

**" The Electric Analogue For Turbulent Flame Studies:
An Investigation Of Its Validity."**

**Ph.D. Thesis Submitted To The
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By**

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CONTENTS.

	<u>Page</u>
Abstract.	6 .
Chapter One,	
Theory of the Electric Analogue.	8
Chapter Two,	
Apparatus and Experimental Procedure.	39
Chapter Three,	
Analogue Predictions.	69
Chapter Four,	
Flame Stability Data.	119
Chapter Five,	
Theoretical Analysis of Flame	
Stabilization in the Wake of An	
Annular Baffle.	163

	<u>Page</u>
Chapter Six,	
Comparisons and Conclusions.	191
Appendix One,	
References.	226
Appendix Two,	
Data for the Calculation of flame	
Stability Parameters.	233
Appendix Three,	
Calculation of the Flow Criterion.	239

Abstract.

The aim of this research was to investigate the validity of the predictions, obtained by the electric analogue technique, regarding the performance of turbulent flames with high chemical loading.

An axi-symmetric analogue model was made; and its predictions were compared with the real flame stability data that could be found in the literature, augmented by some results obtained during the course of this investigation. The types of flame-holders tested were:

1. Annular Baffles,
2. Central Disc Baffles,
3. Hemi-Spherical Baffles,
4. Conical Baffles.

Good qualitative and quantitative agreements between the analogue predictions and the real flame results, were noted in the case of the first two

types of baffle; while the lack of detailed flame stability data prevented a final conclusion to be reached in the case of hemi-spherical and conical baffles. The available data, however, point to a general agreement regarding the influence^{of} the baffle base diameter on the extinction characteristics of the flames stabilised in the wakes of such baffles.

CHAPTER ONE.

THEORY OF THE ELECTRIC ANALOGUE.

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Introduction.

In the first paper on the subject of the electric analogue for the study of steady-state turbulent flame phenomena(Ref.1), Professor D.B.Spalding indicated the aims and general principles of the technique. It was realised that most engineering combustion equipments were expensive to build and test, and yet difficult to design from first principles. He stressed the need for the reduction or elimination of the expensive trial-and-error development process largely used in practice. Even full-scale testing does not give all the information that the designer needs, because of the difficulty of probing very hot gases associated with high-intensity combustion.

The aim of the analogue, set forth in Ref.1, is to reduce the time and cost of combustion chamber development, and to increase the understanding of the

detailed working of combustion phenomena.

The analogue incorporates a model of the combustion chamber design which is under investigation. Air flows into the model, but there is no chemical reaction; instead the reaction is simulated by electrical heating. The absence of high temperatures makes the analogue easy to build and modify; while the observation of the details of the simulated flame is facilitated by the low temperatures involved, and the fact that changes proceed in slow motion.

This type of analogue is concerned with steady-flow combustion equipment of the type used in aircraft gas-turbines and ram-jet engines; but not with combustion chambers where the chemical loading is very low, as in the case of steel-furnaces, for example. In such chambers the physical processes- such as the evaporation of the fuel drops, mixing of the air and fuel, and the heating of the mixture to ignition temperature- control the performance; the chemical reaction taking little time in comparison

with the time required for the transport processes. In the chambers with which the analogue is concerned the transport processes are made much faster, and the maximum space heating rates are aimed for. In such chambers of high loading, the chemical processes are therefore of great importance, especially at critical conditions.

The aim of this Chapter is to enumerate the theoretical bases for the use of the analogue, and the similarity conditions relating it to the combustion chamber which it endeavours to simulate.

Notation.

x, y, z = space rectangular coordinates,

D = a characteristic dimension of the
combustion chamber,

u, v, w = velocity components in the x, y , and
 z directions respectively,

u_0 = velocity of gases approaching the
combustion chamber,

X, Y, Z = the space rectangular coordinates
divided by D ,

U, V, W = the velocity components divided by u_0 ,

ρ = gas density,

k = thermal conductivity,

μ = absolute viscosity,

C_p = specific heat at constant pressure,

T = temperature,

D_j = diffusion coefficient of the gaseous
species j ,

m_j = fractional concentration of species j ,

$\dot{m}'''$ = volumetric rate of material generation, due to chemical reaction,

$\dot{q}'''$ = volumetric rate of heat release due to chemical reaction,

$\dot{q}_{m,m}'''$ = maximum volumetric heat release rate,

$\psi = \dot{q}''' / \dot{q}_{m,m}''' =$ heat release function,

o = suffix indicating reference conditions,

u = suffix indicating unburnt inlet conditions,

b = suffix indicating completely burnt conditions,

$\tau = (T - T_u) / (T_b - T_u) =$ reactedness,

$\Delta T = T_b - T_u =$ temperature rise due to complete adiabatic combustion,

H = heat of combustion,

Pe = the Peclet Number,

Rn = the Reynolds Number,

Pr = the Prandtl Number,

L = the loading factor.

The Differential Equations.

The similarity conditions and the theoretical basis for the use of the electric analogue depend on consideration of the basic equations for the conservation of mass, energy and momentum for steady-state gaseous systems with chemical reaction. These equations are discussed in detail by Spalding (Refs. 2 and 3), and other workers, (Refs. 4 and 11). They are here quoted, together with brief notes on their component parts.

Under certain conditions, the equations for the conservation of mass and energy become quite similar. This similarity is made clear when these equations are expressed in terms of the species fractional concentration and rate of generation on the one hand, and in terms of temperature and volumetric heat release on the other. The momentum equation is the same for the system with and without the chemical reaction.

The conservation equations can be obtained by considering the conditions prevailing in a small

elemental volume, whose dimensions are: $\delta x, \delta y$ and δz , in the x, y and z directions respectively. For a point (x, y, z) the three perpendicular velocity components are given the symbols u, v and w respectively. The conservation equations are discussed below:

1. Conservation of Mass.

The changes in the fractional concentration (m_j) of species j , are due to a balance between diffusion, convection and generation by chemical reaction. In the small elemental volume, the species j is being diffused at the rate of:

$$\left[\frac{\partial}{\partial x} (D_j \delta \frac{\partial m_j}{\partial x}) + \frac{\partial}{\partial y} (D_j \delta \frac{\partial m_j}{\partial y}) + \frac{\partial}{\partial z} (D_j \delta \frac{\partial m_j}{\partial z}) \right] \delta x \delta y \delta z ,$$

it is convected away from the element at the rate of:

$$\left[u \delta \frac{\partial m_j}{\partial x} + v \delta \frac{\partial m_j}{\partial y} + w \delta \frac{\partial m_j}{\partial z} \right] \delta x \delta y \delta z ,$$

and it is chemically generated at a volumetric rate designated by: $\dot{m}_j$.

A mass balance in the elemental volume leads to the equation for the conservation of mass, viz.,

$$\begin{aligned} & \left[\frac{\partial}{\partial x} (D_j \rho \frac{\partial m_j}{\partial x}) + \frac{\partial}{\partial y} (D_j \rho \frac{\partial m_j}{\partial y}) + \frac{\partial}{\partial z} (D_j \rho \frac{\partial m_j}{\partial z}) \right] \\ & - \left[\rho u \frac{\partial m_j}{\partial x} + \rho v \frac{\partial m_j}{\partial y} + \rho w \frac{\partial m_j}{\partial z} \right] \\ & + \left[\dot{m}_j''' \right] = 0 \dots\dots\dots (1) \end{aligned}$$

2. The Conservation of Energy.

The changes in temperature in the elemental volume are due to conduction, convection and heat generation due to chemical reaction. Heat is transferred to the element by conduction at the rate of:

$$\left[\frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (k \frac{\partial T}{\partial y}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) \right] \delta x \delta y \delta z ,$$

the rate of heat transfer by convection is:

$$\left[\rho u c_p \frac{\partial T}{\partial x} + \rho v c_p \frac{\partial T}{\partial y} + \rho w c_p \frac{\partial T}{\partial z} \right] \delta x \delta y \delta z ,$$

and the volumetric rate of heat release by chemical reaction is designated by: $\dot{q}'''$.

A heat balance under adiabatic steady-state conditions leads to the equation for the conservation of energy,

$$\begin{aligned} & \text{viz., } \left[\frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (k \frac{\partial T}{\partial y}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) \right] \\ & - \left[\rho u c_p \frac{\partial T}{\partial x} + \rho v c_p \frac{\partial T}{\partial y} + \rho w c_p \frac{\partial T}{\partial z} \right] + \dot{q}''' = 0 \dots\dots\dots (2) \end{aligned}$$

3. The Conservation of Momentum.

While mass and energy are scalar quantities, momentum is a vector quantity, and therefore its conservation has to be expressed by three equations - one in the direction of each of the rectangular axes x, y and z. It is sufficient for the present purpose to include only one equation, e.g. the one in the x-direction. The other two are quite similar (the three equations are generally called the Navier-Stokes equations). A balance between the inertia forces, the pressure forces and the viscous forces in the x-direction under steady-state conditions leads to the equation:

$$\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} + \rho w \frac{\partial u}{\partial z} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left(\mu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial u}{\partial z} \right) \text{------(3)}$$

¶

The pressure term in Equation (3) may be expressed in terms of velocities and velocity gradients, by the use of Bernoulli's Equation, viz.,

$$-\frac{\partial p}{\partial x} = \rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial x} + w \frac{\partial w}{\partial x} \right) \text{------(4)}$$

Remarks on the Equations.

Inspection of the mass and energy conservation equations, (1) and (2) , shows a degree of similarity - particularly if it is noted that the volumetric rates of species and heat generation are related by the equation :

$$\dot{q}''' = H_j \dot{m}_j' \text{ ----- (5)}$$

where :

H_j = heat of combustion of species j .

Equations (1) and (2) in fact become identical if :

- a. The diffusion coefficients of the various species generated by chemical reaction are equal to one another, and to the thermal diffusivity of the mixture, i.e.,

$$D_j = \alpha = k / C_p \rho \text{ .}$$

- b. The chemical reaction always proceeds by the same path.

These conditions do not always obtain, but they are approximately obeyed under certain conditions; and, since their assumption makes the equations of mass and energy identical, all the necessary parameters for the description of the performance of the flow system with chemical reaction may be obtained from consideration of the energy and momentum equations. Such parameters are obtained by expressing the two equations in a dimensionless form. This is done in the section following.

Dimensionless Form of the Energy and Momentum Equations.

The various properties contained in the energy and momentum equations are expressed in a dimensionless form by the use of reference values. These are given as follows:

Coordinates: a characteristic dimension D ,
thus:

$$X = x/D,$$

$$Y = y/D,$$

$$Z = z/D.$$

Velocities: inlet mixture approach velocity, u_o ,
thus:

$$U = u/u_o,$$

$$V = v/u_o,$$

$$W = w/u_o.$$

Density, [values at inlet conditions,
Viscosity, given suffix o,
Sp. Heat, thus:

Thermal Cond- $\sigma = \rho / \rho_o,$

uctivity, $\lambda = \mu / \mu_o,$

$$C = C_p / C_{p_o},$$

$$K = k / k_o.$$

Volumetric [maximum volumetric heat release
Heat Release rate, thus:

Rate: $\psi = \dot{q}''' / \dot{q}_{m}'''$

Temperature is made dimensionless by transforming it
into the reactedness, τ , given by:

$$\tau = (T - T_o) / (T_b - T_o)$$

where:

T = temperature,

b = suffix denoting completely burned conditions,

o = inlet, or unburnt conditions.

With these transformations, the energy and momentum equations become:

$$\frac{1}{R} \left[\frac{\partial}{\partial x} \left(\kappa \frac{\partial \tau}{\partial x} \right) + \frac{\partial}{\partial y} \left(\kappa \frac{\partial \tau}{\partial y} \right) + \frac{\partial}{\partial z} \left(\kappa \frac{\partial \tau}{\partial z} \right) \right] - \gamma C \left[U \frac{\partial \tau}{\partial x} + V \frac{\partial \tau}{\partial y} + W \frac{\partial \tau}{\partial z} \right] + \frac{1}{L} \Psi = 0 \dots\dots(6)$$

and:

$$\gamma \left[U \frac{\partial v}{\partial x} + V \frac{\partial v}{\partial y} + W \frac{\partial v}{\partial z} \right] = \gamma \left[U \frac{\partial v}{\partial x} + V \frac{\partial v}{\partial x} + W \frac{\partial v}{\partial x} \right] + \frac{1}{R_n} \left[\frac{\partial}{\partial x} \left(\lambda \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\lambda \frac{\partial v}{\partial y} \right) + \frac{\partial}{\partial z} \left(\lambda \frac{\partial v}{\partial z} \right) \right] \dots\dots\dots(7)$$

where:

$$Pe = \text{the Peclet Number} = \frac{u_o g_o C_p o D}{k_o}$$

$$L = \text{the loading factor} = \frac{u_o g_o C_p o (T_b - T_o)}{\dot{q}'' D}$$

$$R_n = \text{the Reynolds Number} = \frac{g_o u_o D}{\mu_o}$$

Parameters for the Description of Combustion Chamber
Performance.

In Equations (6) and (7), one notes that the terms K, δ, C, λ depend only on temperature, and so, if the heat release function ψ is explicit in temperature, then Equation (6) will give the temperature pattern in the combustion chamber in terms of Rn, Pe and L . The conditions under which ψ may be a function of temperature alone are discussed by Spalding, (Refs. 3 and 9). These are briefly indicated later in this section. However, it is sufficient here to state that this might hold for the adiabatic one-stream type of combustion chamber. Such a chamber is one in which the air and fuel enter already mixed. e/

The ratio of the Peclet Number to the Reynolds Number, which is equal to the Prandtl Number, ~~which~~ is effectively constant for gaseous systems. Thus the performance of an adiabatic steady-state

one-stream combustion system may be characterized by the parameters L and Rn . In fact, if the flow is fully turbulent, the Reynolds Number loses its influence due to the fact that the viscous stresses become negligible in comparison with the inertia stresses. The performance of a one-stream combustion chamber is therefore mainly characterized by the loading factor L . This is the sixth of seven Useful Half Truths formulated by Professor D.B. Spalding (Ref.6). These "U.H.T."s are statements " which are sufficiently compact to be memorable, sufficiently general to cover most situations, and sufficiently near the truth to form useful approximations to reality." (Ref.6). The sixth, or the one-stream U.H.T. actually states that the state of a one-stream steady flame in a given chamber depends only on its dimensionless loading, L . Symbolically,

$$\text{Flame state} = F(L) \dots\dots\dots(8).$$

In the preceding discussion, reference was made to the conditions under which the volumetric

heat release rate ($\dot{q}'''$) in a one-stream combustion chamber may be considered as an explicit function of local reactedness ($\mathcal{T}$). This relation between ($\dot{q}'''$) and ($\mathcal{T}$) is in fact the fourth U.H.T. (Ref.6). The requirements for its validity, and a theoretical analysis relevant to the chemical conditions are given in Refs. 6,9 and 10. The flow is required to be adiabatic and turbulent, though the U.H.T. is still valid for laminar flow where the diffusion is "normal", i.e., where the diffusion coefficients of the various chemical species present can be assumed equal to each other and to the thermal diffusivity of the mixture. Furthermore, the chemical reaction is assumed to be controlled by one of the following categories:

1. Single-step reactions,
2. Chain reactions obeying the T.Von Karman and S.Fenner steady-state approximation, which assumes that the local radical concentration is determined only by opposed production and destruction reactions.

3. Chain-branching reactions where the chain reaction is very fast compared with the radical generating and chain breaking reactions.

Modelling of the Combustion Chamber.

The conclusion of the previous section suggests that under fully turbulent conditions, similarity between a one-stream combustion chamber and a model of it will obtain if the loading factor is the same in the two systems. Furthermore, if the density changes consequent upon combustion are neglected, the above discussion will also hold for a semi-cold model of the combustion chamber, provided that we introduce into the model working section a heat release rate function dependent on the local temperature in a manner similar to the dependance of the heat release rate on the local temperature in the combustion chamber.

The electric analogue is such a semi-cold model. In it the heat release function, $\dot{Q}$, is produced by means of electric heaters distributed throughout the working section. Their rate of heat generation is controlled from the outside in accordance with the local temperatures measured by thermocouples which are also distributed inside the analogue working section. The details of heaters and thermocouples distribution, and the form of the heat release function, are discussed in Chapter Two. ψ

Similarity Conditions.

The Equation (6) of the conservation of energy will be valid for both model and original when the conditions mentioned in the previous discussion are applicable. Its solution gives the temperature or reactedness distribution, and hence some of the characteristics of the flame under investigation. For example, extinction will be the condition when there is no non-zero solution to the Equation (6). The form of the solution of this equation is the subject

of the next part of the present Chapter; but first we shall briefly enumerate the conditions necessary for similarity to obtain between the original combustion chamber and the analogue model. These conditions are the assumptions and requirements stated, or tacitly assumed in the previous discussions, and are as follows:

1. Geometric similarity must obtain between the combustion chamber and its analogue model,
2. Bouyancy and compressibility effects must be negligible. These require a low Mach Number, and a high Froude Number. Such conditions are often fulfilled in the type of combustion chamber with which the analogue is concerned.
3. Gas dynamic effects, such as pressure waves, and screeching, must be absent. The analogue is not designed to study such effects.

4. Only gaseous materials are assumed to be present in the combustion chamber. In the present work we have only considered one-stream chambers into which the air and the fuel enter already homogeneously mixed.
5. The Reynolds Numbers in the original and in the analogue must be equal. This equality is not necessary if the flow is fully turbulent - that is, when the viscous stresses in it are greatly outweighed by the inertia stresses.
6. The flow is assumed adiabatic; thus the effect of heat losses must be negligible.
7. The effects of density changes due to combustion are neglected.
8. In both the original and its analogue model, similar relations must obtain between the volumetric heat release rate and the local reactedness.

In examining these similarity conditions, we note that the first four are fulfilled by design, because the analogue model is made geometrically similar to the chamber under investigation, and because it is not made to study chambers where appreciable effects of bouyancy, compressibility, screeching or fuel droplets are present. If such effects are appreciable in the original chamber, then the comparison of its performance with the analogue predictions is not valid. Furthermore, in considering the analogue predictions, or comparing them with flame results, care must be taken to ensure that the flow is actually in the regime satisfying the requirements of either fully-developed turbulence or equal Reynolds numbers. In the combustion chambers with which we are here concerned, the requirement of fully-developed turbulence is usually the more relevant.

The lack of fulfillment of the adiabatic flow condition is one of the drawbacks of the analogue technique. It is expected that the discrepancy between the performance of the original chamber and

the predictions of its analogue model becomes, due to this effect, more serious when low pressures are involved than at atmospheric pressures. This expectation is based on the results obtained by Weiss and Spalding, and their associates. The first (Ref. 7), discovered by experiment that even drastic lagging of a can reaction zone did not greatly influence the stability characteristics of the flame they studied; while Spalding et alia, (Ref. 8), found by theoretical analysis, that the effects of radiation heat losses increase as the pressure decreases. The results might be taken to suggest that the lack of fulfillment of the sixth similarity condition is not very important. However, this and the fact that no appreciable density changes occur in the analogue, constitute the chief sources of discrepancy between the analogue and the original chamber performance. On the other hand, they are the main simplifying factors in the

use of the analogue technique. One of the aims of this work is, therefore, to investigate the magnitude of the discrepancy, and to ascertain its effect on the validity of the analogue technique.

As far as the curves of the volumetric heat release rate against the local reactedness are concerned, their similarity in both the original and the analogue model can theoretically be achieved regardless of their complexity. In ~~the~~ present work, a simple curve is introduced into the analogue for reasons of experimental convenience. This simplification introduces an error, which is , however, of secondary importance. The form of this simple curve, and the error involved, are discussed in later Chapters.

The remaining part of this Chapter will be devoted to further discussion of the energy equation, and its relevance to the electric analogue.

Solution of the Energy Equation.

The dimensionless form of the equation for the conservation of energy was given by Equation (6). Analytical solution of this equation is theoretically possible if knowledge of the distribution of δ, C, U, V, W is available throughout the chamber volume. This is very difficult to obtain, however, and even when obtained, the mathematics and computations involved would be prohibitive. It is therefore more profitable to endeavour to design some experimental techniques that will yield solutions for the desired features of the flame under study. One such technique, advanced by Spalding, is the influence coefficient approach, (Ref.6).

A form which the solution to Equation (6) might take (Spalding) is given in integral terms in Equation (9):

$$T_j = \frac{1}{L} \int_0^1 \psi_k f(r_j, r_k) \frac{dr_k}{r} \text{ ----- (9)}$$

where:

τ_j = reactedness at a point j in the chamber volume,

ψ_k = the value at a point k of the heat release function ψ ,

$f(\tau_j, \tau_k)$ = a dimensionless function, containing the aerodynamic effects between the points j and k in the chamber volume.

(This function is designated as the influence coefficient between the two points j and k, and is given the symbol α_{jk}).

v = volume over which the integration is carried, i.e., the volume of the considered working section,

v_k = the elemental volume associated with the point k.

The integral form of the solution (9) is transformed into the summation:

$$\tau_j = \frac{1}{L} \sum_k \alpha_{jk} \psi_k \frac{\Delta \psi_k}{\psi} \text{ ----- (10)}$$

or, introducing a "weighted influence coefficient", β_{jk} , given by:

$$\beta_{jk} = \alpha_{jk} \frac{\Delta v_k}{v} \text{ ----- (11)}$$

Equation (10) is transformed into:

$$\tau_j = \frac{1}{L} \sum_k \beta_{jk} \psi_k \text{ ----- (12)}$$

This form of the solution (Equation 12) is quite adaptable to experiment. What is now required is to obtain sets of β_{jk} for sufficient numbers of pairs of points j and k; and to compute, by an iterative procedure, the highest value of L which allows the τ 's to have non-zero values. This value of L would then be the extinction loading factor, L_{ext} .

Determination of β_{jk} .

In certain cases, where a simplified flow pattern may be postulated, sets of β_{jk} can be calculated, but in general they have to be measured experimentally. This may be done for a pair of points j and k, either by injecting a

tracer gas at the point k, and measuring consequent concentration of the tracer at the point j; or by releasing heat at a known rate at the point k, and measuring the temperature rise at the point j.

The first technique was used for both hot and cold flows at Messrs. Tiltman Langley and Co. in collaboration with Professor D.B. Spalding. The second technique is used in the electric analogue. Only temperature rise will therefore be considered in this discussion. However, the argument for tracer injection technique is quite parallel. The derivation of β_{jk} from the measured temperature rise at j, ΔT_j , is as follows:

The loading factor is re-defined to include the volume of the chamber in its denominator, i.e.,

$$L = \dot{m} C_p \Delta T / \dot{q}_{km}'' v,$$

where:

$\dot{m}$ = mass flow rate of mixture into combustion chamber,

v = volume of the combustion chamber,

$\dot{q}_{km}''$ = maximum volumetric heat release rate,

ΔT = adiabatic combustion temperature rise,

C_p = specific heat at constant pressure.

With this definition of L , substituted in Equation (10) we get a relation for ΔT_j , viz.,

$$\Delta T_j = T_j - T_u = \sum_k \alpha_{jk} \frac{\dot{q}_k}{\dot{m} C_p} \dots\dots\dots(13).$$

where:

$$\dot{q}_k = \dot{q}_k^m \Delta v_k \dots\dots\dots(14).$$

So, if heat is released only at a point k at the rate of $\dot{q}_k$ then the temperature rise at j will be given by:

$$\Delta T_j = \alpha_{jk} \frac{\dot{q}_k}{\dot{m} C_p} \dots\dots\dots(15).$$

This equation also gives the influence coefficient:

$$\alpha_{jk} = \frac{\dot{m} C_p \Delta T_j}{\dot{q}_k} \dots\dots\dots(16).$$

Equation (16) shows how the influence coefficient can be calculated from ΔT_j for any pair of points j and k in the control volume. The weighted influence coefficients can then be calculated from Equation (11).

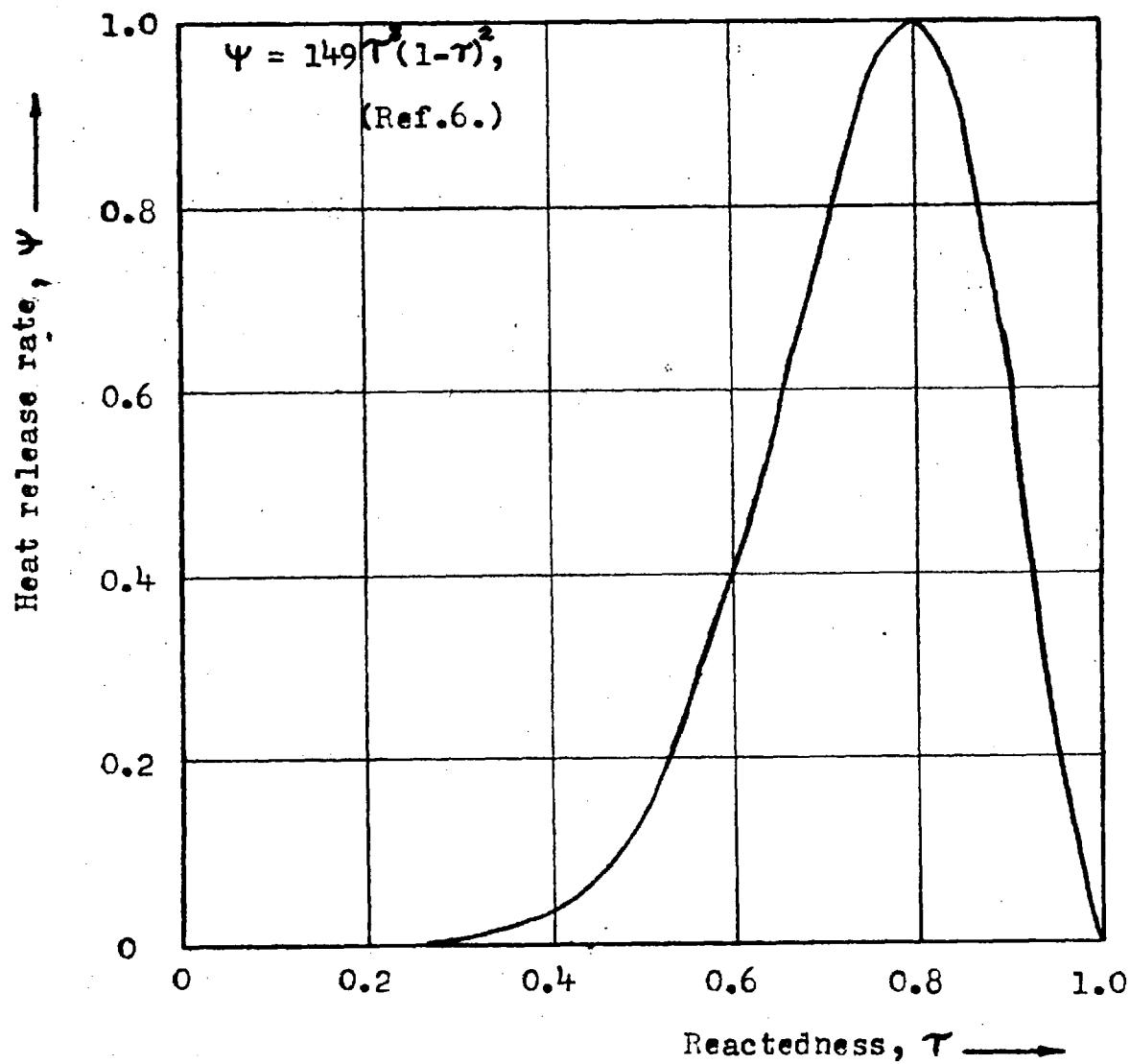
$$\beta_{jk} = \alpha_{jk} \cdot \frac{\Delta v_k}{v} \dots\dots\dots(17).$$

Solution of the Influence Equation (12).

The set of simultaneous influence equations (12) can now be solved by an iterative procedure, if a heat release function ψ explicit in reactedness can be assumed. A "realistic" function (Spalding - Ref.6) used for this purpose is that given by Equation (17) below; (plotted in Fig. 1).

$$\psi = 149 \tau^8 (1-\tau)^2 \text{-----} (17)$$

In view of the necessity for having a reasonably good coverage of the control volume, a large number of influence coefficients has to be measured, giving rise to a correspondingly large number of influence equations. Hence a computer was developed for the purpose by Messrs. Tiltman & Langley Co. Ltd. This computer has been used to solve a few sets of influence equations in the course of this investigation.



"REALISTIC" FUNCTION OF HEAT RELEASE RATE

Fig.1.

The Electric Analogue.

In the electric analogue, the influence coefficients can, of course, be measured for various pairs of points j and k . What the analogue does, however, is both to measure the influence coefficients and also to solve the set of influence equations without the aid of the computer. In other words, it is both a model and a computing machine at the same time.

The construction and experimental procedure of the electric analogue form part of the subject of the next Chapter.

CHAPTER TWO.

APPARATUS AND EXPERIMENTAL PROCEDURE.

-oOo-

Introduction.

The ideal electric analogue for turbulent flames was first described by Spalding in Ref. 1.

A summary of that description will now be given.

A geometrically similar model of the combustion chamber design under investigation is constructed. Air flows through this model in a manner similar to that of the working fluid which would flow through the original combustion chamber. Electrical heating elements are evenly distributed throughout the model volume. Each one can be individually supplied with electric current, the amount of which may be controlled from outside the analogue working section. A thermocouple is associated with each heater, and measures a representative air temperature in the volume element served by the heater. The heaters, thermocouples, and their supports, are sufficiently fine to have no appreciable

effect on the flow pattern in the analogue model.

The thermocouples are connected to the external controls, which operate so that the relation between the rate of heat input from any heater and the local air temperature is similar to the relation between the heat release rate and the local gas temperature in the original combustion chamber.

In terms of the parameters used in Chapter One, such an arrangement of heaters and thermocouples makes possible the introduction throughout the analogue volume, of the volumetric heat release rate function dependent explicitly on the local temperature or reactedness. When the controls are adjusted so that the heat input rate is related to the temperature everywhere in accordance with this ψ - function, the temperatures in the analogue are the non-zero solutions of the set of influence equations, and are therefore the analogue predictions of the flame state which would prevail in the

simulated combustion chamber. The absence of any non-zero temperature differences above the air inlet temperature, indicates the prediction of the extinction condition.

The details of the design and construction of a simple axi-symmetric analogue model forms part of the subject of this Chapter. The other parts give a brief description of:

1. A more complex analogue model built by J. Lucas & Co. Ltd.
(Designated as the Lucas model).
2. A flame stability apparatus from which some flame extinction measurements were obtained during of this investigation.
3. The experimental procedures for running the analogue and the flame stability apparatus.

The Simple Axi-Symmetric Analogue Model.

This electric analogue model is intended to simulate a simple cylindrical combustion chamber with baffles of the bluff-body type, symmetrically disposed about the axis of the chamber, and normal to the flow direction. This analogue is here referred to as the "present analogue model". The principal difference between this model and the ideal electric analogue described in the previous section, is the shape of the heat release function. This shape has, in fact, been used in all the previously constructed analogue models, and is described below.

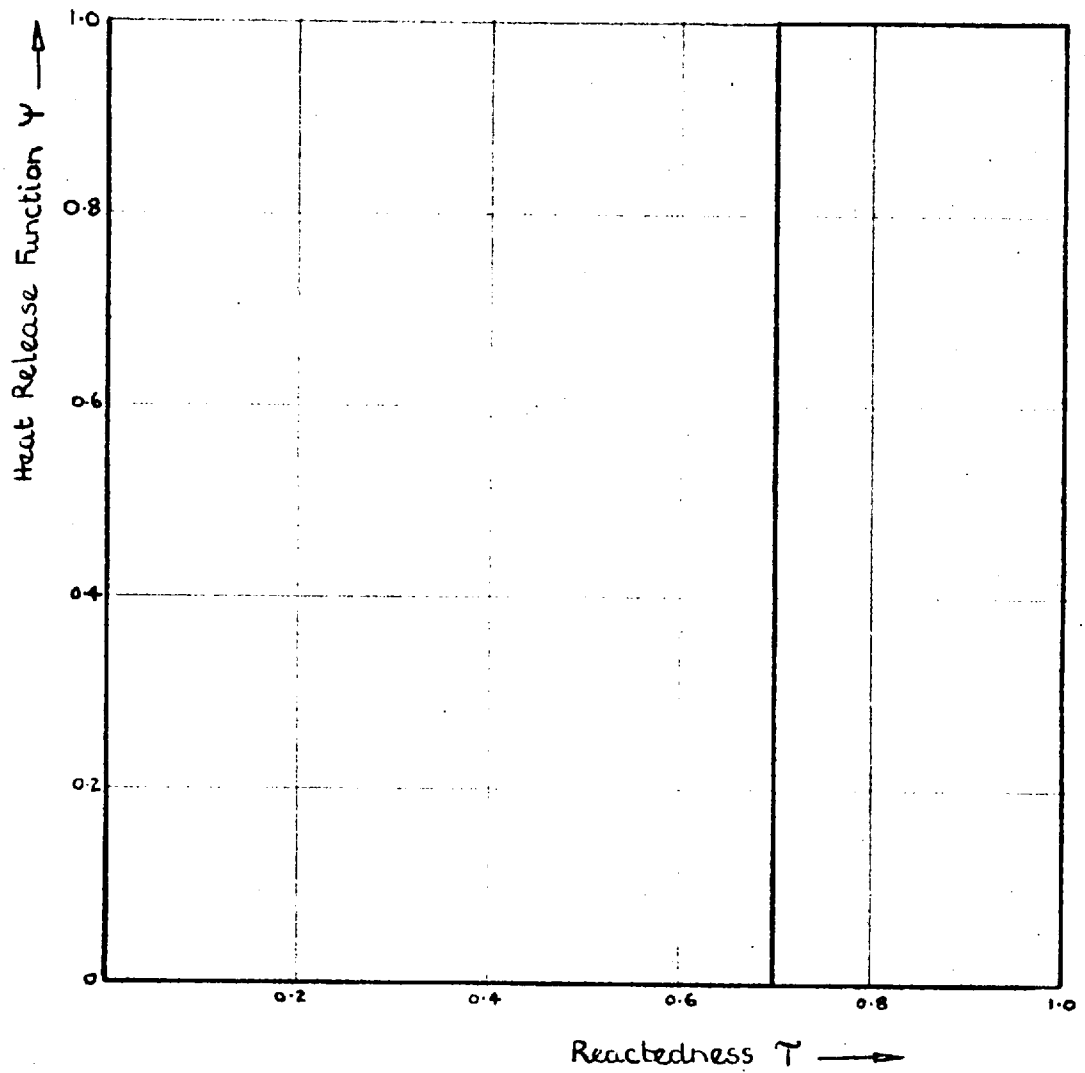
The Heat Release Step-Function.

For purposes of simplicity of construction, instrumentation, and operation, a step-function is used instead of the more realistic continuous heat release function of Fig.1. This

step-function is shown in Fig.2. The volumetric heat release rate is taken to be at its maximum value ($\dot{q}_m'''$) when the local reactedness prevailing in the elemental volume served by a heater is between π and 1; and is taken to be zero when the reactedness is between zero and π_i . The value of π is chosen to give a reasonable approximation to the "realistic" heat release function. A value of 0.7 is given to π_i in this investigation. The step-function is easily produced by an on-off temperature-controlled switch, which may be operated manually, or by an automatic apparatus.

Overall Features of the Present Analogue Model.

The present analogue model was designed to study the stabilization characteristics of flames in the wake of annular and central baffles in an axi-symmetric combustion chamber; and to ensure the inclusion, in the working section,



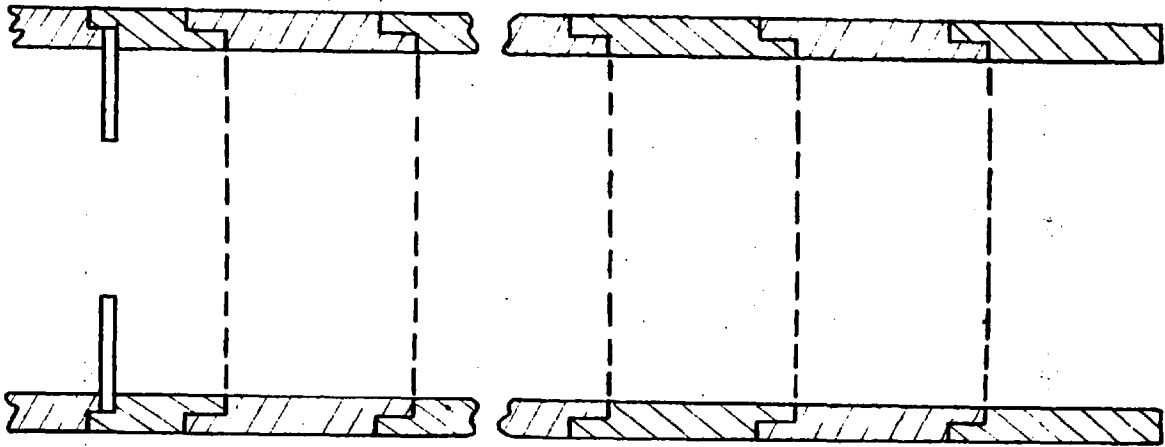
THE HEAT RELEASE RATE FUNCTION USED IN
THE ANALOGUE EXPERIMENTS.

FIG. 2.

of the entire recirculation zone which lies behind each baffle. The working section is cylindrical in shape, with an inside diameter of $6\frac{3}{16}$ inches, and a length of 32 inches. It is comprised of sixteen elements, each having an axial length of 2 inches, and incorporating six heaters, and six thermocouples, together with their supports. The walls of the adjacent elements fit into one another to form the whole length of the working section. Baffles are placed normal to the chamber axis at a fixed distance from the heaters of the first element. These overall features are shown in Fig.3, while the detailed construction of the components of an element are discussed below.

The Wall and Carriers.

Fig.4 shows a section through the wall of one element of the analogue. The edges of the wall are made to enable the adjacent elements to

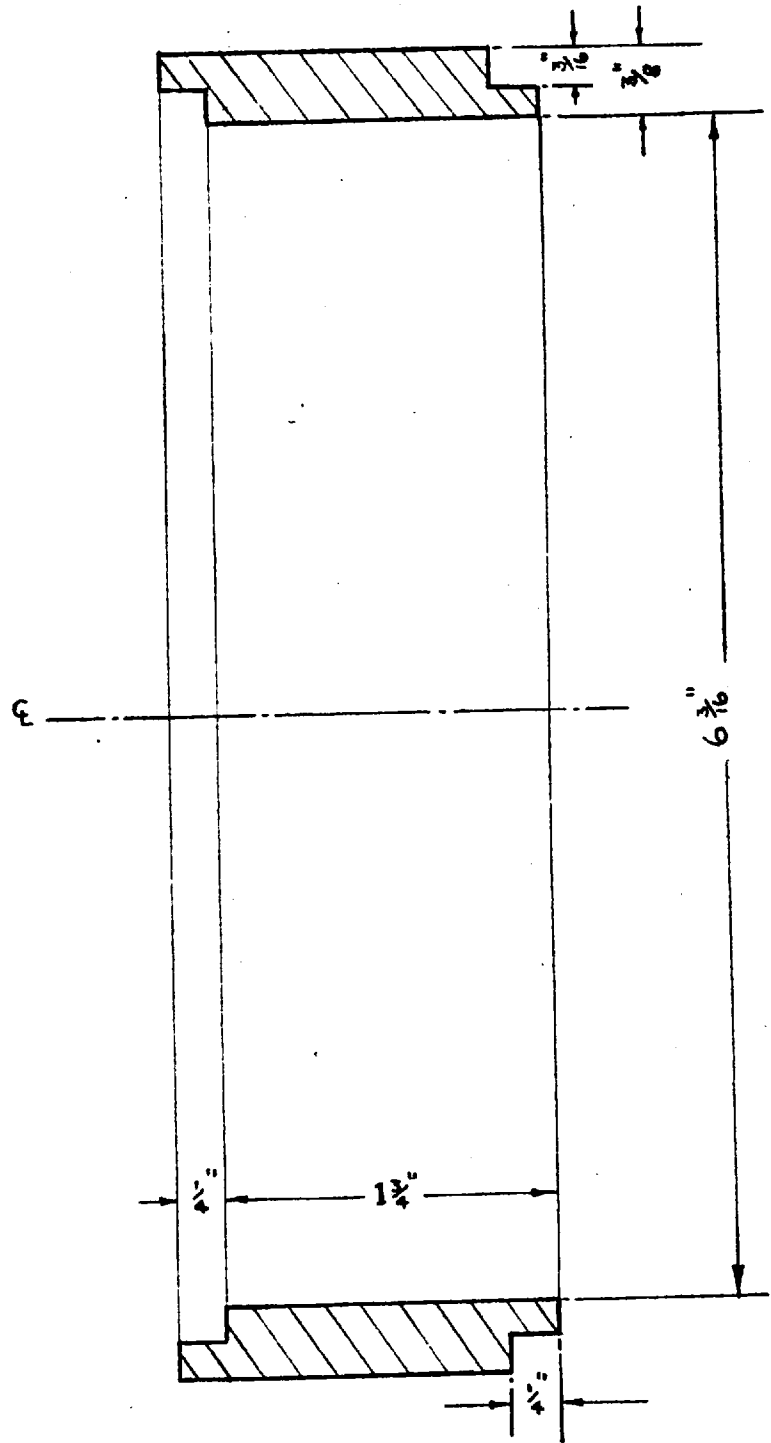


A SKETCH SHOWING THE ARRANGEMENT OF THE WALL
ELEMENTS AND THE POSITION OF AN ANNULAR BAFFLE.

Fig.3.

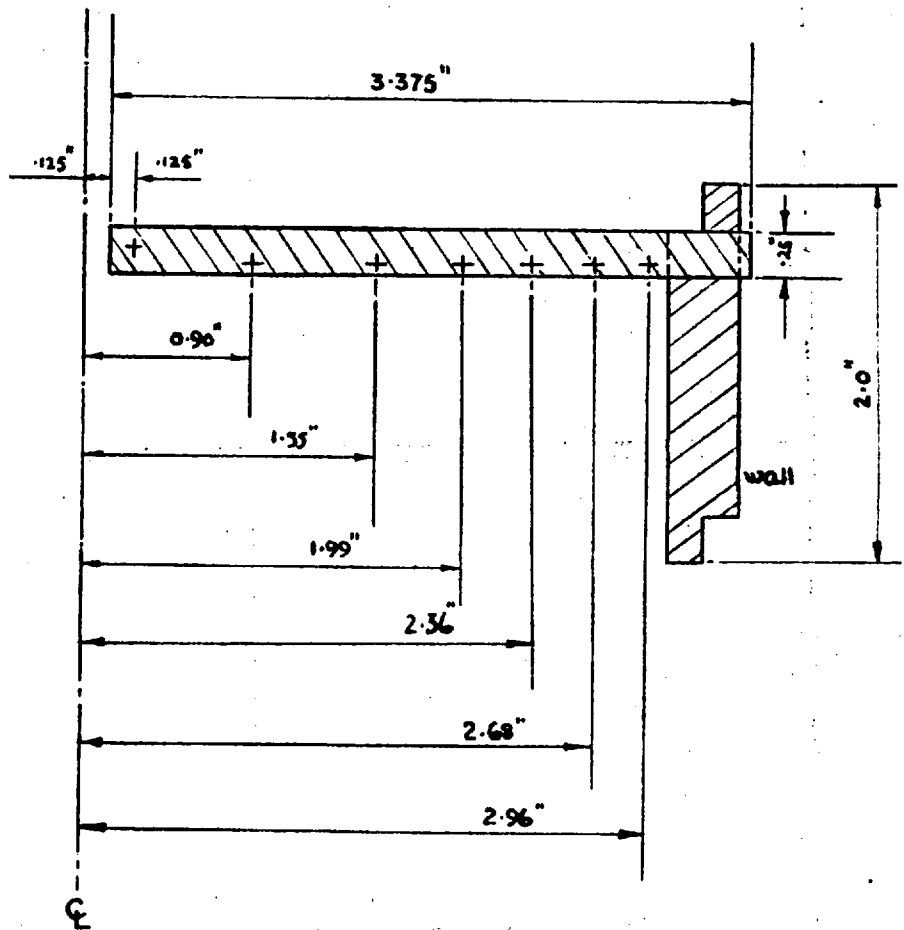
fit into each other, forming the whole length of the analogue working section. Fig. 4 does not indicate the manner in which the carriers of heaters and thermocouples are held in position. This is shown in Fig.5, which gives a diagram of part of a section through the wall and one of the carriers, in particular it shows the cut in the wall where one of the carriers is placed. There are six of these cuts in every element, each accommodating one of the six carriers. The cuts are disposed at equal angular intervals around the wall. There are seven small holes in each carrier. Through the one nearest the centre of the model passes a circular metal ring which holds the six carriers of the element firmly in position. Through each of the other six holes passes one of the six circular heater wires. The heaters are fixed to the carriers at the points of contact. The construction ensures that the arrangement is firm, and able to stand the pressure exerted by the

passing air over the whole range of air-flow rates of interest in this investigation. Each one of the six carriers also holds in position a thermocouple corresponding to one of the heaters. The thermocouple wires are led along the carrier, and through the cut in the wall to the controls outside the working section. The two ends of one heater are silver-soldered to copper wires, and similarly led along the carriers towards the outside of the analogue. The carriers were made of "Tufnol" - a hard insulating material.



SECTION THROUGH WALL OF
AN ANALOGUE ELEMENT.

FIG. 4.



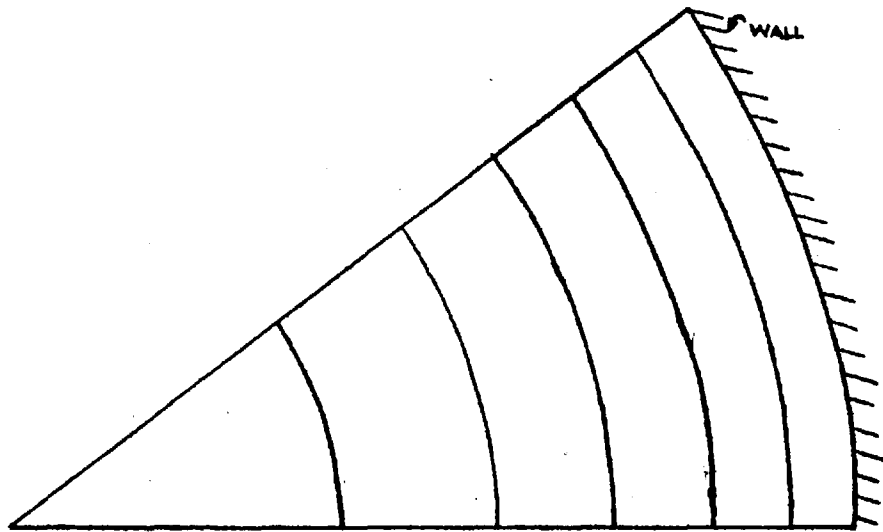
SECTION THROUGH WALL
AND CARRIER.

FIG. 5.

The Heater Elements and their Distribution.

There are ninety-six heaters in the analogue working section; six in each one of the sixteen elements. All these heaters are made of nickel-chrome wires of equal resistances (6.7 Ohms); but of lengths and gauges differing according to their radial position in the element. This radial position is given by (r_m) , the radius of the circle that the n th. heater has to describe. The suffix n takes the values 1 - 6 from the centre of the element towards its wall. The value of (r_m) is calculated on the basis that the element is considered to be divided into six concentric parts, each being called a " cell ". The cells are made equal in volume, and each one is served by a heater and a thermocouple.

A segment of the element, showing the radial boundaries of a cell, is given in Fig.6.



A SEGMENT OF AN ELEMENT, SHOWING THE
OUTER BOUNDARIES OF THE CELLS.

FIG. 6.

The outer radius of the nth. cell is given by:

$$R_m = R_c \sqrt{m/6} \dots\dots\dots(18).$$

where:

R_m = outer radius of the mth. cell,

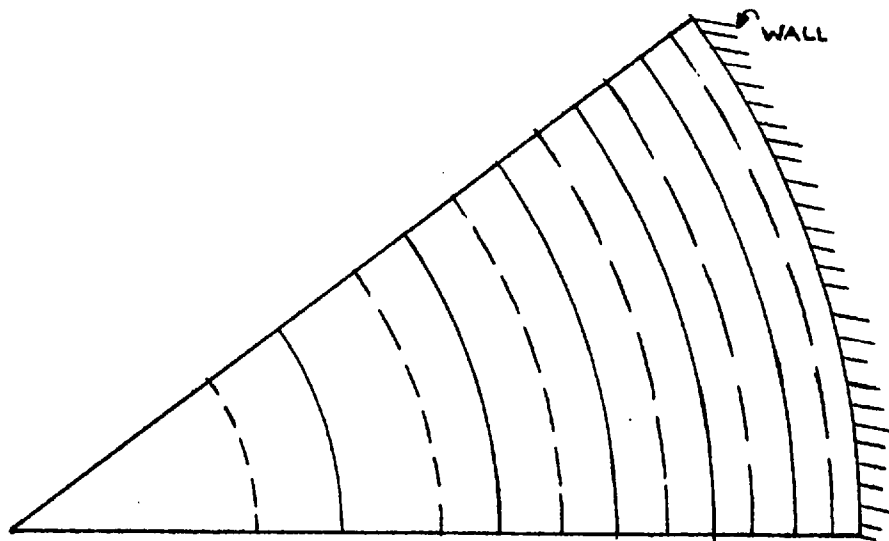
R_c = inner radius of the analogue chamber.

The heaters are placed in one plane normal to the chamber axis; each one being along the centre of area of the cross-section of the cell at that plane. This arrangement is sketched in Fig.7; the solid lines being the cell boundaries, while the dotted lines represent the positions of the heaters. The radius of the mth. heater, r_m , is obtained by taking moments about the centre, thus:

$$\int_{R_{m-1}}^{R_m} 2\pi x^2 dx = r_m \cdot \pi (R_m^2 - R_{m-1}^2) \dots\dots\dots(19)$$

This leads to:

$$r_m = \frac{2}{3} \left[\frac{R_m^2 + R_m R_{m-1} + R_{m-1}^2}{R_m + R_{m-1}} \right] \dots\dots\dots(20)$$



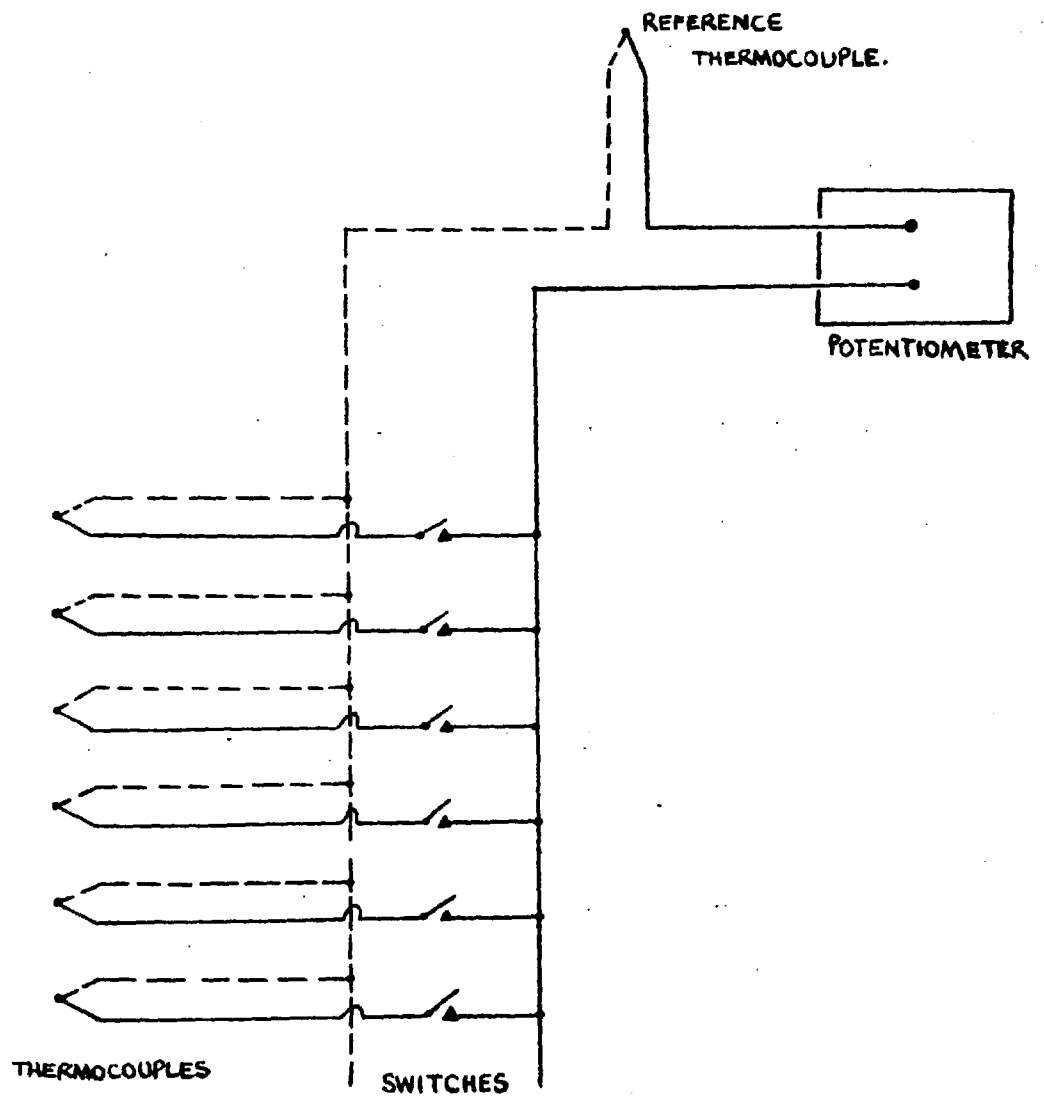
A SEGMENT OF AN ELEMENT, SHOWING THE
POSITIONS OF HEATERS AND BOUNDARIES OF CELLS.

FIG. 7.

The Thermocouples.

The thermocouples are of the copper-constantan type, with a linear electro-motive force / temperature characteristic at the range of temperatures involved in the analogue experiments. The output e.m.f.'s of the thermocouples are measured by a potentiometer. At the entrance of the analogue working section, a thermocouple, called the " reference thermocouple " is placed to measure the temperature of the air approaching the analogue.

As shown in Fig.8, the reference thermocouple is connected to each one of the thermocouples inside the analogue working section. Thus, there is actually a system of differential thermocouples; so that when a thermocouple (j) is in the potentiometer - thermocouple circuit, the reading of the potentiometer is a measure of the temperature difference ($T_j - T_u$), where:
 T_j = temp. at point j in the working section,
 T_u = temp. of the air approaching the analogue.



THE THERMOCOUPLES CIRCUIT.

FIG. 8.

The Electric Circuit.

The electric circuit is required to provide low voltage alternating current to ninety-six electric resistors inside the analogue working section. The voltage has to be kept constant, regardless of the number of heaters that are switched on. As a refinement, there is a display panel of red lights to show which heaters are on, and which are off. This panel shows the pattern of the stable flame predicted by the analogue for a particular loading.

The normal working voltage is 12 volts. It is fed to the heaters from the 230 volts A.C. mains via a step-down transformer. A variable auto-transformer (Variac) is placed between the mains and the step-down transformer, to enable any adjustment which might be made necessary by any fluctuation in the mains voltage. The circuit of each heater resistance also contains an on-off switch, and a red light bulb in

the display panel. As the number of heaters switched on varies from one loading to another, the output voltage of the step-down transformer will also vary. Ballast heaters are added to the circuit to offset this voltage variation. To each heater in the working section, there is a corresponding ballast heater outside, equal to it in resistance, and so arranged that when the heater is off, the ballast heater is on, and vice-versa. The electric circuit is shown in Fig.9.

Air Supply and Flow Measurement.

Air is passed through the analogue working section by a Blackman blower, which is driven by a 20 amps. - 240 volts D.C. motor. The air flow rates through the apparatus can be varied by means of a throttle-valve in the suction side of the blower. These flow rates are measured by the pressure differentials across

a standard orifice plate, in accordance with the British Standards Specifications, Ref.12.

The Baffles.

Annular and central baffles have been tested in the analogue. The geometrical range covered is given below:

a. Annular Baffles:

d/D ratio from 0.112 to 0.89,
 d being the baffle hole diameter.

b. Central Circular Discs:

d/D ratio from 0.325 to 0.89,
 d being the diameter of the disc.

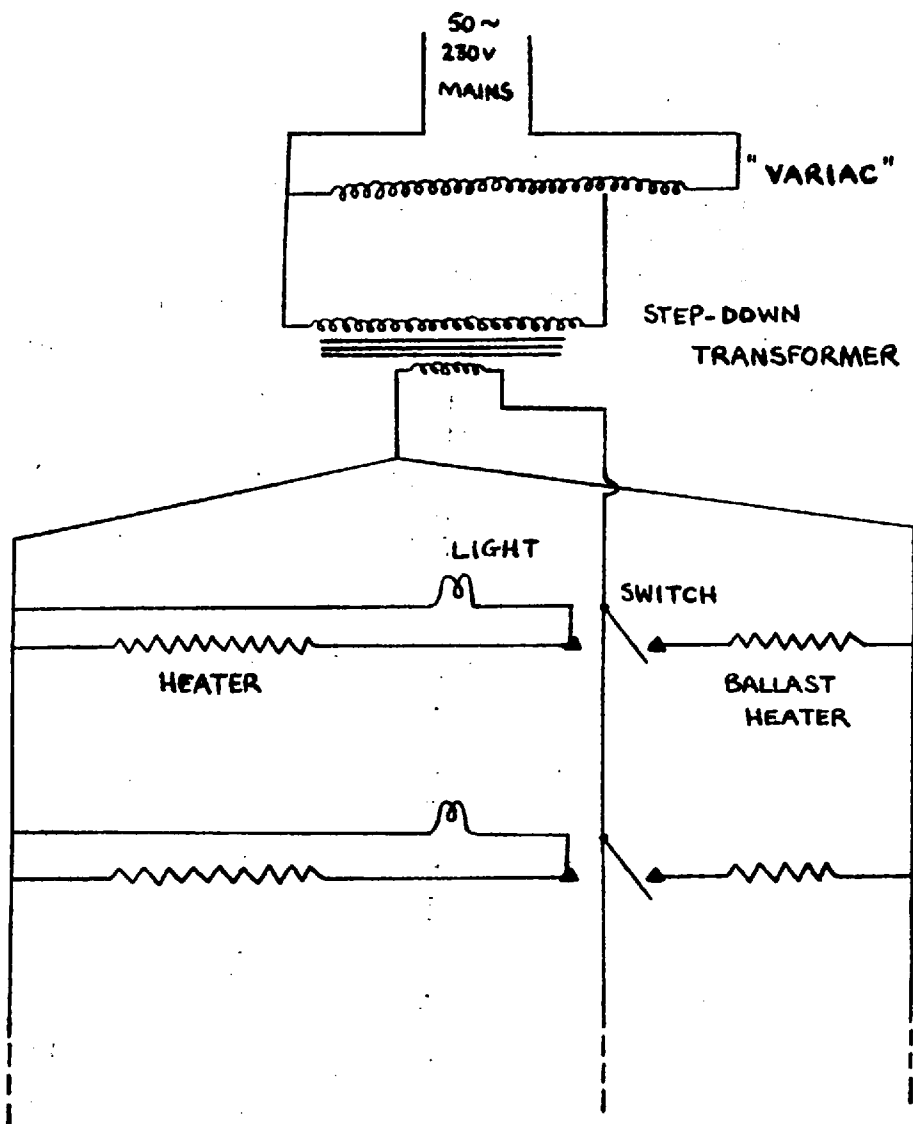
c. Hemi-spheres :

d/D ratio from 0.385 to 0.730,

d. Cones:

d/D ratio from 0.415 to 0.689,

θ , the cone apex angle, from 40° to 120° .



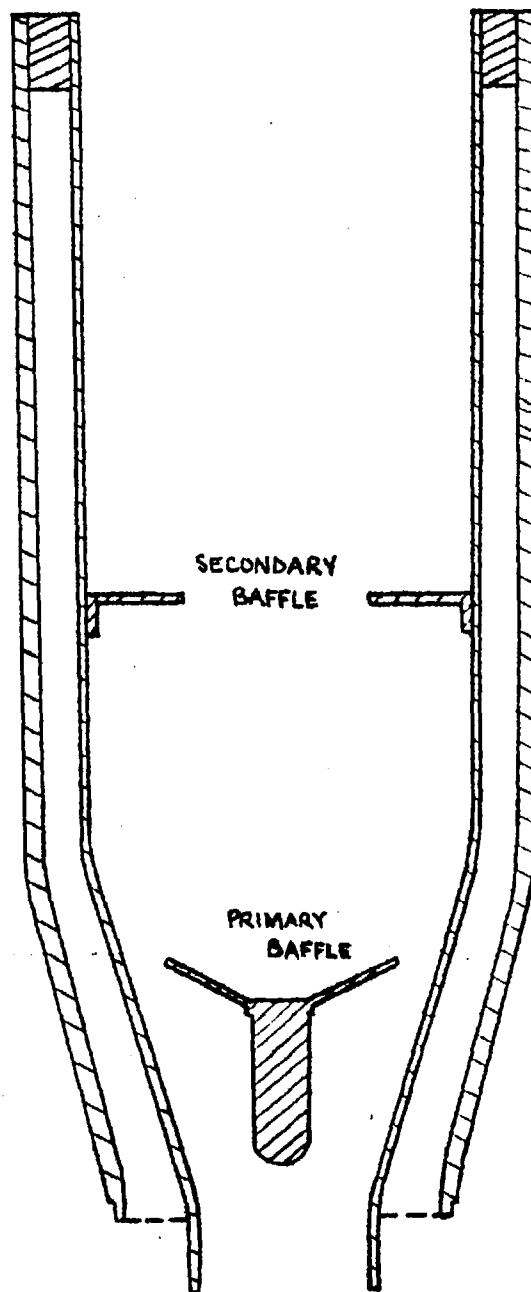
ELECTRIC CIRCUIT

FIG. 9.

The Lucas Analogue Model.

This model was described in Refs. 1 and 13. It has an axi-symmetric working section, with a primary and a secondary baffle. The primary baffle is conical in shape, and fixed in dimensions; whereas the secondary baffle is flat and annular, with a variable hole diameter. Fig.10 shows a view of the working section. Apart from the geometrical configuration of this model, it is designed on the same principles as the model built for this investigation (the " present analogue model ").

The procedure of operation for both electric analogue models is also similar. The Lucas model was used in the early stages of this research to study the effect of the secondary baffle hole diameter on the extinction loading of the chamber. The results are reported in Chapter Three.



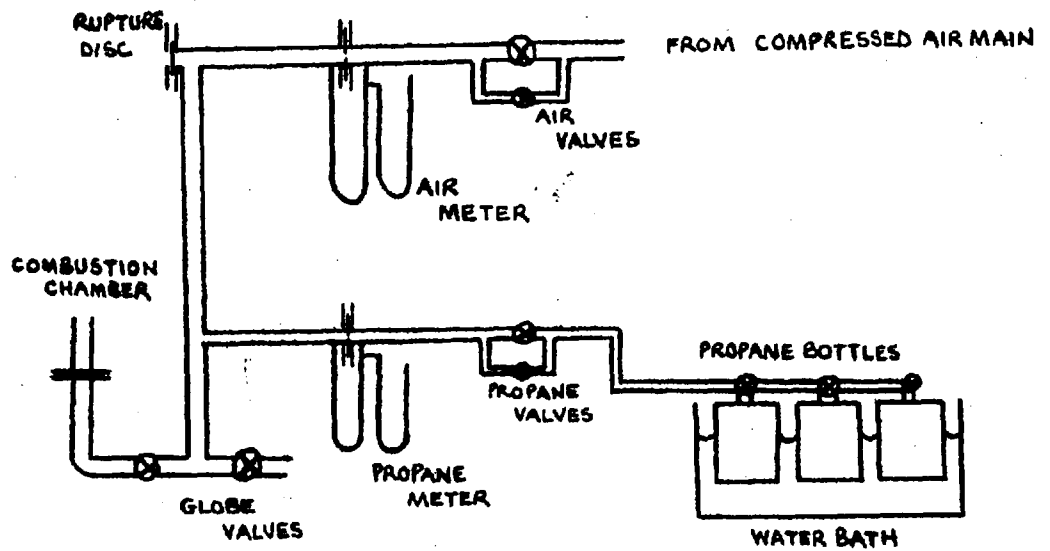
SECTION THROUGH THE
"LUCAS ANALOGUE MODEL."

The Flame Stability Apparatus.

A schematic diagram of this apparatus is given in Fig.11. The combustion chamber is cylindrical in shape, with a diameter of 2.0 inches, and a length of 4.5 inches. Air was supplied from the Imperial College's new compressed air unit at a constant gauge pressure of 42 (or 60) p.s.i. Before entering the combustion chamber, the air is mixed with commercial propane supplied by three bottles in a water-bath. The flow rates of air and propane were measured, before mixing, by means of two B.S.S. orifice plates (Ref.12).

The flow rates, and hence the composition of the mixture, were controlled by means of coarse and fine needle-valves, the latter being used near the extinction condition of the flame.

Extinction loops were obtained for four central circular discs, and the lean portion of the loops for a similar number of annular baffles. The results are reported in Chapter Four.



FLAME STABILITY APPARATUS.

FIG. 11.

Experimental Procedures.

a. The Electric Analogue:

The procedure for a test on a particular baffle in the electric analogue is as follows:

1. The baffle is placed in position at the entrance of the analogue working section.
2. The blower is started, and sufficient time is allowed to elapse for steady-state conditions to obtain; all the heaters being off at this stage.
3. The pressure differential across the B.S.S. orifice plate is recorded, giving the air-flow rate through the analogue.
4. An equal voltage is supplied to all the heaters in the working section.
5. The readings of all thermocouples are recorded

6. In the lower range of the measurements recorded in 5, a particular value is chosen for T_1 (See Fig.2).
7. All the heaters with associated thermocouple readings below the temperature chosen in 6, are switched off.
8. A higher value is chosen for T_1 , and 7 is repeated.
9. Step 8 is repeated until all the heaters are switched off, signifying the extinction condition.
10. The whole procedure is repeated for other values of air-flow rate.

The above procedure provides the data necessary for the calculation of the Reynolds Number, the loading factor, and the analogue predictions for the combustion efficiency. These parameters are defined for the analogue in the next Chapter.

b. Flame Stability Apparatus:

The aim of the experiments on the flame stability apparatus was to obtain stability loops, i.e. to obtain the flow rates and the fuel-air ratio for the condition at which extinction occurs. The procedure was, therefore, to establish a stable flame, and then to extinguish it by gradually increasing the flow rates of the air, or of the fuel. The availability of fine as well as coarse valves in the flow lines of both air and fuel, made it possible to effect small adjustments, and thus to avoid sudden or big changes in the fuel-air ratio at the critical points near the extinction condition. The flow rates of air and fuel at extinction are calculated from the pressures upstream of, and the pressure differentials across, the B.S.S. orifice plates in the air and fuel flow lines respectively.

CHAPTER THREE.

ANALOGUE PREDICTIONS.

-oOo-

Introduction.

This Chapter contains the results obtained by using the electric analogue models described in Chapter Two. These results are mainly **concerned** with the analogue predictions for the loadings above which extinction of the flame in the simulated chamber will occur. The flame holders investigated here are of the bluff-body kind; the baffle geometry types being:

1. Annular Baffles,
2. Central Circular Discs,
3. Central Hemi-Spherical Baffles,
4. Central Conical Baffles

The parameters used in the presentation of the results are those discussed in Chapter One, and are given in the following Notation.

Notation.

- Rn = the Reynolds Number = $u D \rho / \mu$,
- u = average approach velocity of air
upstream of baffle,
- D = inside diameter of analogue chamber,
- ρ = density of approaching air,
- μ = absolute viscosity of approaching air,
- Lv = loading factor based on chamber volume,
 $= \dot{m} C_p \Delta T / \dot{Q}_m v$,
- $\dot{m}$ = mass flow rate of air into the analogue
chamber,
- C_p = specific heat at constant pressure
of the approaching air,
- $\Delta T = \Delta T_1 (1/0.7)$,
- $\Delta T_1 = T_1 - T_u$,
- T_1 = the point on the abscissa of the step
heat release rate function at which
the heat release rate increases from
zero to its full value, (see Fig.2.),
- T_u = temperature of the approaching air,

$\dot{q}_{m}^{*}$ = the maximum volumetric rate of heat release, from the analogue heaters,

v = volume of the analogue chamber, (in the case of the Lucas analogue model, v is the volume of the chamber downstream of the annular secondary baffle),

L_D = loading factor based on a characteristic length of the chamber,

$$= V \rho C_p \Delta T / \dot{q}_{m}^{*} D,$$

V = average velocity of the air past the baffle,

D = diameter of the analogue chamber,

l = length of chamber,

θ = apex angle of a conical baffle,

ϵ_a = analogue prediction for combustion efficiency,

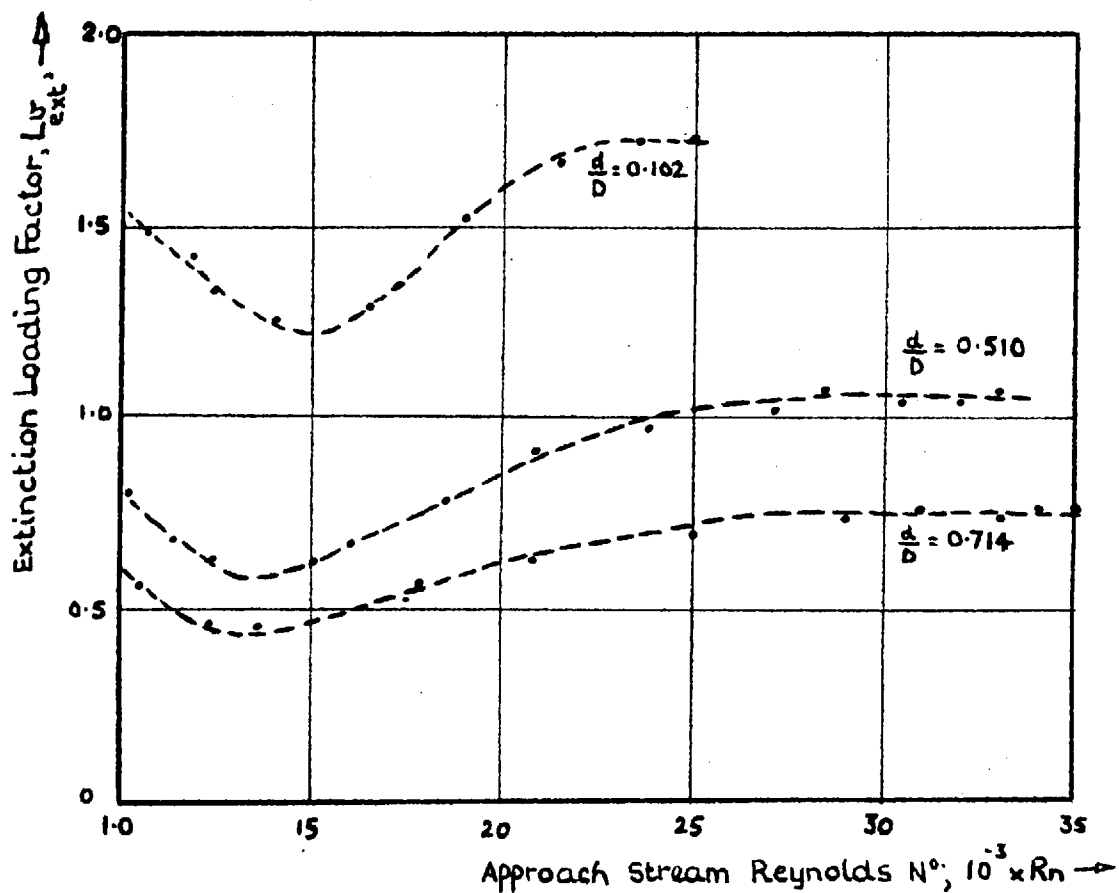
ext. = suffix added to indicate values at the extinction condition.

Predictions of the Lucas Analogue Model - Annular
Secondary Paffles.

The necessity to verify ^{that} the flow rate into the chamber is in the regime where the Reynolds Number ceases to have an important influence on the extinction characteristics, was discussed in Chapter One. The influence of the Reynolds Number on the extinction loading factor predicted by this analogue model was therefore established for each one of the baffles tested. Typical graphs are plotted in Fig.12. This shows the variation of the extinction loading factor based on the chamber volume downstream the secondary baffle, L_v , with the approach stream Reynolds Number, R_n . The parameters L_v and R_n have been defined in the preceding notation. The flow rate was varied from small values to the maximum value attainable by the blower in use. Fig.12 indicates that in the regime of low Reynolds Numbers, the extinction loading factor decreases as

the Reynolds Number is increased. A minimum is reached at a Reynolds Number value of about 1.5 (based on the diameter and an average velocity upstream the secondary baffle). The extinction loading factor then rises with the Reynolds Number, until a regime is reached where this has no more appreciable effect on the extinction loading factor. The last feature is quite clear in the case of baffles with large hole diameters; the maximum flow rates attainable with the smallest hole diameter baffle seem to be on the verge of the regime where the Reynolds Number is of no further influence on the extinction loading factor. It is thought, however, that no appreciable changes in the extinction loading factor would take place if the flow rate were further increased.

From graphs such as those in Fig.12, the extinction loading factor at the maximum Reynolds Numbers are obtained and plotted in Fig.13, against the ratio (d/D) of the baffle hole diameter to the diameter of the analogue chamber.



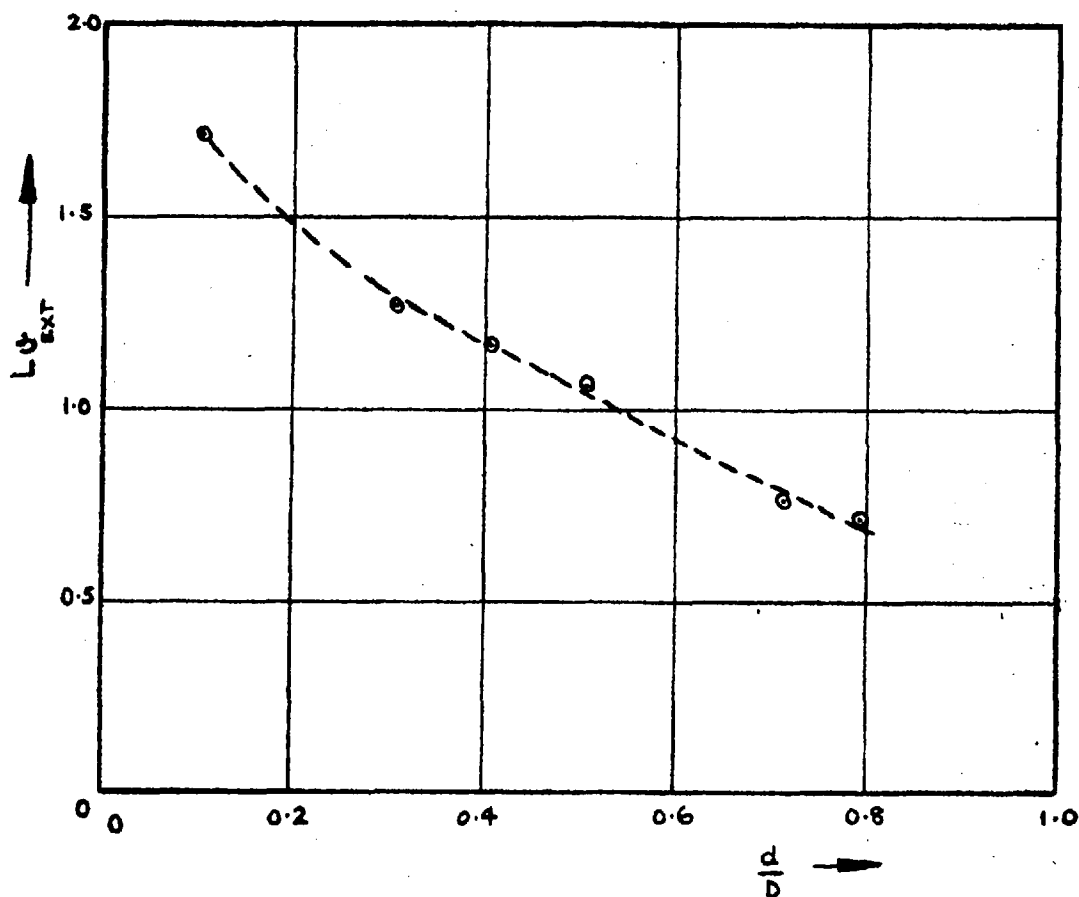
VARIATION OF EXTINCTION LOADING FACTOR
WITH APPROACH STREAM REYNOLDS NUMBER.

ANNULAR SECONDARY
BAFFLES IN LUCAS
ANALOGUE MODEL.

$$Lv_{ext} = \frac{\dot{m}_{ext} Cp \Delta T}{\dot{q}_m U}$$

$$Rn = \frac{4 \dot{m}}{\pi \mu D}$$

Fig. 12.



VARIATION OF EXTINCTION LOADING FACTOR WITH SECONDARY
BAFFLE HOLE DIAMETER UNDER TURBULENT CONDITIONS.

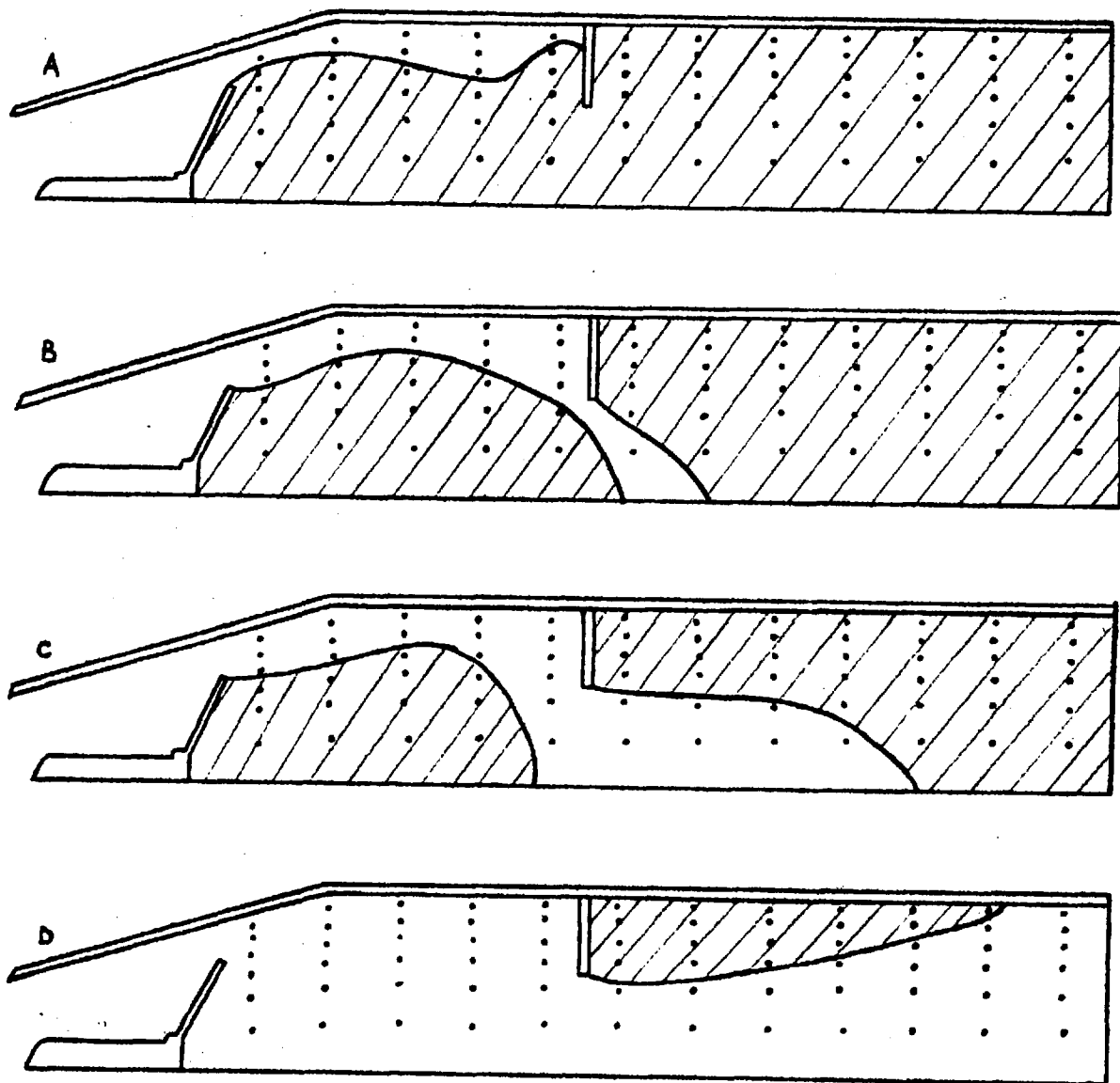
PREDICTIONS FOR ANNULAR
SECONDARY BAFFLES IN THE
LUCAS ANALOGUE MODEL.

$$L^*_{EXT} = \frac{\dot{m}_{EXT} C_p \Delta T}{\dot{q}_m u}$$

FIG. 13.

Fig.13 shows that the extinction loading factor decreases as the ratio d/D is increased.

A further point of interest noted during the experiments on the Lucas analogue model, is the gradual contraction of the regions within the chamber where a stable flame exists as the loading factor is gradually increased. This contraction is illustrated by the series of sketches contained in Fig.14. The loading factor is smallest in Sketch (A), and highest in Sketch (D), where it is indicated that the only region where there is still a persistent flame is in the wake of the secondary annular baffle. This result was noted with all the tests carried out on the Lucas analogue model.



SKETCHES SHOWING THE EFFECT OF INCREASING THE LOADING
FACTOR ON THE SHAPE OF THE FLAME. (SHADED AREAS INDICATE
POSITION OF FLAME).

SOME RESULTS FROM THE
LUCAS ANALOGUE MODEL.

FIG. 14.

LOADING FACTOR INCREASING
FROM A - D.

Predictions of the Simple Axi-Symmetric Analogue
Model.

This model is the circular cylindrical model described in detail in Chapter Two. It is to be remembered that this analogue model differs from the Lucas model in that the latter has a primary baffle and a secondary baffle, while the present model has only the baffles indicated below. Four types of baffle were tested, and the results are given in the next four sections of this Chapter.

1. Annular Baffles.

The experiments carried out on annular baffles are reported below:

a. Combustion Efficiency.

The aforementioned procedure for obtaining the extinction loading factor for a particular baffle, and with a particular flow rate, approaches the extinction condition through a succession of stable " flames ", characterized by a number of

heaters being left on in the analogue working section. Thus, below the loading that causes extinction, a graph may be plotted showing the variation with loading, of the ratio between the number of heaters left on in a stable " flame " and the total number of heaters in the analogue. This ratio is given the symbol ϵ_a . It is to be noted that ϵ_a represents the ratio between the heat input rate at a particular loading, and the maximum possible heat input rate. It is therefore a measure of the efficiency of combustion predicted by the analogue.

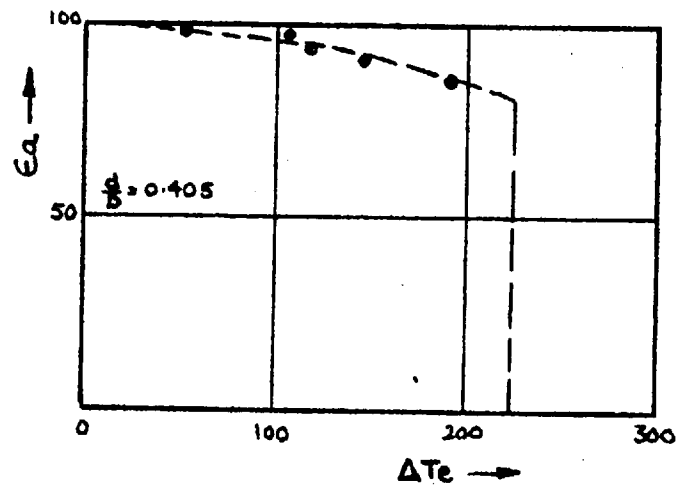
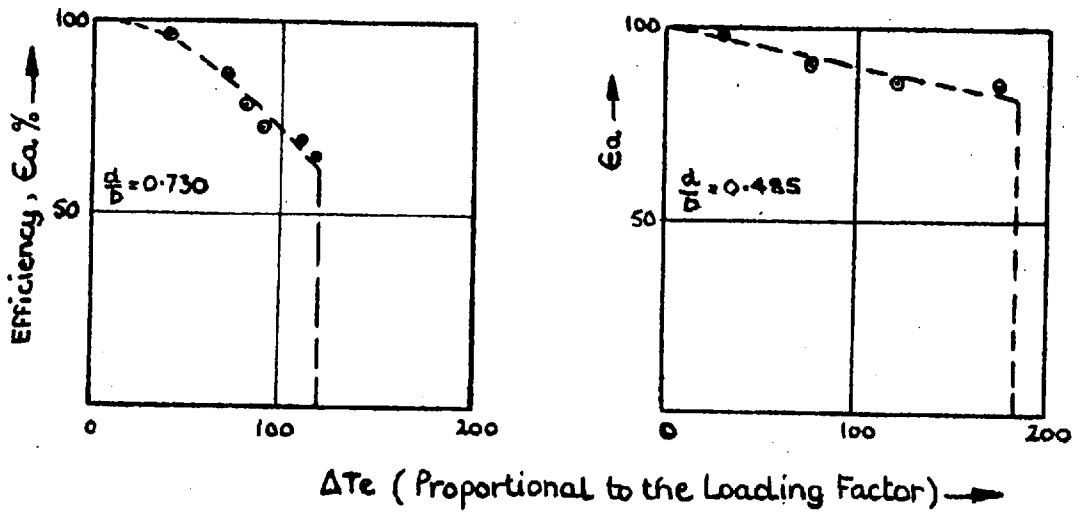
Fig.15 contains samples of the variations of ϵ_a with a parameter proportional to the loading factor based on the chamber volume. The three samples are for three annular baffles with different hole diameters at the maximum attainable flow rates. These graphs show that ϵ_a decreases as the loading factor is increased. When it reaches a value of between 0.65 and 0.85

(depending on the geometry of the baffle)
then any further increase in the loading factor requires all the remaining heaters to be switched off, indicating the sharp drop of the efficiency to zero, i.e. the extinction of the " flame ".
The loading factor at which the combustion efficiency drops to zero is the extinction loading factor.

b. Effect of Heater Voltage.

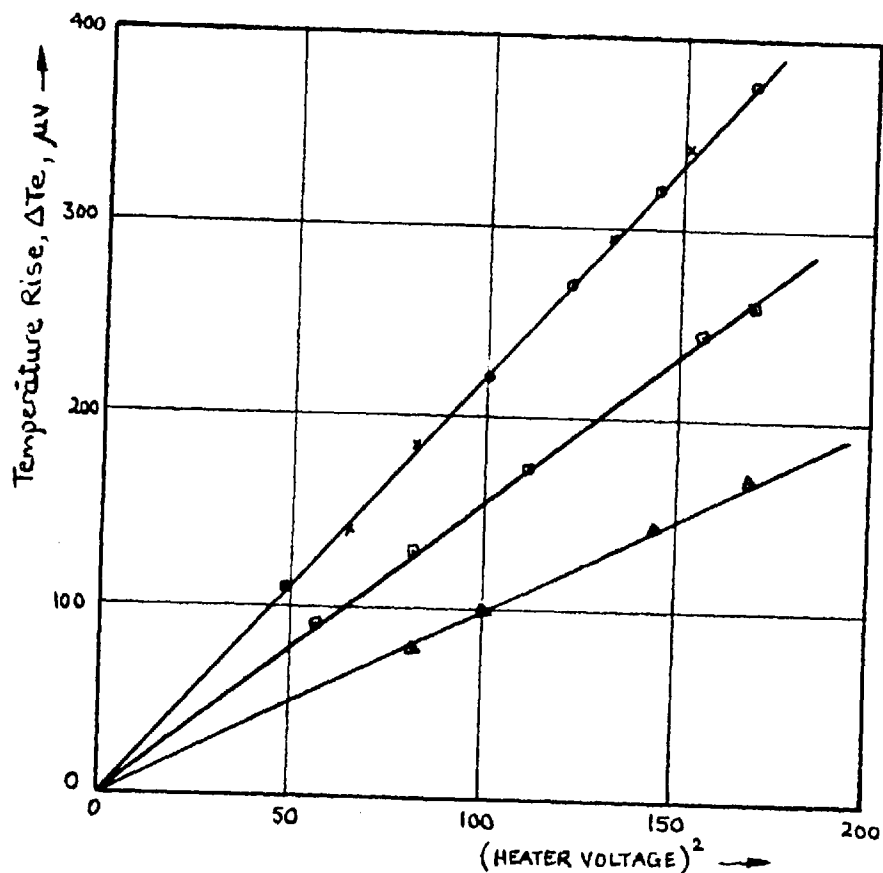
Tests were carried out on the analogue to establish the influence on its predictions due to changes in the voltage applied to the heater elements. For a given baffle and air flow rate, the voltage is changed , and the extinction temperature rise ΔT_e is recorded. Fig.16 shows some typical results plotted as ΔT_e against the square of the heater voltage, the latter being proportional to the heat input rate. The Figure indicates that the extinction temperature rise is proportional to the heat

input rate. This means that the extinction loading factor is independent of the heater voltage in the range covered by these tests. The voltage which was usually chosen in the analogue experiments was 12 volts.



VARIATION OF COMBUSTION EFFICIENCY
WITH LOADING FACTOR. (ANALOGUE PREDICTIONS).

FIG. 15.



VARIATION OF EXTINCTION TEMPERATURE RISE WITH THE
SQUARE OF THE VOLTAGE APPLIED TO THE ANALOGUE HEATERS.

RESULTS OF TESTS ON SOME
ANNULAR BAFFLES IN THE PRESENT
ANALOGUE MODEL. FLOW RATES
BEING HELD CONSTANT AT THE
VALUES IN ADJACENT KEY.

KEY	m lb/sec.	d inches.
○	.464	2.0
Δ	.775	3.5
◻	.522	3.5
x	.376	3.5

FIG. 16.

c. The Effect of Reynolds Number.

The effect of the Reynolds Number on the extinction loading factor for each one of the annular baffles tested has been established. Some of the results are plotted in Fig.17, where the extinction loading factor based on the chamber volume, is plotted against the Reynolds Number. The loading factor and the Reynolds Number are given by:

$$Lv = \dot{m} C_p \Delta T / \dot{q}''' v, \text{ and}$$

$$Re = u \rho D / \mu,$$

the symbols being as defined in the notation.

Fig.17 shows that the maximum attainable mass flow rates are sufficient in all cases to produce flow regimes in which the loading factors at extinction are independent of the Reynolds Number. The baffles whose hole diameter is not larger than 3.5 inches, are seen to show an increase of the extinction loading factor with the Reynolds Number,

the former becoming ultimately constant at the highest flow rates. The graphs for larger hole diameters show only the constant part of the curves, and may be presumed to have similar trends at the lower values of flow rates.

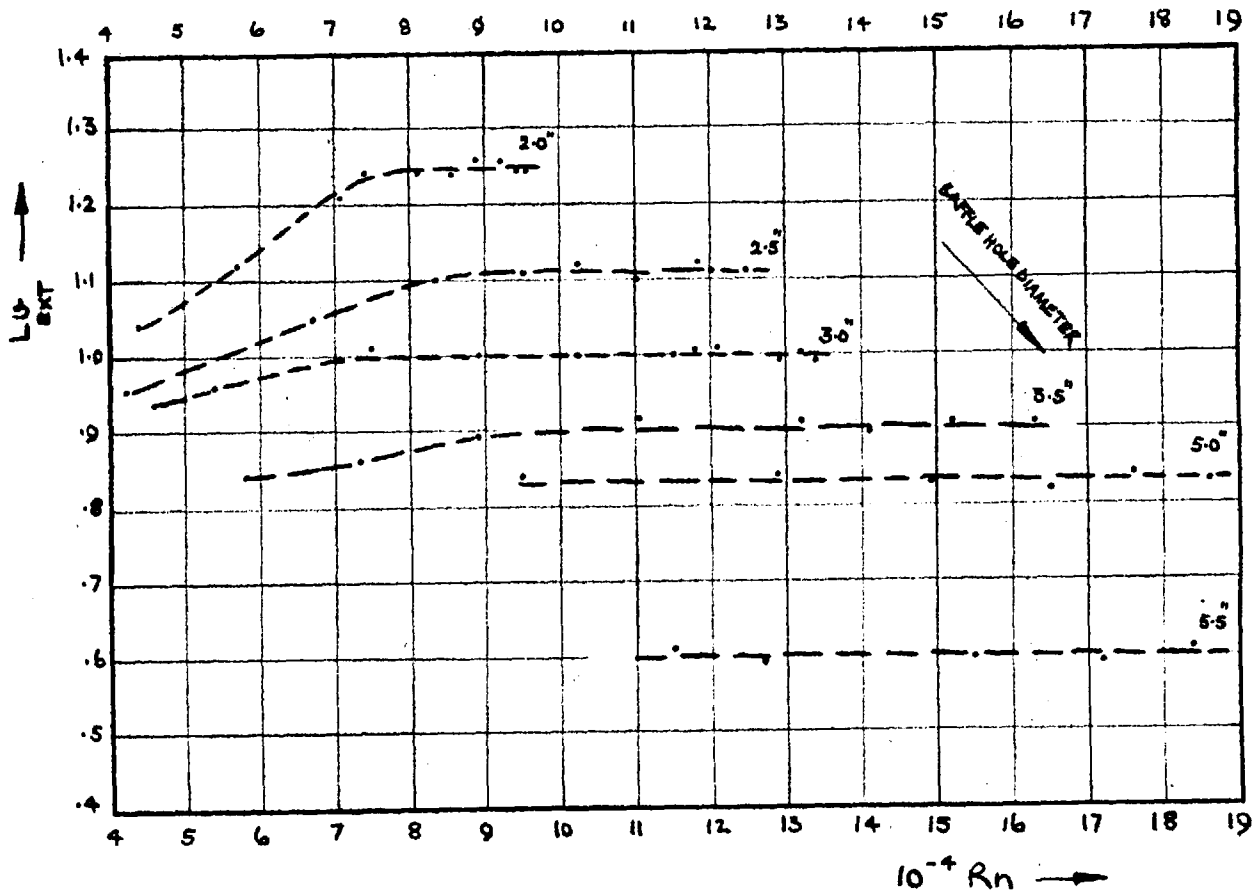
d. The Baffle Hole Diameter Effect.

This effect is considered at the highest attainable Reynolds Numbers. Fig.18 indicates the variation of the extinction loading factor at the highest Reynolds Number, with the ratio of the baffle hole diameter to the diameter of the chamber. The values of $L_v \text{ ext.}$ were obtained for each baffle from the graphs of Fig.17.

It is seen from Fig.18 that, as the baffle hole diameter is increased, the extinction loading factor decreases, except in the region where the ratio d/D is roughly between 0.60 and 0.75. In this region, the loading factor does not appreciably change with the baffle hole diameter;

while beyond it, the drop in $L_{ext.}$ is rather sharp. The results of Fig.18 will now be compared with the results predicted by the Lucas analogue model for annular baffles. This comparison is suggested by the fact that the geometry of the primary baffle in the Lucas model is fixed, and the behaviour noted during extinction tests on the Lucas model, viz., the fact that the last region where the flame persists is in the wake of the secondary annular baffle, and not between the primary and secondary baffles, (see Fig.14).

For this comparison, Fig.19 is presented containing the predictions of both analogue models for the variation of the extinction loading factor with the hole diameter of the annular baffles. The two sets of results are seen to be so close to each other, that one is justified in considering them to form a single curve with some experimental scatter. The part of the curve in Fig.18 where $L_{ext.}$ is indicated to be constant, lies in the region



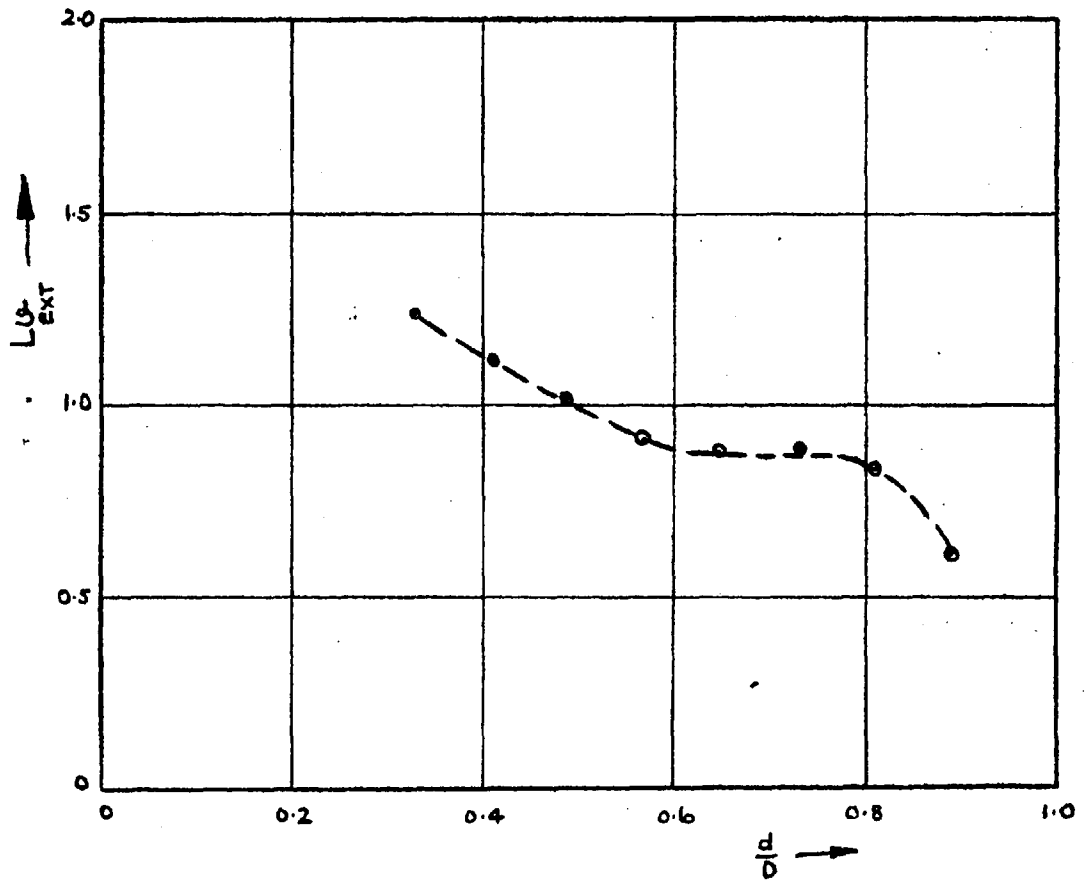
VARIATION OF EXTINCTION LOADING FACTOR WITH APPROACH
STREAM REYNOLDS NUMBER

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL ~ ANNULAR
BAFFLES.

$$Lg = \frac{\dot{m} c_p \Delta T}{\dot{q}_{m \psi}}$$

$$Rn = \frac{4 \dot{m}}{\pi \mu D}$$

FIG. 17.



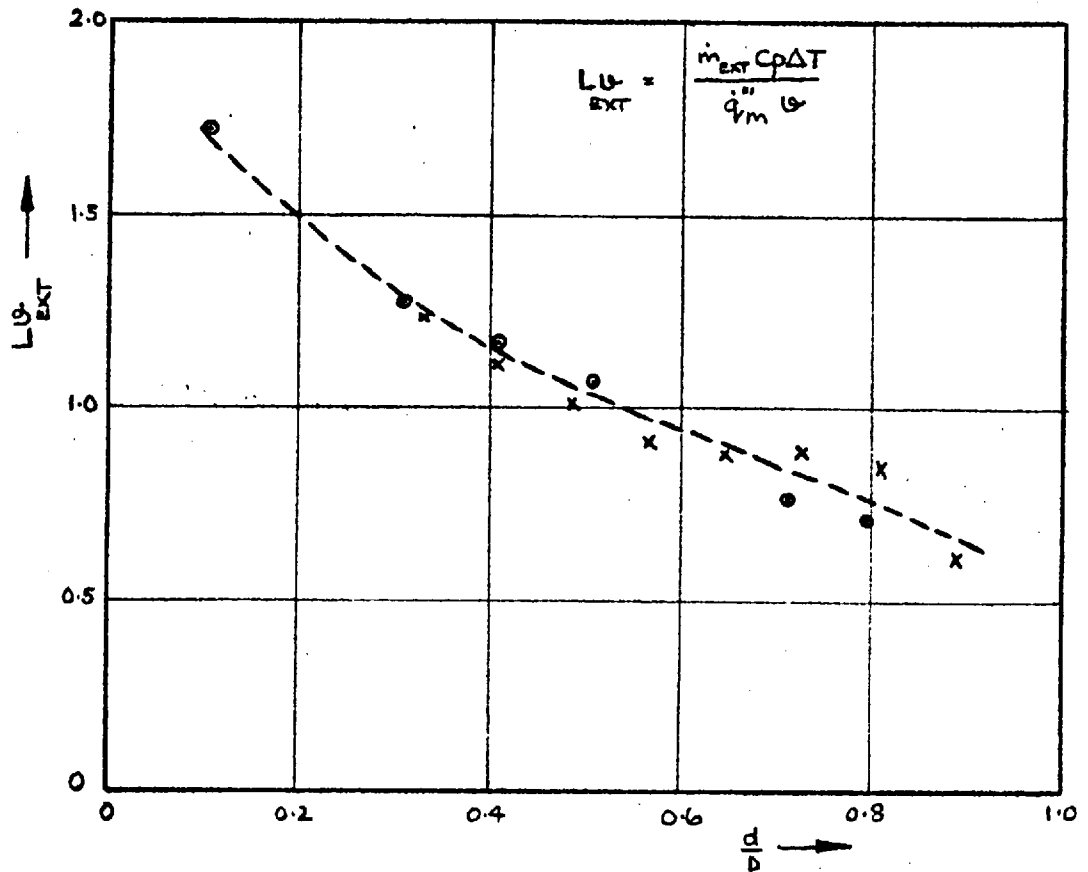
ANALOGUE PREDICTIONS ~ ANNULAR BAFFLES.

($L_{ext} \sim \frac{d}{D}$)

SOME RESULTS FROM THE
PRESENT ANALOGUE MODEL.

$$L\theta = \dot{m} c_p \Delta T / \dot{q}_m \theta.$$

FIG. 18.



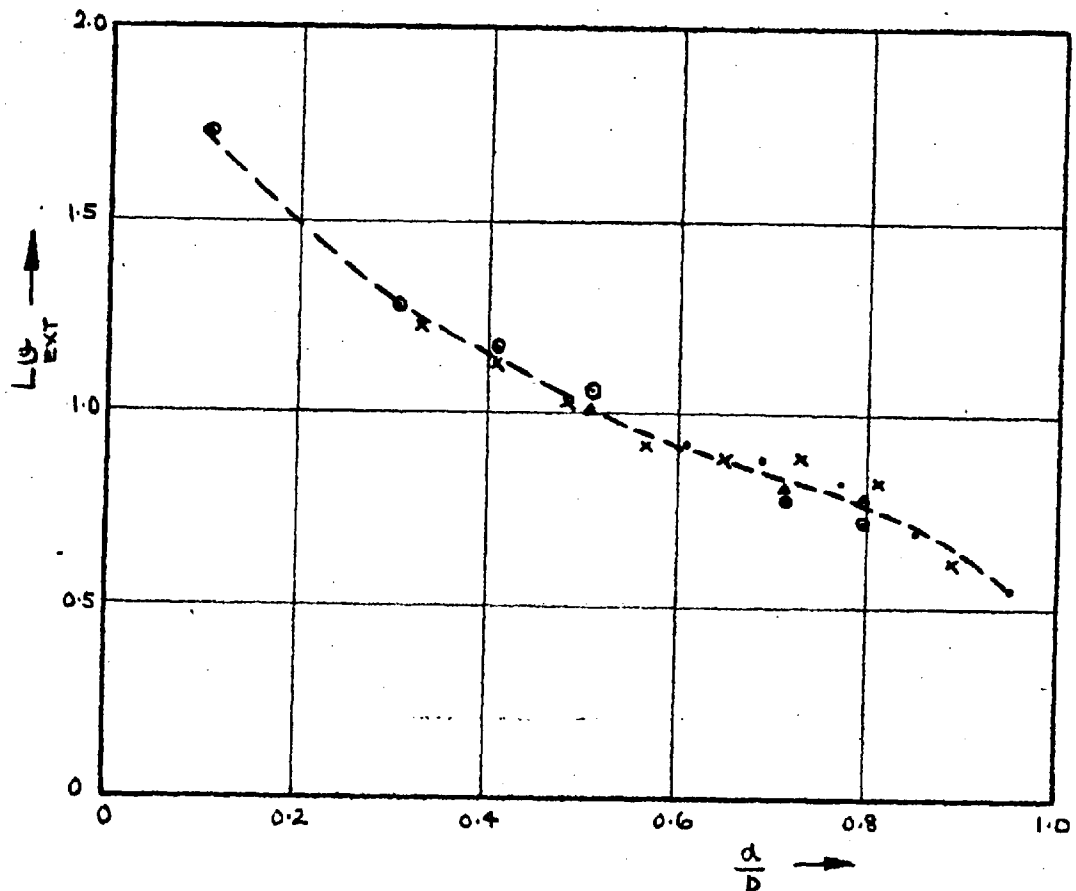
VARIATION OF EXTINCTION LOADING FACTOR WITH
ANNULAR BAFFLE HOLE DIAMETER.

- PREDICTIONS OF LUCAS MODEL
- x PREDICTIONS OF PRESENT MODEL.

FIG.19.

where the scatter is largest in Fig.19. It was therefore considered necessary to obtain more predictions from the present analogue model for some intermediate values of baffle hole diameter. These were obtained at the highest attainable flow rates, which have already been proved to produce Reynolds Numbers high enough to make the extinction loading factor independent of them. The results for the intermediate points are superimposed in Fig.20 on the results previously obtained from the two analogue models. This Figure shows also the results obtained from the present analogue model when it has been fitted with a primary and a secondary baffle, in a manner similar to that of the Lucas analogue model.

Fig.20 confirms the Lucas analogue predictions, and shows that the two models predict similar results as far as the effect of the change in the hole diameter of the annular baffle is concerned. The trend of the analogue predictions for the variation of the extinction loading factor with



ANALOGUE PREDICTIONS ~ ANNULAR BAFFLES
($L_{ext} \sim \frac{d}{D}$)

- Results of Fig. 13.
- x Results of Fig. 18.
- Further points on the present model.
- ▲ Results by present model fitted with primary and secondary baffles.

$$Lg_{EXT} = \frac{\dot{m}_{EXT} Cp \Delta T}{\dot{q}_m Lg}$$

FIG. 20.

the d/D ratio, is clearly defined in Fig.20.

It is this curve which will be used in Chapter Six for comparison with the flame stability results.

2. Central Circular Disc Baffles.

In the presentation of the analogue predictions for central discs, the Reynolds Number and the loading factor are defined as follows:

The Reynolds Number:

This is taken as the approach stream Reynolds Number. It is based on the average velocity and conditions of the air approaching the analogue, together with the diameter of the analogue chamber, i.e.,

$$R_n = u \ D / \nu \dots\dots\dots(21).$$

where:

u = average velocity of air upstream of
baffle,

D = diameter of analogue chamber,

ν = kinematic viscosity of air at inlet
temperature and pressure.

The Loading Factor:

The loading factor is defined in terms of a characteristic velocity and a linear dimension. In the correlation of data for flame stabilization in the wake of bluff-body flameholders, the velocity past the baffle is usually employed. Thus the average velocity of the air past the disc is used here in the definition of the loading factor. The use of a velocity demands the inclusion of a linear dimension in the denominator of the loading factor. The diameter of the analogue chamber is taken for such a linear dimension. The loading factor is therefore defined by:

$$L = V \rho C_p \Delta T / \dot{q}_m D \dots \dots \dots (22).$$

where:

- V = average velocity of air past the baffle,
- ρ = density of air approaching analogue,
- Other symbols being as indicated in notation.

The tests carried out on central disc baffles are as follows:

a. Heater Voltage Effect.

The effect of the voltage applied to the analogue heaters on the extinction loading factor predicted by the analogue for discs with various diameters and at various flow rates, was found to be similar to the effect noted in the case of annular baffles. Within the voltage range of interest in the analogue experiments, the voltage was again found to have no noticeable effect on the analogue predictions.

b. The Reynolds Number Effect.

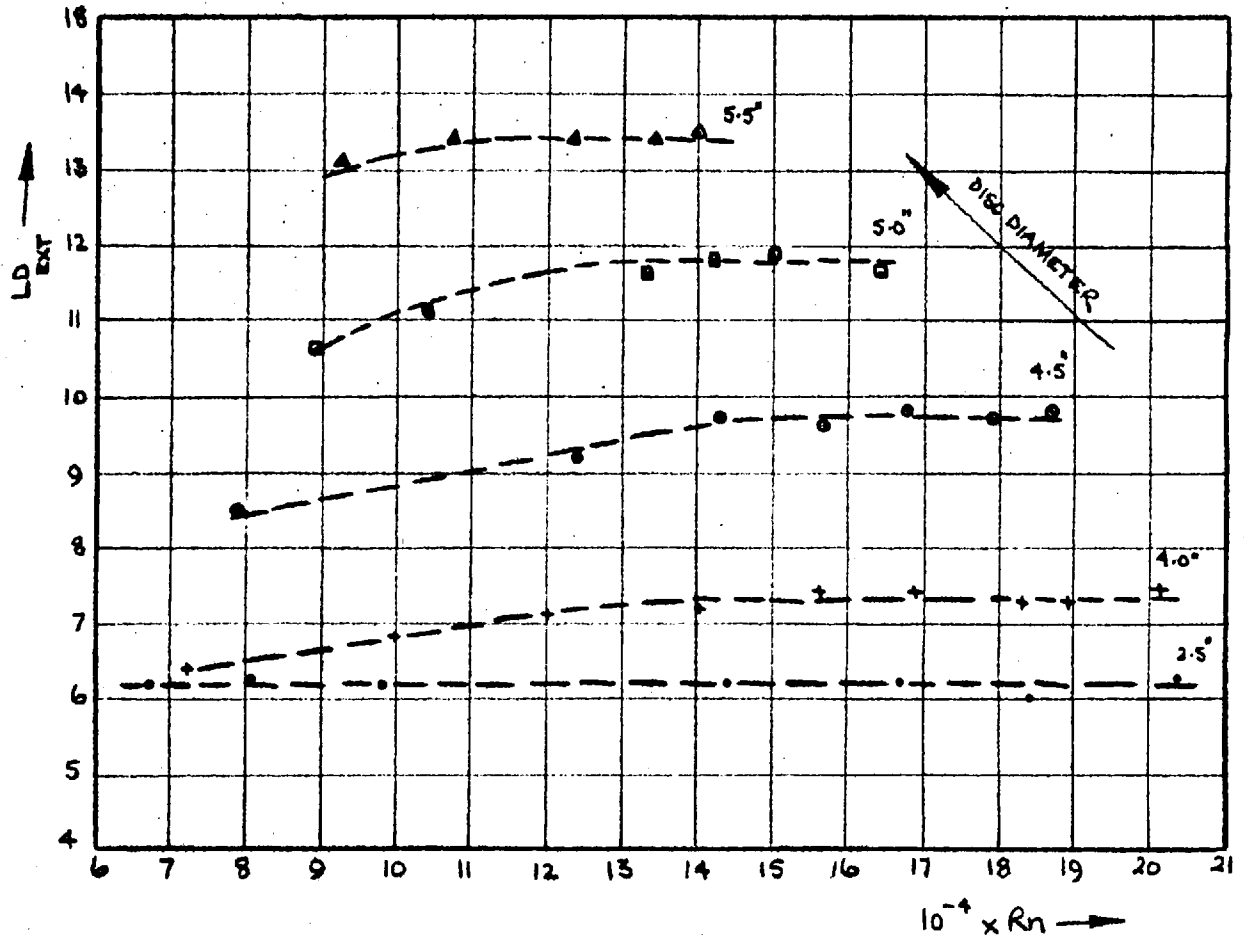
The variation of the extinction loading factor predicted by the analogue with the approach stream Reynolds Number has been established for discs with diameters of 2.0, 2.5, 3.0, 4.0, 4.5, 5.0, and 5.5 inches. It is seen in Fig.21 that the extinction loading factor increases as

the Reynolds Number is increased, until a flow rate is reached where the extinction loading factor remains unchanged. The graphs indicate a stronger Reynolds Number effect at low flow regimes with the larger discs than they do with the smaller discs.

The shape of the graphs in Fig.21 makes it reasonable to conclude that the maximum flow rates attainable by the blower in use are high enough, with all these diameters within the range covered by Fig.21, to make negligible the Reynolds Number effects on the extinction loading factor.

c. The Disc Diameter Effect.

Fig.22 shows the analogue predictions for the variation of the extinction loading factor with the disc diameter under turbulent conditions. The loading factors are obtained from the graphs of Fig.21 at the maximum Reynolds Numbers, and from tests at the highest attainable flow rates on discs



VARIATION OF EXTINCTION LOADING FACTOR WITH APPROACH
STREAM REYNOLDS NUMBER

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL ~ CENTRAL
CIRCULAR DISCS.

$$LD = \frac{V \beta u_c p \Delta T}{\dot{q}_m'' D}$$

$$Rn = \frac{4 \dot{m}}{\pi \mu D}$$

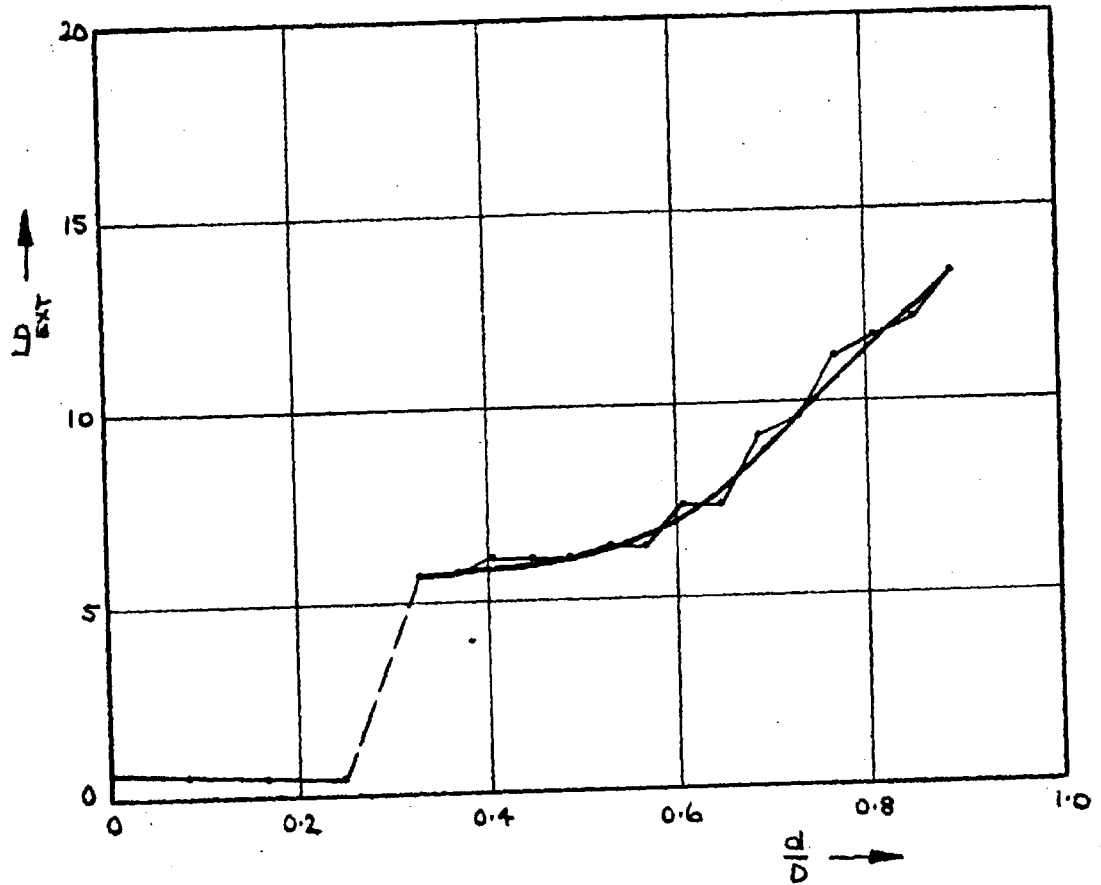
FIG. 21

with intermediate diameters within the range covered in Fig.21.

When the disc diameter is made smaller than $1\frac{3}{4}$ inches, the extinction loading factor (based on the chamber diameter and the velocity past the disc) predicted by the analogue, suddenly drops almost to zero from a value of more than 5.0 in the case of the disc with a 2.0 inches diameter. This drop is due to the distribution of the heaters in the elements of the analogue working section; the heater nearest the axis having a diameter of about $1\frac{3}{4}$ inches. Thus, with discs having diameters below this size, the analogue has no means of simulating the heat release in the recirculation zone behind the baffles. The very small values of $L_{ext.}$ for discs with diameters smaller than $1\frac{3}{4}$ inches, must actually be regarded as an experimental error (a negligible one) due to the fact that the thermocouples are actually downstream of the heaters, which are all switched on at the beginning of the tests.

It can therefore be concluded that the analogue results for discs with a diameter of less than 2.0 inches, may be ignored. These results are joined by a broken line in Fig.22. For discs having larger diameters, the results predicted by the analogue (indicated in Fig.22 by an unbroken line) show that the loading factor at extinction increases with the disc diameter. The curve is rather shallow at its left-hand end, and becomes steeper for larger d/D ratios.

The unbroken line of Fig.22 is the curve which will be used in Chapter Six for comparison with the flame stability results.



VARIATION OF EXTINCTION LOADING FACTOR
WITH DISC DIAMETER

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL ~ CENTRAL
CIRCULAR DISCS.

$$LD = V S u C_p \Delta T / \dot{q}_m''' D$$

FIG. 22.

3. Hemi-Spherical Baffles.

The Reynolds Number and the loading factor are defined for the hemi-spherical baffles in a manner similar to that used in the case of the central circular discs. The results are also presented in a similar manner. Fig.23 shows the dependence of L_{ext} . upon the Reynolds Number.

The extinction loading factor increases with the Reynolds Number, until the latter reaches values where the first becomes independent of it. This trend is seen to be clearly established in Fig. 23 for the hemi-spherical baffles with d/D ratios of 0.385, 0.507, 0.618, and 0.730.

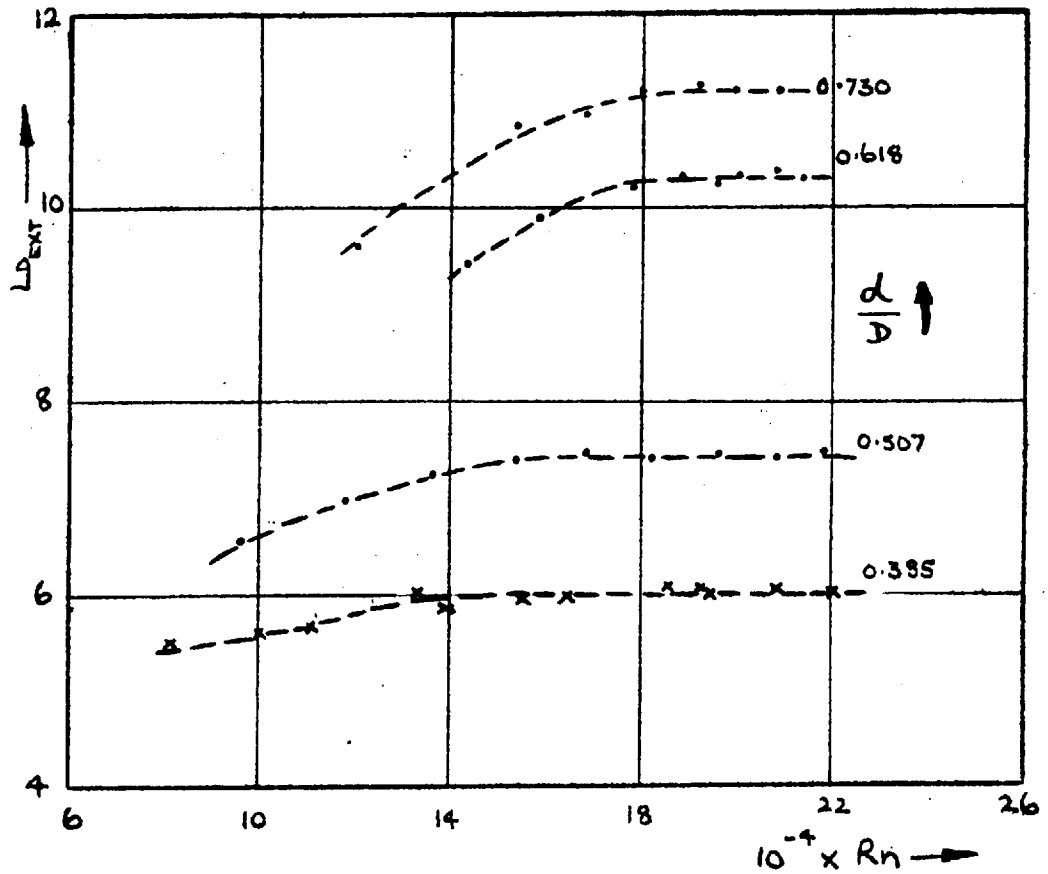
Three more hemi-spherical baffles with d/D ratios between 0.385 and 0.730, were tested at the highest flow rates attainable by the blower. Inspection of Fig.23 suggests that these flow rates are high enough to justify the assumption that the extinction loading factors predicted by the analogue for these three baffles are in the regime of negligible Reynolds Number effect, and could thus be

considered together with the results of Fig.23.

The analogue predictions for the variation of the extinction loading factor with the hemi-spherical baffle diameter is shown in Fig.24. This variation is here considered at the highest Reynolds Numbers. The unbroken lines in Fig.24 connect the analogue prediction points for baffles with d/D ratios in the range of 0.385 to 0.730.

When a baffle of 1.5 inches diameter was tested, the loading factor at extinction was seen to have dropped almost to zero. The explanation for this was given in the presentation of the analogue predictions for central disc baffles.

The curve drawn through the points joined by unbroken lines in Fig.24, indicates the trend that the extinction loading factor for flames in the wake of hemi-spherical baffles, increases with the baffle diameter.



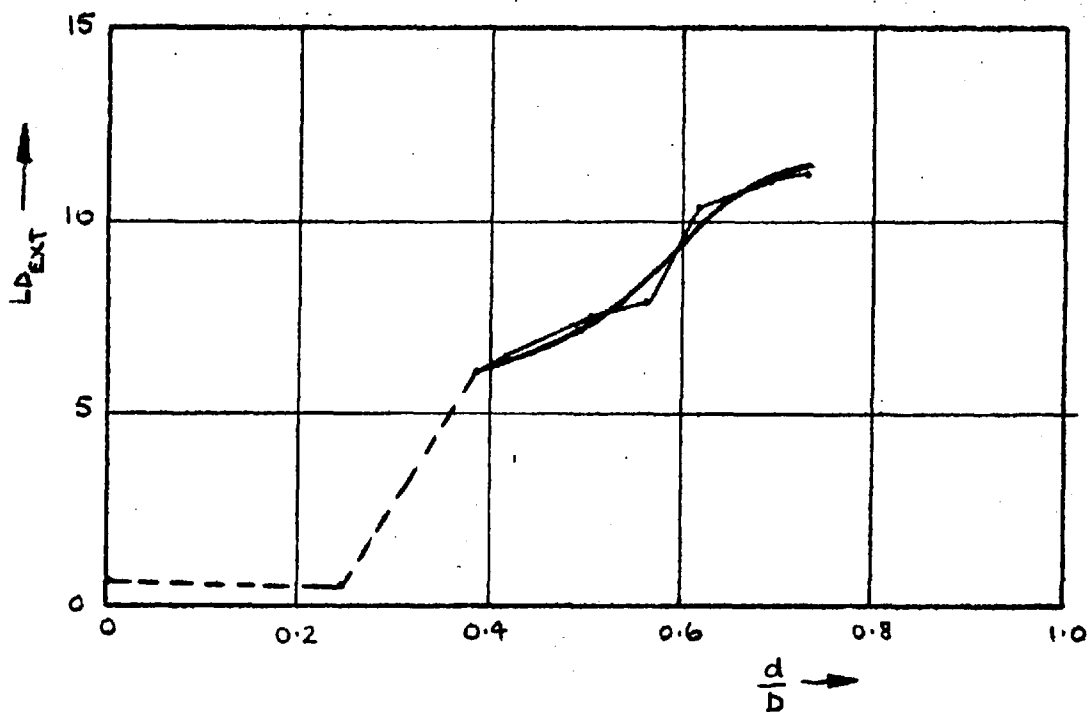
VARIATION OF EXTINCTION LOADING FACTOR WITH
APPROACH STREAM REYNOLDS NUMBER

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL ~ HEMI-SPHERICAL
BAFFLES.

$$LD = VS \frac{C_p \Delta T}{\dot{q}_m} D$$

$$Rn = \frac{4 \dot{m}}{\pi \mu D}$$

FIG. 23.



VARIATION OF EXTINCTION LOADING FACTOR
WITH BAFFLE DIAMETER

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL - HEMI-SPHERICAL
BAFFLES.

$$LD = \frac{V S_u C_p \Delta T}{\dot{q}_m D}$$

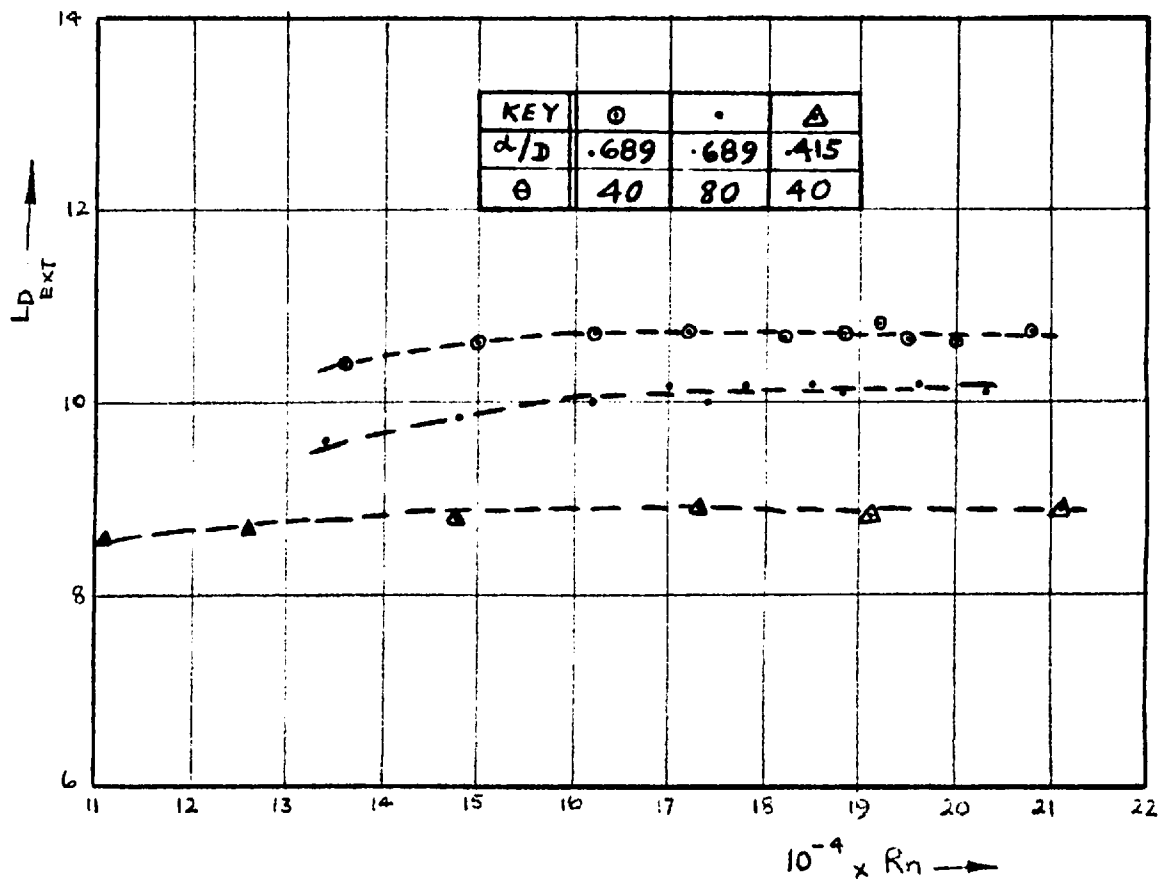
FIG. 24.

4. Conical Baffles.

a. Reynolds Number Effect:

It has already been established that in the cases of the central discs and hemispherical baffles, the maximum flow rates of air that could be passed through the analogue by the blower in use were sufficiently high to produce a Reynolds Number at which the extinction loading factor is independent of it. It is therefore reasonable to assume that the same is true in the case of conical baffles. Thus, from all the conical baffles tested in this series of experiments, only three were fully tested from the point of view of the Reynolds Number effect. The results, plotted in Fig.25, justify the aforementioned assumption, and the test procedure adopted in the present series.

This procedure is to find the extinction loading factor for each baffle at a small number of



VARIATION OF EXTINCTION LOADING FACTOR WITH
APPROACH STREAM REYNOLDS NUMBER.

PREDICTIONS OF THE
PRESENT ANALOGUE
MODEL ~ CONICAL
BAFFLES.

$$LD = V S u C_p \Delta T / \dot{q}_m D$$

$$Rn = 4 \dot{m} / \pi \mu D$$

FIG. 25.

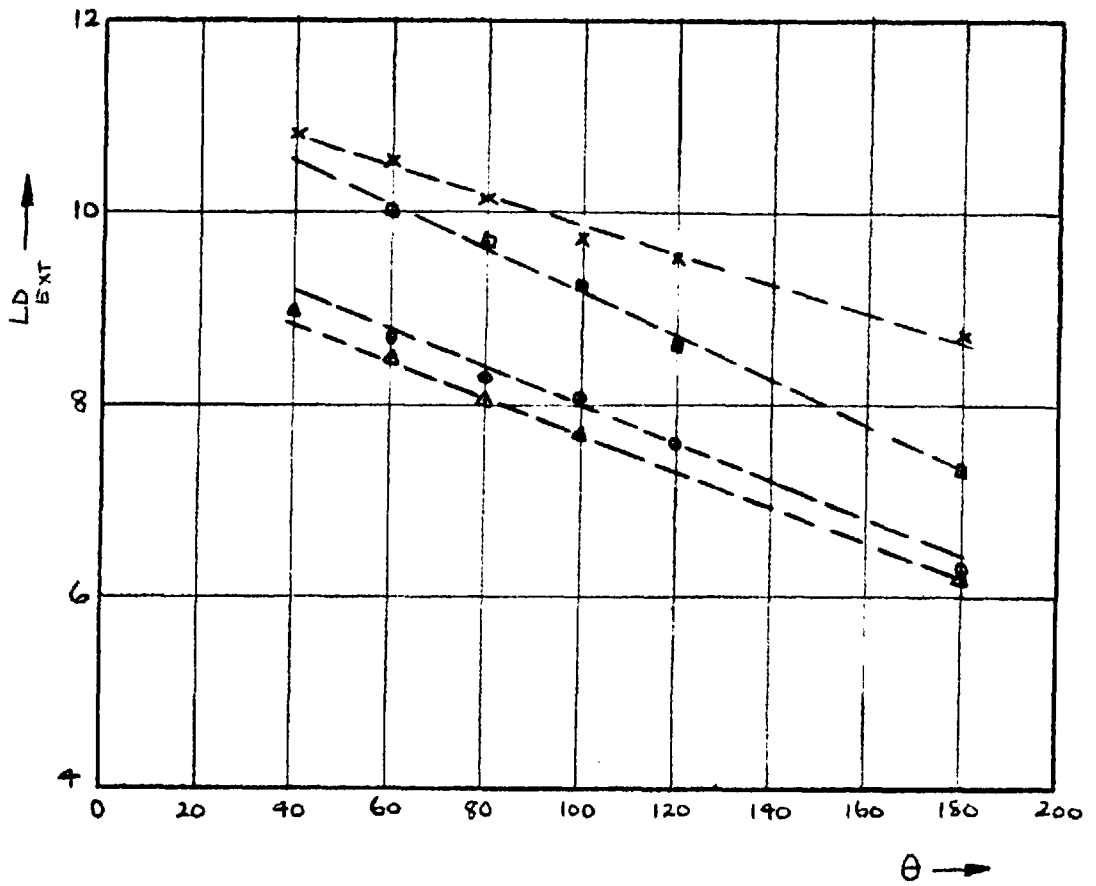
flow rates on and near the upper limits of the Reynolds Number, and to take the average as the analogue prediction of $I_{ext.}$ for the particular baffle. The differences in these measured values of $I_{ext.}$ are as small as those noted at the higher flow regimes in Figs.21 and 23 for central discs and hemi-spherical baffles respectively.

b. Cone Angle Effect:

The above averaged measurements of the extinction loading factor are plotted in Fig.26 against the cone apex angle. The loading factor is defined as for central discs and hemi-spherical baffles, namely,

$$L = V \rho_u C_p \Delta T / \dot{q}_m'' D,$$

with the symbols having the same significance as before. The conical baffles tested had apex angles of 40, 60, 80, 100, and 120 degrees, but Fig.26 also contains results for cone angles of 180 degrees. These last points are the analogue predictions for central discs taken from Fig.22.



VARIATION OF EXTINCTION LOADING FACTOR
WITH CONE APEX ANGLE.

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL ~ CENTRAL
CONICAL BAFFLES.

$$LD = V S u C_p \Delta T / \dot{q}_m D$$

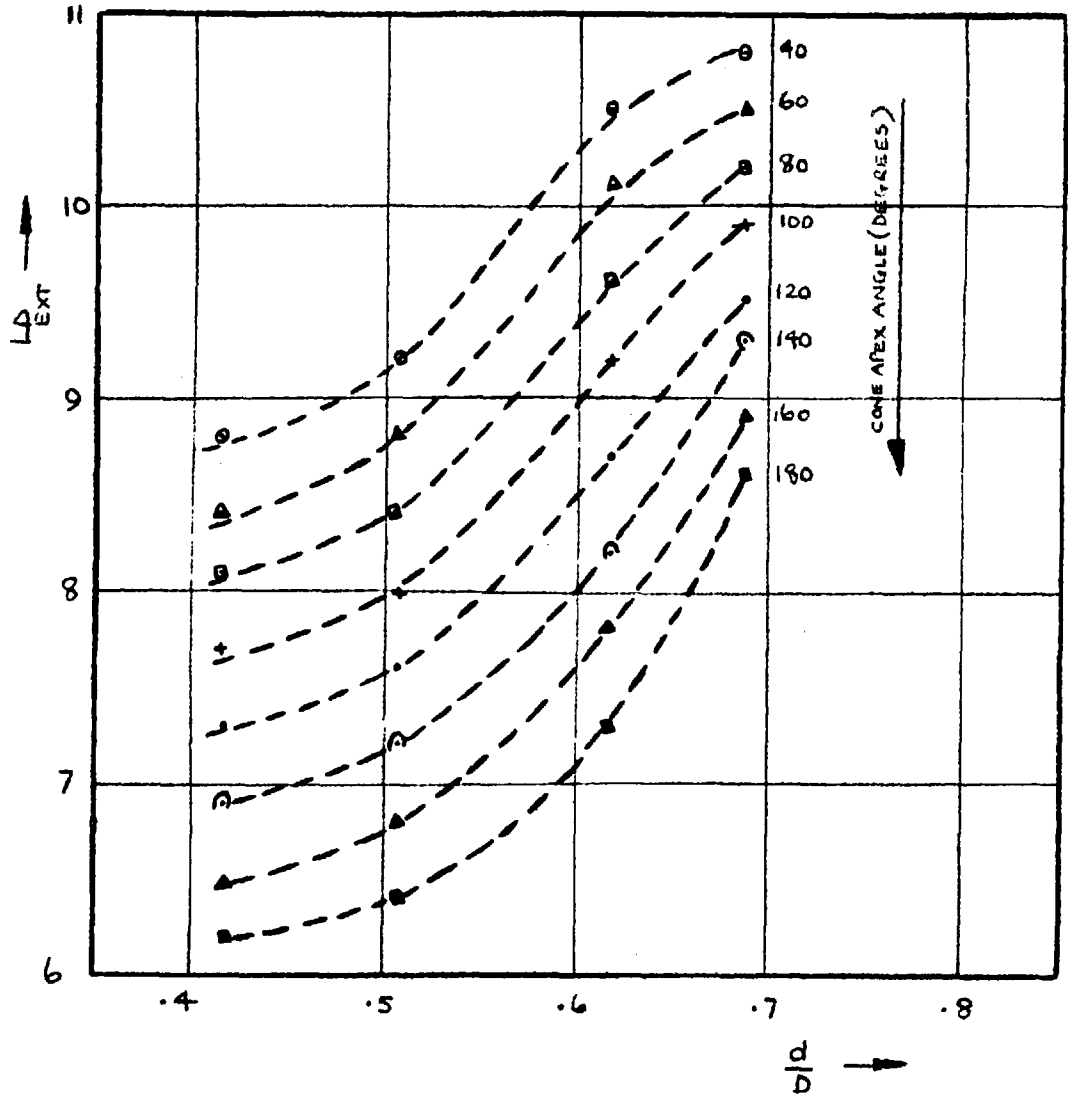
KEY	x	□	○	△
d/D	.689	.618	.507	.415

FIG. 26.

It is seen from Fig.26 that the analogue predicts a decrease in the extinction loading factor as the cone angle is increased. A straight line correlates the analogue predictions to a good approximation for the various base diameters tested.

c. Effect of Cone Base Diameters:

The base diameter effect on the extinction loading factor is indicated in Fig.27, which is based on the results of Fig.26. The loading factor at extinction is seen to increase with the cone base diameter. Within the range covered by Fig.27, there seems to be a tendency for the curves to be less steep at the higher values of d/D when the apex angle is small.



VARIATION OF EXTINCTION LOADING FACTOR
WITH CONICAL BAFFLE BASE DIAMETER.

PREDICTIONS BY
THE PRESENT
ANALOGUE MODEL.

$$LD = V S u C_p \Delta T / \dot{q}_m''' D$$

FIG. 27.

The Flow Criterion, Kf.

In addition to information about the performance of a given combustion system from the reaction point of view, which is characterised by the loading factor, the designer of the system also requires information about the pressure drops encountered when a certain mass of working fluid passes through the combustion chamber. A relation is therefore needed to connect the pressure drop Δp , with the mass flow rate $\dot{m}$, and the other factors that will influence it.

Spalding (Ref.5) used the theory of dimensions to obtain such a relation. The factors included in the analysis were:

1. The air inlet temperature, T_1 ,
2. The outlet gas temperature, T_2 ,
3. Atypical dimension of the chamber, X ,
4. The initial air density, ρ_1 ,
5. The initial sonic velocity, a_1 ,
6. The initial air viscosity, μ_1 .

The relation obtained by dimensional analysis is given by Equation 23 below:

$$\frac{\dot{m}^2}{\Delta p \, g_o \, x^4} = f \left\{ \frac{\dot{m}}{x \, \mu_1}, \frac{T_2 - T_1}{T_1}, \frac{\dot{m}}{g_1 \, x^2 \, \alpha_1} \right\} \dots\dots\dots(23)$$

where:

g_o = the constant in Newton's Second Law
of Motion (32.2 lbm. ft. / lbf. sec.).

Inspection of the three terms in the right-hand side of Equation 23, reveals that the first term has the nature of a Reynolds Number; but since with fully turbulent flows, the influence of this parameter on the flow pattern is negligible, this first term may be ignored. When discussing the second term in the R.H.S. of Equation 23, Spalding suggested the relegation of the group $(T_2 - T_1) / T_1$ to a second level of interest, since the actual range of variation of this group is small in practical combustion chambers, and its effect is similar for nearly all designs.

Thus the ratio of the temperature rise to the initial temperature is also neglected. The third group in the R.H.S. of Equation 23 has the same significance as the Mach Number: it represents the influence of compressibility effects. In the systems where this effect is small because of the low Mach Number, this group may also be ignored.

Equation 23 thus becomes:

$$K_f = \frac{\dot{m}^2}{\Delta p \rho_1 g_0 x^4} \text{-----} (24)$$

where:

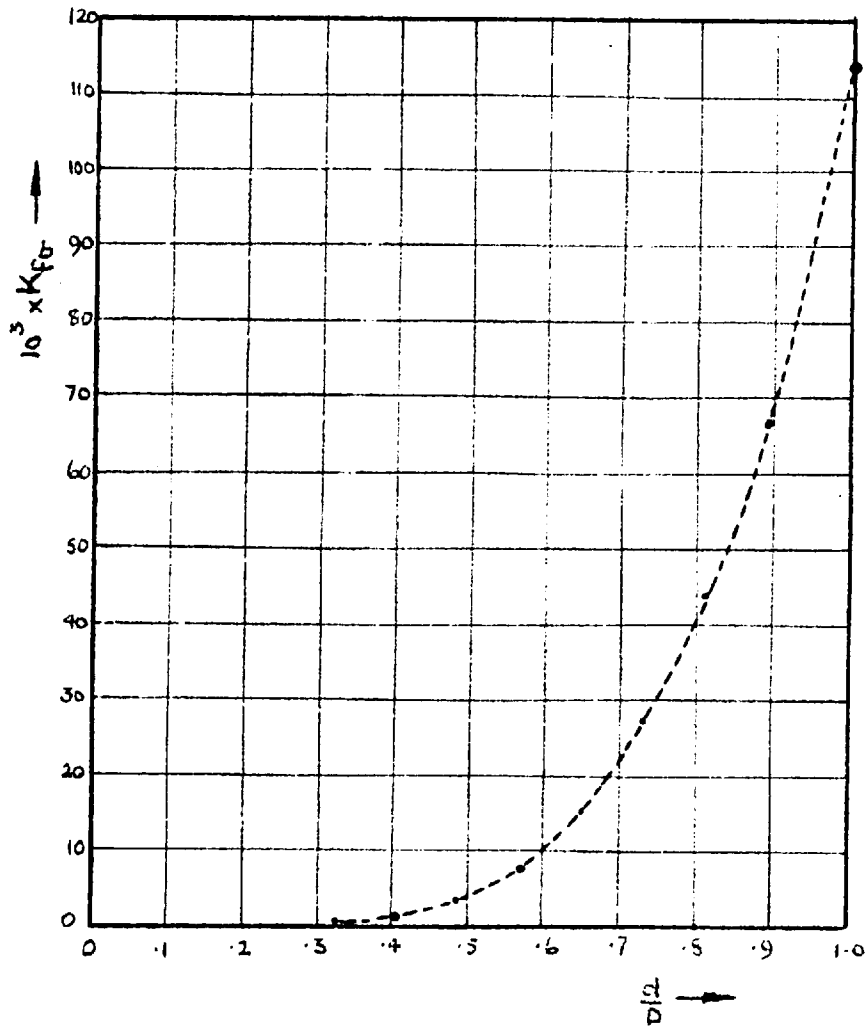
K_f = the flow criterion, the term given to the group in the L.H.S. of Equation 23, and is approximately constant for a given chamber design.

It is, of course, possible to choose other factors to characterize the chamber dimensions. For example, the volume, v , may be chosen. This changes the definition of the flow criterion into:

$$K_{fv} = \frac{\dot{m}^2}{\Delta p \rho_1 g_0 v^{4/3}} \text{-----} (25)$$

The suffix (v), added to the flow criterion indicates that the volume is the characteristic geometrical factor. This form of the flow criterion is being used below to present measurements made on the analogue model with annular baffles. The total pressure drop suffered by the air in passing through the analogue chamber was measured for various flow rates. The fact that the heaters were either on or off was seen to have no appreciable influence on the measured pressure drops.

The variation of the flow criterion with the approach stream Reynolds Number, is negligible for a given chamber configuration. The averaged K_{f_0} values are plotted against the baffle hole diameter in Fig. 27a, which shows that the flow criterion improves as the baffle hole diameter is increased.



VARIATION OF THE FLOW CRITERION
WITH THE BAFFLE HOLE DIAMETER

MEASUREMENTS ON
THE PRESENT
ANALOGUE MODEL.

$$K_{F0} = m^2 / \Delta \phi S_1 g_0 \alpha^{2/3}$$

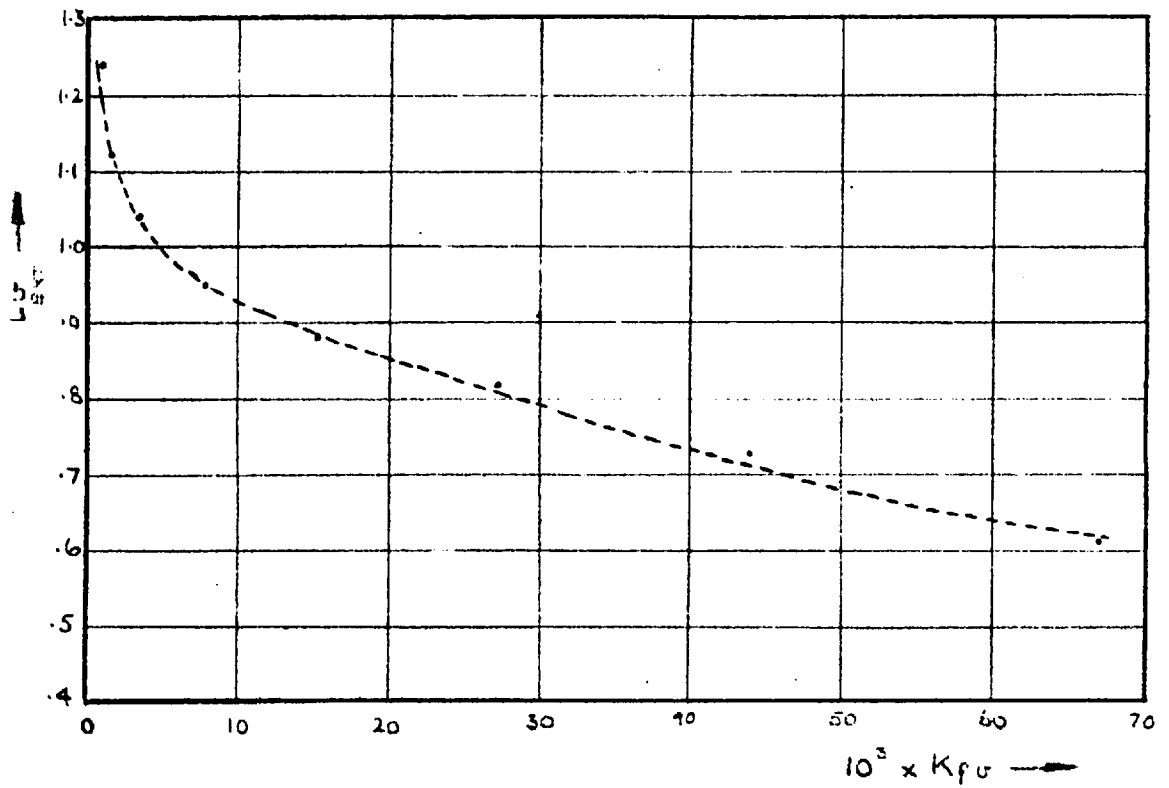
FIG. 27 (a)

The Extinction Loading Factor and the Flow Criterion.

As previously stated, the designer of a combustion system needs to view the performance from both the reaction and the flow viewpoints. He requires high values for both the extinction loading factor and the flow criterion, but finds that these requirements are not compatible, as illustrated by Fig. 20 and Fig. 26. A more useful way to consider the results would be a plot of:

$$L_{ext.} \sim K_{fo} .$$

According to the requirements of the system, the designer can choose the maximum K_{fo} , thereby having a measure of the $L_{ext.}$ associated with this flow criterion. The predictions by the present analogue model for annular baffles are plotted in this manner in Fig. 27b.



VARIATION OF THE EXTINCTION LOADING FACTOR
WITH THE FLOW CRITERION

PREDICTIONS BY THE
PRESENT ANALOGUE
MODEL - ANNULAR
BAFFLES.

$$L_0 = \dot{m} C_p \Delta T / \dot{q}'''_m D$$

$$K_{FU} = \dot{m}^2 / \Delta p S_{190} \phi^{4/3}$$

FIG. 27(b).

CHAPTER FOUR.

FLAME STABILITY DATA.

-oOo-

Introduction.

The aim of this Chapter is to introduce and discuss some data on the stabilization of flames in the wake of bluff bodies. The relevant data are to be used in Chapter Six for comparison with the predictions by the electric analogue for similar systems. It is therefore necessary to keep this comparison in mind when discussing any available flame stability results. These results are usually presented as plots of velocity, or mass flow rate at the extinction condition, against the overall fuel/air ratio, or the equivalence ratio; the latter being equal to the fuel/air ratio divided by the stoichiometric fuel/air ratio. Some workers, however, present their results in terms of correlations involving the velocity, a linear dimension, and the pressure. For example, Mestre (Ref.14) correlated his results for discs, cones and hemispheres in a rectangular chamber by a plot of:

$$\frac{V}{D} \sim A/c \text{(26)}$$

where:

V = velocity past the baffle,

D = a linear dimension, being the base diameter
of the baffle,

A/C = air-fuel ratio.

DeZubay (Ref.15), on the other hand, correlated
his results for discs in a cylindrical chamber by
a plot of:

$$w_f/w_a = f \left\{ \frac{V_D}{P_D^f D^h} \right\} \dots\dots\dots(26)$$

where:

Wf/WA = mass fuel-air ratio,

VD = velocity through annular area,
formed by test duct and disc,

P = pressure in plane of disc,

D = disc diameter,

f, h = exponents used in the correlation.

Such correlations, however, are of no
direct use for the purpose of comparison with the

predictions of the electric analogue. This applies to any presentation of flame stability data which does not indicate the independence of the flame performance of the Reynolds Number. This question was discussed in Chapter One, in the section dealing with similarity conditions; and was taken into consideration in Chapter Three, when the analogue predictions were presented in terms of the extinction loading factor at a Reynolds Number high enough to make the extinction loading factor independent of it.

The flame stability results must therefore first be shown to satisfy this independence condition before comparing them with the analogue predictions. Similarly, any analogue predictions must be taken to be relevant only to systems where the extinction loading factor is independent of the Reynolds Number. The available flame stability results are discussed in the present Chapter, with special regard to the fulfilment of this condition. More data were obtained during this research to

augment the data of other workers. The results discussed in this Chapter are as follows:

1. DeZubay's results for discs in a circular burner,
2. Author's results for discs in a circular burner,
3. Martin and Webb's results for a burner with a fixed primary baffle, and a secondary annular baffle with a variable hole diameter,
4. Martin and Rushton's results for annular baffles in a circular burner,
5. Author's results for annular baffles in a circular burner,
6. Mestre's results for hemi-spherical baffles in a rectangular burner,
7. Results of Barrere and Mestre for conical and disc baffles in a rectangular combustion chamber.

Notation.

L_D = loading factor based on burner diameter,

$$= V \rho_u C_p \Delta T / \dot{q}_m D,$$

V = velocity of fuel and air mixture past
the baffle,

ρ_u = density of the mixture at inlet conditions,

C_p = specific heat at constant pressure,

ΔT = adiabatic combustion temperature rise,

$$= T_b - T_u,$$

T = temperature,

u = suffix denoting unburnt conditions,

b = suffix denoting completely burnt conditions,

$\dot{q}$ = volumetric heat release rate,

m = suffix denoting the maximum,

D = diameter of burner,

d = diameter of baffle,

Rn = the Reynolds Number,

FAR = fuel/air ratio,

ϕ = equivalence ratio,

ext. = extinction conditions.

1. Flame Stability Results - Central Disc Baffles.

A. DeZubay's Results.

These results are reported in Refs. 15 and 17. The first reference contains flame stability curves plotted as blow-out velocity versus the fuel/air ratio, for three circular discs having diameters of 0.25, 0.50, and 1.00 inches, placed normal to the flow direction in a circular duct of 2.75 inches diameter. The fuel used contained at least 95% propane, and not more than 5% butane and/or ethane. The pre-mixed fuel and air enter the chamber at a temperature of about 550° R; while the pressure was varied between atmospheric and 3 psia. All the results were found to correlate when plotted as

$$FAR = f \left\{ \frac{V}{P^{.95} d^{.85}} \right\} \text{-----}(26)$$

The second reference contains the results of the first, plus additional data obtained with the same apparatus and fuel, but with discs of diameters 1.50 and 2.00 inches. The correlation (26) is also used, and the smooth best line is drawn through the experimental points. No indication is given of the various pressures, as in the case of the results of Ref. 17. It is therefore not possible to analyse the results for the 1.50 and the 2.00 inch discs in the same manner as the results for the other discs.

The velocities and fuel/air ratios at the blow-out condition are used for each disc to calculate the extinction loading factor, $L_{D \text{ ext.}}$, which is defined by:

$$L_{D \text{ ext.}} = \frac{V_{\text{ext.}} \rho_u C_p \Delta T}{\dot{q}''_m D} \text{-----}(27)$$

where:

V = average velocity past the disc,

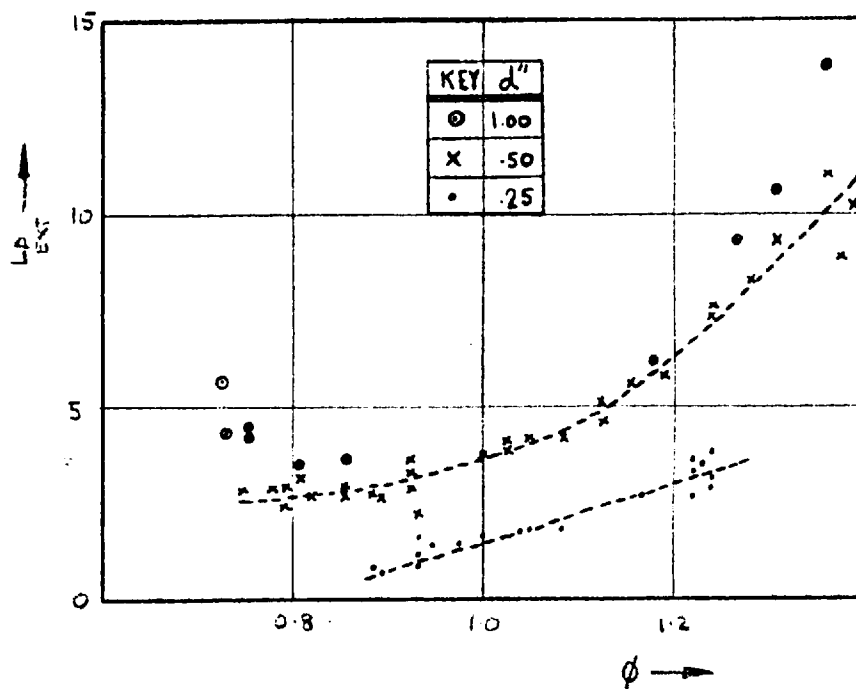
D = duct diameter,

other symbols as defined in the notation.

The method of calculation of the loading factor is described in Appendix 2. The resultant variation of the extinction loading factor with the equivalence ratio, ϕ , is indicated in Fig.28. The actual experimental points were taken from the graphs of Ref.15 as the basis for the calculation of Fig.28.

This figure indicates that for discs with diameters of 0.25 and 0.50, the extinction loading factor increases continuously with the equivalence ratio, although the increase for the half-inch disc is not large when ϕ is between 0.75 and the stoichiometric ratio. In the case of the 1.00 inch disc, only few points are available, and the trend of the results is not clear.

The significance of plots such as Fig.28, lies in the fact that ϕ may be considered as an indication of the Reynolds Number. In the weak fuel/air ratio regime, for example, a bigger ϕ



VARIATION OF THE EXTINCTION LOADING
FACTOR WITH THE EQUIVALENCE RATIO.

RESULTS FOR CENTRAL
DISCS, CALCULATED
FROM DATA OF REF. 15.

$$LD = V S_u C_p \Delta T / \dot{q}'''_m D$$

FIG 28.

is associated with a larger Reynolds Number. The absence of a horizontal straight line in Fig.28, therefore indicates that the results of DeZubay for the two smallest discs are not independent of the Reynolds Number, while the results for the 1.00" disc are not conclusive. Thus, these results cannot be used for comparison with the electric analogue predictions for central discs.

The correlating curve reported in Fig. of Ref.17 for the results of all five discs tested by DeZubay will now be used to construct a graph similar to those of Fig.28, for a particular pressure and disc diameter, e.g.,

Pressure = atmospheric

Disc diameter = 1.00 inches ($d/D = 0.363$).

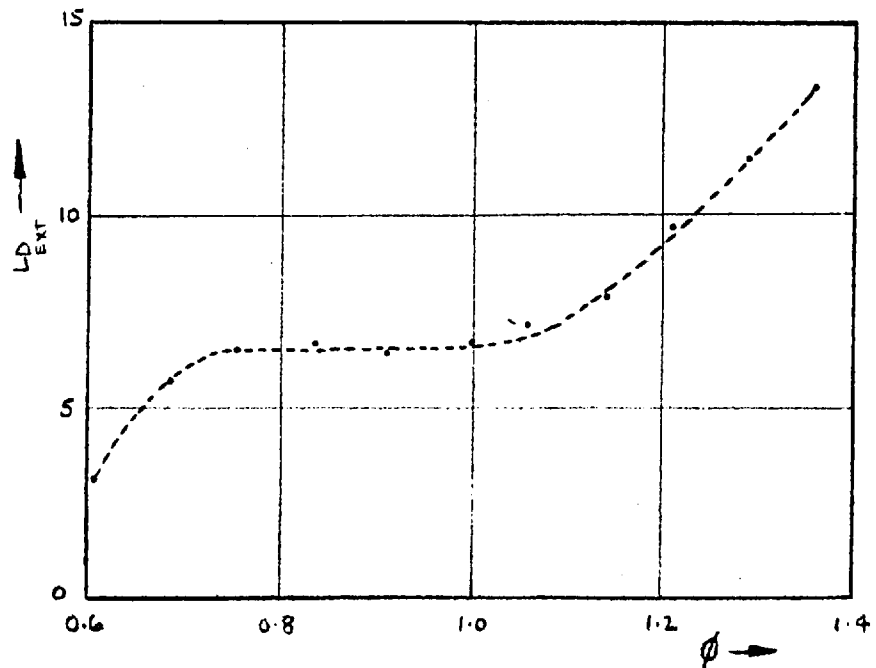
The calculated variation of the extinction loading factor, $L_D \text{ ext.}$, with the equivalence ratio, for this particular example, is given in the graph of Fig.29. This graph shows that the $L \text{ ext.}$ is constant for values of ϕ in the range between the

stoichiometric ratio and about 0.75. Outside this range, the extinction loading factor increases with the equivalence ratio.

In order to show the Reynolds Number effect, the points of Fig.29 are presented as a plot of $L_D \text{ ext.}$ against R_n in Fig.30, R_n being the Reynolds Number based on the approach stream velocity and conditions, and on the disc diameter. Fig.30 shows that the extinction loading factor increases with the Reynolds Number for weak mixtures, until a Reynolds Number is reached above which the loading factor at extinction is independent of it. For rich mixtures, $L_D \text{ ext.}$ decreases (with increased Reynolds Number) towards the constant value reached by the weak mixtures.

The comparisons with the electric analogue are only valid for the part of the weak regime in which $L_D \text{ ext.}$ is independent of R_n . However, because of the considerable experimental scatter around the smooth curve on which Fig.30 is based,

it was decided to obtain more experimental flame results with discs of larger disc/duct ratios than DeZubay's 1.00 inch disc. These results are reported in the following section.

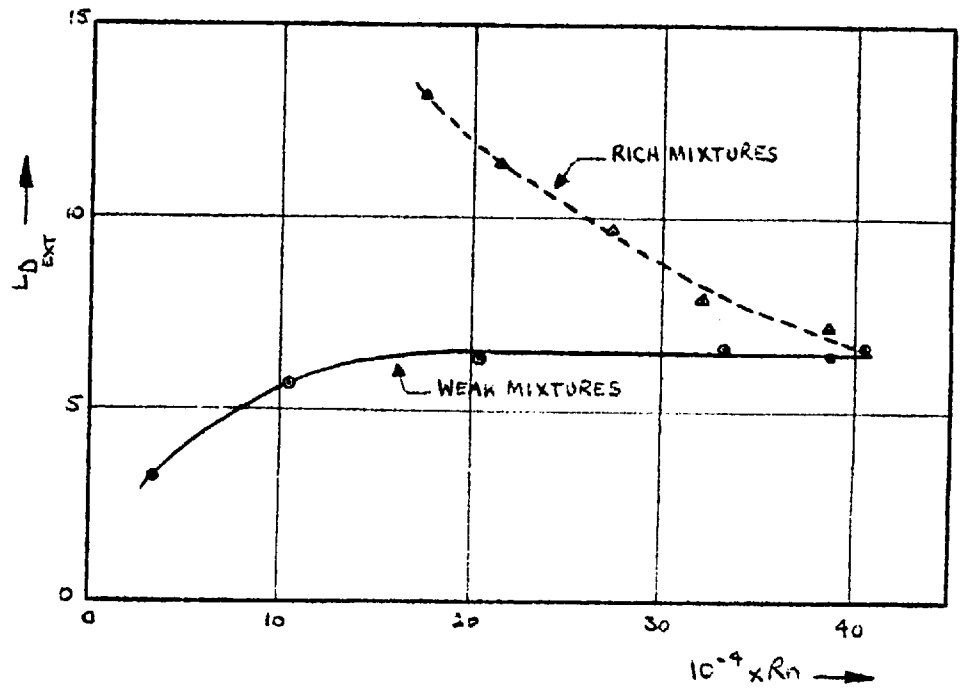


VARIATION OF THE EXTINCTION LOADING
FACTOR WITH THE EQUIVALENCE RATIO.

RESULTS FOR CENTRAL
DISCS, BASED ON THE
CORRELATION OF REF. 17.

$$LD = V S_u C_p \Delta T / \dot{q}_{f,m}''' D$$

FIG. 29.



VARIATION OF THE EXTINCTION LOADING FACTOR WITH THE REYNOLDS NUMBER.

RESULTS FOR CENTRAL DISCS, BASED ON THE CORRELATION OF REF 17.

$$Rn = u d / \lambda$$

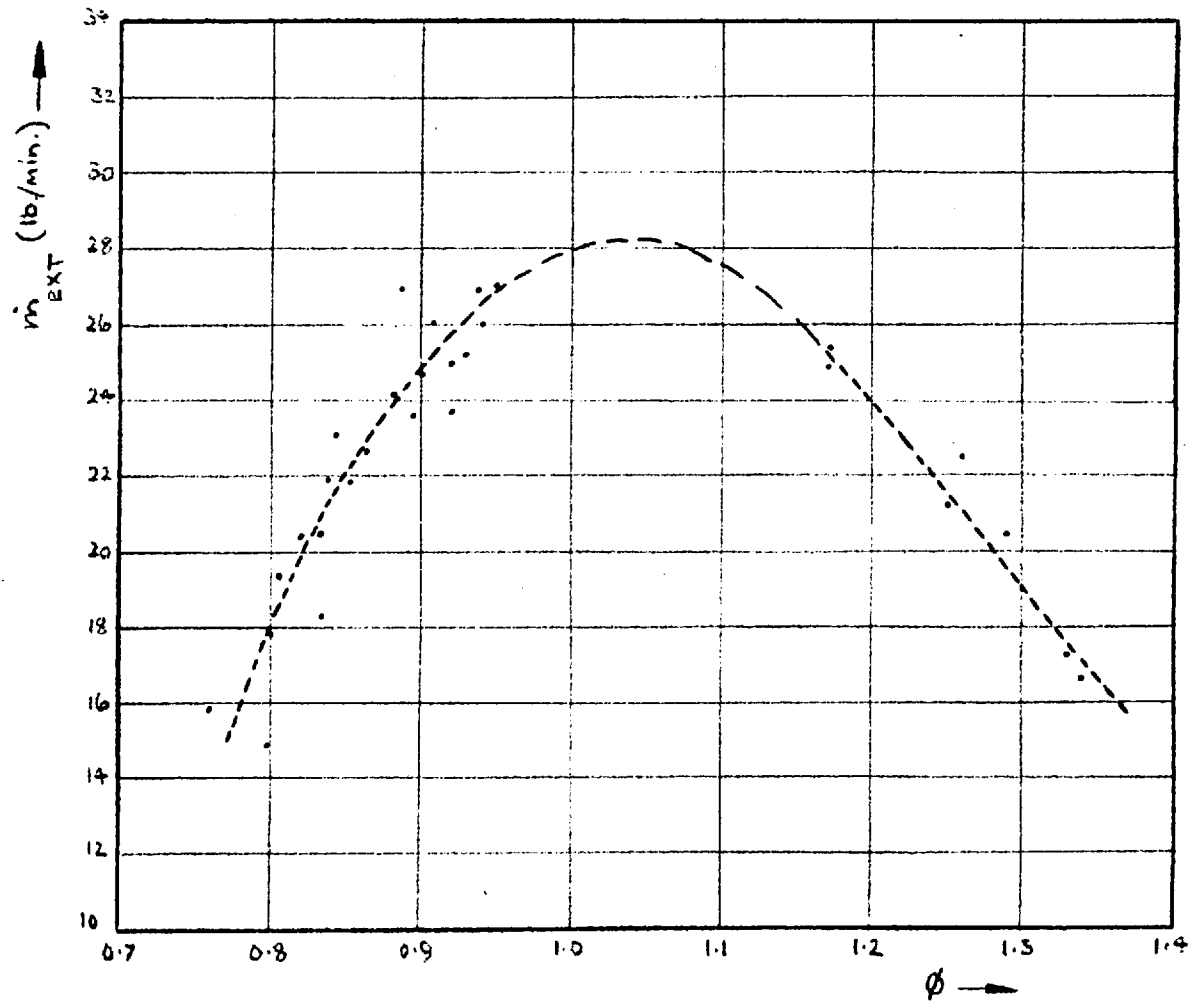
$$L_D = V 8 u c_p \Delta T / q_{m}''' D$$

FIG. 30.

b. Author's Results.

The apparatus used in obtaining the flame stability results of this section was described in Chapter Two. The four circular discs tested had diameters of 0.75, 1.00, 1.25, and 1.50 inches; the duct diameter being 2.00 inches. Commercial propane was used as fuel. It entered the combustion chamber pre-mixed with air, at room temperature. The results of the tests on the four discs are plotted in Figs. 31, 32, 33, and 34, as the mass flow rate at blow-out against the equivalence ratio. The actual experimental points of these graphs are used, in the manner described in Appendix 2, to calculate the extinction loading factor. This is defined by:

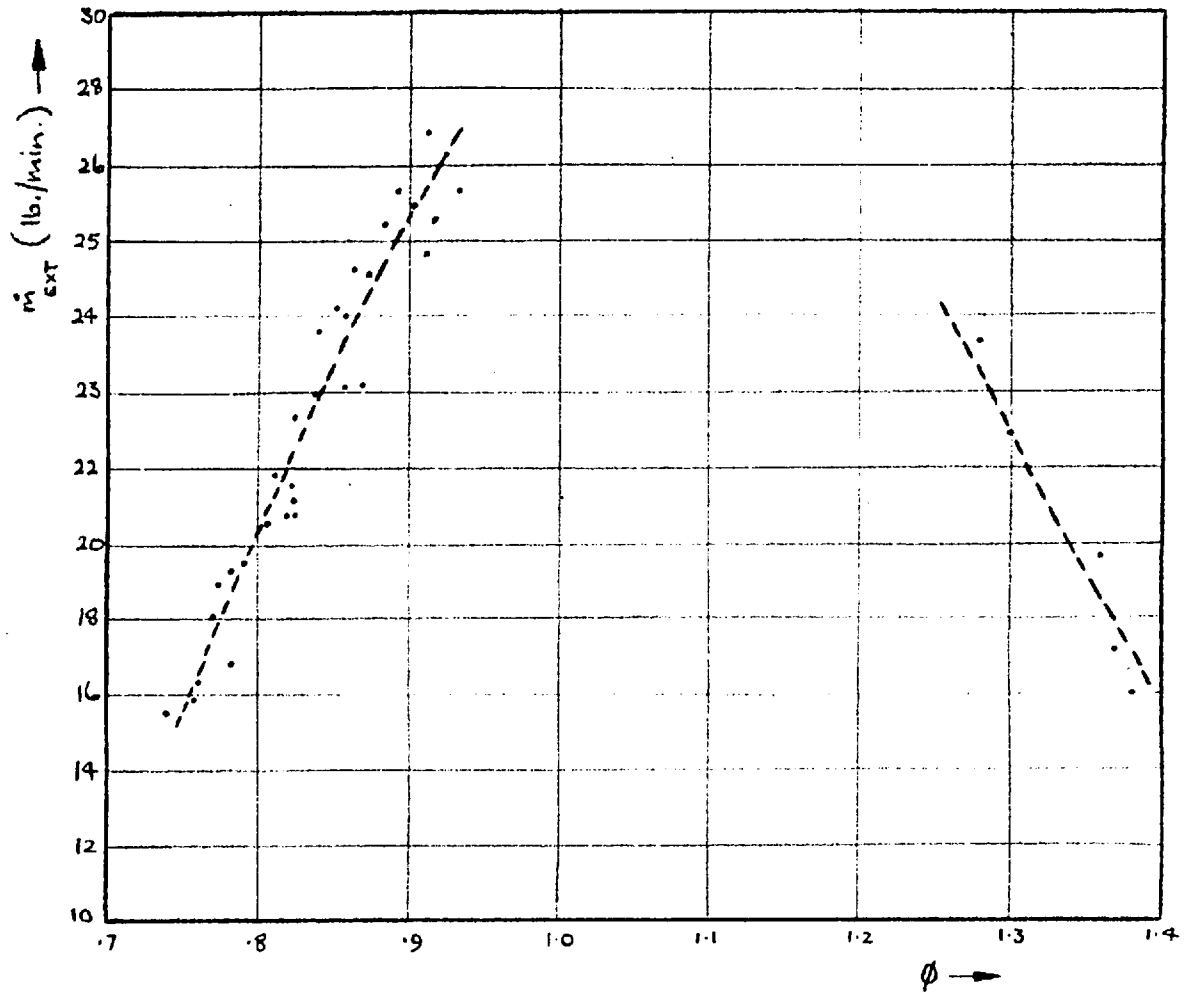
$$L_{D_{ext.}} = \frac{V_{ext.} \rho_u \dot{Q} \Delta T}{\dot{q}_{m D}} \text{ ----- (27)}$$



FLAME STABILITY LIMITS FOR CENTRAL DISCS
($d = 0.75$ INCHES)

AUTHOR'S RESULTS

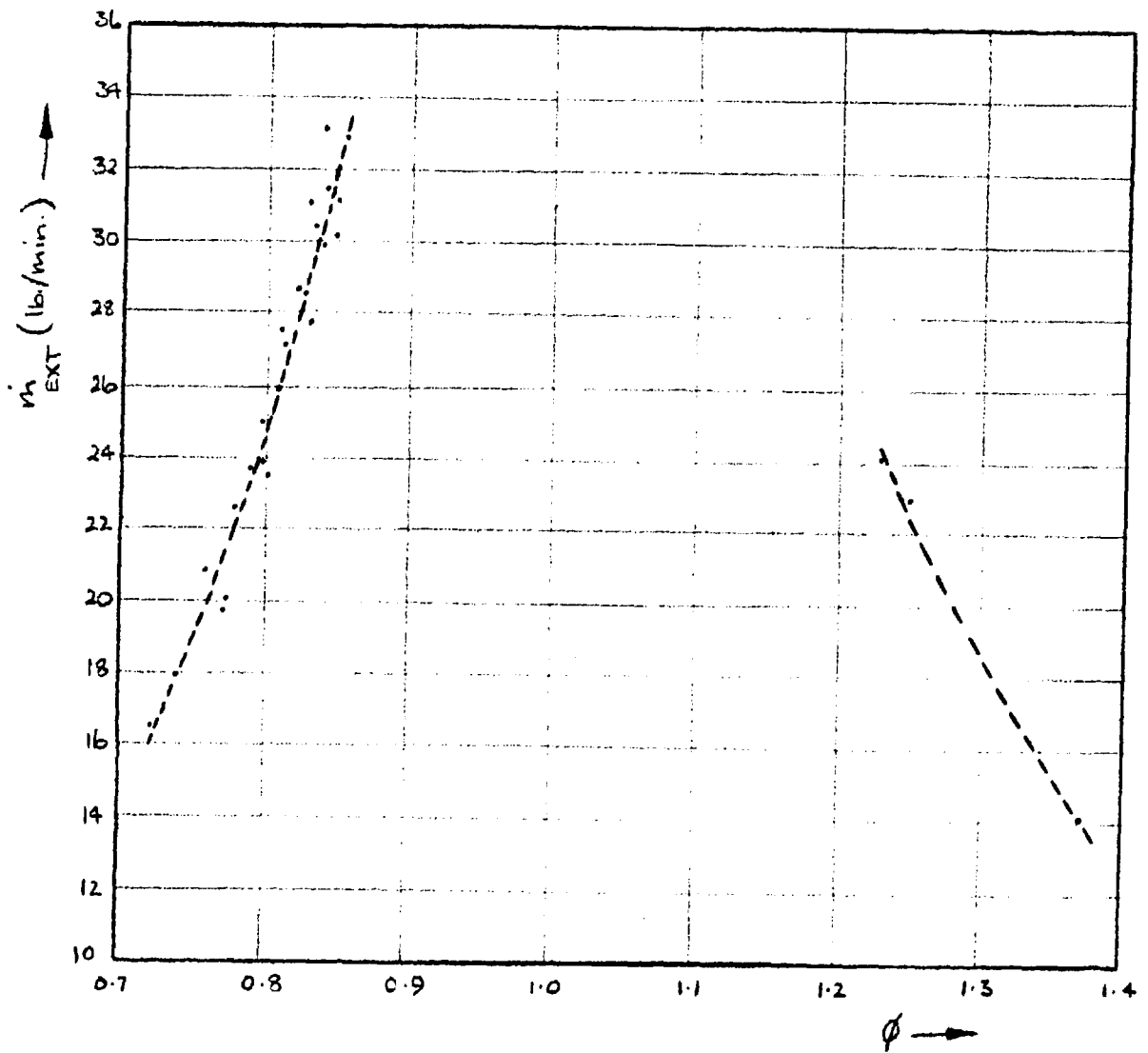
FIG. 31.



FLAME STABILITY LIMITS ~ CENTRAL DISCS
($d = 1.00 \text{ INCH}$)

AUTHORS' RESULTS

FIG. 32.

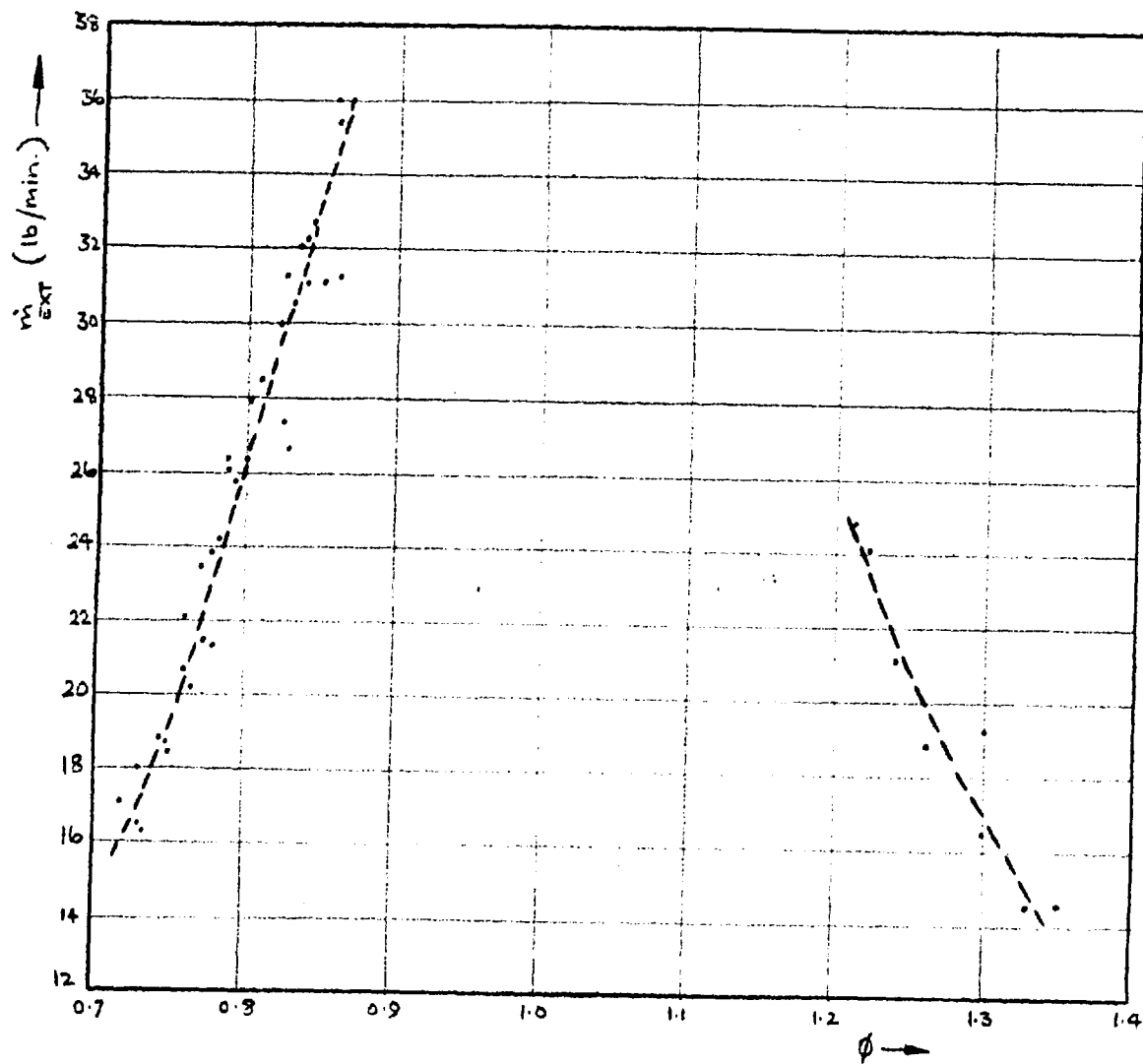


FLAME STABILITY LIMITS ~ CENTRAL DISCS.

($d = 1.25$ INCHES)

AUTHOR'S RESULTS

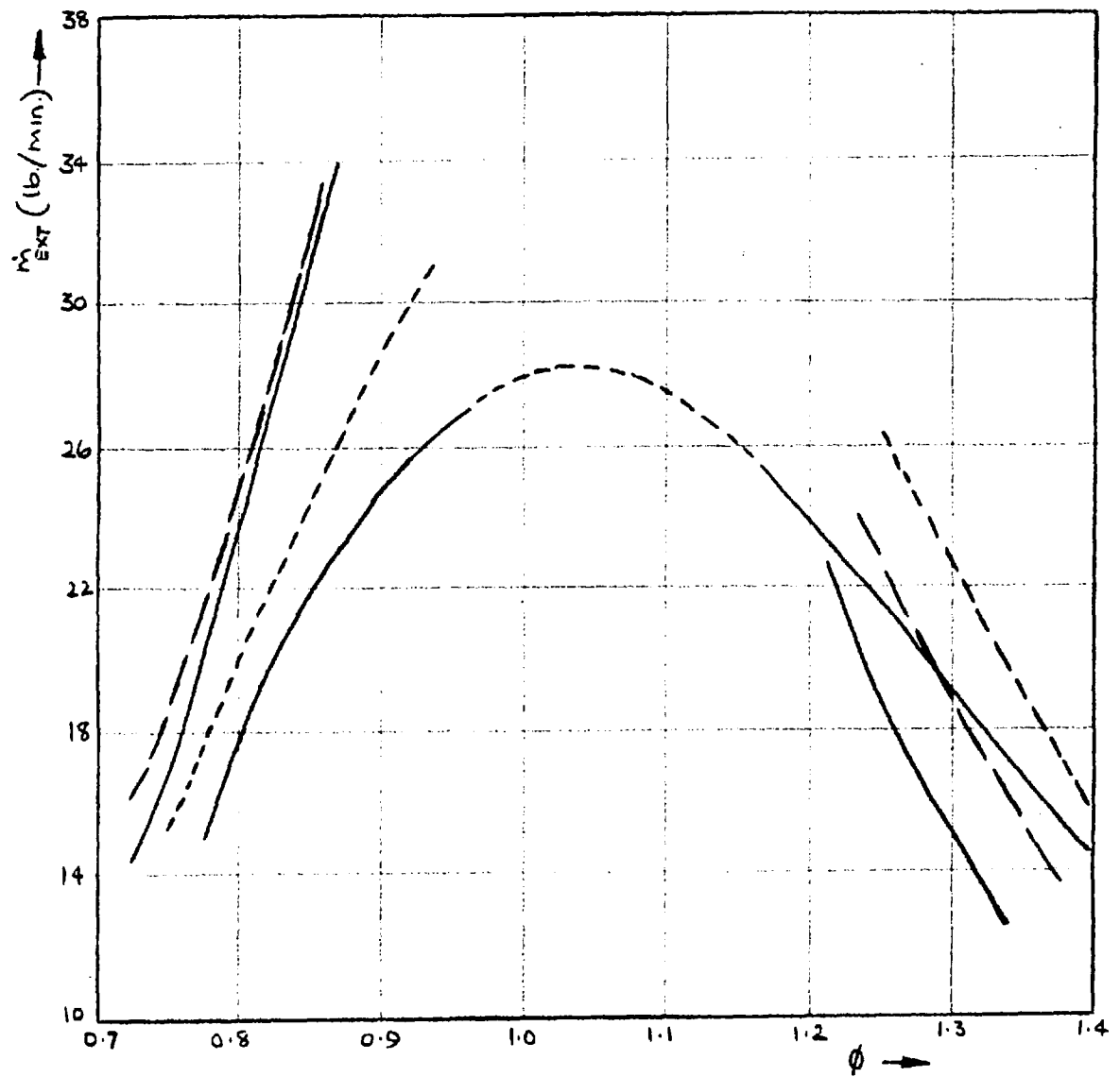
FIG. 33.



FLAME STABILITY LIMITS ~ CENTRAL DISCS
($d = 1.50$ INCHES)

AUTHOR'S RESULTS

FIG. 34.



FLAME STABILITY LIMITS ~ CENTRAL DISCS.

LINES DRAWN THROUGH
POINT OF FIGS. 31, 32, 33 & 34.

FIG. 35

where:

Vext. = the average velocity past the disc
at the extinction condition,

D = duct diameter,

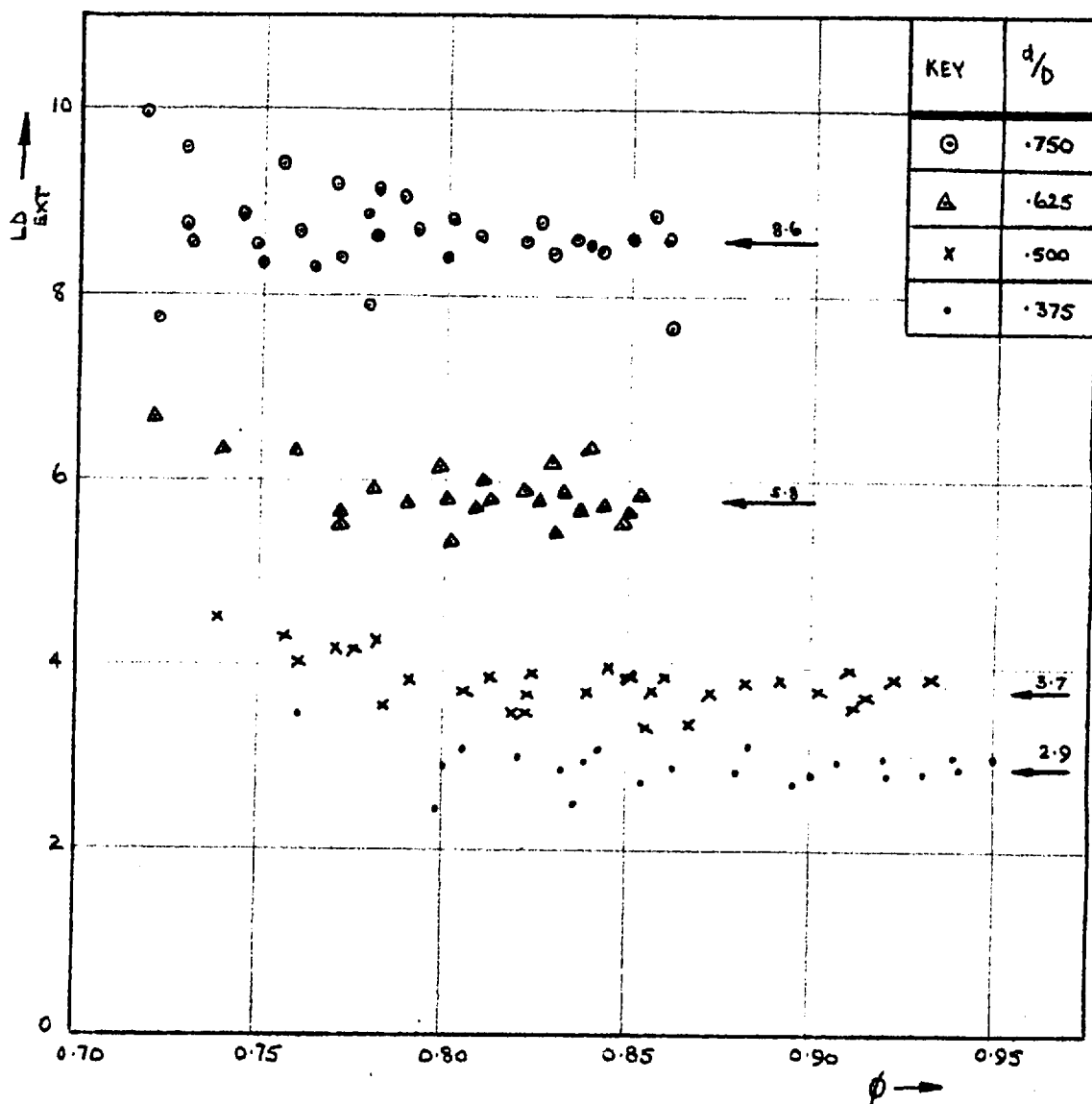
other symbols as defined in notation.

The resultant calculated loading factors are plotted in Fig.36 for the weak mixtures, and in Fig.37 for the rich mixtures, against the equivalence ratio. (Not many rich extinction points were obtained, mainly due to the expectation that the results would not be of direct use for comparison with the electric analogue predictions. This was suggested by the analysis of DeZubay's results in the previous section, and confirmed even by the few points obtained during these tests.)

Fig.36 shows that the loading factor at extinction remains constant for the four discs tested when the mixtures are weak, and with an equivalence ratio higher than about 0.77. On the

other hand, Fig.37 shows that when the mixture is rich, there seems to be no region of constant extinction loading factor. To the right of the experimental scatter for each disc in Fig.36, there is an arrow indicating an approximate average of the constant values of the loading factor. These values are plotted against the disc diameter in Fig.38.

The Reynolds Number has a negligible effect on the points of Fig.38, and therefore this curve is the one that will be used in Chapter Six for comparison with the predictions of the electric analogue for circular disc baffles.

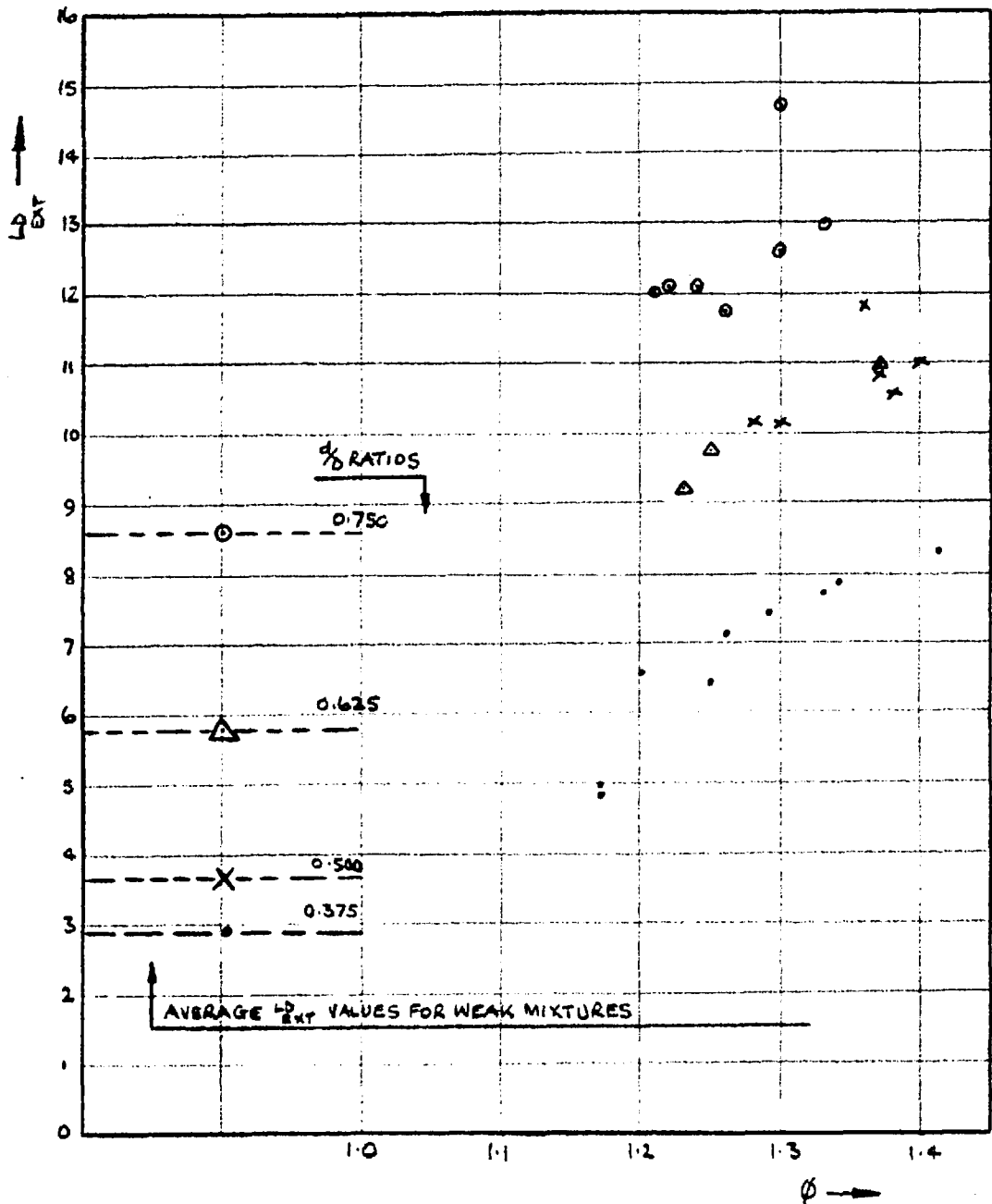


VARIATION OF EXTINCTION LOADING FACTOR
WITH WEAK EQUIVALENCE RATIO

AUTHOR'S FLAME
STABILITY RESULTS
FOR CENTRAL DISCS.

$$L_D = V S u C_p \Delta T / q''_{m D}$$

FIG. 36.

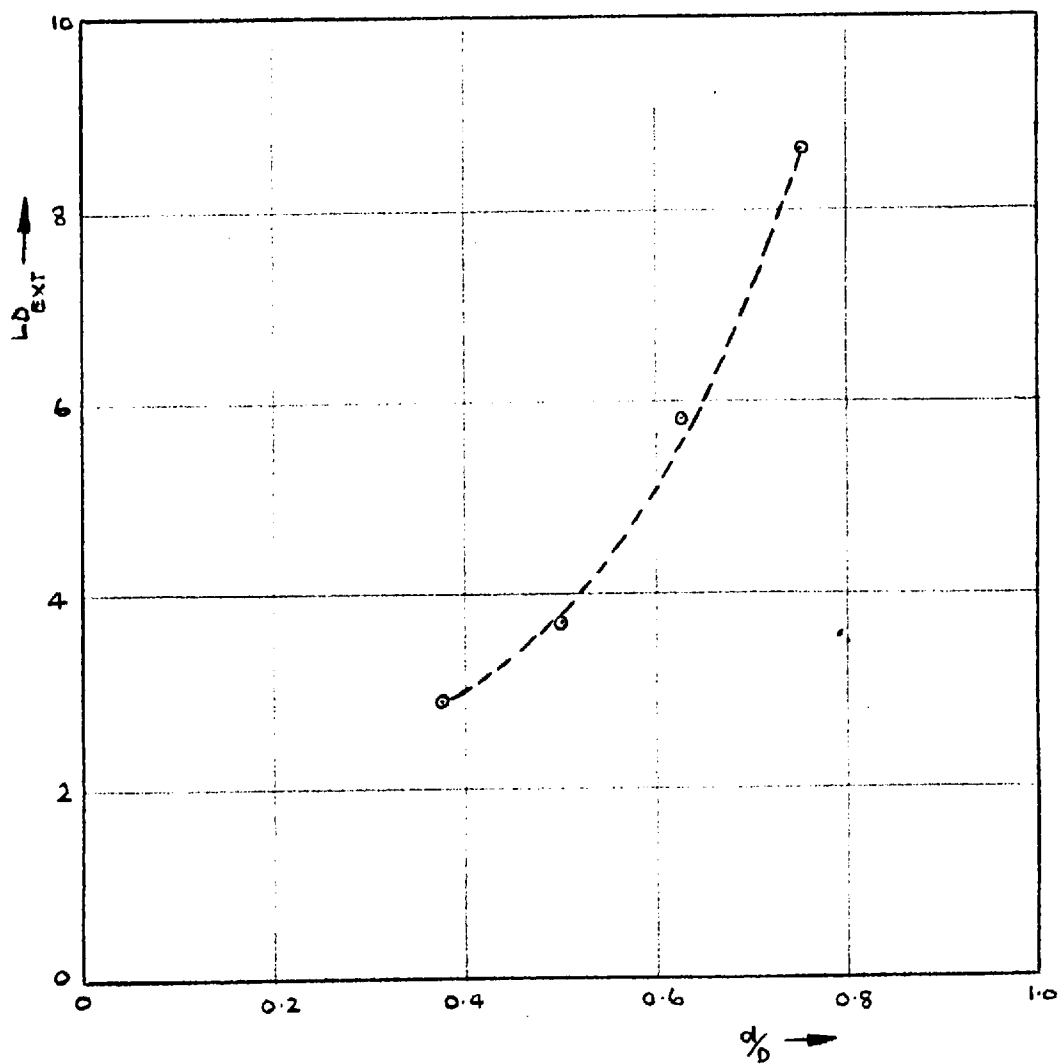


VARIATION OF EXTINCTION LOADING FACTOR
WITH RICH MIXTURE EQUIVALENCE RATIO

AUTHORS RESULTS
FOR DISC BAFFLES

$$L_D = \frac{V S_u C_p \Delta T}{\dot{q}_{m D}}$$

FIG. 37.



VARIATION OF EXTINCTION LOADING FACTOR
WITH DISC DIAMETER.

AUTHOR'S FLAME STABILITY
RESULTS WITH NEGLIGIBLE
REYNOLDS NUMBER EFFECT.

$$L_D = \frac{V S u C_p \Delta T}{\dot{q}''_m D}$$

FIG. 38

2. Flame Stability Results - Annular Baffles.

All the data analysed in this section were obtained by students at the Imperial College of Science and Technology, under the supervision of Professor D.B.Spalding. The analysis is again concerned in providing possible basis for the comparison of flame stability results with analogue predictions for annular baffles in axi-symmetric combustion chambers.

a. The Results of Martin and Webb.

These results are reported in Ref.18. The burner used is geometrically similar to the Lucas analogue model which was described in Chapter Two. The diameter of the burner is 2.0 inches, and the diameter of the hole of the secondary annular baffle was varied between 0.625 and 1.500 inches, but the reported flame stability limits graphs for baffles with small hole diameter, show a shift towards the rich mixtures region. This indicates

that there was an appreciable entrainment of ambient air by the jet issuing from the hole of the secondary baffle. The reported flame results for these baffle sizes were therefore excluded from the present discussion. Only the results for the remaining baffles are here presented. These had hole diameters of 1.00, 1.25, and 1.50 inches.

The results for these baffles, as reported in Ref.18, show little scatter. The smooth line through the experimental points has therefore been taken here as the basis for calculating the variation of the extinction loading factor with the inlet mixture equivalence ratio. The definition of the loading factor used in the present section is similar to that used in the presentation of the analogue predictions for annular baffles in Chapter Three, namely,

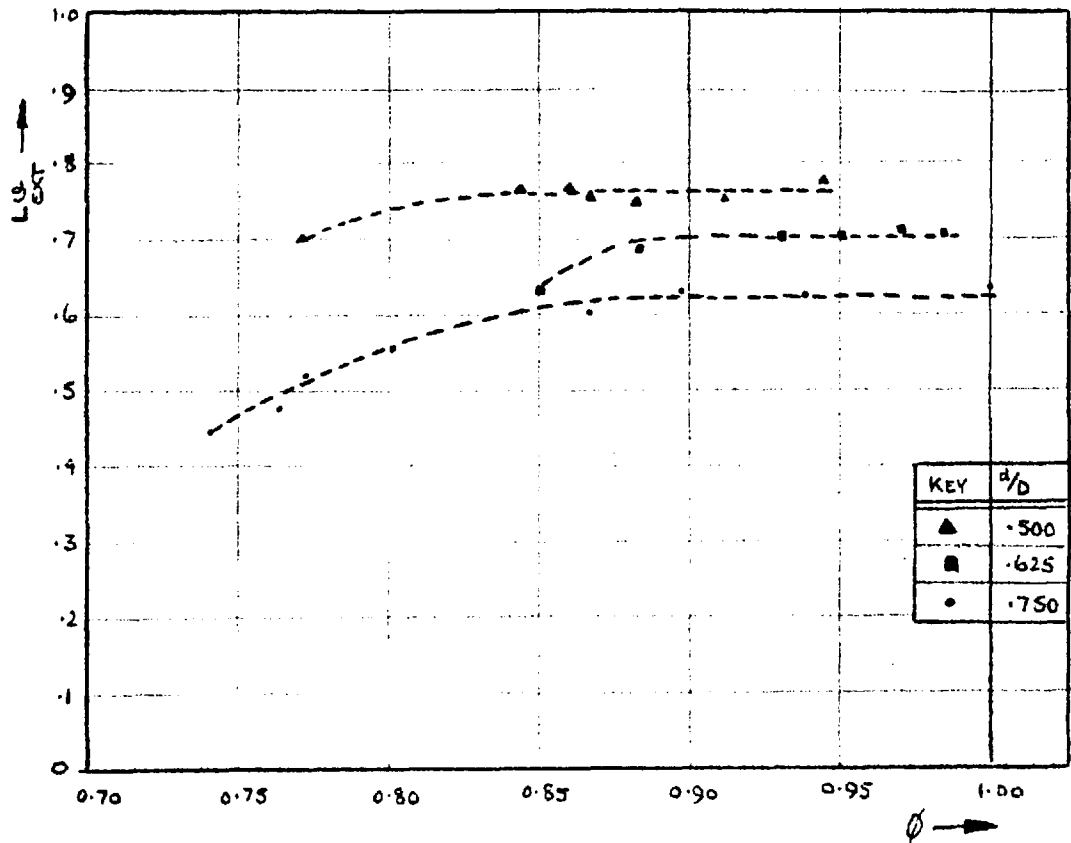
$$L_e = \frac{\dot{m} Q \Delta T}{q_m v} \text{-----} (28)$$

where:

$\dot{m}$ = mass flow rate of the fuel and air mixture,
 v = volume of combustion chamber, - other symbols
as in notation.

Fig.39 represents the variation of the extinction loading factor with the weak equivalence ratio near the stoichiometric condition, and it reveals that for all the three baffles, there is a region where the loading factor at extinction is independent of the equivalence ratio. This in fact means that the influence of the Reynolds Number on the stabilization characteristics of the baffle may be ignored. The constant values of L_{ext} in Fig.39 are plotted against the ratio of the baffle hole diameter to the burner diameter in Fig.40, showing the decrease of the extinction loading factor as the annular baffle hole diameter is increased.

It is the graph in Fig.40 which will be used, in Chapter Six, for comparison with the predictions of the electric analogue for annular baffles.

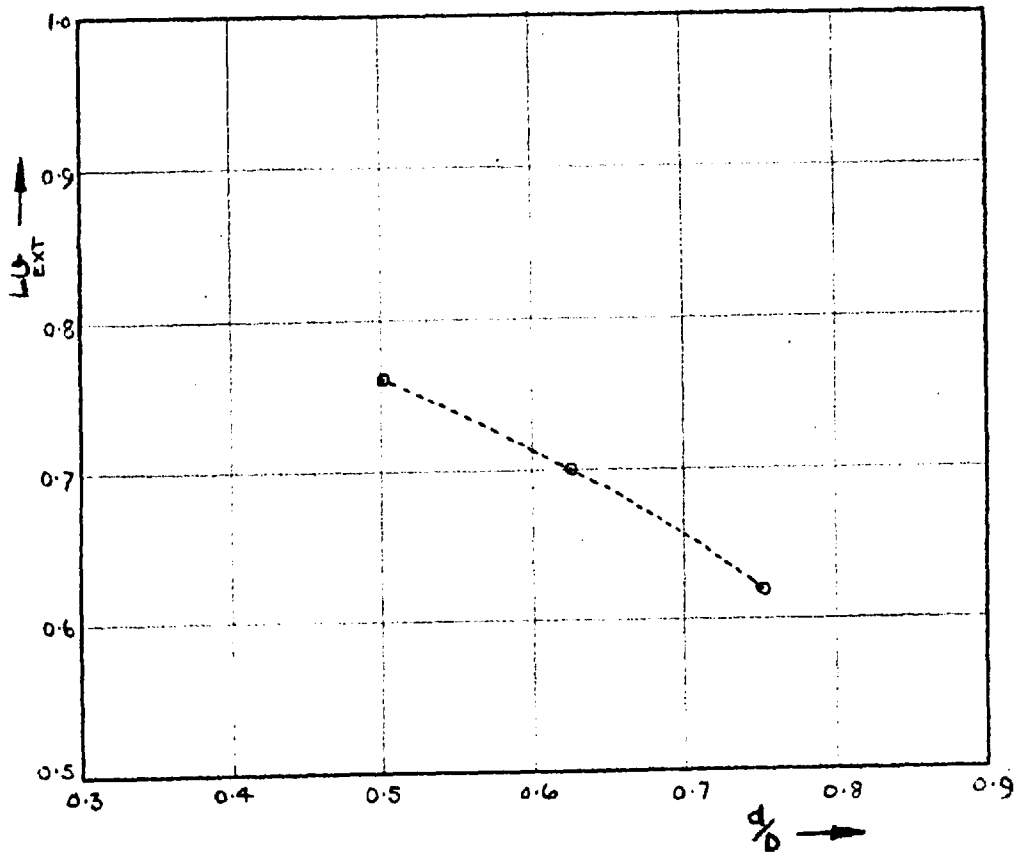


VARIATION OF THE EXTINCTION LOADING FACTOR
WITH THE WEAK EQUIVALENCE RATIO

RESULTS CALCULATED
FROM DATA OF REF. 18 FOR
ANNULAR SECONDARY BAFFLES.

$$L_g = \frac{\dot{m} c_p \Delta T}{\dot{q}_{r_m}'' D}$$

FIG. 39.



VARIATION OF EXTINCTION LOADING FACTOR
WITH ANNULAR BAFFLE HOLE DIAMETER

POINTS TAKEN
FROM FIG. 39.

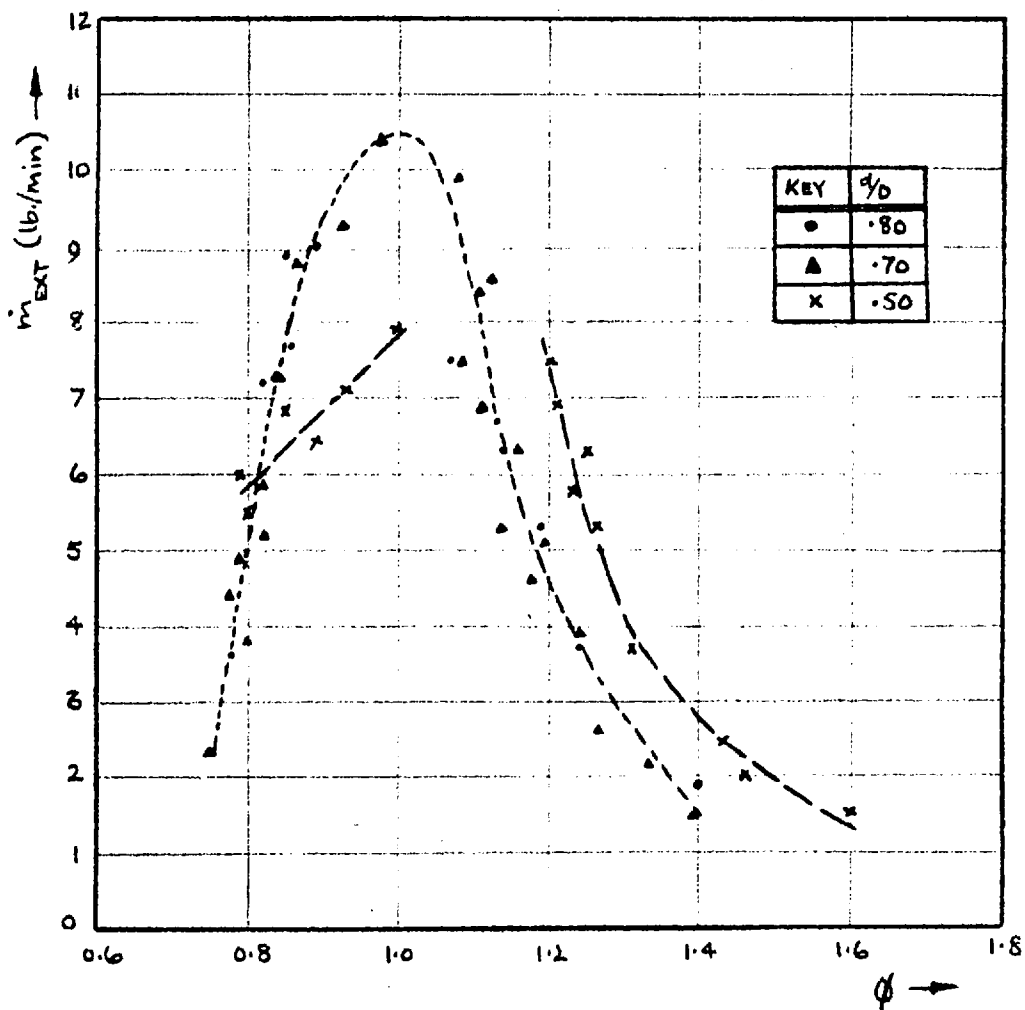
$$Lg = \frac{\dot{m} C_p \Delta T}{\dot{q}_{m b}}$$

FIG. 40.

b. The Results of Martin and Rushton.

These results are reported in Ref.19. The burner used was a simple cylindrical duct of 1.5 inches diameter, and 7.5 inches in length. The fuel used was commercial propane premixed with air at room temperature. Three annular baffles were tested, the ratio of their hole diameter to the diameter of the duct being 0.5, 0.7, and 0.8.

The flame stability limits presented in this section have been calculated according to the British Standards Specifications, from the basic measurements of pressure differentials, across the B.S.S. orifice plates reported in Ref.19. The results are plotted in Fig.41 as the mass flow rate of the fuel and air mixture at the extinction condition against the mixture equivalence ratio. The graph for the baffle with d/D ratio of 0.5 is seen to be shifted bodily into the rich regime, suggesting a considerable entrainment of surrounding air by



FLAME STABILITY LIMITS ~ ANNULAR BAFFLES.

RESULTS BASED ON
DATA FROM REF. 19.

FIG. 41.

the burning jet. The results for this baffle are therefore discarded.

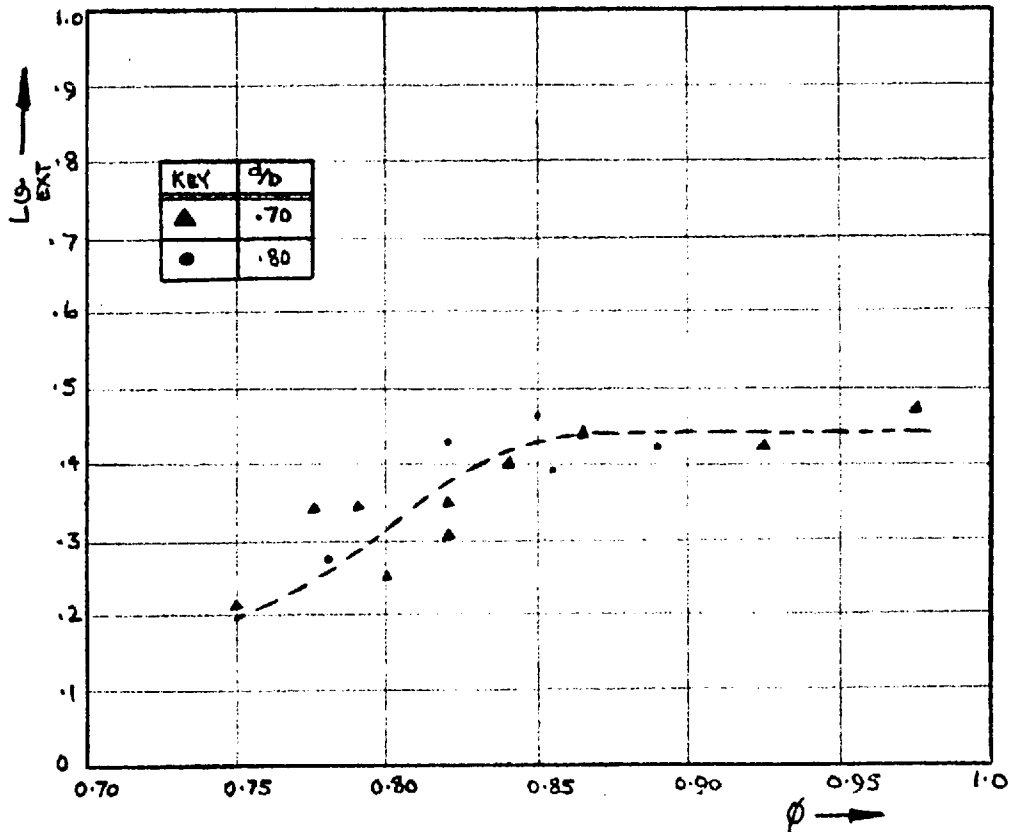
Fig.42 contains the results for the other two baffles, plotted as the extinction loading factor against the weak equivalence ratio. A region is seen to exist where the loading factor is independent of the equivalence ratio near the stoichiometric condition. This again indicates that the loading factor is independent of the Reynolds Number.

Fig.42 is taken to give the following result:

Annular baffles with d/D ratios of 0.7 and 0.8,

$$Lv_{\text{ext.}} = 0.44.$$

This result is to be included in the comparison of the electric analogue predictions with the real flame data in Chapter Six.



VARIATION OF THE EXTINCTION LOADING
FACTOR WITH THE WEAK EQUIVALENCE RATIO

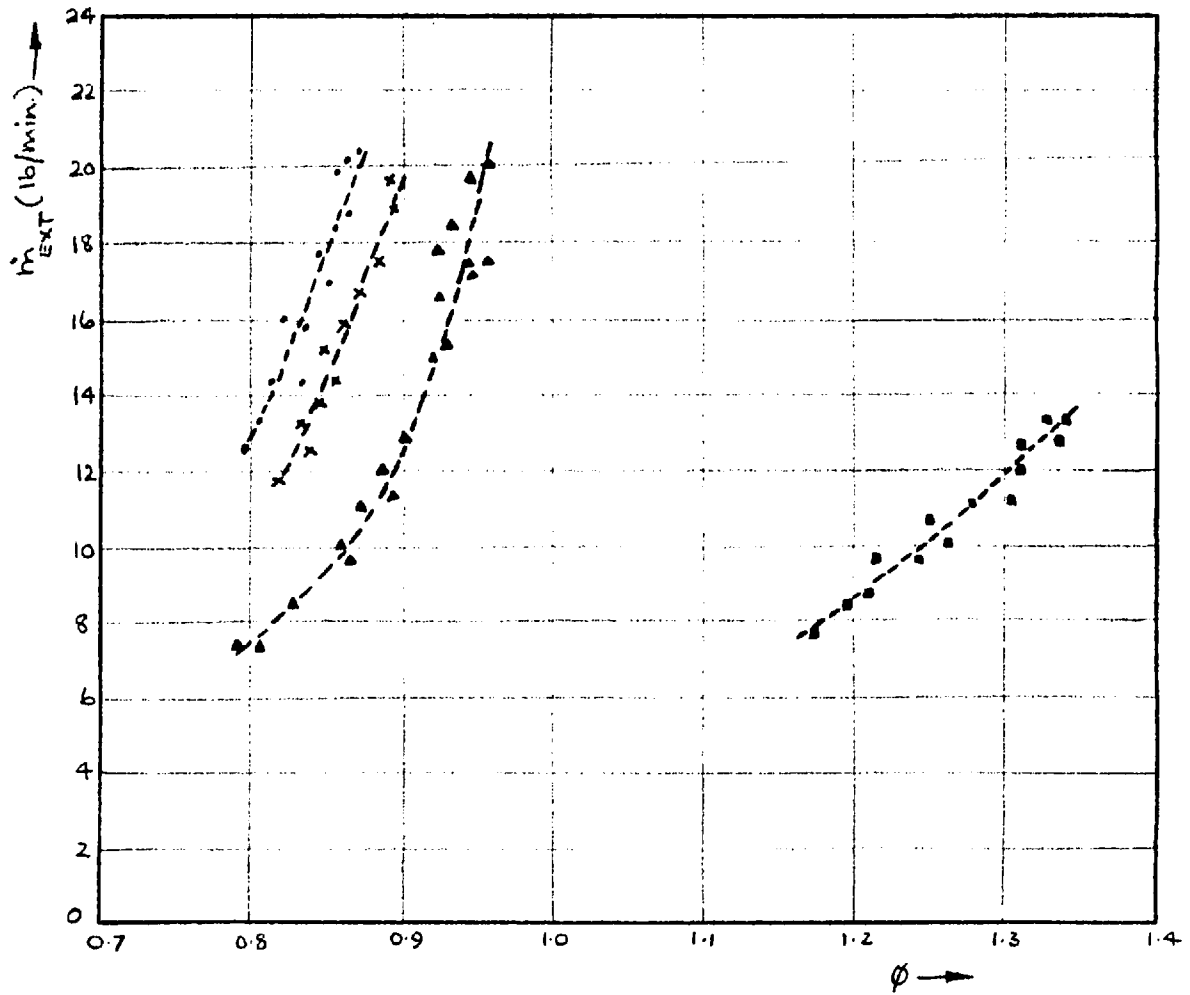
FLAME STABILITY RESULTS
FOR ANNULAR BAFFLES,
CALCULATED FROM THE DATA
OF REFERENCE 19.

$$L_0 = \frac{\dot{m} C_p \Delta T}{\dot{q}''_m \phi}$$

FIG. 42.

c. Further Results.

During this investigation, an attempt was made to get flame stability results for annular baffles with d/D ratio of less than 0.70, in a chamber similar to the one used by Martin and Rushton. Four annular baffles were tested in a burner having a 2.00 inch diameter. Only the weak part of the flame stability limits were obtained, and plotted in Fig.43 as the mass flow rate at extinction versus the mixture equivalence ratio. The figure indicates that the results for the baffle with d/D ratio of 0.50 have shifted into the rich regime, while those for the baffle with d/D ratio of 0.625 have shifted towards this region. This suggests entrainment of ambient air by the burning jet, the effect being larger for the baffle with smaller hole diameter. Increasing the length of the duct downstream the baffle was found to cause rough burning. Even the results for the baffles



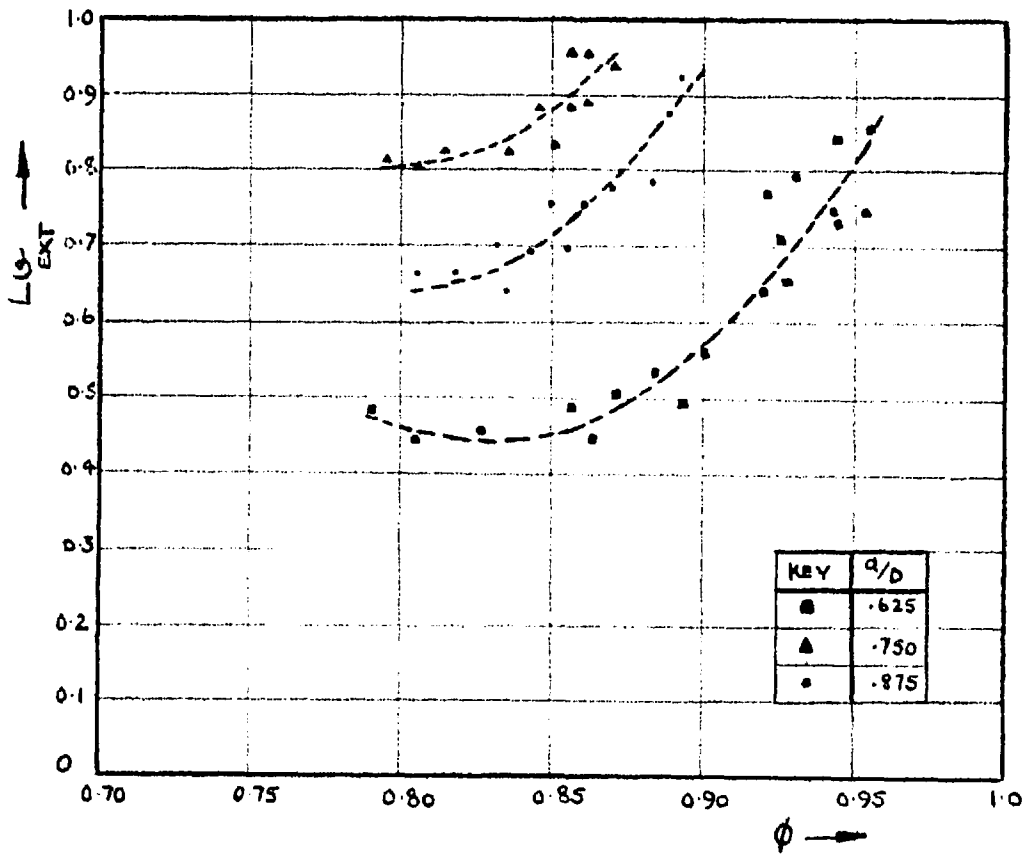
FLAME STABILITY LIMITS ~ ANNULAR BAFFLES

AUTHOR'S RESULTS.

FIG. 43.

KEY	d/D
•	.750
▲	.625
■	.500
×	.875

with d/D ratios of 0.750 and 0.875 cannot be used for comparison with the analogue predictions. The reason for this is shown in Fig. 44, where these results are plotted as the extinction loading factor versus the weak equivalence ratio. The loading factor at extinction is seen to increase with the weak equivalence ratio at the high values of the latter, indicating that there is still a Reynolds Number effect on the extinction loading. It was therefore decided to confine the comparison to the results discussed in sections (a) and (b).



VARIATION OF THE EXTINCTION LOADING FACTOR
WITH WEAK EQUIVALENCE RATIO

FLAME STABILITY
RESULTS FOR ANNULAR
BAFFLES. FROM DATA
OF FIGURE 43.

$$LG = \frac{\dot{m} c_p \Delta T}{q''_{m \phi}}$$

FIG. 44

3. Flame Stability Results - Hemi-Spherical Baffles.

Results of Mestre:

The available data on the stabilization of flames in the wake of hemi-spherical baffles due to Mestre, reported in Ref.14, were obtained in a combustion chamber rectangular in cross-section. The fuel used was kerosene, which entered the chamber already well mixed with air, at an inlet temperature of 573° K. The diameters of the hemi-spheres tested were 15, 20, and 25 millimetres; the chamber cross-section dimensions being 100 x 150 millimetres. i/

The flame stability limit curves reported in Ref.14 are not complete; the weakest rich point having an air/fuel ratio of about 12.5 and the richest weak point having an air/fuel ratio of about 18.5. These air/fuel ratios correspond to equivalence ratios of about 1.18 and 0.80 respectively. In this section, attention will again be confined to the weak stability limits. The extinction

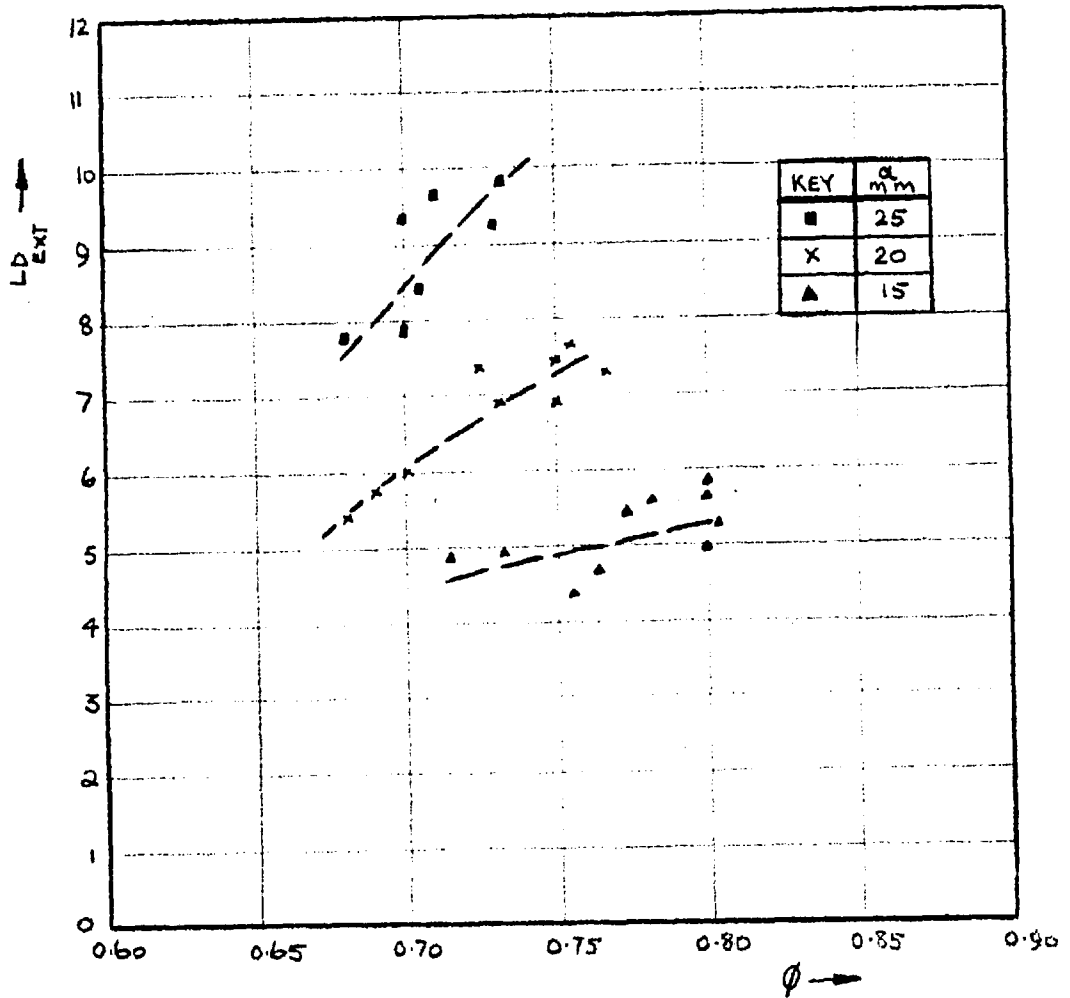
loading factor was calculated for the experimental points of Ref.14, and the results are plotted against the mixture equivalence ratio in Fig.45.

The loading factor is defined as follows:

$$L_D = V S_a C_p \Delta T / \dot{q}_m D \dots\dots\dots(27).$$

where: the symbols have the significance given in the notation.

Fig.45 may be taken to suggest that in the range of air/fuel ratios covered, the highest was not enough to make the extinction loading factor independent of the Reynolds Number. Thus the results of Fig.45 cannot be used quantitatively for the purpose of comparison with the electric analogue predictions for hemi-spherical baffles. However, it may be surmised from Fig.45 that, under fully turbulent conditions, the extinction loading factor increases with the hemi-spherical baffle diameter.



VARIATION OF THE EXTINCTION LOADING FACTOR WITH
THE WEAK EQUIVALENCE RATIO.

FLAME STABILITY
RESULTS FOR HEMI-
SPHERICAL BAFFLES.
BASED ON DATA OF
REFERENCE 14.

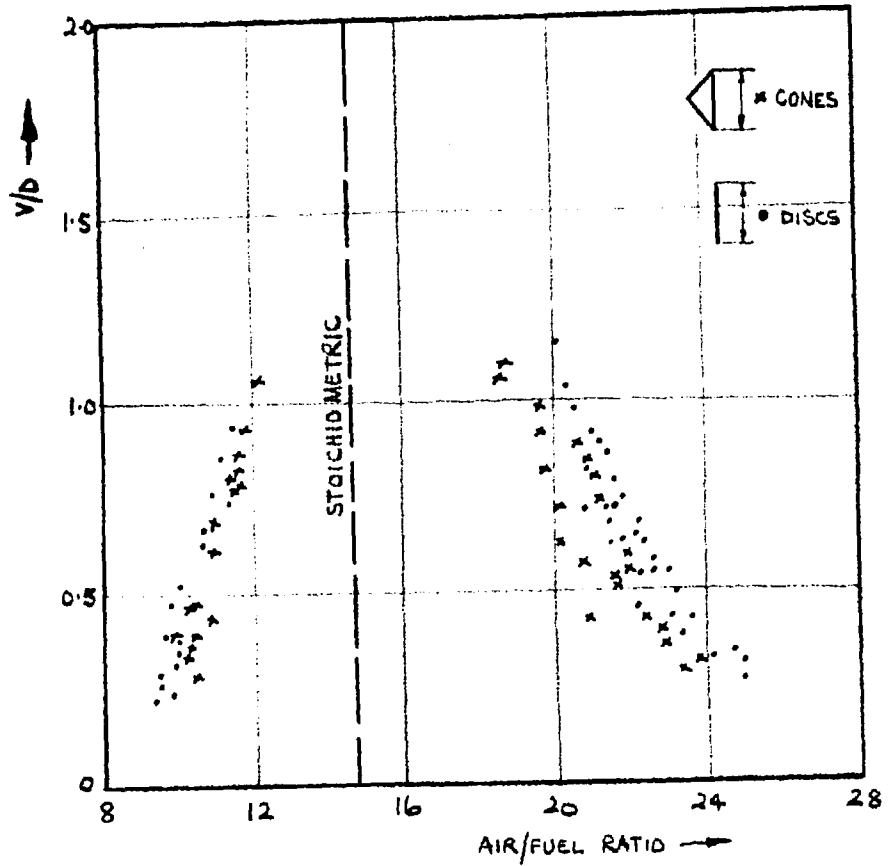
$$L_D = V S u_c p \Delta T / \dot{q}_{f_m}'' D$$

$D \equiv$ MEAN HYDRAULIC
DIAMETER OF CHAMBER,

FIG. 45.

4. Flame Stability Results - Conical Baffles.

The data available under this heading are those of Ref.16 due to Mestre. The combustion chamber used in obtaining the results for the hemi-spherical baffles was also used to obtain data for the stabilization of flames in the wake of circular discs and right-angled cones. The diameters of the discs and cones tested were 15, 20 and 25 millimetres; while the dimensions of the chamber rectangular cross-section were 100 x 150 millimetres. The results are correlated in Ref.16, in terms of the ratio of the gas velocity to baffle diameter against the air/fuel ratio. But there is no means of separating the extinction points for individual discs or cones. These results, cannot therefore be analysed in terms of the loading factor and the influence on it due to the Reynolds Number. Data of Ref.16 for cones and discs are copied in Fig.46.



CORRELATION OF STABILITY LIMITS.

FROM FIG. 10 b
REFERENCE 16.

FIG. 46.

CHAPTER FIVE.

THEORETICAL ANALYSIS OF FLAME STABILIZATION
IN
THE WAKE OF AN ANNULAR BAFFLE.

-oOo-

Introduction.

The performance of a burning flow system is the result of the interaction between the physical and chemical processes associated with it. In a one-stream system, the physical processes are characterized by the flow pattern; while the chemical processes may be represented by local volumetric heat release rate. It is primarily the complexity of the flow pattern which is the main obstacle to the prediction of the performance of the system by means of theoretical analysis. In certain cases, however, this problem has been solved. For example, Spalding (Ref.20) analysed a simple two-dimensional recirculation system, the spherical reactor used by Longwell and Weiss in their determination of heat release rates (Ref.21), and the diffusion-controlled spherical reactor.

Spalding also suggested an analysis of the flame stabilization in the wake of annular baffles

in an axi-symmetric chamber. The details of this analysis, and the solution of the resulting differential equations, make the subject of this Chapter.

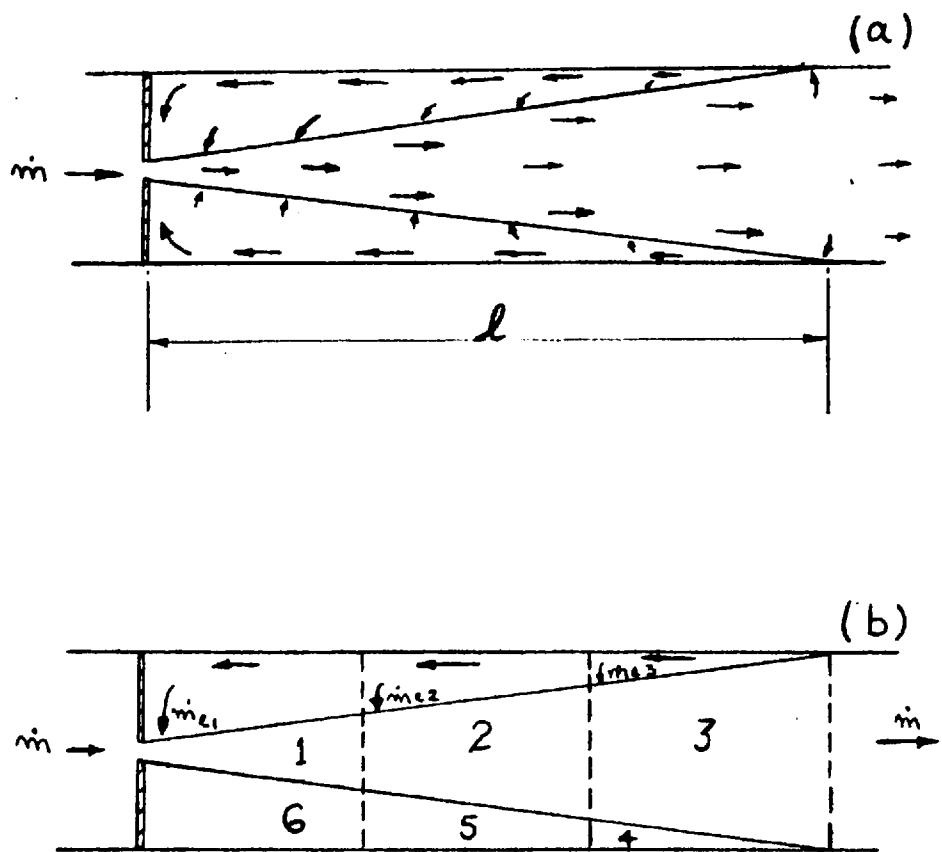
A flow pattern in the wake of the annular baffle is first postulated, and found to be a reasonable approximation to reality by comparing its consequences in terms of influence coefficients with those measured on the electric analogue apparatus. The suggested flow pattern is then used in conjunction with an assumed heat release rate function, to derive equations the solution of which gives the performance of the system in terms of the loading factor, the combustion efficiency, and a parameter related to the geometry of the combustion chamber.

The Postulated Flow Pattern.

The following are the features of the flow pattern, which is assumed to obtain in the wake of an annular baffle placed normal to the

axis of an axi-symmetric combustion chamber:

1. The system is regarded as adiabatic, and the flow as being fully turbulent.
2. The premixed fuel and air enter the chamber through the hole of the baffle at the rate of m lbs. per sec., forming a jet which spreads in a conical shape symmetrical about the chamber axis. The jet meets the wall of the chamber at a given distance downstream the baffle. At that point, a proportion of the gases are reversed into the recirculation zone, as indicated in Fig.47.
3. The temperature of the gases entering the recirculation zone is the same as the temperature of the gases continuing to flow in a stream-wise direction. This temperature is here characterised by ΔT_e (the temperature rise above that of the entering mixture).



POSTULATED FLOW PATTERN

FIG. 47

4. The recirculating gases are entrained by the jet as suggested by the arrows in Fig.47.
5. As far as the rate of entrainment and the flow in the jet are concerned, the laws of free jets are assumed to apply. In particular, the mass rate of jet flow at a distance x downstream the baffle is given by the approximation

$$\dot{m}_{jx} = k \frac{x}{d} \left(\frac{\rho_b}{\rho_u} \right)^{1/2} \dot{m} \text{ ----- (29)}$$

Where:

k = a constant empirically determined from free jet data,

d = the baffle hole diameter,

ρ_b , ρ_u = densities of the burning jet fluid and the incoming gas mixture respectively.

Equation 29 may be written as:

$$\dot{m}_{j\mu} = R \mu \dot{m} \text{ ----- (30)}$$

where:

$$R = \frac{R l}{d} \left(\frac{S_b}{S_u} \right)^{1/2},$$

$$\mu = \frac{x}{l}$$

R is designated as the recirculation parameter,
l is the length of the jet up to the place where
it hits the chamber wall.

validity of the Postulated Flow Pattern.

Before using the flow pattern as a basis for analytical study of the system, it is necessary to confirm its validity as a reasonable approximation to reality. This is done by comparing measured influence coefficients between pairs of points behind the annular baffle, with the values given by calculations based on the postulated flow pattern.

a. Calculation of Influence Coefficients.

It was shown in Chapter One that the influence coefficient between a pair of points

j and k, when heat is released at the point k, is given by:

$$\alpha_{jk} = \frac{\dot{m} c_p \Delta T_j}{\dot{q}_k} \text{-----(31)}$$

where:

ΔT_j = temperature rise at the point j,

$\dot{q}_k$ = heat input rate at the point k,

other symbols as before.

In an adiabatic chamber, ΔT_e (defined above) is given by:

$$\Delta T_e = \frac{\dot{q}_k}{\dot{m} c_p} \text{-----(32)}$$

Equations 31 and 32 give the following definition for the influence coefficient:

$$\alpha_{jk} = \frac{\Delta T_j}{\Delta T_e} \text{-----(33)}$$

Equation 33 affords a means for calculating the influence coefficient from the postulations made about the flow pattern. For the purpose of this calculation, the chamber volume between the baffle and the point where the jet meets the wall, is divided into six parts, as shown in Fig.47. These

parts are made up by dividing the chamber into three equal sections along its axis. Volumes 1, 2 and 3 are thus in the jet, and volumes 4, 5, and 6 are in the recirculation zone. The first three parts entrain gases from their corresponding recirculation zone parts at the rates determined by Equation 30, which was assumed to hold for the jet issuing from the hole of the annular baffle. The calculation of the influence coefficients is now only a matter of heat balance considerations; the exit temperature rise, ΔT_e , being taken as the temperature rise in part 3 of the chamber. Table (3) contains the resulting calculated α_{jk} values for an annular baffle with a hole to duct diameter ratio $d/D = 0.405$.

Measurement of Influence Coefficients.

The measurements are performed on the present model of the analogue. This is considered to be divided into six sections, as in Fig. 47, where a section is called " the source point k ", it

		SOURCE POINT K					
		1	2	3	4	5	6
P O I N T S J	1	1.00	.34	.34	.43	.50	1.00
	2	1.00	1.00	.67	.86	1.00	1.00
	3	1.00	1.00	1.00	1.00	1.00	1.00
	4	1.00	1.00	1.00	1.28	1.00	1.00
	5	1.00	1.00	1.00	1.28	1.50	1.00
	6	1.00	1.00	1.00	1.28	1.50	2.97

CALCULATE α_{jk}

$\frac{d}{D} = 0.405$

TABLE [3]

means that the heaters in this part of the analogue working section are switched on. The heat input rate at the point k is thus the rate of heat generation by these heaters. The temperature rise at a point j, ΔT_j , is likewise the average of temperature rises recorded by thermocouples in the section j.

In addition to ΔT_j , the other quantity that has to be measured in order to obtain the influence coefficient, α_{jk} , is the exit temperature rise, ΔT_e . This is taken as the average temperature rise recorded by the six thermocouples of the first analogue element downstream the plane where the jet hits the wall of the analogue chamber. The influence coefficient is then calculated from the formula:

$$\alpha_{jk} = \Delta T_j / \Delta T_e.$$

The measured influence coefficients that correspond to the values given by Table (3), which was calculated on the basis of the postulated flow pattern,

		POINTS K					
		1	2	3	4	5	6
P O I N T S J	1	1.00	.30	.26	.33	.55	1.22
	2	1.03	1.00	.62	.72	1.32	1.26
	3	.96	1.00	1.00	1.00	1.00	1.00
	4	.94	.93	1.00	1.22	1.00	.88
	5	1.15	1.20	1.10	1.34	1.80	1.14
	6	.94	.89	.90	1.11	1.78	3.58

MEASURED α_{jk}

$$\frac{1}{D} = .405$$

$$\dot{m} = .37 \text{ lb/sec.}$$

TABLE [4]

		POINTS K					
		1	2	3	4	5	6
POINTS J	1	1.00	.28	.25	.31	.59	1.10
	2	1.04	1.00	.56	.84	1.22	1.24
	3	.93	.94	1.00	1.00	1.00	1.00
	4	.93	.99	1.00	1.45	1.00	.90
	5	1.10	1.15	1.06	1.60	1.75	1.04
	6	1.00	.93	.96	1.33	1.72	3.45

MEASURED α_{jk}

$$\frac{d}{D} = 0.405$$

$$m = 0.415 \text{ lb/Sec.}$$

TABLE [5]

		POINTS K					
		1	2	3	4	5	6
P O I N T S J	1	1.00	.29	.28	.29	.62	1.10
	2	1.04	1.00	.59	.71	1.28	1.24
	3	.85	.99	1.00	1.00	1.00	1.00
	4	.85	.90	1.00	1.06	.83	.90
	5	1.12	1.20	1.15	1.22	1.66	1.07
	6	1.16	.92	.82	.87	1.62	3.45

MEASURED α_{jk}

$$d/b = 0.405$$

$$m = 0.464 \text{ lb/sec.}$$

TABLE [6].

are given in Tables (4), (5) and (6). These are all for a baffle with a similar d/D ratio of 0.405, but for mass flow rates of 0.37, 0.414 and 0.465 lbs./sec. respectively.

Comparison Between Measured and Calculated Influence Coefficients.

This comparison is carried out by inspection of Table (3) on the one hand, and Tables (4), (5) and (6) on the other. The former contains the calculated values of the influence coefficients, while the latter contains the measured values. It is seen that the values of Δ_{jk} measured for the various flow rates differ from each other, and from the calculated influence coefficients by less than 25% of the calculated values. In view of the crude sub-division of the chamber for the purposes of calculation and measurement, this proximity may be taken to indicate:

1. That the flow rates used during the

measurements of α_{jk} are high enough to make these almost independent of the Reynolds Number.

- 2 . That the postulated flow pattern is a reasonable approximation to reality.

The first of these conclusions is confirmed by reference to Fig.17, where it is seen that the extinction loading factor becomes independent of the Reynolds Number when the latter exceeds a value of about 8×10^4 , for a baffle with d/D ratio of 0.405. The Reynolds Number corresponding to the flow rates of Tables 4,5, and 6, are 8.2×10^4 , 9.2×10^4 , and 10.3×10^4 respectively. e/

The second conclusion justifies the use of the postulated flow pattern as a basis for an approximate theoretical analysis to predict the performance of the flames in the wake of annular baffles. This is done in the next section of this Chapter.

Theoretical Analysis.

In this analysis, the reaction in the jet is neglected to simplify the computations necessary for the solution of the resulting equations. This approximation will be kept in mind when using the results of this analysis for comparisons in the next Chapter. The following sections will now be devoted to the derivation and solution of the differential equations governing the performance of the system:

The Equations.

An element of the volume at a distance x from the baffle, and having a thickness dx , will be considered. Some of the symbols used are shown in Fig. 48. A heat balance for the shaded part the elemental volume gives:

$$-\dot{m}_{rx} C_p dT = \dot{q} A_x dx \dots\dots\dots(34)$$

where:

T = temperature of recirculating gases at x ,

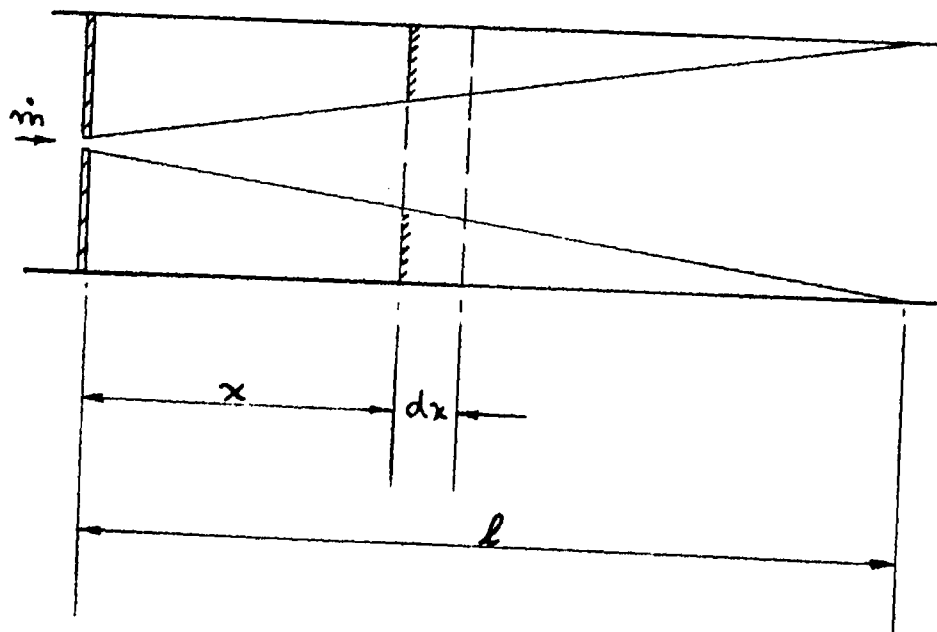


DIAGRAM FOR THEORETICAL ANALYSIS

FIG. 48.

$\dot{m}_{rx}$ = mass rate of flow of recirculating gases
at a distance x from baffle,
 $\dot{q}'''$ = local volumetric rate of heat release,
 Ax = shaded area shown in Fig.48.

With the introduction of the heat release function
 ψ , and the dimensionless temperature and distance (τ and μ), Equation 34 becomes:

$$\frac{d\tau}{d\mu} = \frac{\psi(1-\mu^2)}{L(R\mu-1)} \dots\dots\dots(35)$$

where:

$\tau = (T-T_u)/(T_b-T_u)$,
 $\psi = \dot{q}''' / \dot{q}'''_{\max.}$,
 $\mu = x/l$,
 $L = \dot{m} C_p \Delta T / \dot{q}'''_{\max.}$ v = loading factor,
 $v = (\pi/4) D^2 l$ = the volume,
 $R = k l / d$ = recirculation parameter.

The heat balance carried over the whole chamber volume gives the equation :

$$\dot{m} C_p (T_e - T_u) = \int_0^l \dot{q}''' Ax dx \dots\dots\dots(36)$$

T_e being the exit temperature.

Equation 36 may be transformed into the non-dimensional form given by:

$$L T_e = \int_0^1 (1 - \mu^2) \psi d\mu \text{ ----- (37)}$$

The simultaneous equations that have to be solved are Equations 35 and 37. The term τ which denotes the reactedness of the gases at the exit plane, may be taken as the combustion efficiency. This is seen from the definition of τ_e , viz.,

$$\tau_e = (T_e - T_u) / (T_b - T_u),$$

which is the ratio between the actual temperature rise of the gases in passing through the combustion chamber, and the temperature rise for complete combustion.

Solution of the Equations.

The heat release function ψ to be used in the solution of Equations 35 and 37, is given by:

$$\psi = 149 \tau^8 (1 - \tau^2) \text{ ----- (38)}$$

Equation 38 is plotted in Fig.1.

The procedure followed in the solution is as follows:

1. Equation 35 is considered as a finite difference equation, i.e.,

$$\Delta T = - \frac{\psi (1-\mu^2)}{L(R\mu-1)} \Delta \mu \text{ ----- (39)}$$

2. The length of the chamber is divided into ten equal parts, thus:

$$\mu_0 = 0, \mu_1 = 0.1, \mu_2 = 0.2 \dots \dots \mu_{10} = 1.0, \\ \text{or, } \Delta \mu = 0.1.$$

3. For a given recirculation parameter (R), and loading factor (L), a value is assumed for the exit reactedness T_e . This reactedness is for $\mu = \mu_{10} = 1.0$.

4. From Equations 39 and 38, ΔT is calculated for $\mu = 1.0$.

5. From steps 3 and 4, a value is obtained for τ at $\mu = 0.9$.
6. Steps 4 and 5 are repeated until τ values are obtained for all the μ values.
7. The Simpson Rule for the calculation of area under a curve is used to calculate the R.H.S. of Equation 37.
8. The L.H.S. of Equation 37 is calculated from the assumption of step 3.
9. If the results of steps 7 and 8 coincide, then a solution has been obtained; otherwise the procedure from step 3 onwards is repeated until a solution is reached.
10. The procedure is repeated for other values of L and R.

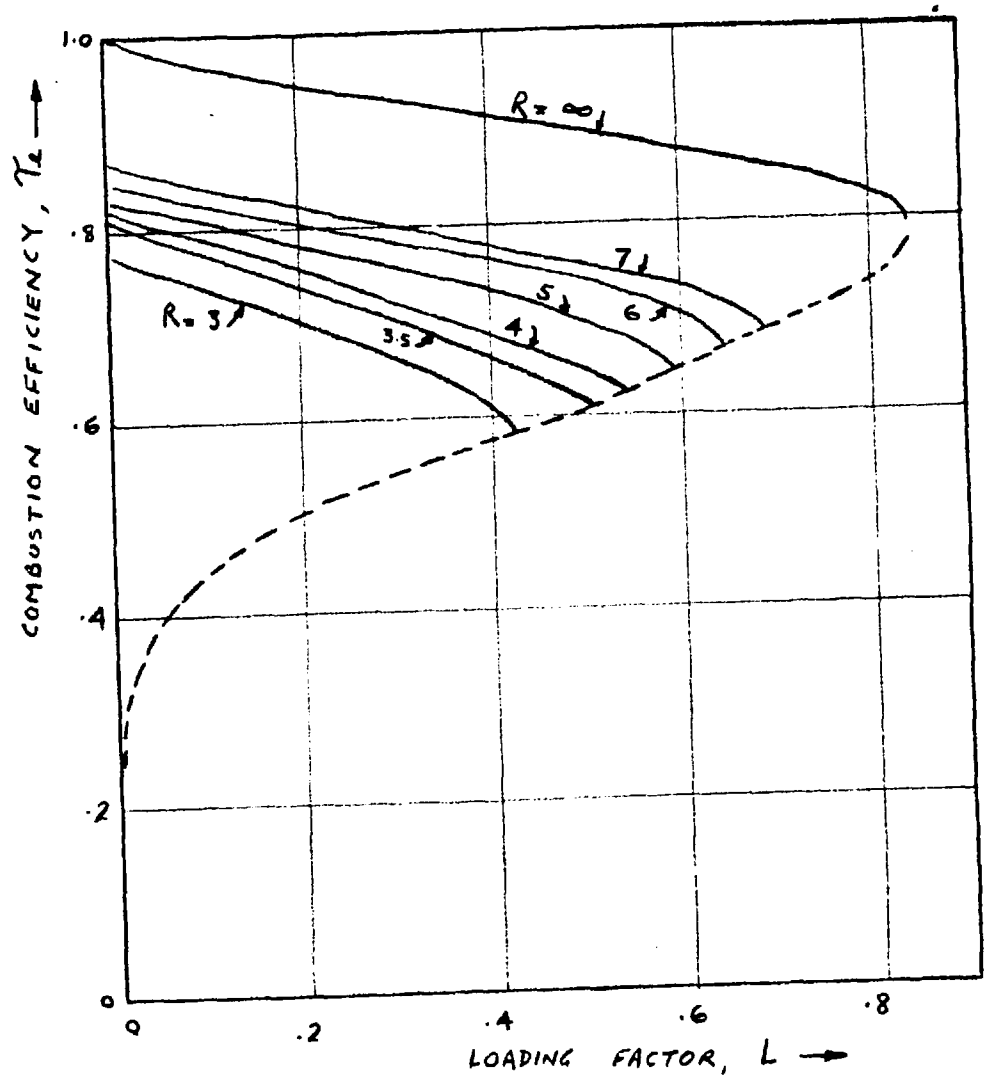
The procedure does not converge when a certain value of L is exceeded for a particular R . This value of L is the extinction loading factor. When it converges, the procedure gives values of the combustion efficiency for particular loading factors and recirculation parameters. The solution also gives the variation of the reactedness in the recirculation zone with the distance from the chamber entrance. The computed results are given in Figs. 49 and 50. From the former, a graph (Fig. 51) is cross-plotted for the variation of the extinction loading factor with the recirculation parameter. This graph shows that the extinction loading factor increases with the recirculation parameter. Since $L_{ext.}$ is zero when R is zero, the broken line of Fig. 51 is drawn as an extension to the computed part of the curve.

Consideration of the definition of the recirculation parameter allows a cross-plot to be made from Fig. 51 in terms of the extinction

loading factor and the annular baffle hole diameter, the latter being expressed as the ratio d/D . Such a plot is given in Fig.52, which indicates the decrease of L_{ext} . as d/D is increased.

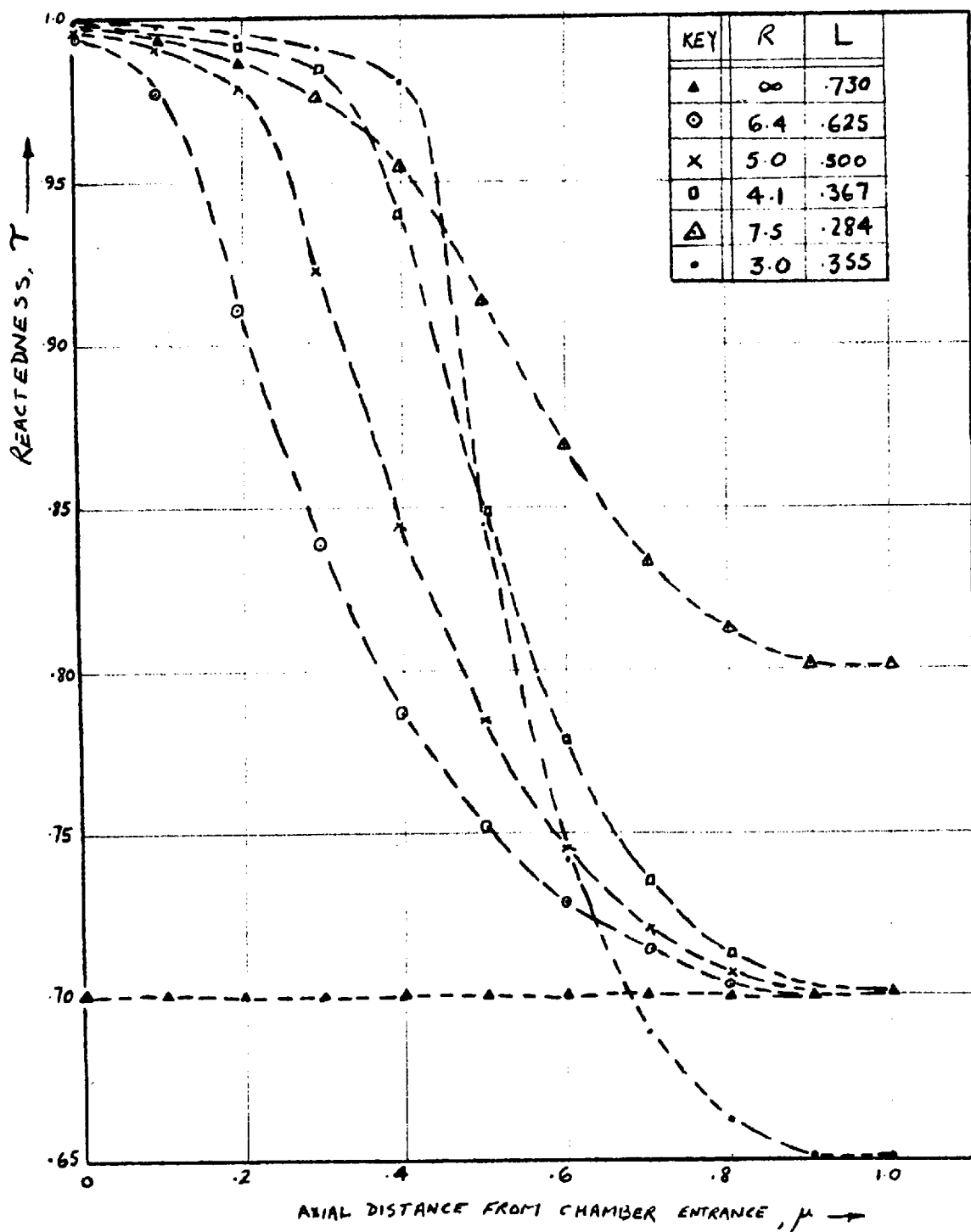
Calculation Of The Pressure Loss.

To complete the picture of calculations for the system with annular baffles, Appendix Three is added on Page 239. It contains calculated values of the flow criterion K_{fv} , introduced in Chapter Three. The resultant K_{fv} values can be used in conjunction with the results of the theoretical analysis of this chapter to complete the information about the performance of the combustion chamber.



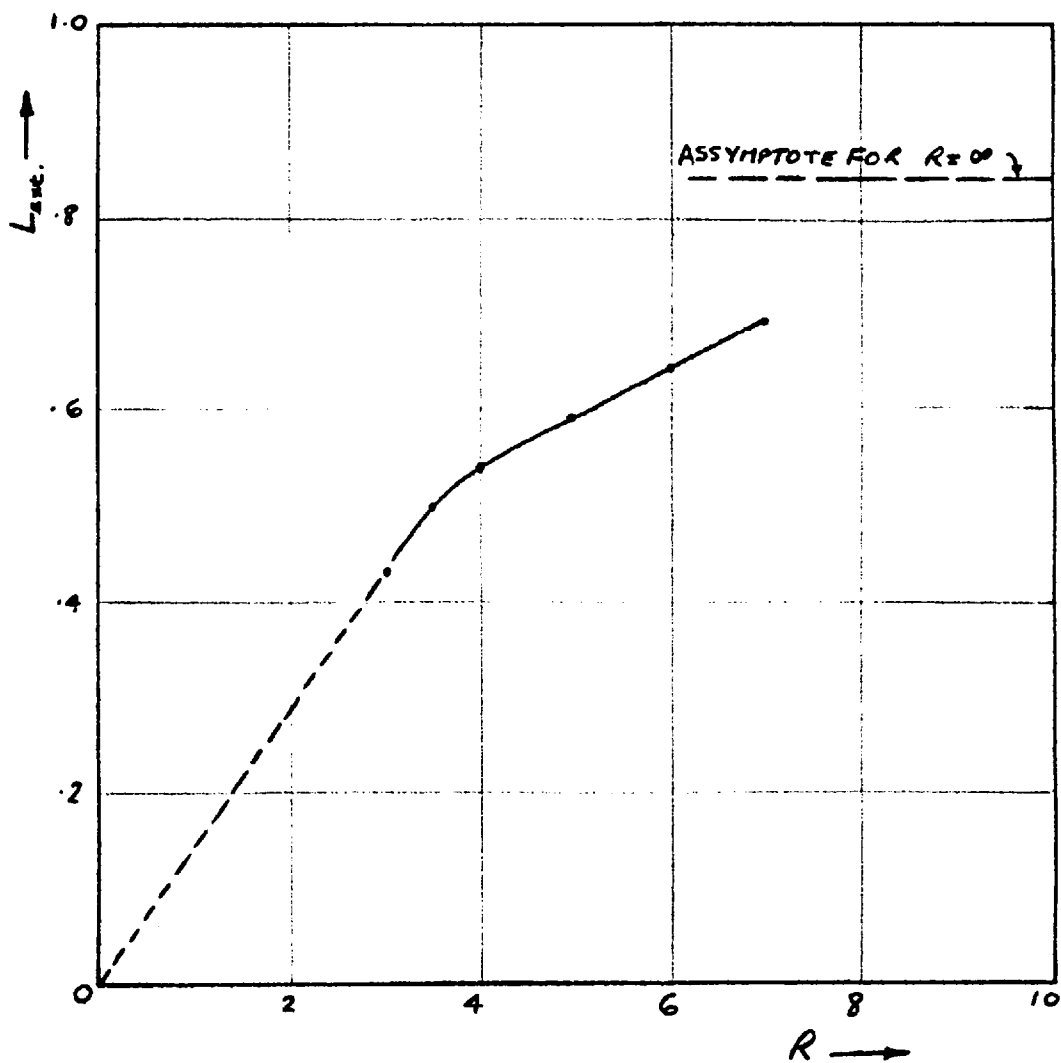
RESULTS OF THEORETICAL ANALYSIS

FIG. 49



REACTEDNESS IN THE RECIRCULATION ZONE [SMILES].

FIG. 50.



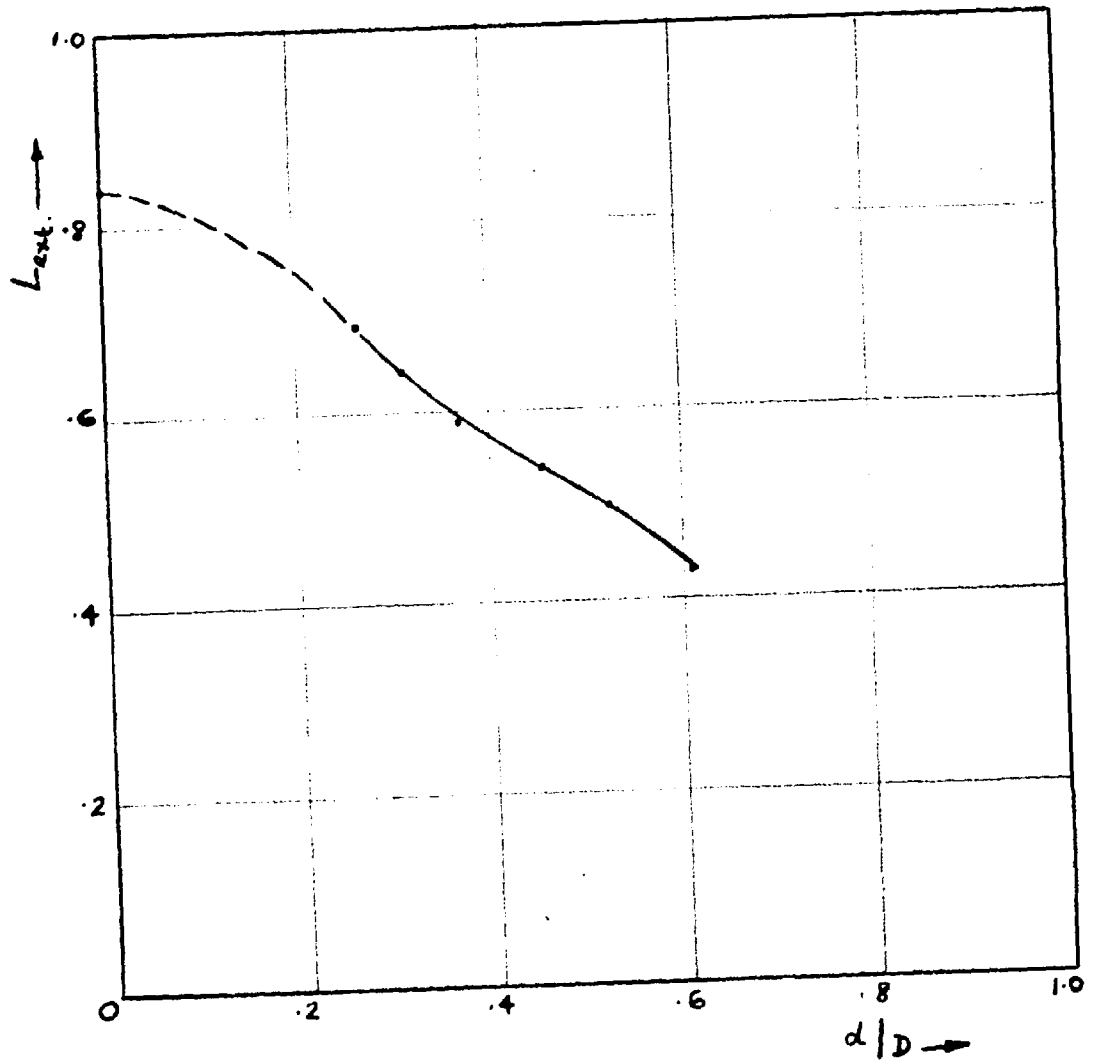
VARIATION OF EXTINCTION LOADING FACTOR
WITH RECIRCULATION PARAMETER.

RESULTS OF THEORETICAL
ANALYSIS. FROM FIG. 49.

$$L = m \phi \Delta T / \dot{q}_{fm}^{\prime\prime} U$$

$$R = k l / d$$

FIG. 51.



VARIATION OF EXTINCTION LOADING FACTOR
WITH ANNULAR BAFFLE HOLE DIAMETER.

RESULTS OF THEORETICAL
ANALYSIS.

$$L = m C_p \Delta T / \dot{q}_m \text{ or}$$

FIG. 52.

CHAPTER SIX.

COMPARISONS AND CONCLUSIONS.

-oOo-

Introduction.

The data presented and discussed in Chapters Three, Four and Five included:

1. The analogue predictions for the flame extinction in the wake of a variety of baffles,
2. Some real flame extinction data,
3. The results of an approximate theoretical analysis of the stabilization of flames in the wake of annular baffles.

In this Chapter, the validity of the analogue predictions is investigated by comparing the resultant trends and when possible, the magnitude of the parameter characterising the performance of the one-stream combustion chambers fitted with the various baffle types.

Hemi-Spherical and Conical Baffles.

The analogue predictions for the variation of the extinction loading with the geometry of baffles of the hemi-spherical and conical types, under fully turbulent conditions, were presented in Chapter Three. The author knows of no published flame stability data corresponding to geometrically similar combustion systems. The available data, included in Chapter Four, were obtained in combustion chambers with a rectangular cross-section. It is therefore impossible to carry out a detailed quantitative comparison between the analogue predictions and the flame stability results. The comparison has to be confined to the general and qualitative aspects of the results.

Hemi-Spherical Baffles.

p 105

The analogue predicts (Fig. 24) that the loading factor at the extinction condition

increases with the hemi-spherical baffle diameter. The magnitude of the loading factor lies between 5 and 12 when defined as:

$$L_D = V \rho u C_p \Delta T / \dot{q}_w''' D$$

the symbols having the significance stated in Chapter Three.

^{p160}
Fig. 45, on the other hand, shows the results of Mestre (Ref.14) plotted in terms of the extinction loading factor and the weak equivalence ratio. It is to be noted that the graphs of Fig. 45 do not suggest that the loading factor becomes independent of the equivalence ratio when this has high values below the stoichiometric. This means that the loading factor is still dependent on the Reynolds Number, and therefore the results cannot be directly compared with the analogue predictions. Nevertheless, Fig. 45 clearly suggests that the trend of the flame stability data is that the extinction loading factor increases with the

baffle diameter. This is the same as the trend predicted by the analogue.

A further point that might be noted is that the extinction loading factors have the same orders of magnitude in both the analogue predictions and the flame stability results. However, this point is not necessarily significant in view of the fact that the flame stability results were not obtained under fully turbulent conditions, and that the combustion chamber used by Mestre was rectangular in cross-section, while the analogue chamber had a circular cross-section.

Conical Baffles.

The analogue predictions and the available real flame results for the extinction of flames in the wake of conical baffles, were introduced in Chapters Three and Four respectively. One trend suggested by the electric analogue is that under fully turbulent conditions, the extinction loading

factor decreases as the baffle apex angle is increased while keeping the base diameter of the baffle constant. Another trend suggested is that for a given apex angle, the extinction loading factor increases with the base diameter. The geometric ranges covered by the analogue experiments were:

- a. Apex angle between 30 and 180 degrees,
- b. Ratio of base diameter to chamber diameter approximately between 0.4 and 0.7.

As far as the change of extinction loading with the base diameter is concerned, the very fact that the real flame results of Mestre could be correlated in terms of V / D suggests that the trend predicted by the analogue is the same as the trend of the real flame results. On the other hand, the change of the extinction loading with the cone apex angle is not clearly established in the case of the real flame results. Mestre's results might

be taken to suggest that the disc has a higher extinction loading than the right-angled cone, while the analogue predicts the reverse. The points that make the flame results doubtful as a basis for comparison with the analogue may be summarised as follows:

1. The chamber used by Mestre had a rectangular cross-section, while the analogue chamber was axi-symmetric.
2. The range of the weak mixtures covered in the real flame experiments lies between equivalence ratios of about 0.6 and 0.8. It is unlikely that the loading factors in this range will be independent of the Reynolds Number. This is suggested by the analysis of the flame results for central discs and hemi-spherical baffles in Chapter Four.
3. The baffles tested during the flame extinction experiments were too small, and fall outside the range covered by the analogue experiments.

The cones tested by the analogue had a ratio of base diameter to duct diameter of approximately between 0.4 and 0.7; while the corresponding equivalent range in the case of the flame experiments was between 0.125 and 0.208.

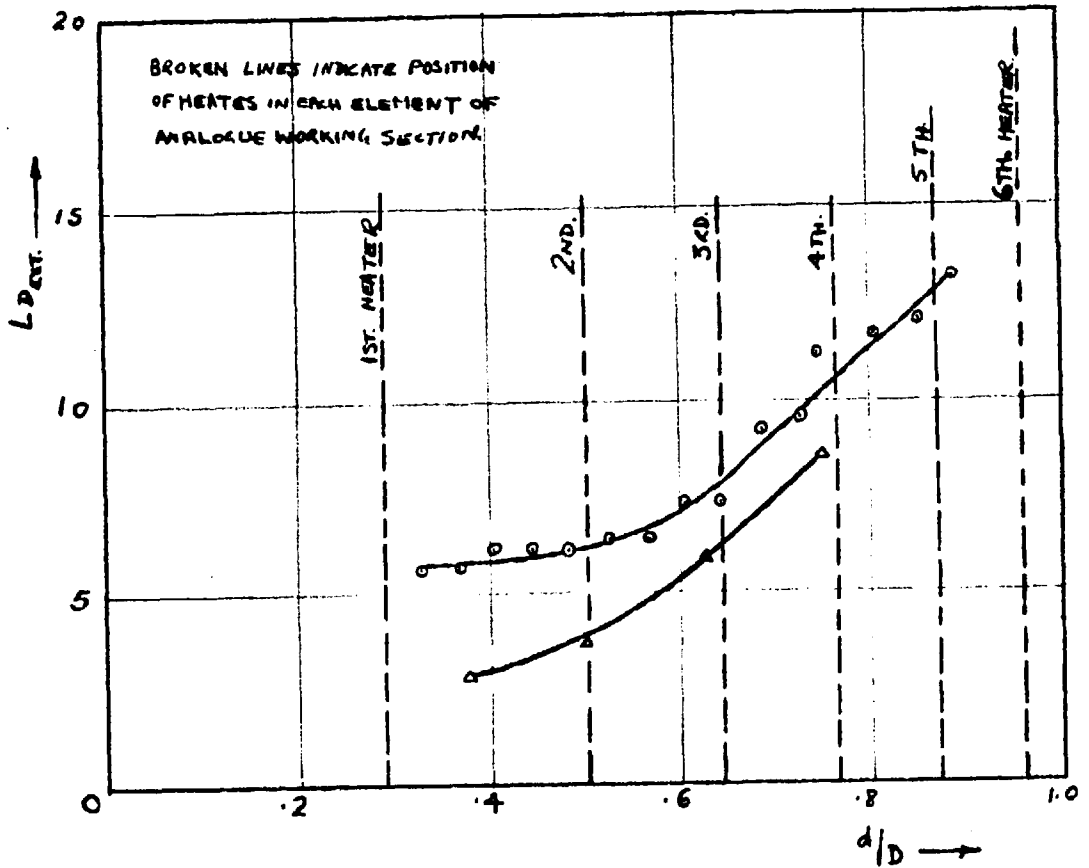
4. There is a considerable amount of scatter in the real flame results.

Thus, as far as the effect of change in the cone angle is concerned, no conclusion can be safely drawn from the comparison between the analogue predictions and the real flame results. Further detailed flame extinction tests are required before such a conclusion may be reached.

Central Discs.

The comparison of the analogue predictions for central discs with the real flame results, is carried out under fully turbulent conditions. When such conditions prevail, the performance of the flame, and the analogue simulation of it, is independent of the Reynolds Number. The flame results and the analogue predictions have already been given in previous Chapters; the former in Fig. 38, and the latter in Fig. 22. These two figures contain graphs showing the variation of the extinction loading factor with the ratio of disc-to-duct diameters; the loading factors being at the flow regime where they become independent of the Reynolds Number. The two graphs are plotted together in Fig. 53 for the purpose of the present comparison. The features to be noted in Fig. 53 are:

1. Qualitatively, the analogue predicts that the extinction loading factor increases with the



COMPARISON OF ANALOGUE PREDICTIONS WITH REAL FLAME RESULTS

CENTRAL CIRCULAR DISC BAFFLES.

○ ANALOGUE PREDICTIONS

FROM FIG. 22.

△ REAL FLAME RESULTS

FROM FIG. 38.

$$L_D = V S_u C_p \Delta T \left| \frac{q_m}{q_m} \right|^{0.11}$$

FIG. 53.

disc diameter. This agrees with the trend shown by the flame results.

2. Quantitatively, the analogue predictions and the real flame results have the same order of magnitude.
3. The analogue predictions are higher than the real flame results.
4. The magnitude of the quantitative difference between the analogue predictions and the real flame results decreases as the disc-to-duct diameter ratio becomes higher, until the latter reaches a value of about 0.6. For d/D ratios higher than this, the discrepancy becomes almost constant; the analogue predictions being then about 25% higher than the real flame results.
5. Below the d/D ratio of 0.6, the analogue predictions show only a small change of

extinction loading factor with disc diameter. Bigger changes are noted in this region in the case of the flame results.

Thus it is seen that the analogue predicts the performance of central discs as flame stabilizers to a good degree of approximation, except in the case of small disc diameters. This insensitivity of the analogue predictions to changes of disc diameter at small d/D ratios may be explained by reference to the distribution of the heaters in the elements of the analogue working section which was described in Chapter Two. It was stated there that in each analogue element there are six heaters, forming six concentric circles in one plane normal to the axis of the analogue. The ratios between the heater circles diameters and the inside diameter of the analogue chamber, are indicated by the vertical dotted lines in Fig. 53. The line marked " first heater " corresponds to the heater nearest the

axis of the analogue chamber, while that marked " sixth heater " corresponds to the heater nearest the wall of the analogue. The remaining dotted lines indicate the four intermediate heaters.

Fig. 53 shows that the increase of the difference between the analogue predictions and the real flame results , occurs in a range of disc to duct diameter ratios where there are not more than two heaters from each element included in the recirculation zone behind the discs. The reason for the deterioration of the analogue predictions for the smaller disc diameter range is, therefore, the crude sub-division of the analogue working section. It may thus be expected that a better agreement between the analogue predictions and the real flame results would be achieved with a finer sub-division.

The conclusions to be drawn from the discussion of this section, are that the analogue can successfully predict the performance of central discs as flame stabilizers under fully turbulent

conditions; that the agreement is both qualitative and approximately quantitative; and that the approximation may be improved with a finer sub-division of the analogue working section.

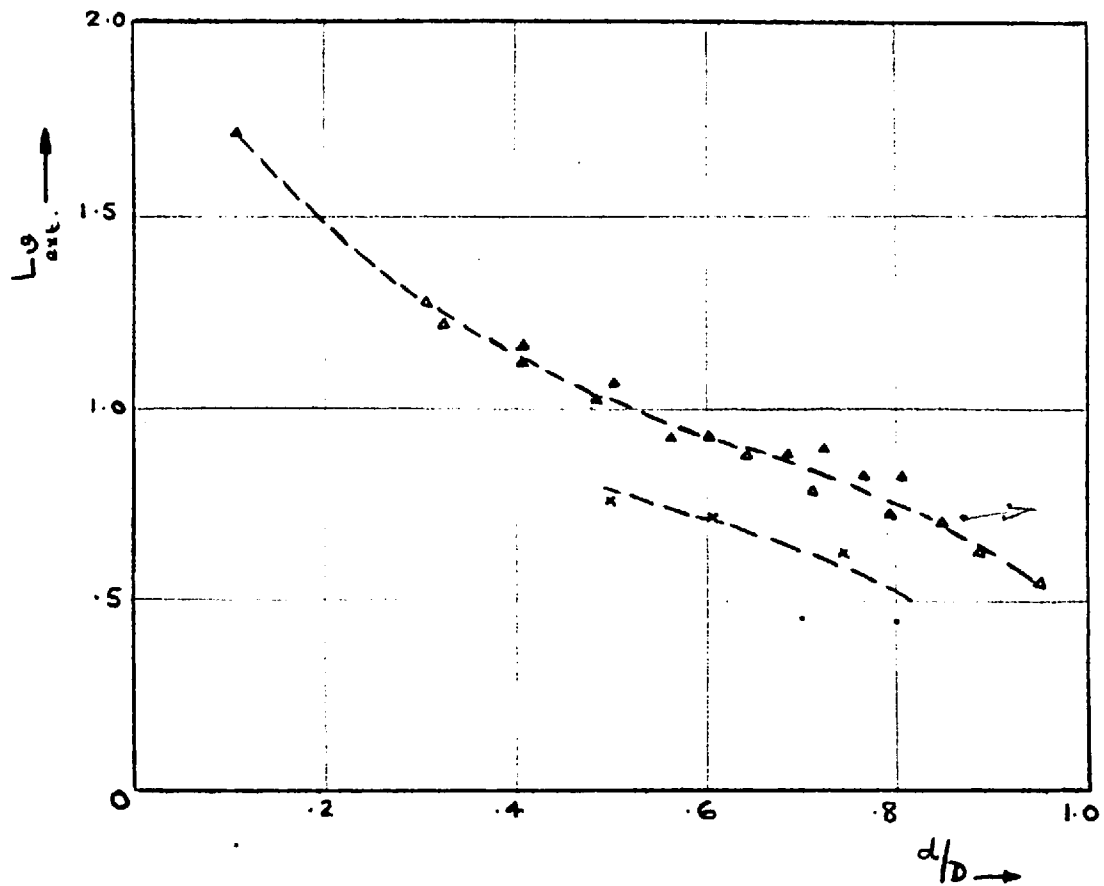
Annular Baffles.

The results being compared in this section were obtained under fully turbulent conditions. This has been ascertained in the discussions concerned with these results in Chapters Three and Four. The analogue predictions for the variation of the extinction loading factor with the ratio of the annular baffle hole diameter to the diameter of the chamber, were given in Fig. 20. The corresponding real flame results were recorded in Figs. 40 and 42. Both the analogue predictions and the real flame results are plotted together in Fig. 54.

A good qualitative agreement is seen between the analogue and the real flame results. The analogue predictions are seen to be some 30 - 40% higher in magnitude than the real flame results. It is to be noted that this percentage discrepancy is somewhat higher than that found in the case of central disc baffles. The reason for this increase

in the magnitude of the discrepancy may be due to the nature of the real flame results, rather than an inherent analogue tendency towards more accurate predictions in the case of central baffles, than in the case of annular baffles. Where annular baffles are concerned, the real flame results may suffer from a small amount of entrainment of surrounding air by the jet issuing from the hole of the baffle. The results which suffer from greater and more serious entrainment effects have already been discarded during the analysis of the available flame stability data in Chapter Four. With central baffles, on the other hand, the entrainment effects are not as important as with annular baffles; partly because there is no jet in the case of central baffles, and partly because the recirculation zone is longer in the wake of the annular baffle than in the wake of the central baffle.

This entrainment effect causes a decrease in the magnitude of the extinction loading factor,



COMPARISON OF ANALOGUE PREDICTIONS WITH REAL FLAME RESULTS.

ANNULAR BAFFLES.

△ ANALOGUE PREDICTIONS
FROM FIG. 20

x REAL FLAME RESULTS
FROM FIG. 40.

• REAL FLAME RESULTS
FROM FIG. 42.

$$L_0 = \frac{m c_p \Delta T}{\dot{q}_m \theta}$$

FIG. 54.

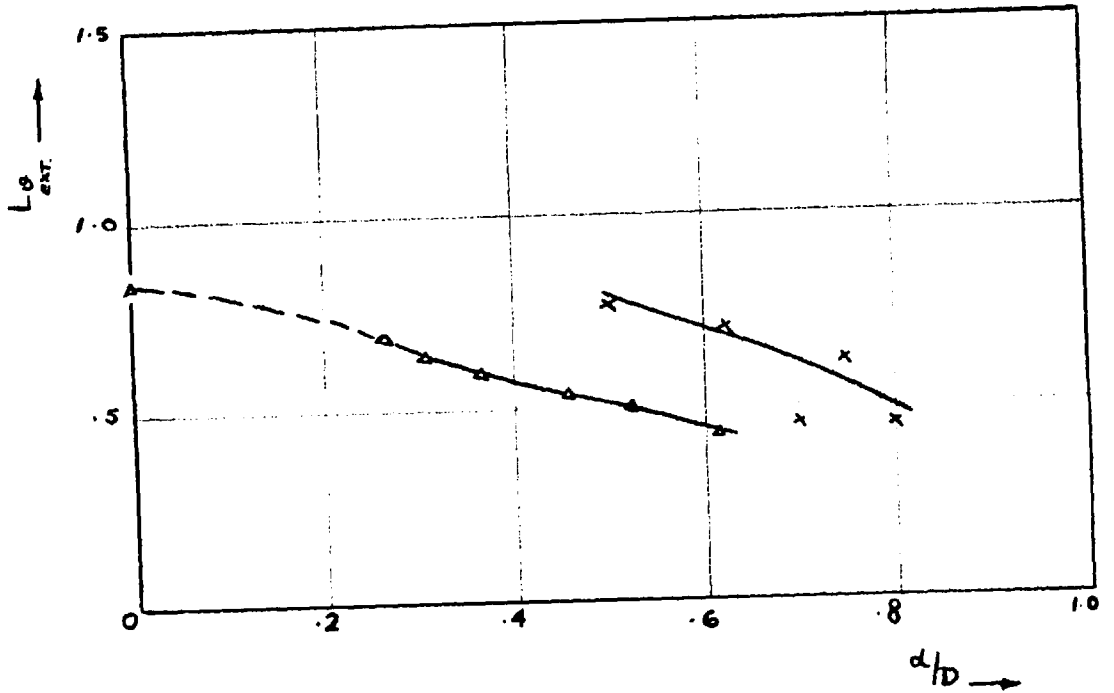
when this is calculated on the basis of the inlet fuel and air flow rates and conditions; hence the increase in the magnitude of the difference between the analogue predictions and the real flame results.

Keeping in mind this entrainment effect, and the similarity conditions which are ignored by the analogue technique, both the qualitative and the quantitative agreements between the analogue and flame results, may be considered satisfactory in the range of annular baffle hole diameters covered by the flame stability tests presented in Chapter Four.

With the air supply capacity available, it was possible to cover a wider range of hole diameters in the tests on the electric analogue. So, in order to provide further basis for comparison with the analogue predictions, the approximate theoretical analysis of Chapter Five will now be used in the discussions following.

Theoretical Analysis and Real Flame Results.

The available data of flame stability for annular baffles cover a range of d/D ratio of between 0.5 and 0.8, while the range covered by the electric analogue for similar systems is much wider, especially in the lower values of d/D . Attempts to obtain flame stability results for baffles with smaller hole diameters have failed in the available flame stability apparatus described in Chapter Two. Hence the theoretical analysis of the system given in Chapter Five. But before comparing the results of this analysis with the predictions of the analogue, it is necessary to compare the analytical results with those of real flames. This comparison is carried out in Fig. 55. It is seen that the two curves are almost parallel to each other when they cover the same d/D ratio, indicating a good qualitative agreement.



COMPARISON OF RESULTS OF THEORETICAL ANALYSIS WITH RESULTS
OF REAL FLAMES ~ ANNULAR BAFFLES.

Δ ANALYTICAL RESULTS.

x REAL FLAME RESULTS.

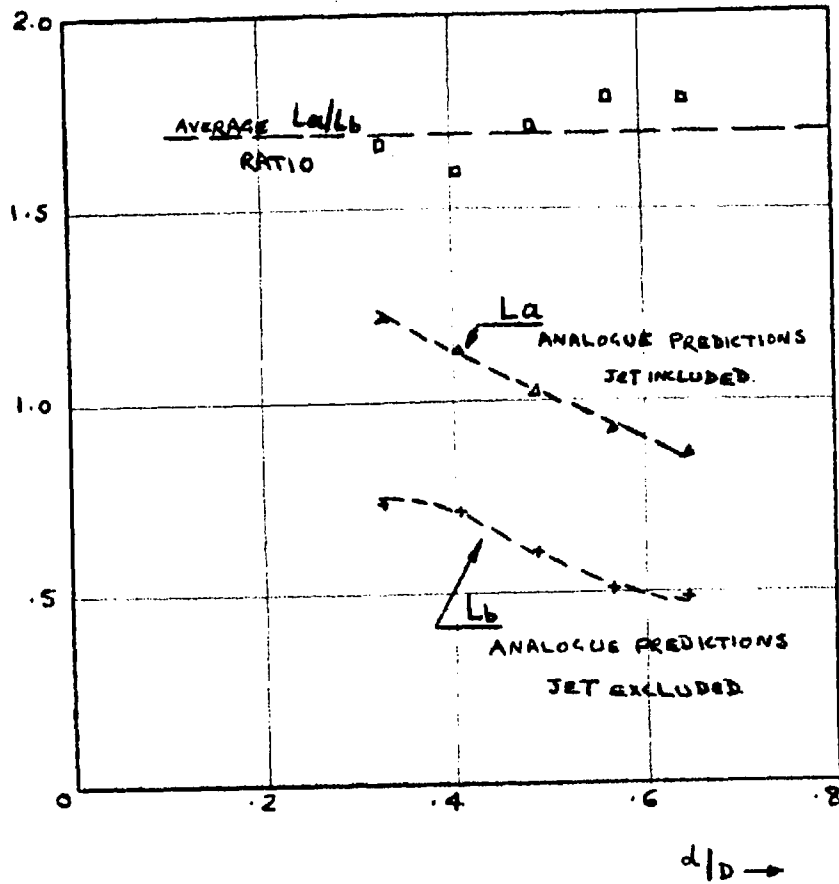
$$L_0 = \frac{\dot{m} C_p \Delta T}{q_{m,0}''}$$

FIG. 55.

Quantitatively, however, the flame stability results are higher than the analytical results by a factor of 1.5 to 1.6. This difference is to be expected, since the analysis ignored the reaction in the jet issuing from the hole of the annular baffle. The magnitude of the effect of the reaction in the jet on the extinction loading, may be estimated by the following experiment on the electric analogue:

Experiment.

The usual procedure for an extinction test on the electric analogue was described in Chapter Two. All the heaters in the working section are switched on at the beginning of the procedure. The present experiment, however, is started with only the heaters in the recirculation zone switched on. The boundaries of the recirculation zone are taken according to the requirements of the postulated flow pattern, which was the basis of the analysis.



EFFECT OF REACTION IN THE JET ON LOADING FACTOR.
(ANNULAR BAFFLES)

RESULTS OF EXPERIMENT
ON ELECTRIC ANALOGUE.

$$L = m \cdot C_p \cdot \Delta T / \dot{q}_m \cdot v$$

FIG. 56.

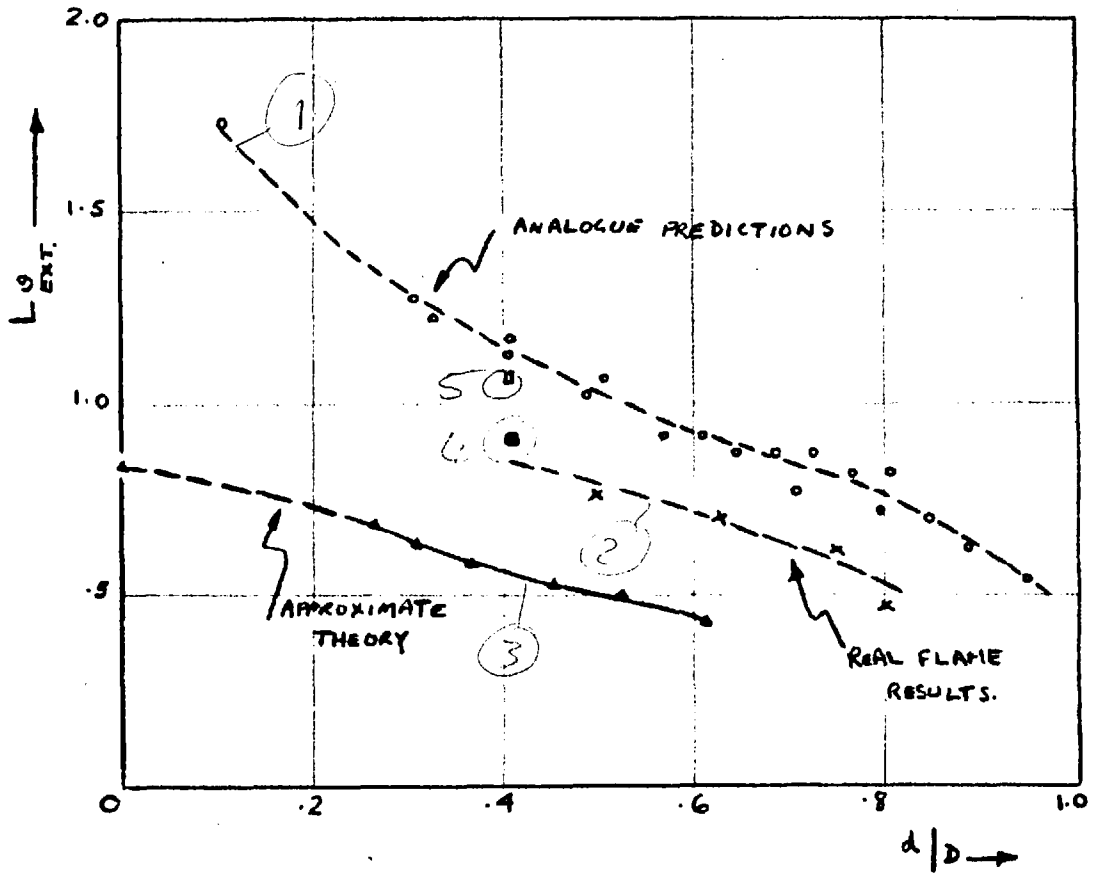
Fig. 56 contains the results of this experiment. The points marked " La " are the extinction loading factors predicted by the analogue at the maximum attainable Reynolds Numbers, these points being taken from Fig. 20. The points marked " Lb " are the analogue predictions at the same Reynolds Numbers, obtained with only the heaters in the recirculation zone being switched on at the start of the experiment. The extinction loading factors La and Lb are both based on the volume of the chamber downstream the annular baffle. The ratio La/Lb for the various baffles tested in this experiment, are also plotted in Fig. 56. It is seen that this ratio has values between 1.6 and 1.8; the average being about 1.7. Thus, a value of 1.7 may be taken as a constant value for the ratio La/Lb.

This result may be taken to indicate the magnitude of the effect of the reaction in the jet on the results of the theoretical analysis. From it may be drawn the conclusion that this approximate analysis gives an under-estimation of the extinction loading factor by about 1 to 1.7. It

should be remembered here that the magnitude of the discrepancy between the approximate analysis and the real flame results, was previously found to be a factor of about ($1/1.5$) to ($1/1.6$). This factor is close to the factor suggested by the experiments ignoring the reaction in the jet.

In fact, this point is further confirmed by the solution of the sets of influence equations by the computer mentioned in Chapter Five. For a baffle with d/D ratio of about 0.4, the extinction loading factor given by the computer (when this is fed by a set of influence coefficients calculated on the basis of the postulated flow pattern) is about 0.9. Such a value of extinction loading factor lies very close to the curve drawn through the results of the real flame extinction tests. This is indicated in Fig. 57, which contains plots of L_{ext} . versus the d/D ratio obtained as follows:

1. The predictions by the electric analogue -
2. The real flame extinction tests results -



RESULTS FOR ANNULAR BAFFLES.

$$L_0 = \frac{m c_p \Delta T}{\dot{q}_m \phi}$$

FIG. 57.

3. The results of the approximate theoretical analysis -
4. The computer solution to a set of influence equations; the influence coefficients being calculated on the basis of the flow pattern suggested in Chapter Five -
5. The average of the computer solutions of sets of influence equations; the influence coefficients being measured on the electric analogue at various flow rates -

The heat release rate used in the analogue experiments was given by the step-function of Fig. 2, while the more realistic function of Fig.1 was used both in the computer and in the solution of the equations resulting from theoretical analysis.

The general picture that emerges from Fig. 57, and from the preceding discussions of this section on annular baffles, is summarised in the following remarks:

1. The same trend of variation of the extinction

loading factor with the annular baffle hole diameter, is shown to be the same by the results of the approximate theoretical analysis, the results of real flame extinction tests, and the predictions of the electric analogue. This conclusion is, in itself, a positive point with regard to the validity of the analogue predictions.

2. The magnitude of the quantitative difference between the results of the theoretical analysis and the real flame results, is explained by the approximation involved in the analysis, namely, the fact that it ignored the reaction in the jet.
3. The computer solution based on the calculated ^{Point 6} influence coefficients, ^{measured RR and} is very close both to the real flame results, ⁽²⁾ and to the results given by the theoretical analysis ⁽³⁾ multiplied by the factor ^{1.7} deduced from the experiment described on Page 211.
4. The average computer solution based on the measured

influence coefficients, is just below the results predicted by the electric analogue for the particular baffle. The difference in this case is due to experimental error, and to the shapes of the heat release functions used. The magnitude of the discrepancy due to the shape of the heat release rate function is discussed later in this Chapter.

5. The general conclusions which may be drawn from the present section, are:
 - a. Qualitatively, the electric analogue successfully predicts the influence of the baffle hole diameter on the extinction characteristics of turbulent flames in the wake of annular baffles,
 - b. The magnitude of the extinction loading factors predicted by the analogue are about 35% higher than the real flame results.

6. With regard to the magnitude of the discrepancy between the analogue predictions for the extinction loading and the available flame results, it is to be noted that the latter may suffer from a small amount of entrainment of surrounding air. Such an entrainment is not present in the electric analogue, and therefore it may be responsible for part of the discrepancy.

Summary of Comparisons Between Real Flame Results
and Analogue Predictions for Central Discs and
Annular Baffles.

The preceding remarks on the comparisons for discs and annular baffles may be taken to indicate the following:

1. That the analogue predictions are qualitatively similar to the real flame results.
2. That the magnitude of the extinction loadings predicted by the analogue is higher than the real flame results by:
 - a. About 25% in the case of central disc baffles,
 - b. About 35% in the case of annular baffles.
3. Because of the small entrainment of the ambient air by jets issuing from the holes of annular baffles, the magnitude of the difference may be

taken as being approximately the same with both annular and central disc baffles; perhaps in the order of 30%.

This quantitative difference between the analogue predictions and the real flame results is partly due to the shape of the heat release rate function used by the analogue, namely the step-function. Some indication of the approximation involved in the use of this function may be obtained by considering the extreme case of a homogeneous reaction zone, in which the volumetric rate of heat release is represented by:

- a. The step-function,

$$\psi = 0 \quad 0 < \tau < 0.7$$

$$\psi = 1 \quad 0.7 < \tau < 1.0$$

- b. The more realistic continuous function,

$$\psi = 149 \tau^8 (1 - \tau^2)$$

Using the usual notation, a heat balance of such a homogeneous reaction zone under adiabatic conditions gives:

$$\dot{q}'' v = \dot{m} C_p (T - T_u) \text{-----(40)}$$

or, in dimensionless form

$$\psi = L \tau \text{-----(41)}$$

where:

$$L = \text{the loading factor} = \dot{m} C_p \Delta T / \dot{q}''_m v.$$

Equation (41) is represented by straight lines through the origin of a $(\psi \sim \tau)$ graph. The slope of these lines increases with the loading factor, and each line is called a "heat demand line" (Ref.3) or, "the line representing the enthalpy change of the gas supply" (Refs. 1 and 22).

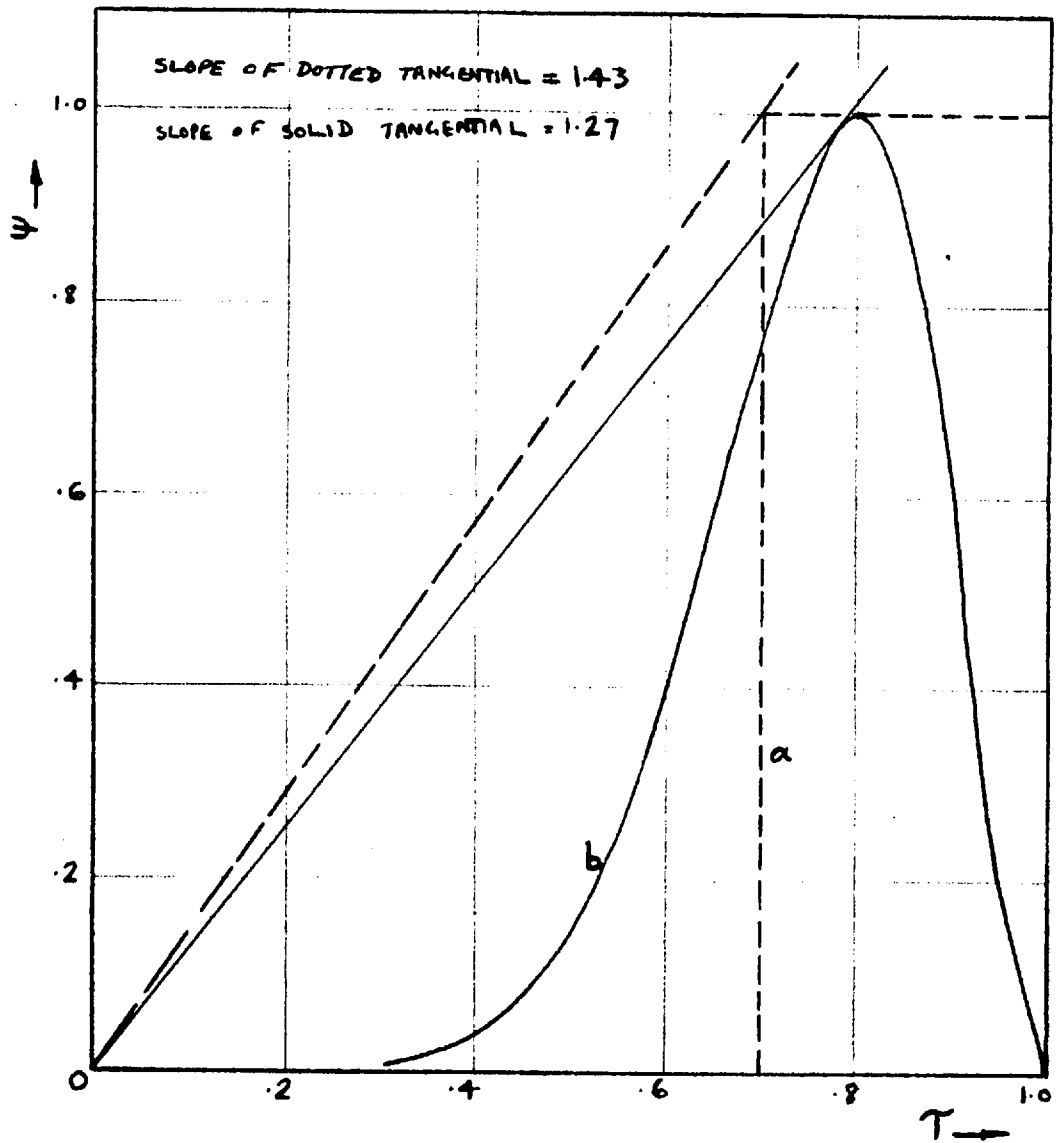
The equilibrium, or stable flame, condition in the zone is determined by the right-hand intersection of the heat demand line with the curve representing the heat release rate function. Extinction of the flame is reached when the loading factor

has increased until the heat demand line becomes tangential with the heat release rate curve. The slope of the heat demand line then gives the extinction loading factor.

Fig. 58 shows the heat release rate functions (a) and (b), together with their respective tangential heat demand lines. The slopes of these two lines are (1.43) and (1.27) respectively; their ratio being (1.12). Thus, the step-function is responsible for an over-estimation, of about 12% in the value of L_{ext} . above the value that would be obtained if the realistic heat release rate function were used.

Apart from this factor, due to the shape of the analogue heat input curve, absolute quantitative agreement between the analogue predictions and the real flame results are not to be expected because of the approximations discussed in Chapter One. The fact that all the approximations made caused no more discrepancy than about 20%, may be taken to affirm

the validity of the analogue technique in the prediction of the stability characteristics of flames in the wake of central discs and annular baffles.



HEAT RELEASE RATE FUNCTIONS AND THEIR CORRESPONDING
TANGENTIAL HEAT DEMAND LINES.

FIG. 58

APPENDIX ONE.

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-oOo-

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APPENDIX TWO.

DATA FOR THE CALCULATION
OF
FLAME STABILITY PARAMETERS.

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The required data for the calculation of flame stability parameters, are the velocity of the fuel and air mixture at the extinction condition (V), the product of the adiabatic combustion temperature rise and the specific heat at constant pressure (C_p) and the maximum volumetric heat release rate ($\dot{q}_{\text{max}}''''$). These were taken as follows:

1. The Velocity, (V).

This was considered to be the average velocity past the baffle. In the case of central baffles, for example, its product with the mixture density upstream the baffle, was taken as:

$$V \rho_u = \frac{\dot{m}}{\frac{\pi}{4} (D^2 - d^2)} \dots \dots \dots (2.1)$$

where:

- $\dot{m}$ = mass flow rate of mixture into the chamber,
- D = diameter of the combustion chamber,
- d = diameter of central baffle.

2. The Product $C_p \Delta T$.

This was calculated from the lower calorific value of the fuel, using the following symbols:

$\dot{m}_f$ = mass flow rate of fuel,

$\dot{m}_a$ = mass flow rate of air,

LCV = lower calorific value of fuel.

The rate of heat release due to adiabatic combustion is given by $\dot{m}_f \times \text{LCV}$. The Law of Conservation of Energy therefore gives:

$$(\dot{m}_f)(\text{LCV}) = (\dot{m}_f + \dot{m}_a) C_p \Delta T \quad \text{-----} \quad (2.2)$$

Equation (22) assumes the same specific heats for the air, the fuel, and the mixture; and for propane as fuel, it leads to:

$$C_p \Delta T = 707 \left(\frac{1}{\phi} + 0.0638 \right) \quad \text{-----} \quad (2.3)$$

LCV was taken from Ref.23, and Equation 23 is plotted in Fig.21 as $C_p \Delta T$ versus ϕ .

3. The Maximum Volumetric Heat Release Rate.

The data of Longwell and Weiss (Ref.21), for iso-octane in a spherical reactor, were taken as the basis for the calculation of $\dot{q}_m'''$ at a given inlet temperature and equivalence ratio. These data have been used because of the lack of detailed similar information for the other fuels used in the reported flame stability results.

Fig.22 shows $\dot{q}_m'''$ at inlet temperature of 400° K. This has been calculated from the above data by Bragg and Holliday (Ref. 4). Fig.23 shows the variation of a factor β with the inlet temperature T_u . β is defined below, using the symbols of Ref.21:

$$\beta = \frac{(N/V_{pp})_{\text{max. at inlet temp. } T_u^{\circ} \text{ K}}}{(N/V_p)_{\text{max. at inlet temp. } 400^{\circ} \text{ K}}}$$

where:

N = molal flow rate of air,

V = volume of the combustor in litres,

p = pressure in atmospheres,

T_u = mixture inlet temperature in degrees K.

Figs.22 and 23 allow the maximum volumetric heat release rate to be estimated for a particular mixture inlet temperature and fuel/air ratio.

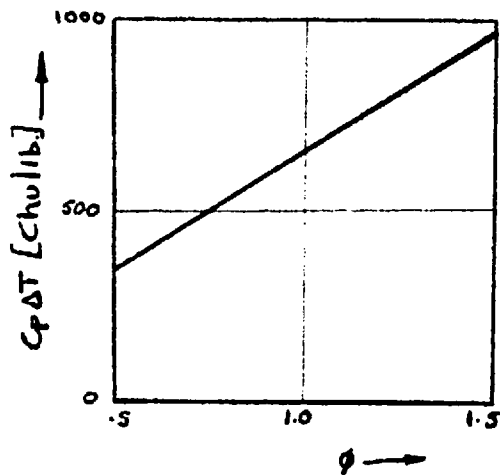


FIG. 2.1.
VARIATION OF $(C_p \Delta T)$ WITH
EQUIVALENCE RATIO ϕ .

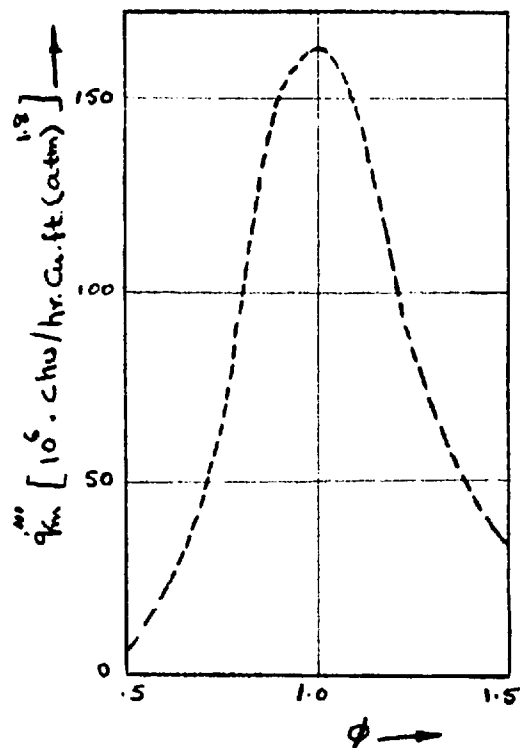


FIG. 2.2
VARIATION OF MAXIMUM VOLUMETRIC HEAT
RELEASE RATE WITH EQUIVALENCE
RATIO. $[T_u = 400^\circ K]$ - REF. 4.

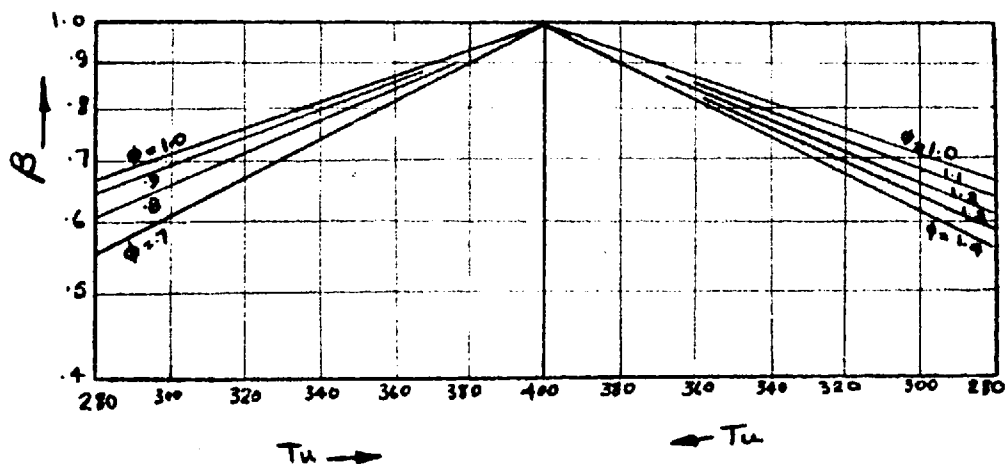


FIG. 2.3 - VARIATION OF β WITH T_u - REF. 21.

APPENDIX THREE

CALCULATION OF THE FLOW CRITERION.

The flow criterion, Kfv, is defined by :

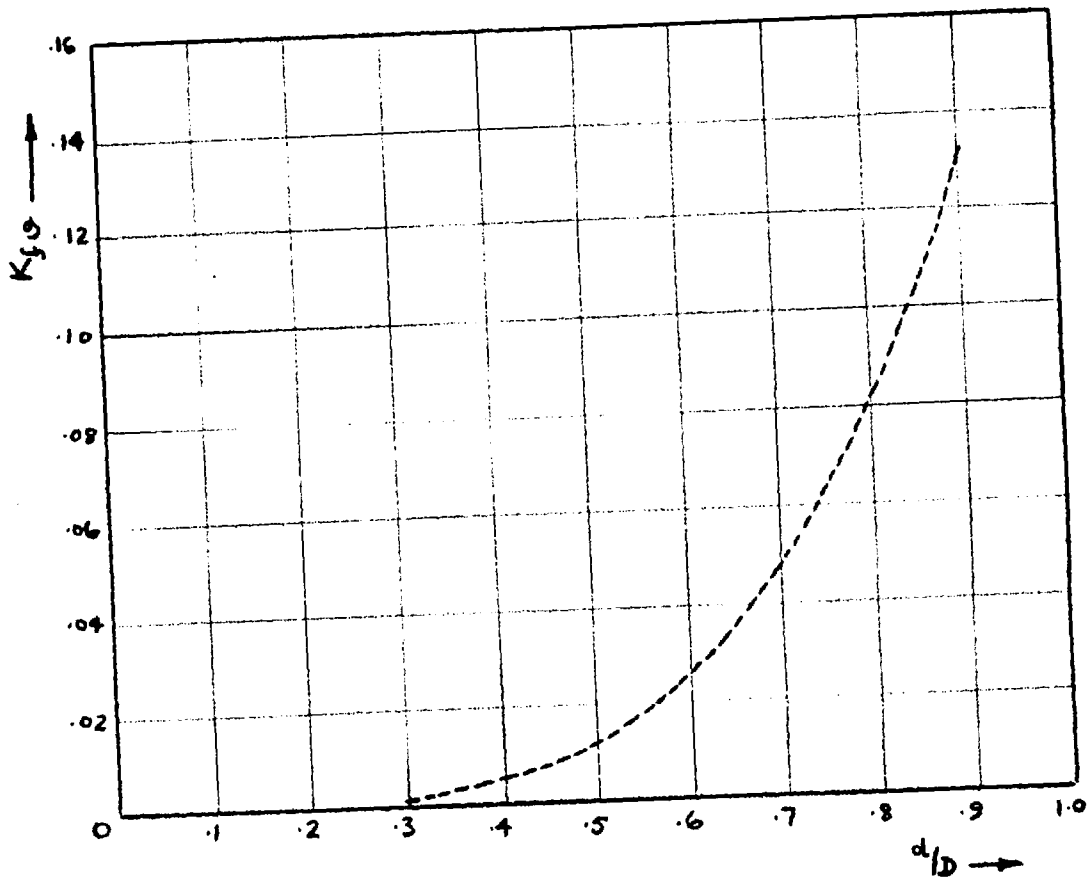
$$K_{fv} = \dot{m}^2 / \Delta P \rho_u g_o v^{4/3}$$

(the notation is the same as in Chapter Three.)

The stagnation pressure drop, Δp , is assumed to be related to the average velocity of the gases entering the combustion chamber; namely, being equal to one velocity head. The flow criterion then becomes :

$$K_{fv} = 0.099 (d/D)^4$$

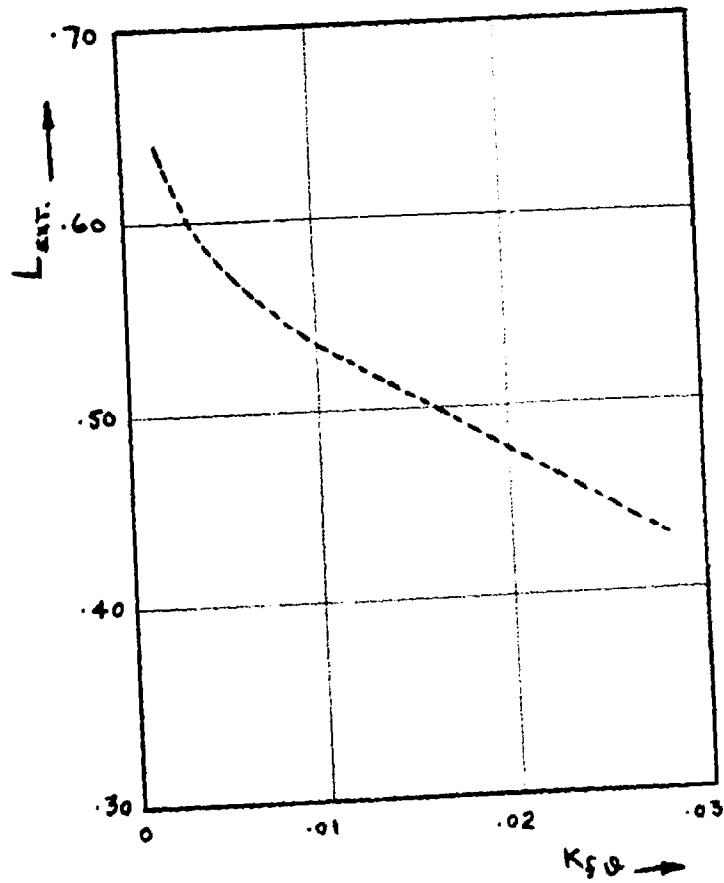
This is plotted in Fig. 3.1, as Kfv v. d/D. Now Fig. 52 gave the results of the theoretical analysis of the system, in terms of the extinction loading factor and the d/D ratio. From Figs. 3.1 and 52, a graph may be constructed showing the variation of Lext. with Kfv. This is drawn in Fig. 3.2. The curve here has the expected shape : it is impossible to combine good combustion characteristics (high Lext.) with low pressure loss (high Kfv.)



VARIATION OF THE FLOW CRITERION WITH THE
ANNULAR BAFFLE HOLE DIAMETER.

$$K_{f0} = \frac{.2}{m} \frac{1}{\Delta p \rho_1 g_0^{4/3}}$$

FIG. 3.1.



VARIATION OF THE EXTINCTION LOADING FACTOR
WITH THE FLOW CRITERION.

$$L = \frac{\dot{m} G \Delta T}{\dot{q}_m \vartheta}$$

$$K_{f0} = \frac{\dot{m}^2}{\Delta p \vartheta \vartheta^{4/3}}$$

FIG. 3.2.

" The Electric Analogue For Turbulent Flame Studies: An
Investigation Of Its Validity. "

ADDENDUM TO THESIS

Covering Points Raised at the Oral Examination held on the
29th March 1961.

Abdulla Abdul Karim El-Bahadli, B.Sc.
September 1961.

ACKNOWLEDGEMENTS.

The author gratefully acknowledges the helpful guidance of Prof. D.B.Spalding, M.A., Ph.D., A.M.I.Mech.E., in the preparation of this Addendum.

Thanks are also due to Dr. D.Vortmeyer of the Imperial College for many useful discussions and assistance during the course of this research and the writing of the Addendum.

1. Influence of Wires on the Flow Pattern.

One of the assumptions implicit in the use of the electric analogue technique is that the heaters, thermocouples, and their supports inside the analogue working section have a negligible effect on the flow pattern of the gases passing through the analogue chamber. It is therefore necessary to demonstrate the validity of this assumption for the analogue model. Convincing evidence of the correctness of the assumption would be the closeness of influence coefficients, α_{jk} , measured for pairs of points j and k inside the working section, under the following conditions;

- a. With the wires and their supports in position,
- b. With the wires removed from the working section.

The definition, and procedures for the measurement of the influence coefficients are given by Spalding in Ref.6., and are quoted in the thesis in Chapters One and Five.

However, the only measurements made on the present axi-symmetric analogue model with the effects of the wires in mind, are the measurements of dynamic and static pressure drops across a length of the working section, with and without the wires and their supports. These measurements are given later in this discussion. It is nonetheless appreciated that such measurements cannot provide a conclusive test of the validity of the assumption about the influence of the wires in the analogue. Together with the other data discussed below, they can only be taken as pointers in the absence of conclusive proof.

The data used here consist of the following:

- a. Predictions by two axi-symmetric analogue models whose working sections contain wires and carriers of different sizes. In the thesis, these models were referred to as the " Lucas " and "present " analogue models.
- b. Dynamic pressure drops along the axis of the working section of the present analogue model with and without the wires.
- c. Static pressure drops along the wall of the present analogue model with and without the wires.

a. Predictions By The Axi-Symmetric Models.

The sizes of the heating wires and the means by which they, and their corresponding thermocouples, are connected to the controls outside the analogue working section, are different in the Lucas and the present analogue models. The working sections of these two models are described in Ref.13. and the thesis respectively. Despite these differences, the two models give similar predictions for the extinction of flames in the wake of annular baffles.

Fig.20. (p. 92. in the thesis) shows the variation of the extinction loading factor with the ratio of the annular baffle hole diameter to the analogue chamber diameter, as predicted by the two analogue models. The predictions were at flow rates high enough to make the extinction loading factors independent of the Reynolds Number. The Figure reveals the closeness of the predictions made by the two models. The relation between the extinction loading factor and the influence coefficients is discussed by Spalding in Ref.6. Accordingly, Fig.20. may be

taken to indicate similar influence coefficients in the wake of annular baffles with equal hole/duct diameter ratios in the two models. This closeness of influence coefficients can signify one of two points:

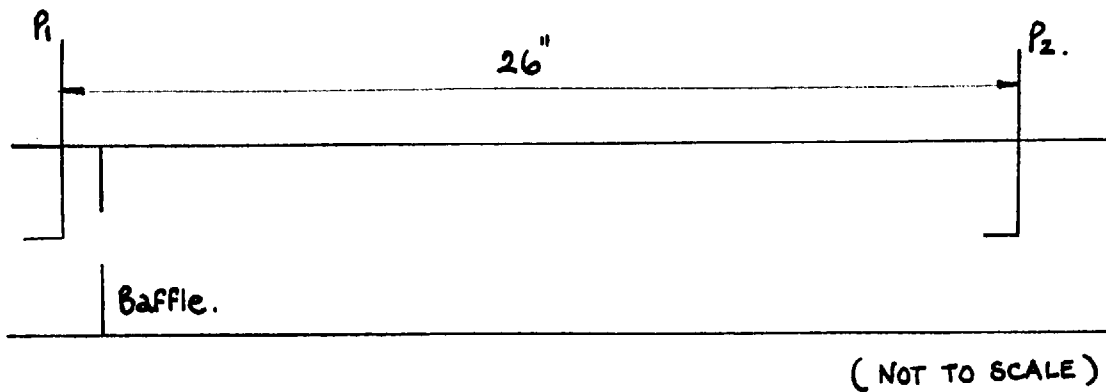
1. That in both models the wires and their supports have negligible influence on the flow pattern,
2. That the influence on the flow pattern is similar in the two models.

Further data are necessary to establish which of these two points is valid; but because of the differences in the size and disposition of the wires and thermocouples in the two models, the first point is perhaps the more likely.

b. Dynamic Pressure Drops Along the Analogue Chamber.

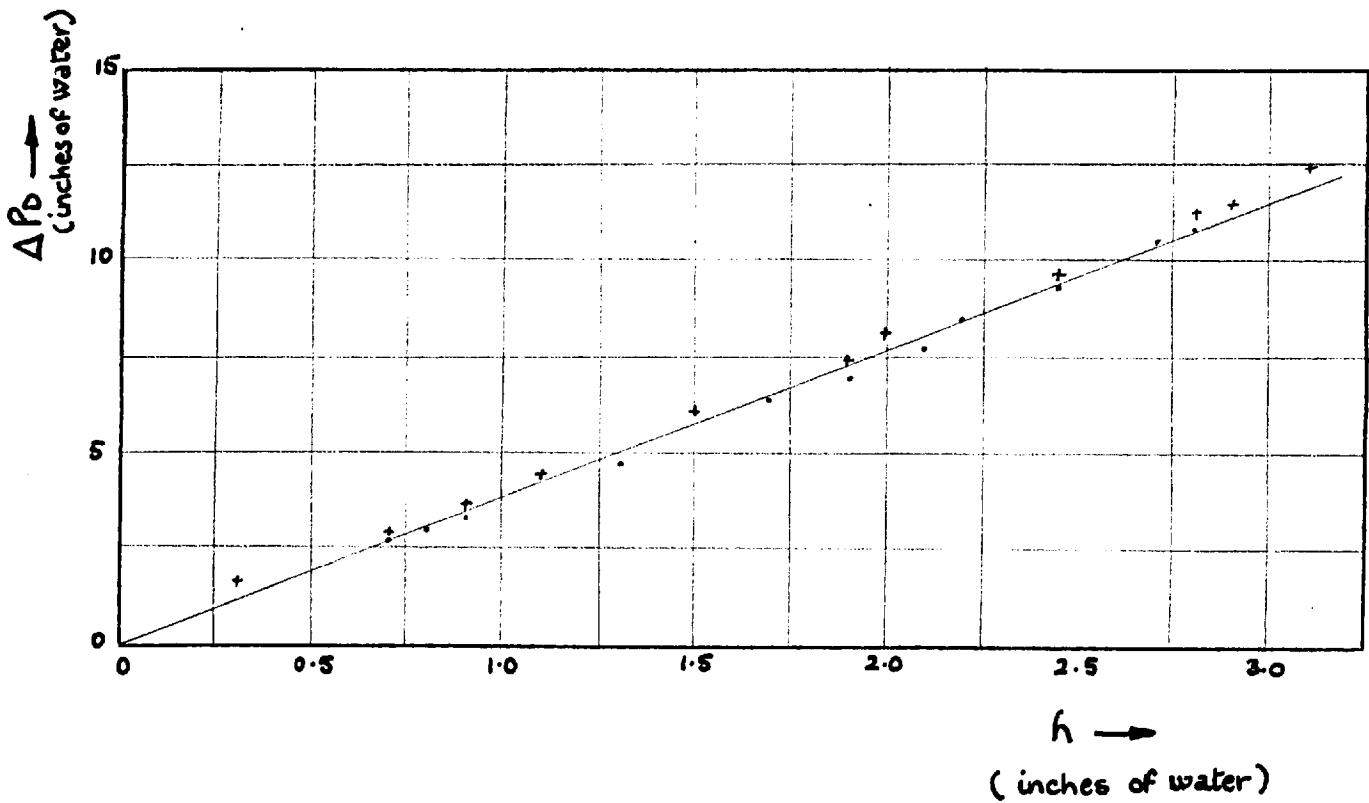
Fig.A1. indicates the experimental set-up. An annular baffle of 2.5" hole diameter was placed in position; P_1 and P_2 being two Pitot tubes placed at distances of 1" upstream and 25" downstream the baffle respectively. As shown in the figure, the small arms of the Pitot tubes are at the axis of the analogue elements; the other ends being connected to the two legs of a manometer whose pressure differential measures the dynamic pressure drop, ΔP_D

Fig.A2. shows the variation of ΔP_D with h , the pressure differential across the British Standard Orifice measuring the air flow rate through the working section. This quantity, h , is proportional to the square of the mass flow rate as shown in point 2 of this Addendum.



SET UP FOR THE MEASUREMENT OF DYNAMIC
PRESSURE DROP ALONG THE ANALOGUE AXIS.

Fig. A1.



VARIATION OF DYNAMIC PRESSURE DROP ALONG THE ANALOGUE
AXIS WITH THE AIR FLOW RATE.

Fig. A2.

In Fig. A2:

- + denotes values measured with the heaters and thermocouples in position, and
- denotes values measured when the heaters and thermocouples were removed from the working section.

It may be concluded from Fig. A.2. that the increase in the dynamic pressure drops, between the two points at the axis of the elements, due to the presence of the heaters and thermocouples is negligible for the range of flow rates covered by these experiments. (This is the same range used in the analogue extinction tests reported in the thesis.)

c. Static Pressure Drops along the Wall of the Analogue Chamber.

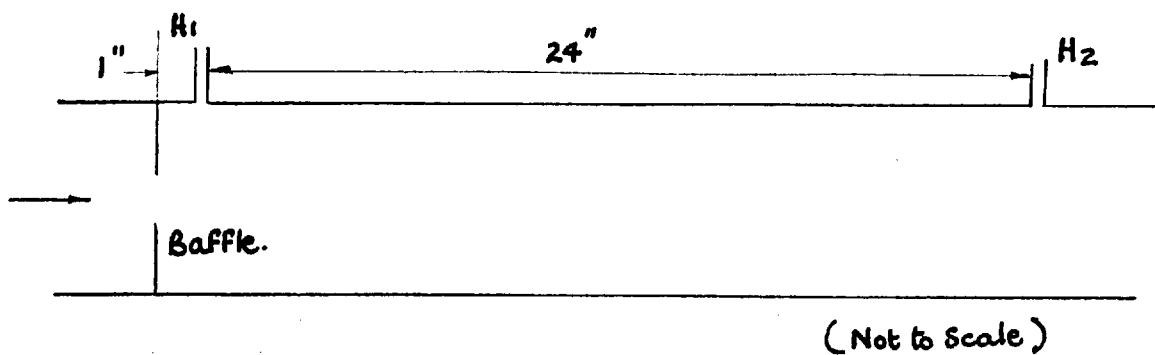
In Fig.A3., H_1 and H_2 are two taps, 24 " apart, with H_1 at a distance of 1 " downstream the annular baffle. H_1 and H_2 are connected by rubber tubing to a manometer which therefore measures the difference in static pressure at the points H_1 and H_2 , ΔP_{sw} . Fig.A4. shows the variation of ΔP_{sw} with h (defined before). In this Fig:

- + denotes values measured with the heaters and thermocouples in position, and
- denotes values measured with the heaters, etc. removed from the working section.

(it is to be noted that the static pressure at H_2 was always higher than that at H_1 , and thus

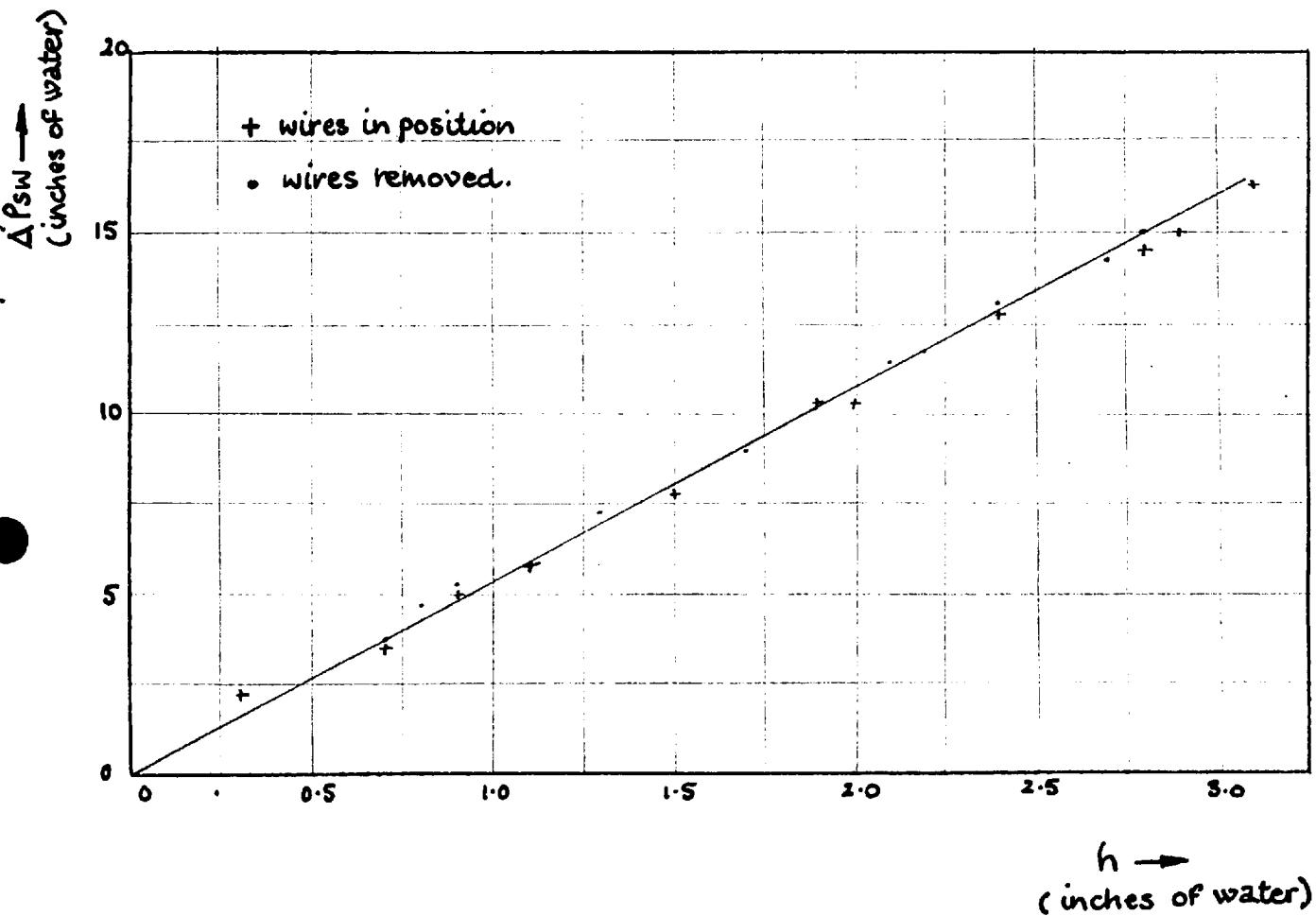
$$\Delta P_{sw} = \text{static pressure at } H_2 - \text{static pressure at } H_1).$$

Fig.A4. indicates that the presence of the heaters, thermocouples, and their supports, have a negligible effect on ΔP_{sw} .



SET UP FOR THE MEASUREMENT OF STATIC
PRESSURE DROP ALONG THE ANALOGUE WALL.

Fig. A3.



VARIATION OF STATIC PRESSURE DROP ALONG THE CHAMBER
WALL WITH THE AIR FLOW RATE.

Fig. A4.

d. Conclusion.

The available data regarding the influence of the wires on the flow pattern are not conclusive. They do not however suggest the invalidity of the assumption that this influence is negligible. Nevertheless, measurements of influence coefficients with and without the wires are still necessary to reach a conclusion on this point.

2. Measurement of Air Flow Rate.

The measurement of Air flow rate in the analogue apparatus is discussed below under two headings:

a. British Standards Requirements.

The measuring device was an orifice plate with D and D/2 taps made in accordance with the British Standards given in B.S. 1042: 1943. Fig A5. is a diagrammatic representation of the metering part of the analogue apparatus.

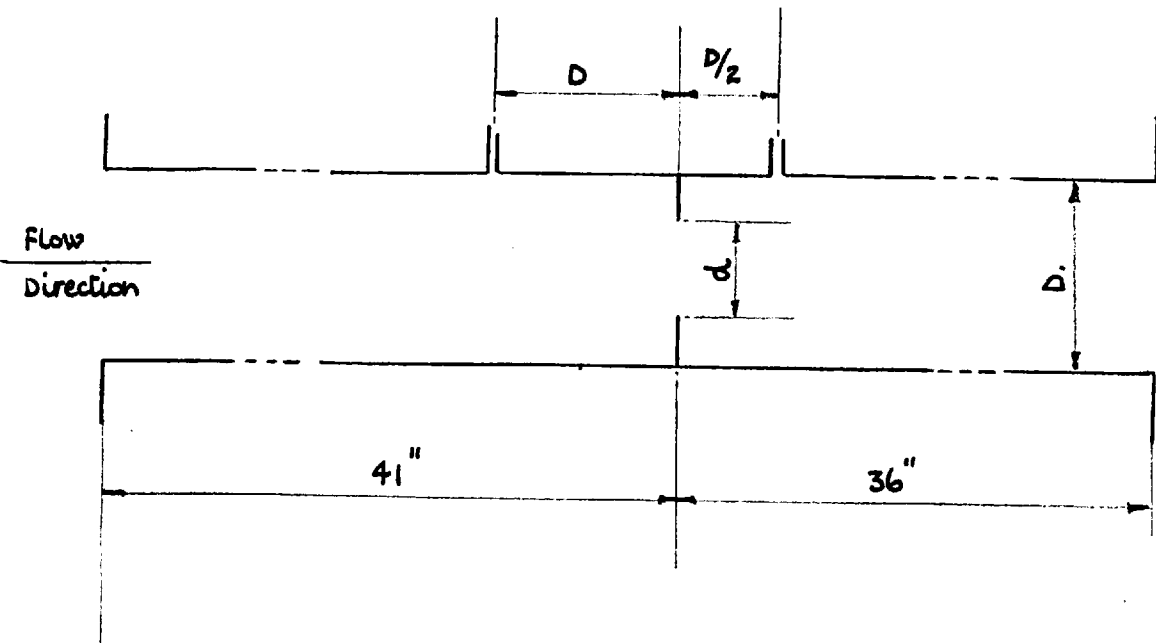
The pipe diameter, $D = 6.75"$

Orifice inside diameter, $d = 4.25"$

the area ratio, $m = (d/D)^2 = 0.395$, is within the requirements of the B.S.S.

The straight pipe requirement upstream of the orifice plate is seen from Fig.7. of the above reference to be about six pipe diameters, i.e., $S_u = 6D = 40.5"$. This requirement is met by the available 41" upstream the orifice (Fig.A5).

The straight pipe requirement downstream the orifice is also given in the above reference; it is $S_d = 5D = 33.7"$. The available pipe length is 36" , which also meets the B.S.S. requirements.



METERING PART OF THE ANALOGUE APPARATUS.

Fig. A5.

b. Formula for Flow Rate Calculation.

The formula given on p.12 of the above reference is

$$L = 359.1 C Z d^2 E \sqrt{h} \sqrt{W}$$

where:

L = mass flow rate in lbs/hr,

C = coefficient of discharge,

Z = a combined multiplier: the product of three separately specified multipliers for viscosity, pipe size, and expansion,

d = orifice diameter in inches,

E = velocity of approach factor,

$$= 1 / \sqrt{1 - m^2}$$

m = area ratio = $(d/D)^2$,

h = differential pressure in inches of water,

W = density of fluid in lbs/cu.ft.

(W has been given the constant value of 0.075 lbs/cu.ft., neglecting the small changes in ambient air temperature and barometric pressure.)

C and Z are obtainable from graphs in the data sheets of the above reference. The resulting formula for the mass flow rate of air through the analogue chamber is approximately

$$\dot{m} = 0.33 \sqrt{h} \quad \text{lbs/sec.}$$

3. Application of the Thring-Newby Similarity Criterion.

a. Introduction.

The similarity conditions between a turbulent jet diffusion flame and a cold model of it were studied by Thring and Newby (Ref. 24.). The result is here quickly derived following a method suggested during the oral examination for the particular case represented in Fig. A6. The figure shows a diagrammatic representation of a jet issuing into a chamber of diameter D , the diameter of the orifice (or the hole of the annular baffle) being d . In the figure V_u and ρ_u are the average velocity and density of the gases entering the chamber, ρ_b is the density of the burnt gases, and V is the average velocity of the gases at the plane where the jet strikes the wall of the chamber (plane X in the figure).

The momenta at the baffle plane and the plane X are

$$\frac{\pi}{4} d^2 \rho_u V_u^2 \text{ and } \frac{\pi}{4} D^2 \rho_b V^2 .$$

These must bear the same ratio for the model and the prototype respectively. For the amount of recirculation to be similar in both prototype and cold model, the ratio V/V_u must be kept constant,

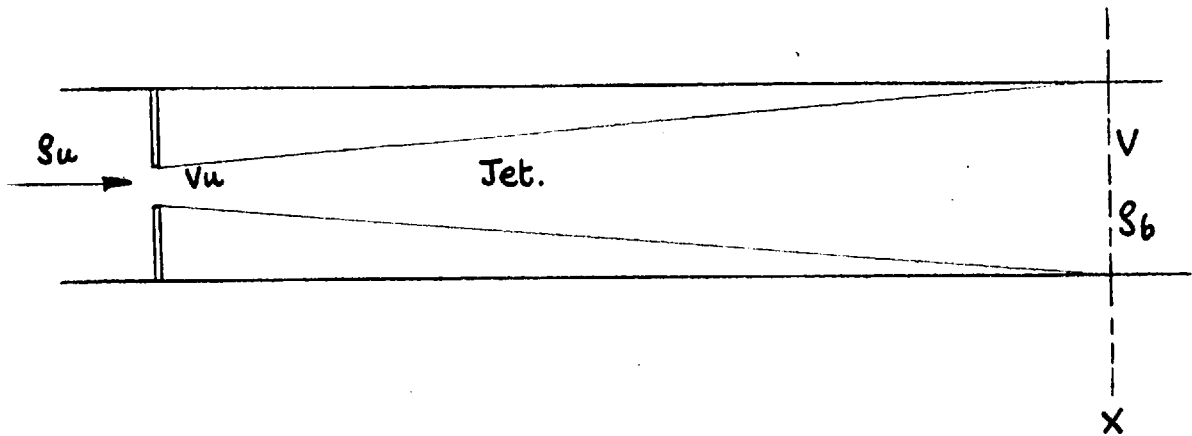
i.e., $d^2 \rho_u / D^2 \rho_b$ should be constant,

or

$$\left[\frac{d}{D} \sqrt{\frac{\rho_u}{\rho_b}} \right]_{\text{MODEL}} = \left[\frac{d}{D} \sqrt{\frac{\rho_u}{\rho_b}} \right]_{\text{PROTOTYPE}}$$

But since there is no change of density in the cold model, the criterion becomes

$$\left[\frac{d}{D} \right]_{\text{MODEL}} = \left[\frac{d}{D} \sqrt{\frac{\rho_u}{\rho_b}} \right]_{\text{PROTOTYPE}}$$



JET ISSUING INTO A CIRCULAR CHAMBER

Fig. A6.

Now $\rho_u > \rho_b$,

$\therefore \left[\frac{d}{D} \right]_{\text{MODEL}}$ is greater than the corresponding (d/D) ratio of the prototype by a factor of $\sqrt{\rho_u / \rho_b}$

i.e., the predictions of the cold model obtained for a particular $\left[d/D \right]_{\text{MODEL}}$ ratio refer to a prototype condition of a smaller d/D ratio, in fact to a ratio equal to

$$\left[\frac{d}{D} \right]_{\text{MODEL}} \times \sqrt{\rho_b / \rho_u}$$

The aim of this point of the Addendum is to apply this criterion to the comparison between the electric analogue predictions and the real flame results in the case where the flameholder is an annular baffle.

b. Application to Analogue Predictions.

Fig.54. (p.207 of the thesis) compares the analogue predictions with the real flame results for the extinction of flames in the wake of annular baffles in an axi-symmetric combustion chamber. The data in the figure are plotted in terms of the extinction loading factor, L_{ext} , and the ratio d/D . The abscissa shows the actual d/D ratios used in the analogue and the real flame tests. The Thring-Newby Similarity Criterion was not taken into consideration in Fig. 54. of the thesis.

We shall now consider whether the application of this criterion improves the agreement between the analogue and flame results. For this consideration the abscissa of either the analogue predictions or the flame results needs to be adjusted in accordance with the criterion stated in the previous part of this section. The analogue is of course being taken as the cold model. It was stated above that the

predictions of the cold model obtained for a particular ratio refers to a prototype condition of a baffle hole to duct diameter ratio of

$$\left[\frac{d}{D} \right]_{\text{MODEL}} \times \sqrt{\rho_b / \rho_u}$$

thus we must multiply the abscissa of the analogue predictions curve by a factor of $\lambda = \sqrt{\rho_b / \rho_u}$ to make the comparison in accordance with the Thring-Newby criterion.

The inlet temperature of the gases used in the real flame tests was about 290° K , and an approximate value of the flame temperature is about 2000° K (Ref.2.). And since at constant pressure the density is inversely proportional to the absolute temperature, the value of the factor λ will be

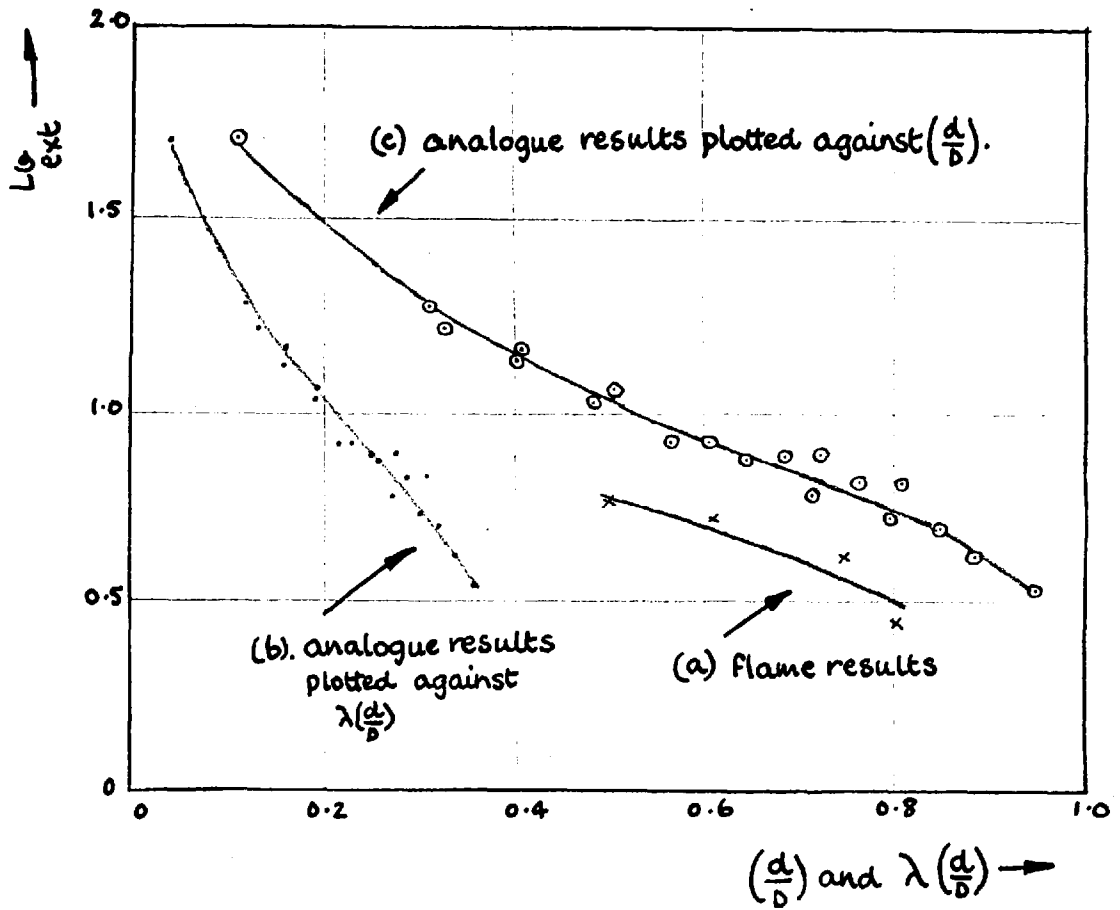
$$\lambda = \sqrt{\frac{290}{2000}} = 0.378$$

For the new comparison Fig. A7 is constructed. It shows:

1. The real flame results as in Fig. 54 of the thesis, (Graph A).
2. The analogue predictions plotted against an effective hole to duct diameter ratio equal to the d/D ratios of Fig. 54 multiplied by 0.378, (Graph b).
3. Also reproduced in this figure is the graph of the analogue predictions as plotted in Fig. 54 of the thesis. This is done to show how the application of the Thring-Newby criterion affects the comparison between real flame results and analogue predictions.

c. Discussion.

It is seen from Fig. A7 that the use of the Thring-Newby criterion does not improve the agreement between the



THE USE OF THE TARING~NEWBY
SIMILARITY CRITERION

Fig. A7.

analogue predictions and the real flame results. A better agreement is obtained by comparing the predictions and the flame results in accordance with the requirements of geometric similarity. Two reasons which are likely to be the cause of this result are:

1. The Thring-Newby criterion is derived for flames with low chemical loading, where combustion is determined by the rate of fuel and air mixing. On the other hand, the electric analogue is concerned with flame conditions under which the chemical loading is high. Thus the criterion is more likely to apply in the case of furnace flames, for example, rather than in the case of a high loading gas turbine combustion chamber.
2. The criterion refers to a similarity between prototype and cold model. The electric analogue, however, is not quite a cold model. It is provided with heaters which are means of simulating the heat generation due to chemical reaction.

It might however be stated that the value taken for the flame temperature (2000° K) is too high in view of the chemical loading in the combustion chamber of the flame experiments; and that for the analogue, the ratio ρ_u/ρ_b is somewhat higher than unity.

If the flame temperature is assumed (as a guess) to be 1500° K , then λ as defined above, is given by

$$\lambda = \sqrt{290/1500} = 0.44$$

and if an allowance is to be made for the deviation from unity of $\left[\frac{\rho_u}{\rho_b} \right]_{\text{ANALOGUE}}$, then the multiplier for the (d/D)

ratios of Fig. 54 in the thesis should be

$$W = 0.44 \left[\sqrt{\rho_u / \rho_b} \right]_{\text{ANALOGUE}}$$

With the temperatures involved in the extinction tests on the analogue, the variation of (ρ_u / ρ_b) varies slightly from unity. A representative value is given by

$$\sqrt{\rho_u / \rho_b} = 1.025$$

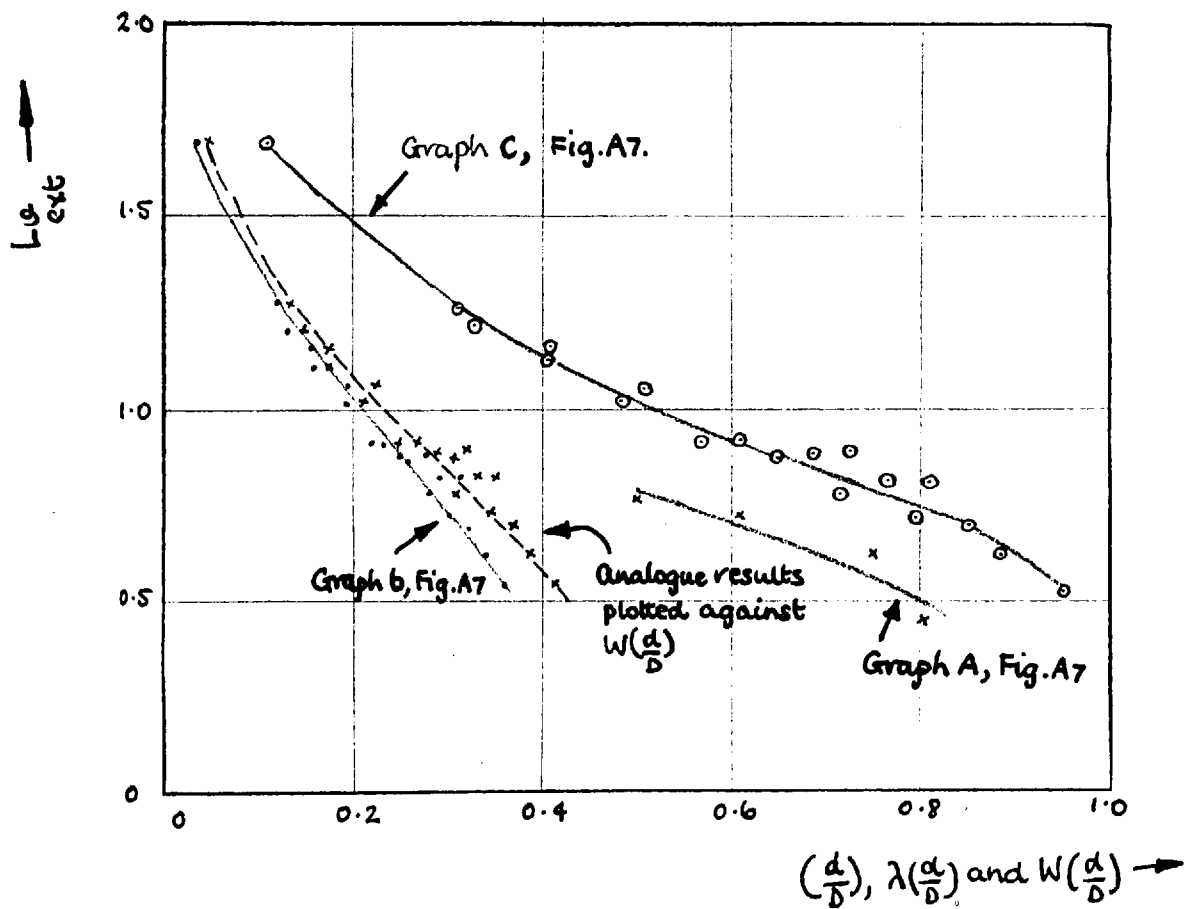
This makes $W = 0.44 \times 1.025$
 $= 0.45.$

The analogue predictions plotted against $W (d/D)$ are presented in Fig. A8 which also contains the graphs of Fig. A7 to show the improvement gained by the above corrections.

4. Photographs of the Apparatus.

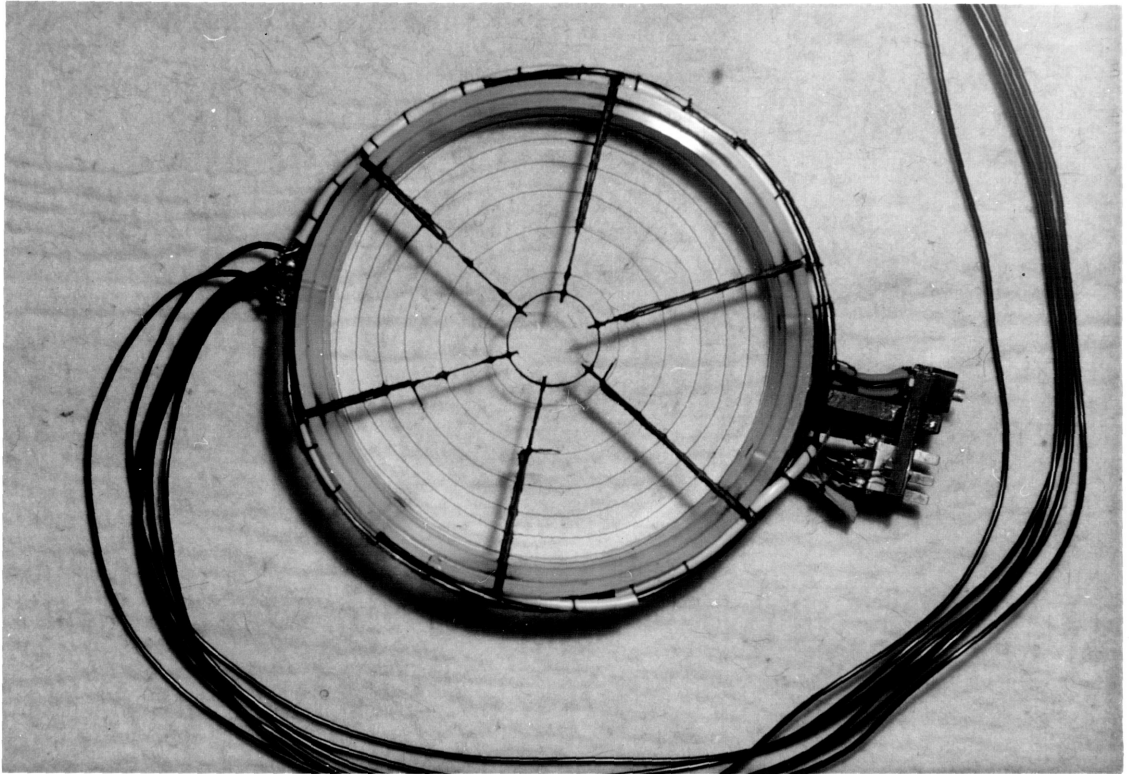
The description of the components of the electric analogue apparatus was discussed in Chapter Two of the thesis. Four photographs are given here to show:

1. A view of an element of the analogue,
2. The working section without any baffle,
3. A view of the working section with an annular baffle placed in position,
4. A general view of the whole apparatus.

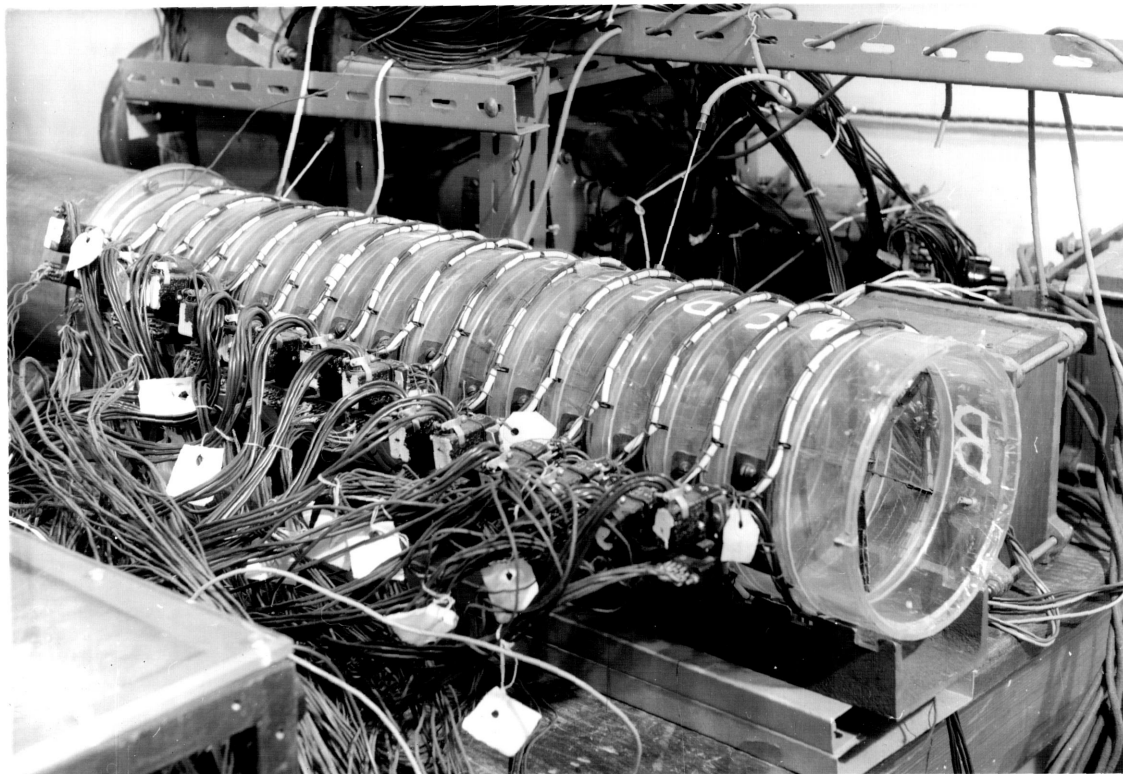


CORRECTIONS TO FIG. A7.

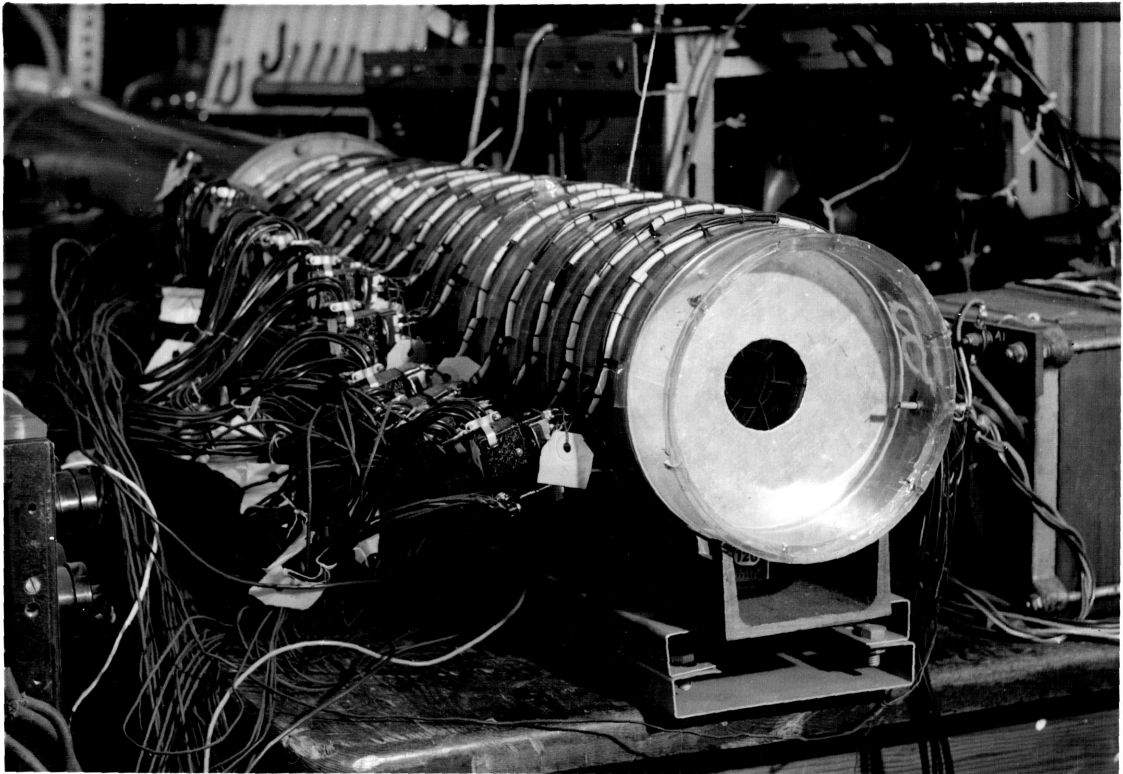
Fig. A8



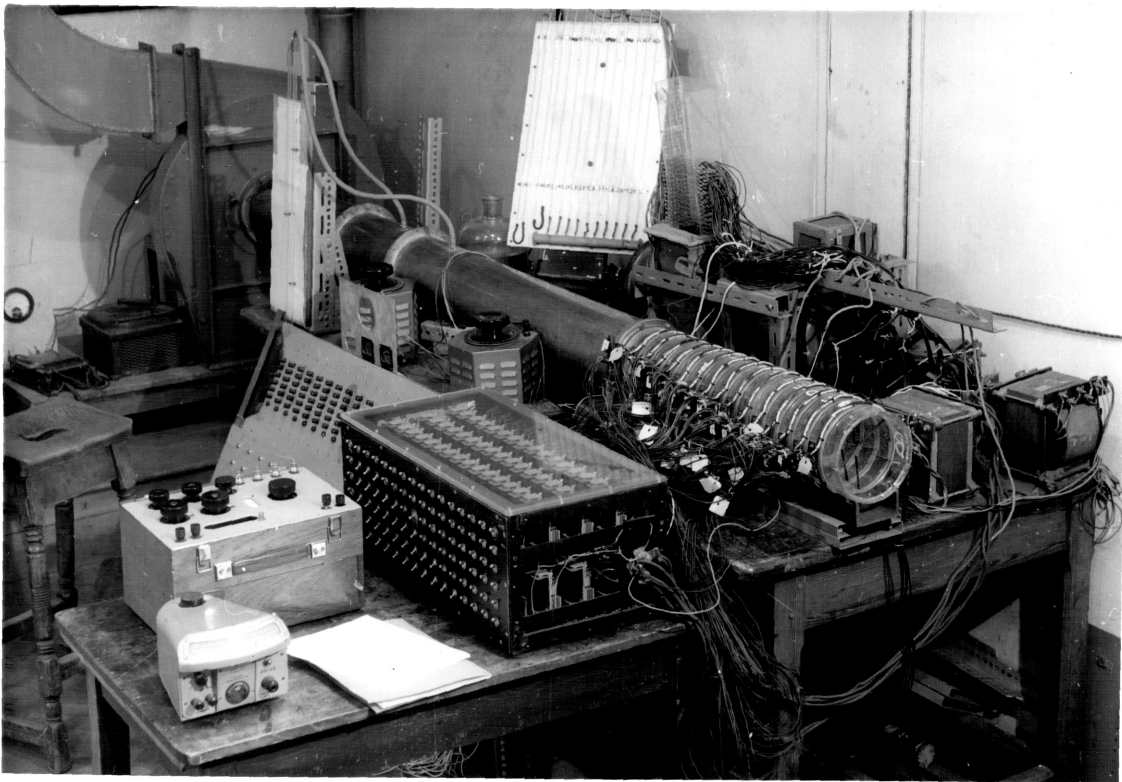
1. VIEW OF AN ANALOGUE ELEMENT.



2. VIEW OF THE ANALOGUE WORKING SECTION (WITHOUT BAFFLE).



3. VIEW OF THE ANALOGUE WORKING SECTION, WITH AN ANNULAR
BAFFLE PLACED IN POSITION.



4. GENERAL VIEW OF THE APPARATUS.

5. The Reynolds Number and the Equivalence Ratio.

In the thesis, discussions of flame stability data (of the author and other workers), the results were plotted in terms of the extinction loading factor, $L_{ext.}$, versus the equivalence ratio, ϕ . For example, the author's results for central discs in a cylindrical chamber were thus presented in Fig. 36 (p. 142) for mixtures with weak fuel air ratio, and in Fig. 37 (p. 143) for rich mixtures. Such a manner of presentation of flame stability data confuses the effects of the fuel air ratio and of the Reynolds Number. For along each graph in Figs. 36 and 37, the Reynolds Number is changing. A new discussion of the flame stability data is therefore given below for the cases where a quantitative comparison was made between the real flame results and the predictions of the electric analogue.

The purpose of the present discussion is to discover whether there is a fully developed turbulent flow condition in the wake of the flame holders. This is important in view of one of the criteria necessary for similarity to obtain between the analogue model and the real flame it endeavours to simulate; namely that either the Reynolds Number in the original and in the analogue be equal, or that the flow is fully turbulent. The analogue results used in the thesis were shown to be independent of the Reynolds Number, but it is still necessary to show that fully turbulent conditions prevailed in the real flame results.

In their study of the role of wake transition in the process of flame stabilisation on bluff bodies, E.E. Zukoski

and F. E. Marble (Ref. 25) established the existence of a critical range of Reynolds Number about 10^4 based on the characteristic dimension of the bluff body and the conditions of the gases passing it. (A similar result was also published, a year earlier, by D.B.Spalding and B.S.Tall in Ref. 18). Below this critical range, molecular transport processes in the wake of the bluff body dominate the flame stabilisation mechanism; while above this critical Reynolds Number range, a transition occurs replacing molecular diffusion by turbulent exchange processes. Above the transition Reynolds Number, a logarithmic plot of the maximum blow-off velocity versus the Reynolds Number, produces a straight line with a positive slope (Fig. 16, Ref. 25) for a variety of baffle shapes. Due to geometric differences, the data plotted had different levels of blow-off velocity; but the variations of the blow-off velocity with the Reynolds Number were comparable. The slopes of the straight lines in these plots were similar.

According to M.V.Herbert (Ref. 26), the safest criterion for wake-boundary transition is presumably the shift of equivalence ratio ϕ at the peak of the blow-off loop, at which point stability is then best correlated. The tops of these loops are however not always covered; but where there is no evidence of the loop peaking far from the stoichiometric fuel air ratio, Ref. 26 suggests that one can employ the free stream ϕ at values other than unity. It is to be remembered that the shift occurs mainly with bluff bodies whose characteristic dimension is so small that the Reynolds Number falls below the transition value, i.e., the flow is mainly laminar

in the critical range between the main stream and the hot wake behind the bluff body. In view of the usual steep shape of the blow-off loop, it is necessary when using a free - stream equivalence ratio other than unity, to guard against choosing too low a value of ϕ ; since if ϕ is too low, the blow-off velocity could be so small as to make the Reynolds Number fall below the critical transition level. Zukoski and Marble (Ref. 25) concluded that as far as the dependence of the blow-off velocity on the flame holder dimension is concerned, the range of ϕ for which this dependence does not appreciably change is given by

Results used in the Thesis.

a. Central Discs.

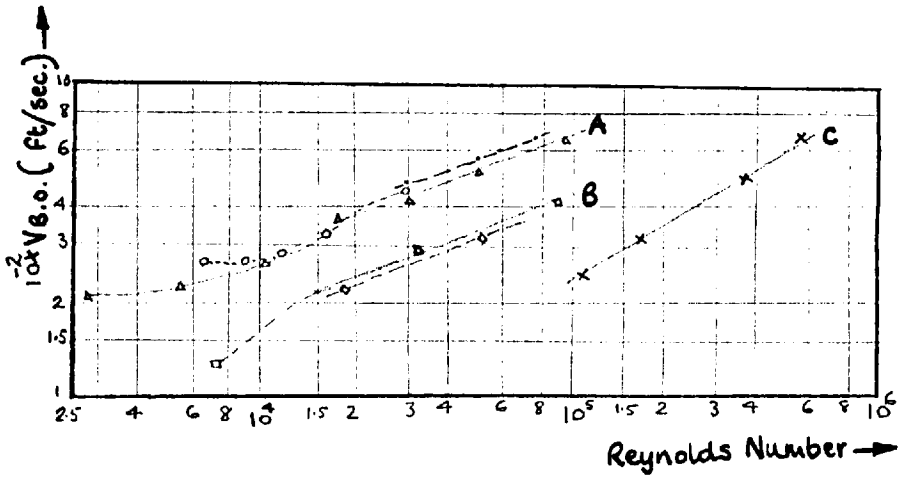
The main data used in the comparison between the analogue predictions and the real flame results in the case of central discs are those of the author (Figs. 31 - 34, pp. 135 - 139 of the thesis). DeZubay's results (Ref. 15) for similar configuration, but with smaller discs, were also discussed. The latter results were also considered by Zukoski and Marble at the condition of maximum blow-off velocity. Because of the fact that, at atmospheric pressure, the loop for the largest disc was not completed at its peak, and that some doubts were cast on the data for the smallest of the three discs tested by DeZubay, Ref. 25 concentrated its discussions on DeZubay's results under non-atmospheric conditions. These results are therefore of no direct use for comparison with the electric analogue predictions for central disc baffles.

The discs tested by the author have larger d/D ratios, but the stability loops of Figs. 31 - 34 in the thesis are not completed at their peaks. We shall therefore plot the blow-off velocity versus the Reynolds Number at an equivalence ratio of 0.85. The resulting graph is presented here in Fig. A9. This figure also contains a copy of Fig. 16 in Ref. 25 which, as stated previously, contains plots of maximum blow-off velocity for a variety of centrally-placed baffles.

The features that may be observed from Fig. A9 are

1. The level of blow-off velocities is generally lower for the author's discs (Graph C) than for the other baffles (graph groups A and B).
2. The Reynolds Numbers are generally higher for the author's discs than for the other baffles.
3. The disc results are correlated by a straight line with a positive slope - just as the results for the other baffles.
4. The Reynolds Numbers for all four discs at this particular equivalence ratio (0.85) are well above the critical transition value.
5. The slope for the straight line correlating the disc results is somewhat higher than that for the other baffles.

The first of the above features may be explained by the shape of the flame stability loops. The loops for the discs do not show evidence of peaking far from the stoichiometric fuelair ratio. Thus, at $\phi = 0.85$ the stability loops have not yet peaked. Hence the lower level of Graph C in Fig. A9.



VARIATION OF BLOW-OFF VELOCITY WITH
THE REYNOLDS NUMBER

GRAPHS A & B - copy of Fig. 16, Ref. 25,
- maximum B.O. velocities.

GRAPH C - central disc results
- present investigation
- $\phi = 0.85$.

Fig. A9.

The second point is explained by reference to the definition of the Reynolds Number which is based on the baffle characteristic dimension, taken as its diameter, and the gas conditions past the baffle. The diameters of the discs tested in this investigation are bigger than the characteristic dimensions of the baffles whose stability data are presented in groups A and B of Fig. A9.

The third and fourth features indicate that the flow in the critical region between the main stream and the hot wake behind the disc is fully turbulent.

The small differences in the slope of the straight line correlating the disc results may be due to the fact that these results were not obtained under stoichiometric conditions. That is, the small difference of slope may express the slight change in the dependence of the flame stability characteristics on the baffle dimension, due to the change in equivalence ratio.

For our purposes, the important conclusion is that the flow is fully turbulent in the critical wake region for all the four discs tested by the author. This is a fortunate result (in view of the introductory remarks about the thesis discussion of the data) since it means that the analogue predictions can be compared with the flame results at the particular equivalence ratio of 0.85. The comparison of Fig. 53 (p. 200 of the thesis) between the analogue and flame results will be valid at the equivalence ratio of 0.85. With possibly some small differences, similar results may be expected to obtain at higher ϕ values; but more flame data are needed to confirm this expectation.

b. Annular Baffles.

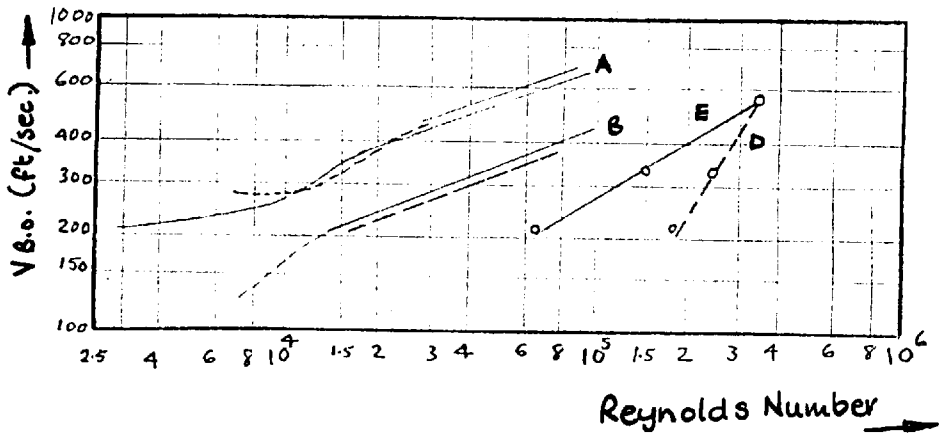
The flame stability results for annular baffles that do not seriously suffer from entrainment of ambient air are those reported in Ref.18. They are part of the results obtained by Martin and Webb. The peak of the stability loop for the baffle with a 1" diameter hole was not covered. We shall therefore plot the results in a manner similar to that followed with the central disc baffles. The value of the equivalence ratio chosen is 0.95.

In the case of the centrally placed baffles, the Reynolds Number was based on the diameter of the baffle; but in the case of the annular baffles we have a choice of two dimensions

1. The baffle hole diameter, d , or
2. $D - d$, where D is the duct diameter.

In Fig. A10 Graphs A and B are those copied from Ref. 25; Graph D correlates the annular baffles results in terms of a Reynolds Number based on d ; while in Graph E the Reynolds Number is based on $(D - d)$. Graphs D and E are both for equivalence ratio of 0.95.

It is noted that the results used in Graphs D and E are correlated by straight lines, that the Reynolds Numbers (however based) are well above the critical transition values, and that Graph D is much steeper than Graph E. Graph E has a slope which is not much different from the slopes of the graphs A and B in Fig. A10. This means that the use of $(D - d)$ is the characteristic dimension in the



VARIATION OF THE BLOW-OFF VELOCITY
WITH THE REYNOLDS NUMBER

GRAPHS A & B - Copy of Fig. 16, Ref. 25

- maximum B.O. velocity

GRAPH D - annular baffles results, Ref. 18,

- R_n based on d ; $\phi = 0.95$.

GRAPH E - same results as GRAPH D

- R_n based on $(D-d)$; $\phi = 0.95$.

Fig. A10.

definition of the Reynolds Number, brings the annular baffle flame stability results in line with the results for the centrally placed baffles.

The annular baffle results (for $\phi = 0.95$) being shown to have been obtained under fully turbulent conditions, may thus be used for comparison with the predictions of the electric analogue. The comparisons in Chapter Six of the thesis for annular baffles may be considered as valid for flame results at an equivalence ratio of 0.95. Some modification may be necessary if the comparison is to be extended to cover the stoichiometric conditions.

6. Experimental Data.

The data used in the thesis are given in the tables after the present notes. The details are as follows:

a. Analogue results

1. Annular baffles I, these are the complete tests on the present analogue model.
2. Annular baffles II, tests at highest flow rates on the Lucas Analogue model.
3. Annular baffles III, tests on intermediate size baffles at highest flow rates, present analogue model.
4. Annular baffles IV, tests at highest flow rates using the present analogue model fitted with a primary and a secondary baffle, as in the case of the Lucas model.
5. Central disc baffles,
6. Hemispherical baffles,
7. Conical baffles.

b. Flame stability data

1. Annular baffles I, results of Ref. 18.
2. Annular baffles II, author's results.
3. Annular baffles III, results of Ref. 19.
4. Central disc baffles, I, results of DeZubay.
5. Central disc baffles, II, author's results.
6. Hemispherical baffles, Mestre's results.

ELECTRIC ANALOGUE PREDICTIONS *

ANNULAR BAFFLES I					.73	.905	170	19.5	.905	CENTRAL DISC BAFFLES					.89	.500	188	10.75	13.4	.73	.552	277	12.0	9.6
$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D		.849	179	12.35	.89	$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D	$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D	$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D
.324	.430	483	9.50	1.25		.762	198	16.45	.89	.405	.955	200	18.45	6.01	.324	.430	483	9.50	1.25	.635	.985	185	21.3	7.8
	.410	525	8.85	1.265		.672	228	14.55	.90	11.5	.951	186	20.6	6.2	.364	.958	180	20.7	5.8	.415	1.015	179	21.9	6.4
	.396	531	8.55	1.24		.603	241	13.0	.855						.445	.951	176	20.6	6.1	CONICAL BAFFLES				
	.372	569	8.05	1.245		.478	313	10.35	.88						.525	.945	168	20.4	6.4	$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D
	.344	612	7.40	1.24	.81	.322	476	6.98	.90						.608	.935	171	20.2	7.4	θ°				
	.326	632	7.05	1.215		.942	150	20.5	.83						.688	.892	184	19.3	9.2	.415	.978	258	21.1	8.9
	.262	730	5.66	1.125		.861	164	18.6	.83						.768	.795	195	17.2	11.1	40	.883	282	19.1	8.8
	.203	874	4.37	1.045		.76	180	16.45	.905						.85	.692	166	15.0	12.1		.800	315	17.3	8.9
	.424	990	9.34	1.25		.684	207	14.8	.83	.485	.948	172	20.5	6.21	HEMISPHERICAL BAFFLES						.687	363	14.8	8.8
.405	.579	328	12.5	1.115		.591	242	12.8	.84	12.6	.891	179	19.35	6.14	$\frac{d}{D}$	$\frac{m}{10^3 \text{ sec}}$	ΔT μV	$\frac{10^4}{R_0}$	L_D		.582	423	12.6	8.7
	.555	341	12.0	1.115		.437	327	9.47	.84						.385	.946	186	20.4	6.04	"	.511	475	11.1	8.6
	.510	369	11.0	1.105	.89	.816	175	17.6	.84							.855	207	18.45	6.09	"	.93	258	20	8.95
	.475	403	10.25	1.125		.99	106	21.3	.615	.868	.934	157	20.2	6.4		.762	228	16.4	6.0	"	"	243	"	8
	.440	425	9.5	1.11		.83	117	19.2	.61	14.2	.822	179	17.75	6.4		.660	257	14.25	5.83	"	"	235	"	7.7
	.386	455	8.34	1.10		.721	142	15.5	.60							.511	321	11.05	5.66	"	"	245	"	8.95
	.313	577	6.76	1.06		.532	194	11.5	.605							.362	443	7.8	5.5	"	"	237	"	8.65
	.194	830	4.2	.955		.587	172	12.7	.595							.892	197	19.3	6.05	"	"	228	"	8.3
.485	.631	268	13.6	.995		.797	126	17.2	.59							.716	243	15.5	6.0	"	"	221	"	8.05
	.597	286	12.9	1.0	ANNULAR BAFFLES II						.685	214	14.8	6.4		.628	271	13.55	5.88	"	"	184	"	6.7
	.558	307	12.05	1.005		.852	122	18.4	.61	.648	.93	157	20.1	7.38		.456	358	9.85	5.6	"	"	230	"	10
	.525	325	11.45	1.005	.102	.0902	612	2.64	1.72	16.5	.844	172	18.25	7.32		.9	193	19.4	5.98	"	"	222	"	9.65
	.473	362	10.2	1.005	.306	.1135	361	3.3	1.27		.722	200	15.6	7.34		1.02	172	22	6.01	"	"	211	"	9.2
	.411	416	8.9	1.005	.408	.117	320	3.42	1.17		.551	256	11.95	7.12	.567	1.01	187	21.75	7.47	"	"	196	"	8.58
	.295	673	5.38	.97	.57	.126	284	3.54	1.07		.33	385	7.14	6.4		.903	203	19.55	7.33	40	.962	200	20.8	10.7
	.614	278	13.25	1.005	.714	.126	194	3.69	.76		.872	164	18.85	7.28		.773	246	16.8	7.53		.93	206	20	10.6
	.221	721	4.6	.94	.796	.125	183	3.66	.71		.781	185	16.9	7.34		.628	289	13.55	7.15		.902	213	19.5	10.65
	.598	312	11.8	1.005	ANNULAR BAFFLES III						.361	295	10.0	6.8		.443	375	9.6	6.94		.89	219	19.2	10.8
	.346	580	7.47	1.02		.642	226	14.0	7.29		.642	226	14.0	7.29		.963	196	20.8	7.43		.877	220	18.9	10.7
.568	.758	206	16.3	.915	.95	1.08	85	23.4	.54	.73	.868	180	18.75	9.72		.842	221	18.2	7.4		.892	228	18.2	10.65
	.705	220	15.2	.910	.85	.96	124	20.9	.70	20.7	.781	200	16.85	9.85		.704	236	15.3	7.37		.797	242	17.2	10.7
	.652	235	14.1	.90	.77	.92	154	19.9	.82		.572	257	12.35	9.2		.542	323	11.75	6.97		.75	258	16.2	10.7
	.411	371	8.9	.895	.69	.84	178	18.2	.88		.361	374	7.8	8.5	.618	.92	221	21.2	10.3		.694	275	15	10.6
	.266	570	5.75	.895	.61	.784	199	17.0	.92		.829	186	17.9	9.68		.869	250	18.75	10.3	"	.631	297	13.6	10.4
	.610	256	13.2	.915	ANNULAR BAFFLES IV						.722	212	15.65	9.61		.732	285	15.8	9.95		.93	204	20	10.55
	.511	305	11.05	.92		.660	235	14.25	9.74		.660	235	14.25	9.74		.922	236	19.9	10.3	"	.94	194	20.3	10.1
	.338	435	7.32	.865	.79	.92	144	20	.78	.81	.693	200	14.95	11.85		.823	265	17.75	10.25	80	.909	203	19.6	10.2
.648	.781	192	17.0	.89	.71	.88	157	19.1	.81	27.9	.761	179	16.45	11.6		.66	300	14.25	9.4		.87	210	18.8	10.1
	.728	206	15.65	.88	.51	.657	262	14.2	1.01		.613	221	13.25	11.6		.963	230	20.8	10.4		.858	213	18.5	10.15
	.596	249	12.85	.87	* UNDER ROOM TEMPERATURE AND PRESSURE						.573	243	12.4	11.9		.91	240	19.6	10.25		.822	222	17.8	10.15
	.511	292	11.05	.88	40 $\mu V = 1^\circ C$						.658	212	14.2	11.8	.73	.97	184	20.9	11.2		.804	225	17.4	10
	.362	413	7.83	.88							.478	275	10.88	11.1		.921	194	13.9	11.2		.785	233	17	10.15
	.276	523	6.0	.865							.411	305	8.9	10.6		.887	202	19.2	11.25		.748	241	16.2	10
	.836	178	18.1	.875						.89	.643	148	14.0	13.5		.835	213	17.0	11.2		.686	258	14.8	9.8
	.656	228	14.2	.88						46.9	.618	152	13.4	13.4		.779	225	16.8	11.0	"	.612	280	13.4	9.6
	.456	329	9.85	.88							.571	164	12.3	13.4		.712	243	15.4	10.85	"	.93	185	20	9.5

* UNDER
ROOM TEMPERATURE
AND PRESSURE
40 $\mu V = 1^\circ C$

FLAME STABILITY RESULTS

ANNULAR BAFFLES I			CENTRAL DISCS I			CENTRAL DISCS II			HEMISPHERICAL BAFFLES		
d ins	φ	m _{ext} lbs/sec	d ins	φ	V _D F.P.S.	d ins	φ	m _{ext} lbs/min	d ins	φ	V _{m/sec}
1.5	.741	.066	1.25	.943	19.7	1.0	.73	248	1.0	.824	21.2
	.763	.083		.931	18.5		.756	290		.812	27.1
	.772	.096		.955	17.5		.756	304		.81	27.5
	.802	.127		.945	17.2		.73	319		.808	25.9
	.866	.182		.943	17.5		.808	338		.799	25.0
	.898	.202		.922	17.9		.858	448		.79	23.7
	.938	.207		.925	16.6		1.18	456		.78	22.6
	1.0	.207		.928	15.3		1.265	440		1.37	14.2
				.92	15.0		1.36	408		1.23	24.2
				.90	12.9		1.305	404		1.25	23.0
1.25	.85	.175	1.0	.866	12.0	.50	.742	177	.25	.73	18.0
	.881	.218		.893	11.3		.793	225		.718	17.1
	.928	.227		.872	11.0		.78	237		.732	16.4
	.95	.231		.857	10.0		.793	262		.76	22.1
	.984	.24		.863	9.6		.82	267		.772	21.4
				.827	8.4		.808	315		.786	26.3
				.805	7.4		.855	321		.791	25.7
				.79	7.4		.898	352		.778	23.8
							.855	360		.787	26.1
							.854	363		.77	23.4
1.0	.772	.130	ANNULAR BAFFLES III			CENTRAL DISCS II			.25	.857	26.0
	.842	.209	1.05	.921	.245	.75	.76	15.88		.856	26.2
	.86	.225		.943	.257		.805	19.41		.812	21.8
	.866	.230		.925	9.3		.853	21.88		.805	20.5
	.881	.233		.865	8.8		.855	26.95		.79	19.5
	.921	.245		.84	7.3		.907	26.02		1.38	16.0
	.943	.257		.82	5.9		.835	18.28		1.4	15.3
				.82	5.2		.843	23.08		1.37	17.1
				.79	4.9		.80	17.9		1.36	19.7
				.775	4.4		.798	14.9		1.3	23.0
.75	.890	19.7	1.05	.922	404	1.0	.922	450	1.25	.743	18.0
	.893	18.9		.922	450		.922	500		.722	16.5
	.884	17.5		.865	8.8		.922	500		.759	20.9
	.87	16.7		.84	7.3		1.0	517		.772	20.0
	.86	15.8		.82	5.9		1.025	517		.772	19.8
	.843	15.2		.82	5.2		1.05	517		.799	23.9
	.855	14.4		.79	4.9		1.025	527		.822	28.6
	.845	13.8		.775	4.4		1.125	498		.832	30.4
	.832	13.2		.80	3.8		1.085	474		.848	30.1
	.837	12.5		.75	2.3		1.156	465		.828	31.2
.50	.818	11.7	1.2	.89	9.0	1.25	.92	25	1.0	.801	23.5
	.805	11.2		.85	8.9		.92	25		.83	27.8
	.87	20.4		.855	7.7		.90	24.6		.839	33.2
	.862	20.2		.82	7.2		1.29	20.5		.853	32.9
	.855	19.8		.78	3.6		.895	23.6		.85	31.1
	.862	18.8		.79	6.0		.82	20.4		.858	29.9
	.855	18.4		.80	5.5		.862	22.7			
	.845	17.7		.85	6.8		1.25	21.3			
	.85	17.0		.89	9.0		1.17	24.8			
	.835	15.8		.85	6.4		1.33	17.3			
	.814	14.3		.93	7.1		1.26	22.6			

c. Influence coefficients.

The tables of influence coefficients contain calculated and measured values of α_{jk} and β_{jk} (defined in the thesis) for the flow in the wake of an annular baffle with d/D ratio of 0.405. The resulting influence equations have been solved on the computer to establish the extinction loading factors. The resultant solutions are:

From calculated influence coefficients, $L_{ext} = 0.90$,

From measured influence coefficients, $L_{ext} = 1.07$.

The latter value is the average of the values obtained when feeding the computer with the measured β_{jk} in Tables I, II, and III.

(The actual respective L_{ext} values are about 1.08, 1.06 and 1.07).

d. Formulae.

The formulae and other data used in the calculation of the experimental results are as follows: (the symbols are defined in the thesis)

1. Lucas Analogue model tests:

$$\dot{q}_{vm}''' = 0.25 \text{ ~~ch/ft}^2 \text{ sec}~~ \text{ chu/sec. ft.}^3$$

(voltage applied to the heaters = 4.0 volts)

2. Present analogue model tests:

Heater voltage = 12.0 volts

$$\dot{q}_{vm}''' = 1.96 \text{ ~~ch/ft}^2 \text{ sec}~~ \text{ chu/sec. ft.}^3$$

Mas flow rate, neglecting the changes in ambient air temperature and barometric pressure, is given by

$$\dot{m} = 0.33 \sqrt{h} \text{ lb/sec.}$$

P O I N T S j	SOURCE POINT K					
	1	2	3	4	5	6
	1	1	.34	.34	.43	.50
	2	1	1	.67	.86	1
	3	1	1	1	1	1
	4	1	1	1	1.28	1
	5	1	1	1	1.28	1.5
	6	1	1	1	1.28	1.5

CALCULATED d_{jk}

j	K					
	1	2	3	4	5	6
	1	1.0	.30	.26	.33	.55
	2	1.03	1.0	.62	.72	1.32
	3	.96	1.0	1.0	1.0	1.0
	4	.94	.93	1.0	1.22	1.0
	5	1.15	1.2	1.1	1.34	1.8
	6	.94	.89	.9	1.11	1.78

MEASURED d_{jk} I

$$\dot{m} = 0.37 \text{ lb/sec}$$

j	K					
	1	2	3	4	5	6
	1	1.0	.28	.25	.31	.59
	2	1.04	1.0	.56	.84	1.22
	3	.93	.94	1.0	1.0	1.0
	4	.93	.99	1.0	1.45	1.0
	5	1.1	1.15	1.06	1.6	1.75
	6	1.0	.93	.96	1.33	1.72

MEASURED d_{jk} II

$$\dot{m} = 0.414 \text{ lb/sec.}$$

j	K					
	1	2	3	4	5	6
	1	1.0	.29	.28	.29	.62
	2	1.04	1.0	.59	.71	1.28
	3	.85	.99	1.0	1.0	1.0
	4	.85	.90	1.0	1.06	.83
	5	1.12	1.2	1.15	1.22	1.66
	6	1.16	.92	.82	.87	1.62

MEASURED d_{jk} III

$$\dot{m} = 0.464 \text{ lb/sec}$$

j	K					
	1	2	3	4	5	6
	1	.09	.06	.09	.04	.09
	2	.09	.17	.17	.08	.17
	3	.09	.17	.25	.09	.17
	4	.09	.17	.25	.12	.17
	5	.09	.17	.25	.12	.26
	6	.09	.17	.25	.12	.26

CALCULATED β_{jk}

j	K					
	1	2	3	4	5	6
	1	.09	.05	.06	.03	.13
	2	.09	.17	.15	.06	.22
	3	.08	.17	.25	.09	.17
	4	.08	.16	.25	.11	.17
	5	.10	.20	.30	.12	.36
	6	.08	.15	.22	.10	.37

MEASURED β_{jk} I

$$\dot{m} = 0.37 \text{ lb/sec.}$$

j	K					
	1	2	3	4	5	6
	1	.09	.04	.06	.03	.10
	2	.09	.17	.14	.07	.20
	3	.08	.16	.25	.09	.17
	4	.08	.17	.25	.13	.17
	5	.10	.19	.26	.14	.33
	6	.09	.16	.24	.12	.33

MEASURED β_{jk} II

$$\dot{m} = 0.414 \text{ lb/sec.}$$

j	K					
	1	2	3	4	5	6
	1	.09	.04	.07	.03	.11
	2	.09	.17	.15	.06	.21
	3	.07	.17	.25	.09	.17
	4	.07	.15	.25	.09	.14
	5	.10	.20	.28	.11	.36
	6	.10	.15	.20	.08	.37

MEASURED β_{jk} III

$$\dot{m} = 0.464 \text{ lb/sec.}$$

TABLES OF CALCULATED INFLUENCE COEFFICIENTS
(ANNULAR BAFFLE WITH $d/D = 0.405$)

h represents the pressure differential across the B.S. orifice in inches of water.

Reynolds Number:

$$R_D = \frac{u \rho_u D}{\mu_u} = 21.7 \times 10^4 \dot{m}$$

Loading Factor:

a. Based on chamber volume

$$L_v = 5.89 \times 10^{-3} \dot{m} \Delta T$$

ΔT being measured in microvolts

b. Based on chamber diameter

$$L_D = \alpha \dot{m} \Delta T, \text{ where}$$
$$\alpha = 29.1 \times 10^{-3} / (1 - (d/D)^2)$$

3. DeZubay's real flame results for discs:

The Reynolds Number is based on the disc diameter and the conditions of the gas passing the baffle, i.e.

$$R_d = u_d \rho_u / \mu_u$$
$$= \beta V \quad (\text{both } \beta \text{ and } V \text{ are given in the Tables})$$

4. The author's real flame results for discs:

A similar definition of Reynolds Number is used here as for DeZubay's results, but in terms of the mass flow rate, $\dot{m}$, and γ given in the tables.

$$R_d = \gamma \dot{m}$$

5. Hemispherical baffle results: (Mestre)

For these results:

$$R_d = \Delta V \quad (\text{both } \Delta \text{ and } V \text{ are tabulated})$$

7. The Combustion Efficiency

On page 80, Chapter Three of the thesis, a parameter called the analogue combustion efficiency was introduced and given the symbol ϵ_a . For a stable flame predicted by the electric analogue, ϵ_a was defined as the ratio between the number of heaters left on in the " flame " and the total number of heaters in the analogue working section.

An accepted form of the definition for an overall combustion efficiency ζ in a burning system may be expressed as

$$\zeta = \left(\text{actual total head temperature rise of the gases in passing through the combustion chamber} \right) / \left(\text{total head temperature rise for complete combustion} \right).$$

The numerator in the definition of ϵ_a may be taken as proportional to the actual total head temperature rise of the air passing through the analogue chamber; but the denominator has no similar relation to the total head temperature rise for complete combustion. That is, while the numerators in the definitions of ϵ_a and ζ are proportional to each other, there is no such proportionality in the case of the denominators. Thus ϵ_a cannot be said to represent an analogue prediction of combustion efficiency. It is in fact a measure of the heat input rate in the analogue chamber during the existence of a predicted stable flame.

8. The Navier-Stokes Equations.

Equation 3 (p. 17 of the thesis) was given as the Navier-Stokes Equation in the x-direction. The viscous term in Equation 3 is in fact incomplete. In their full form, these equations are given by:

$$\rho \frac{du}{dt} = \rho F - \nabla p + \mu \nabla^2 u + \frac{1}{3} \mu \nabla \nabla u \dots\dots\dots (X)$$

where

ρ = gas density,

u = velocity,

t = time,

F = external force on unit volume of fluid,

p = pressure,

μ = viscosity,

and

$$\frac{du}{dt} = \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z}$$

$$\nabla = \frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z}$$

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$$

In the steady-state conditions, and in the absence of external forces on the fluid, both $\frac{\partial u}{\partial t}$ and F are equal to zero.

The dropping of the last term in the R.H.S. of equation x (see Equation 3 of the thesis) does not change the conclusions of Chapter One, since both $\mu \nabla^2 u$ and $\frac{1}{3} \mu \nabla \nabla u$ in Equation x suggest the same dimensionless group, namely the Reynolds Number.

9. Equation 5. of Thesis.

This equation on page 18 of the thesis had a printing error.

It should read:

$$\ddot{q} = \sum H_j \ddot{m}_j$$

10. References.

Some references were not clearly stated in the thesis. This is amended in the following list, giving these references their original numbers in the thesis. The list also includes the references used in the discussions of this Addendum.

<u>No.</u>	<u>Author.</u>	<u>Reference.</u>
6.	D.B.Spalding.	Some Thoughts on Flame Theory, Combustion and Propulsion, Third AGARD Colloquim, Pergamon Press, p.269, 1958.
Q/11.	B . Thwaites. (Editor)	Incompressible Aerodynamics, Oxford University Press, 1960, p. 32.
23.	H.M.Spiers. (Editor)	Technical Data On Fuel, Fourth Edition, 1943.
24.	M.W.Thring & M.P.Newby.	Combustion Length of Enclosed Turbulent Jet Flames, 4th. Symposium (International) on Combustion, p.789, Williams & Wilkins Co., Baltimore 1953.

<u>No.</u>	<u>Author.</u>	<u>Reference.</u>
25.	E.E.Zukoski & F.E.Marble	The Role of Wake Transition in the Process of Flame Stabilization on Bluff Bodies, AGARD Combustion Researches and Reviews, 1955, Butterworths Scientific Publications, London. p.167.
26.	M.V.Herbert.	Aerodynamic Influences on Flame Stability, Progress in Combustion Science and Technology, Volume I, p.61, Pergamon Press.