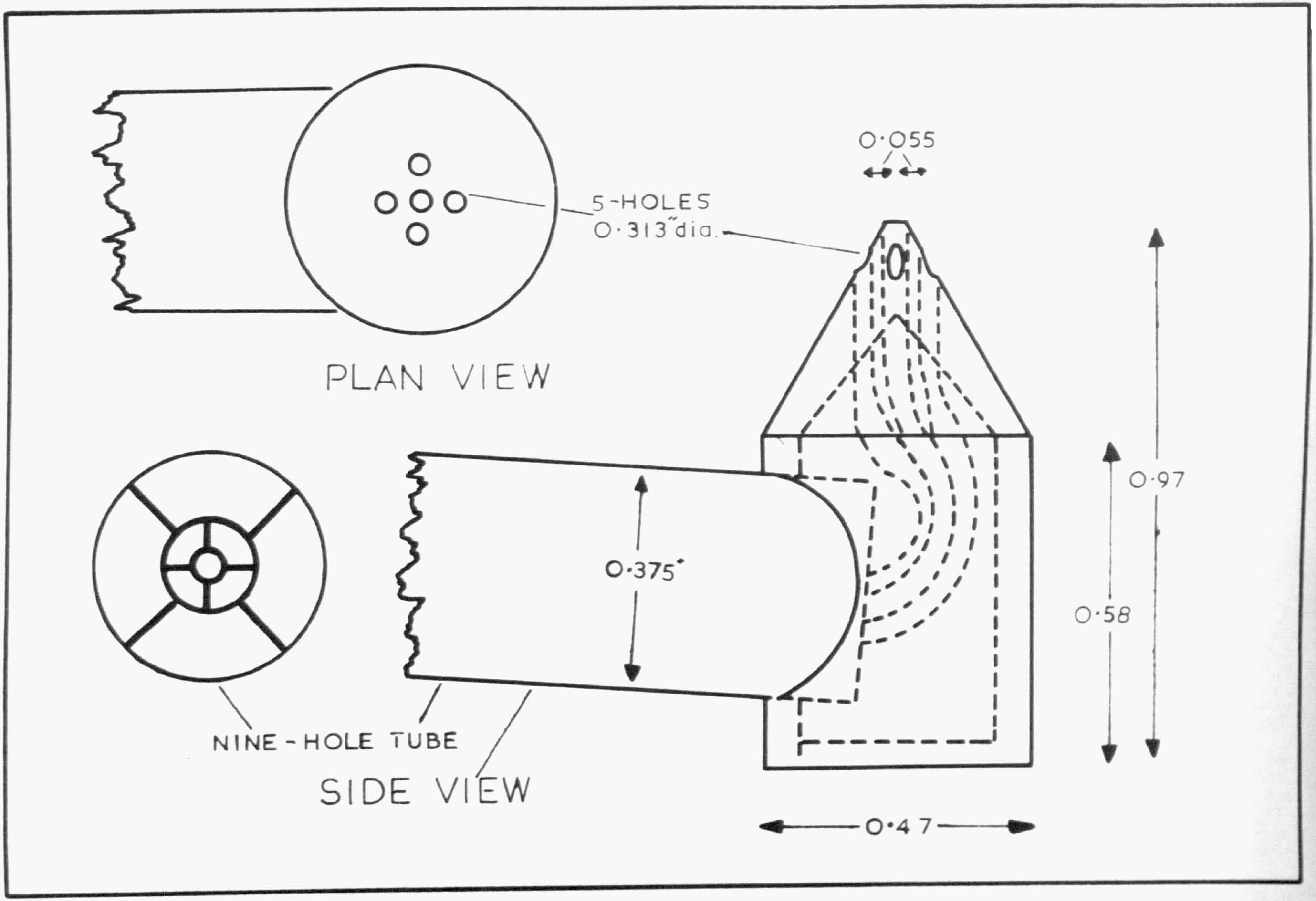
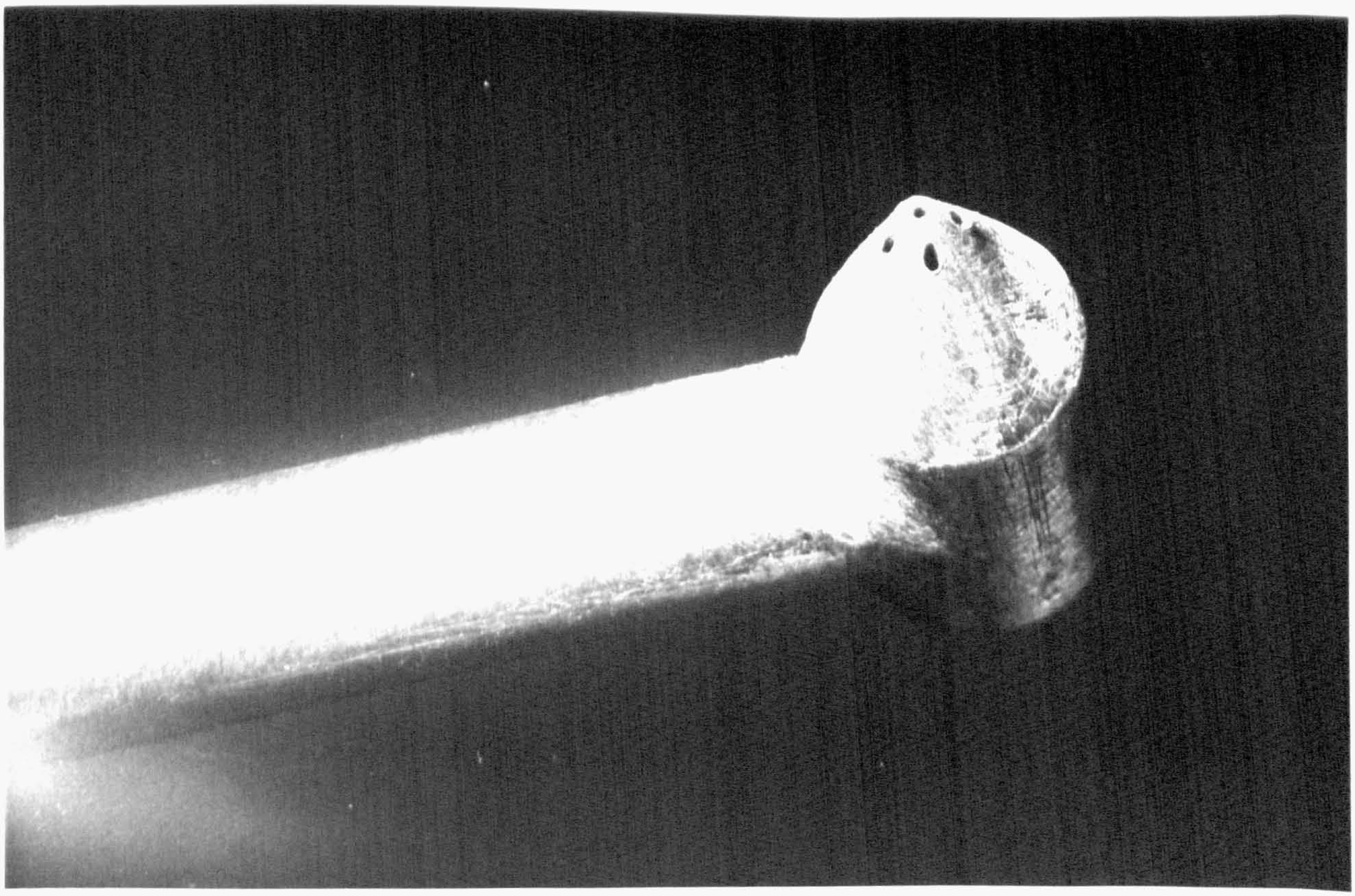
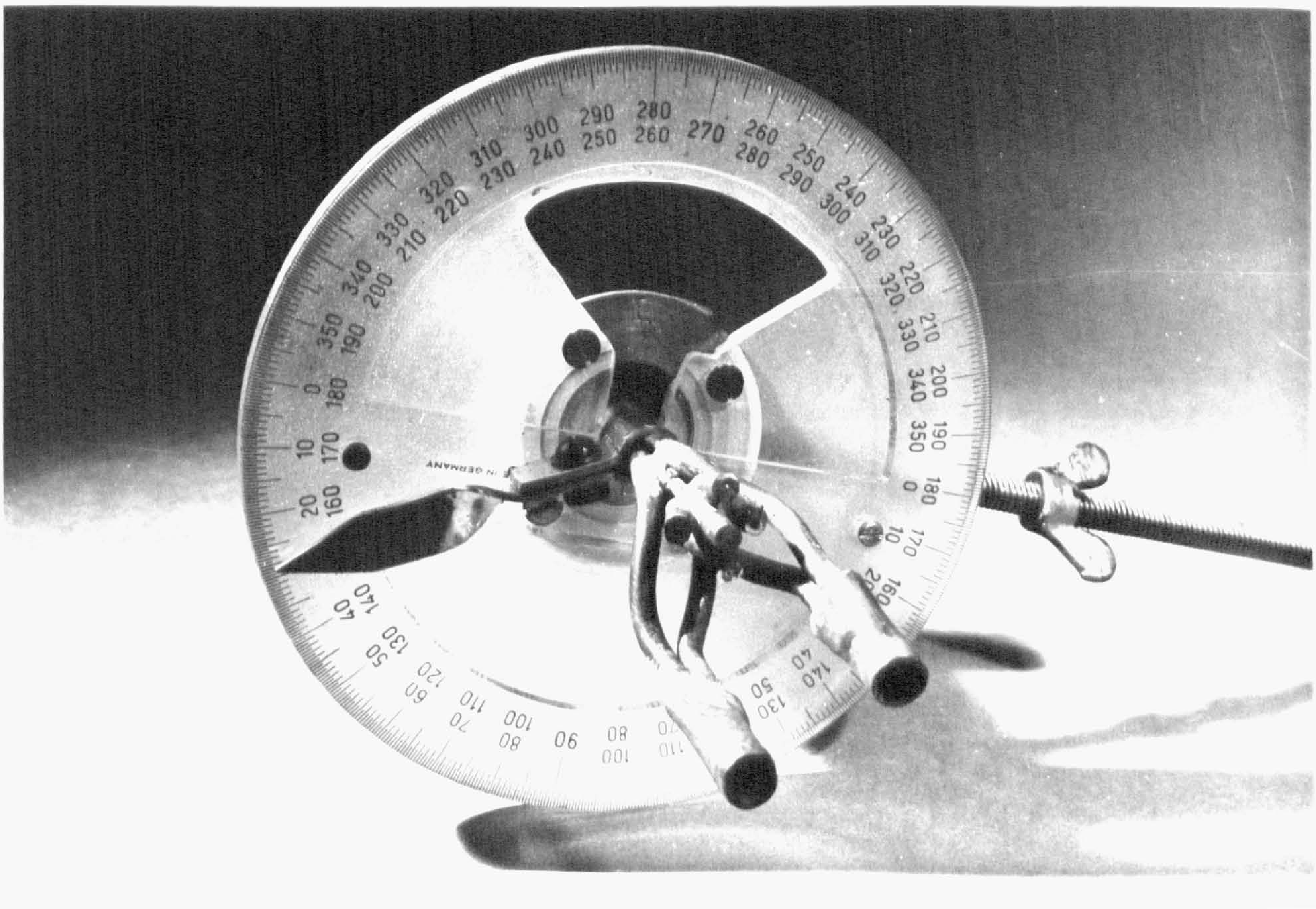



FIG. VII.5(a) CONICAL HEAD FIVE-HOLE PITOT. (x 4)

STAINLESS STEEL - WATER COOLED.





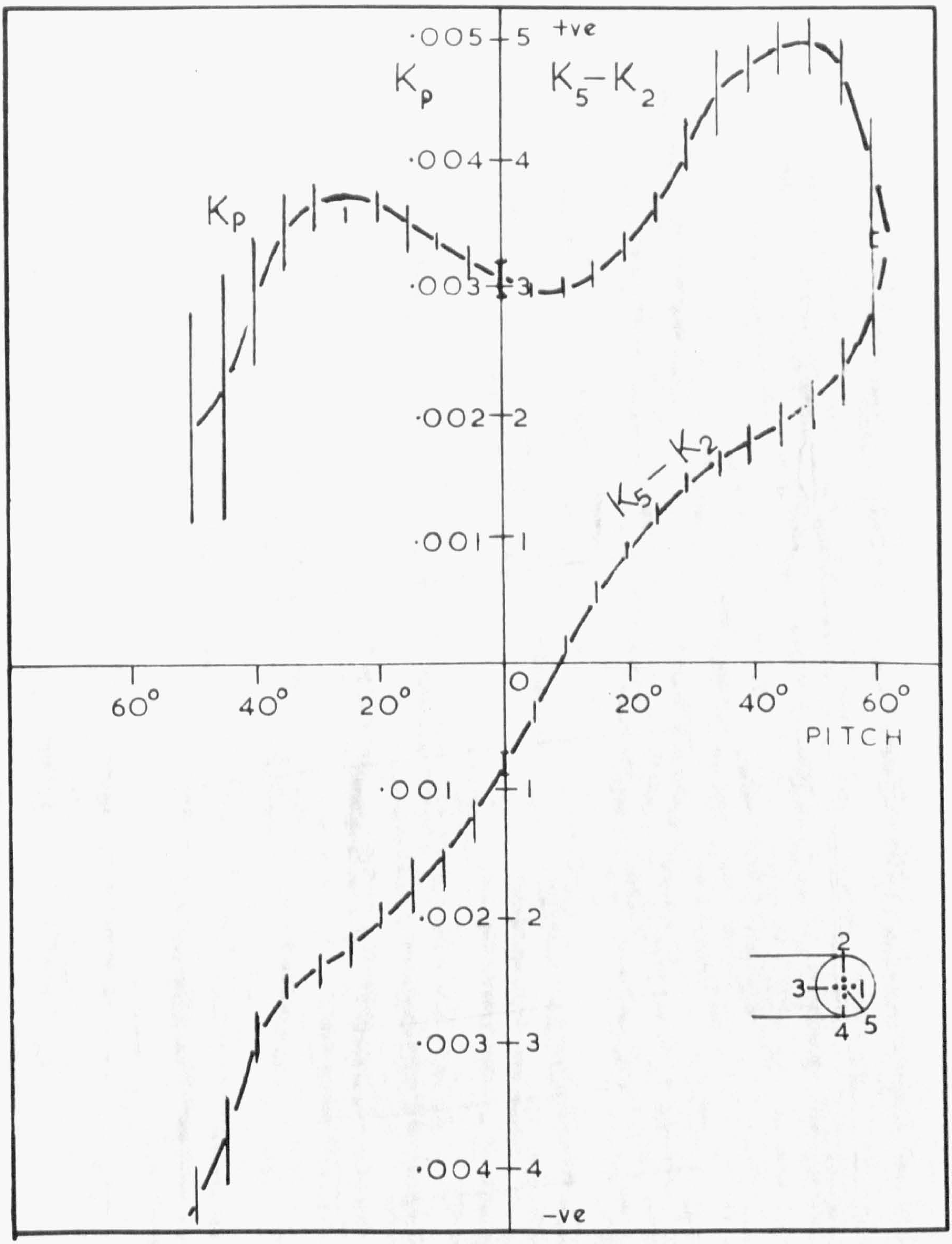



FIG. VII.6. CALIBRATION CURVES FOR CONICAL HEAD PITOT.

(FOR VALUES OF $1/2\ \rho v^2$ IN THE RANGE 9·7 TO 96·9)

$$k_1 = \frac{p_1 - p_s}{q}; \qquad k_2 = \frac{p_2 - p_s}{q} \text{ etc.} \qquad k_t = \frac{p_t - p_s}{q}$$

$$k_{54} = k_5 - k_4; \qquad k_p = \frac{k_3 - k_1}{k_5 - k_4}$$

where $q$ = velocity head, $p_{1,2,3,\text{etc.}}$ = individual pressure readings.

These pressures measured at each tapping are given by rearranging the above equations

$$p_1 = p_s + q\,k_1 \qquad p_{2,4} = p_s + q\,K_{2,4} \text{ etc.}$$

Thus, eliminating $p_s$ the velocity head $q$ is given by

$$\tfrac{1}{2}\rho v^2 = q = \frac{p_5 - p_1}{k_5 - k_1} = \frac{p_5 - p_{2,4}}{k_5 - k_{2,4}} = \frac{p_5 - p_3}{k_5 - k_3} \quad \text{etc.}$$

and the static pressure by

$$p_s = p_1 - qk, = p_{2,4} - q\,k_{2,4} \text{ etc.}$$

Only the pitch angle has to be determined from these measurements as $p_2 = p_4$ by setting

$$k_p = \frac{k_3 - k_1}{k_5 - k_{2,4}} = \frac{p_3 - p_1}{p_5 - p_{2,4}}.$$

When both yaw and pitch are determined by differential pressure measurements $k_p$ and $k_y$ can be represented by $k_p = \dfrac{p_5 - p_1}{p_5 - p_3}$ $k_y = \dfrac{p_5 - p_4}{p_5 - p_2}$. In the subsequent calibrations, plots are given of $k_5$; $k_5 - k_{2,4}$; $k_p$ against angle of pitch for different values of the velocity head.

# APPENDIX VIII.

## Calculation of Equivalent Diameter for the Burners Used.

The concept of equivalent diameter $d_o'$ has been discussed in section 2.7.2., where it is shown how this is related to the combined momenta of the oil spray and air jet. Where there is a density difference, a second equivalent radius $d_o''$ is necessary. This is related to $d_o'$ by the following expression:-

$d_o'' = d_o' \sqrt{\rho_o/\rho_g}$ Or where it is more convenient to use measured data and allow for combined momenta of concentric jets, and actual values for discharge coefficients.

$$d_o'' = \frac{\dot{m}_o}{\sqrt{\rho_g \pi G_o}} \qquad \text{where } G_o = \text{momentum.} = mv. = \frac{4m_o^2}{\rho \pi d_o^2 C}$$

$C$ = discharge coefficient.

$d_o$ = $2r_o$ = actual nozzle diameter

$d_o'$ = $2r_o'$ = equivalent nozzle, diameter at same density.

$d_o''$ = $2r_o''$ = equivalent nozzle, diameter at different density.

$$\rho_g = \text{density of hot gases} = \frac{xCO_2 + yCO + zO_2 + sH_2 + qN_2}{100}$$

where x,y,z,s,q represent densities of the various components at the temperature of the gases, and $CO_2$, CO etc. are volume %'s. For the flames considered these are given in Table C.

TABLE C.

| Flame | | Fuel Rate lb./hr. | Equivalence Ratio | $d_o'$ ins. | $d_o''$ ins. |
|---|---|---|---|---|---|
| Full size Burner | $60^{\circ}$ Duplex Full Power. | 2175 | 1.28 | 15.46 | 29.3 |
| | $80^{\circ}$ Spill Full Power | 2175 | 1.38 | 15.4 | 29.2 |
| | Low Power | 575 | 1.98 | 15.3 | 24.2 |
| Model Burner | "Low-Power" Case Scaling Case 1. | 46 | 1.98 | 3.25 | 6.5 |
| | "Full Power" Case Scaling Case 2. | 50 | 1.28 | 4.16 | 7.1 |

Reference $d_o$ is taken as the final outlet diameter of the burner nozzle.

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