MEASUREMENT OF RECIRCULATION IN FLAME STABILIZATION

Glenn N. Adams

Thesis submitted to the Faculty of Graduate Studies and Research in partial fulfillment of the requirements for the degree of Doctor of Philosophy.

Gas Dynamics Laboratory
McGill University
August, 1953

ACKNOWLEDGEMENTS

The author greatly appreciates the guidance given by the research supervisors, Professor J.S. Marshall of the Physics Department and Professor D.L. Mordell, Department of Mechanical Engineering. Particular thanks are due Professor Mordell, Director of the Gas Dynamics Laboratory, and the Defense Research Board of Canada for the generous provision of research facilities.

At various times assistance and helpful advice have been received from almost every member of the laboratory. Mr. Robin Fancott, a mechanical engineering undergraduate, assisted in the experimental work on stabilization by jets, carried out most of the calculations involved in Sections 2 and 3, and assisted with some of the photographic work. Miss Marianne Macdonald typed the thesis.

The author wishes to express particular thanks for two Studentships and a Fellowship awarded by the National Research Council of Canada.

SUMMARY

An idealized combustion chamber is considered in which a fraction of the combustion products is returned through a separate passage to heat the incoming mixture. The four factors determining the behaviour of the flame are recirculation, reaction, upstream conduction, and losses to the walls. If conduction is neglected, the equation relating the remaining terms is soluble graphically to give a curve of temperature against time or distance. For given inlet conditions a critical degree of recirculation is specified, below which combustion will not be maintained.

Experiments were carried out with jets directed upstream in a parallel tube to entrain combustion products and return them to the pre-combustion region. Initially, increasing jet flow increases the jet penetration and extends the range of fuel/air ratios for which combustion is maintained. However at higher jet flows the walls of the surrounding tube begin to have an unfavourable effect and penetration is reduced. Eventually increasing jet flow leads to decreasing stability.

Two refractory stabilization ducts were built in which burnt gases were recirculated through a separate tube, which was calibrated for flow measurement in the second duct. The recirculation was induced and controlled by means of an ejector. Extraneous stabilizing effects and material failures prevented any useful measurements being made. The construction of the ducts is described in detail, particularly the casting of complex shapes.

Recommendations are made for the construction of improved ducts which might permit satisfactory measurements of recirculation to be made. Alternatively, it may be necessary to use a less direct approach, such as photographic observation of the flow patterns in eddies caused by air injection or obstacles in the stream.

CONTENTS

1. INTRODUCTION

Self-maintaining combustion consists of an exothermic chemical reaction proceeding at such a rate that heat is produced at least as fast as it is carried away. In mixtures of hydrocarbon gases and air the reaction proceeds so slowly at normal temperatures that the heat is carried away as quickly as it is produced and there is negligible temperature rise. If, however, the mixture is heated, the reaction proceeds more rapidly (roughly according to exp $\left(-\frac{E}{R_0 T}\right)$, the fraction of particle collisions which involve a joint energy greater than E, R_0 being the gas constant and T the temperature) and eventually heat is produced faster than it is lost. The reaction accelerates rapidly, finally slowing down as it nears completion because collisions between suitable particles become less frequent. The rapid phase of the reaction during which most of the combustible material is burnt, is accompanied by emission of light, forming the visible "flame".

If the initial heating is local, heat is conducted from the region of combustion to the adjoining mixture, accompanied by the diffusion of active radicals, and the flame propagates at a speed which depends on many factors, particularly the thermal conductivity, density, and specific heat of the gases involved, as well as the initial and final temperatures and the rate of heat loss. It is generally assumed that radical diffusion can be allowed for by changing the numerical factor in the heat conduction equations, since similar laws are followed.

In the ideal case of combustible mixture flowing in a parallel tube with no heat loss or addition a flame will be stationary if the speed of the fresh mixture moving down the tube is equal to the speed at which the flame propagates upstream. In general, a flame in a moving stream is said to be "stabilized" if it remains stationary relative to the environment.

The above type of stabilization is unattainable in practice because any change in the mixture speed or burning speed causes the flame to travel upstream or downstream with a speed equal to the difference, and the flame "blows back" or "blows out". If, however, the tube has diverging walls the speed of the incoming mixture varies down the tube and the flame may find a location where the speeds are equal. Changing conditions cause the flame to fluctuate slightly upstream and downstream. This kind of stabilization can only exist for flow rates such that the normal flame speed, usually of the order one metre per second (4), is not greatly exceeded.

At low flows a flame can be stabilized in a conducting parallel tube by means of heat carried upstream by the walls. If the wall temperatures are fixed the flame is also fixed spatially.

At high flows heat cannot be conducted back by the gases, nor even the walls, quickly enough to warm the incoming mixture adequately. It is then necessary to separate some of the hot combustion products, carry them back to the pre-combustion region, and allow them to heat the fresh gases in mixing with them. This is called "recirculation".

Recirculation can be brought about in many ways: for example, by inserting an obstacle in the stream to produce eddies, by directing a jet upstream to entrain combustion products and carry them to the cold region, by providing a vortex around the combustion region which when collapsing produces a pressure gradient directing the hot gases upstream along the axis, or by introducing air through openings in the side of the combustion tube so as to cause eddies. The latter two techniques are commonly used in gas turbine combustion chamber design.

Stabilization on obstacles has been thoroughly investigated by Williams (12). Behind such obstacles eddies exist into which hot gases are drawn at the downstream end to be expelled at the upstream end, heating the fresh mixture. It was found that the stability limits are determined by the size of the obstacle and not the shape (except for streamlined obstacles, where no eddy exists). Williams reports that good correlation was obtained by plotting $\frac{V_{BO}}{n^{0.45}}$ against air/fuel ratio, where V_{BO} is the blowout velocity and n is a characteristic dimension.

Controlled injection of air to produce eddies is widely used. The holes through which secondary air enters are located by trial and error, until the combustion chamber will operate over an acceptable range of flows and mixture ratios. The flow patterns are almost impossible to predict. Bailey and Figueroa (1) made a study of a simplified form of this type of chamber. There was qualitative agreement with their theory.

A theory relating quantity of recirculation to range of stability would be very useful. Fig. 1 shows a model combustion chamber proposed by Mordell (6). Air enters and is joined by hot recirculated gases, then fuel is injected and vaporizes. There follows a delay period while the reaction builds up, and a short combustion region. Finally the combustion products leave the chamber, part being led back to mix with the entering air.

Knowing the final temperature reached for a given fuel/air ratio it is possible to make a heat balance at the point where the recirculated gases are mixed with the fresh air, to obtain the temperature when combustion begins. Then, from spontaneous ignition delay data, it is possible

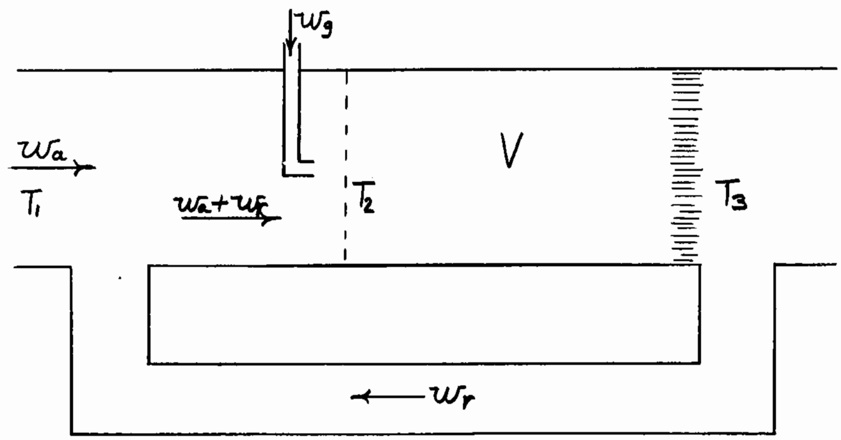


Fig. 1. Mordell's Model

to say whether or not there will be time for combustion to occur before the mixture reaches the end of the combustion space. In this way stability limits can be determined.

Let w_a and w_g be the mass flows of air and gas respectively, and w_r the recirculated mass flow. Let T_1 be the temperature of air and gas at inlet to the combustion chamber, T_2 the temperature after the recirculated gases have been added, and T_3 the temperature when combustion is completed. If A, l, and V are the area, length, and volume of the combustion space (between the point where air, gas, and recirculated products are mixed and the point where recirculation begins),

$$(w_a + w_g + w_r) = \rho_2 A \frac{l}{t} = \rho_2 \frac{V}{t} = \frac{P}{RT_2}\frac{V}{t} \,. \qquad (1.1)$$

This assumes that the density ρ is nearly constant during the time that the gases take to travel through the combustion space: this is justified because the temperature rises slowly at first, most of the combustion occurring in a short time at the end. The pressure is assumed constant. Solving the above equation,

$$t = \frac{PV}{RT_2}\frac{1}{w_a + w_g + w_r} \,. \qquad (1.2)$$

The final temperature T_3 is obtained from an overall energy balance:

$$(w_a + w_g) c T_3 = (w_a + w_g) c T_1 + w_g \eta H \; . \qquad (1.3)$$

Here c is the mean specific heat, η is the combustion efficiency, and H is the heating value of the fuel. Introducing the fuel/air ratio

$$\omega = \frac{w_g}{w_a} \; , \qquad (1.4)$$

we have

$$T_3 = T_1 + \frac{\omega}{1+\omega} \frac{\eta H}{c} \; . \qquad (1.5)$$

The temperature T_2 is found by a heat balance:

$$(w_a + w_g + w_r) T_2 = (w_a + w_g) T_1 + w_r T_3 \; . \qquad (1.6)$$

Let us define a recirculation parameter

$$\mu = \frac{w_r}{w_a} \; . \qquad (1.7)$$

(The ratio of recirculation to total combustion section flow is

$$\mu' = \frac{w_r}{w_a + w_g + w_r} = \frac{\mu}{1 + \omega + \mu} \doteq \frac{\mu}{1+\mu} \; .)$$

Then

$$T_2 = \frac{(1+\omega) T_1 + \mu T_3}{1 + \omega + \mu} \; . \qquad (1.8)$$

Substituting for T_3 from (1.5),

$$T_2 = \frac{(1+\omega+\mu) T_1 + \frac{\omega\mu}{1+\omega} \frac{\eta H}{c}}{1 + \omega + \mu} \; . \qquad (1.9)$$

If we define

$$\alpha = 1 + \omega + \mu \qquad (1.10)$$

and

$$\beta = \frac{\omega\mu}{1+\omega} \frac{\eta H}{c} \; , \qquad (1.11)$$

$$T_2 = \frac{\alpha T_1 + \beta}{\alpha} \; . \qquad (1.12)$$

Studies of spontaneous ignition delay, e.g. Mullins (7), show that when fuel is injected into heated air there is a delay while the reaction slowly accelerates before the visible flame accompanying rapid combustion appears. The delay time τ is related to the initial temperature T_2 by the equation

$$\log \tau = A + \frac{B}{T_2} \quad , \tag{1.13}$$

where A and B are empirical constants.

The stability limit is obtained by identifying this delay time with the time taken for the gases to flow through the combustion section, equation (1.2). Substituting from equations (1.2), (1.10) and (1.12),

$$\log \left[\frac{PV}{R w_a} \frac{1}{\alpha T_1 + \beta} \right] = A + \frac{\alpha B}{\alpha T_1 + \beta} \quad . \tag{1.14}$$

This can be solved to give the inlet mass flow w_a at extinction as a function of the fuel/air ratio ω and the recirculation parameter μ, which are contained in the variables α and β:

$$\log w_a = \log \frac{PV}{R} - A - \log(\alpha T_1 + \beta) - \frac{\alpha B}{\alpha T_1 + \beta} \quad . \tag{1.15}$$

Given an air mass flow and fuel/air ratio, this equation can be solved (graphically) for the required recirculation. When this calculation is carried out for a Rolls-Royce "Dart" combustion chamber operating at a kerosene/air ratio of 0.055, μ is found to be 0.46, and the ratio of recirculation to total flow is

$$\mu' = \frac{\mu}{1+\mu} = 0.32 \quad ,$$

which is not unreasonable. However, with a fuel/air ratio of 0.027, near the weak operating limit, the required recirculation is much larger: $\mu = 1.85$, or $\mu' = 0.65$, which does seem excessive.

This theory assumes that no combustion occurs until the recirculated gases have mixed with the incoming air and fuel. In an actual combustion chamber it is almost certain that at least some of the gas and air would be mixed before being joined by the recirculated flow. As soon as mixing began, some of the fresh mixture would be heated greatly, almost to T_3 , and would begin to burn. The flame would then spread through the remaining mixture. Thus only enough recirculation to heat up a small part of the mixture would be required, and the fraction of the total flow recirculated could be much smaller.

It is evident that experimental investigations are required to provide some insight into the mechanism of stabilization by recirculation. Rogers (8) used an arrangement in which a jet of air was blown upstream towards the incoming air (into which gaseous fuel was injected), Fig. 2a, with the object of entraining combustion products and carrying them to the pre-combustion region. Actually, because of poor mixing and unsteady flow patterns, the combustion took place right at the gas jets, indicating that small eddies were producing the stabilization. The apparatus was modified, Fig. 2b, so that separately burnt gases were supplied close to the gas jets. With this arrangement it was found that increasing "recirculation" led to greater stability, over most of the range investigated. The objection to this method is that the recirculated products should come from the actual reacting gases so as to have the same temperature and composition.

Some further investigations of this nature were carried out (Section 3), but it became apparent that stabilization by jets is a complicated process, and attention was turned to a more positive method of measuring recirculation.

If a combustion chamber could be made similar to the idealized model in Fig. 1, but with the fuel and air pre-mixed, it should be pos-

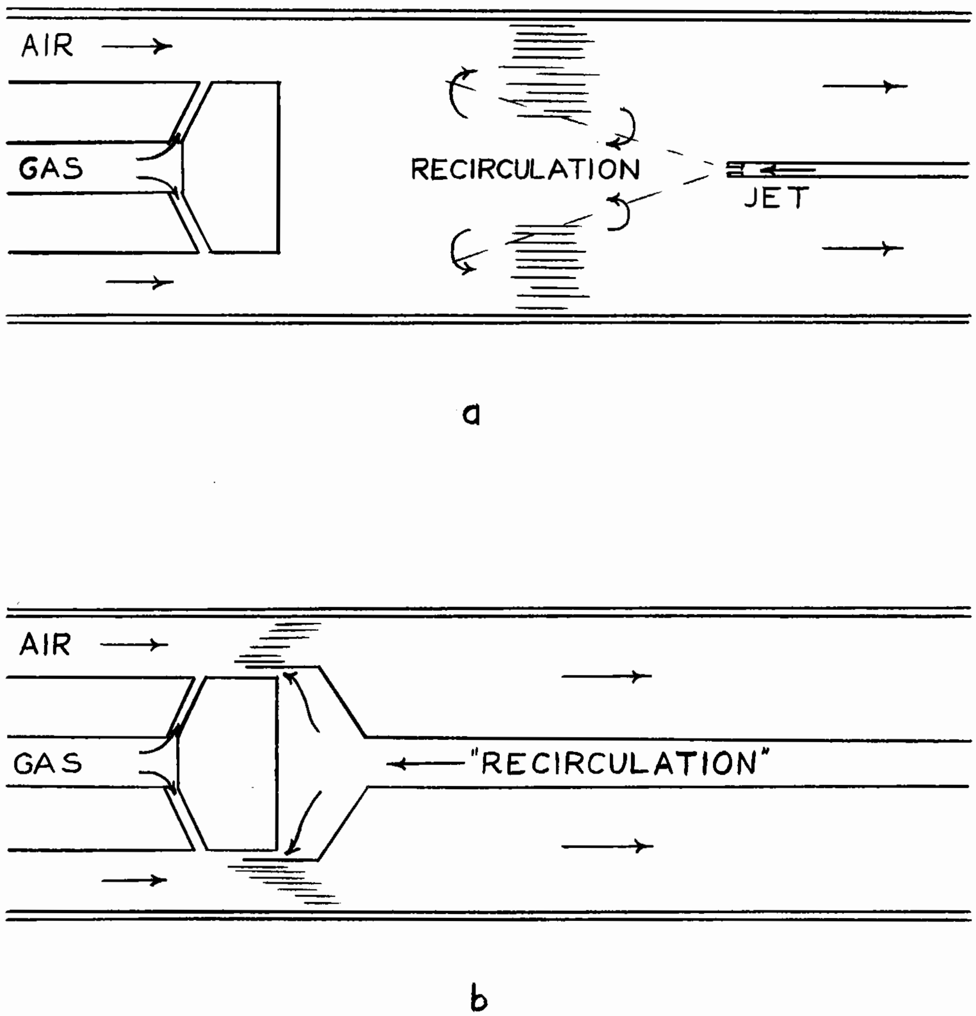


Fig. 2. Rogers' apparatus

sible to measure the amount of recirculation required to stabilize the flame. The essential point is that the recirculated combustion products must be delivered to the incoming mixture at the same temperature they had when they were taken from the exhaust, or as near to it as possible. Otherwise, their effectiveness will be greatly reduced, particularly in the light of the exponential effect of temperature. This means that the duct walls must be very near the flame temperature (say 2200 K). This rules out metal construction, and leads to the use of refractories.

The difficulties in the construction and operation of such a combustion duct are large, but it was considered that an attempt should be made. This was not very successful. Much trouble was encountered with extraneous stabilizing effects and with local failures of the refractory at high temperatures. However, experience was gained in the handling of castable refractories and recommendations are made for subsequent work.

2. THEORY

Some idea of the behaviour of the type of duct employed can be obtained from the following discussion, which makes use of the concepts introduced by Mallard and Le Chatelier (5). Pre-mixed fuel and air enter at a mass rate w with temperature T_0 and speed v_0, Fig. 3a. When the mixture is heated to the ignition temperature T_{ig} by recirculation and conduction, combustion occurs in a thickness δ, the final temperature being T_b.

A mass flow w_r is recirculated and mixes with the incoming mixture at a constant rate so that mixing is complete after a distance L (Fig. 3b). Therefore at a distance l after mixing begins, a fraction $\lambda = \frac{l}{L}$ of the recirculated gas has been added. (A similar argument can be followed on a time basis; distances are easier to visualize.) Conduction is assumed to be significant only in a short distance negligible compared with L. At the distance l, the temperature T_r is found by heat balance:

$$\left(w + \frac{l}{L} w_r\right) T_r = w T_0 + \frac{l}{L} w_r T_b . \qquad (2.1)$$

We may define a recirculation parameter $\mu = \frac{w_r}{w}$. Then

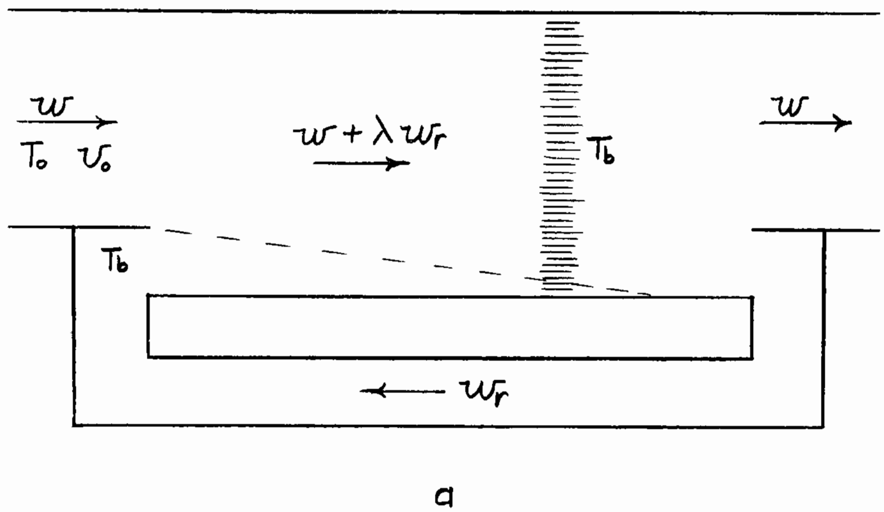
$$T_r = \frac{T_0 + \lambda \mu T_b}{1 + \lambda \mu} . \qquad (2.2)$$

The heating by conduction is, per unit area,

$$k \frac{T_b - T_{ig}}{\delta} = \rho_0 v_0 c \left(T_{ig} - T_r \right), \qquad (2.3)$$

where c is the specific heat and k may be greater than the thermal conductivity if there is turbulent flow. Substituting T_r from (2.2),

$$k \frac{T_b - T_{ig}}{\delta} = \rho_0 v_0 c \left[T_{ig} - \frac{T_0 + \lambda \mu T_b}{1 + \lambda \mu} \right], \qquad (2.4)$$



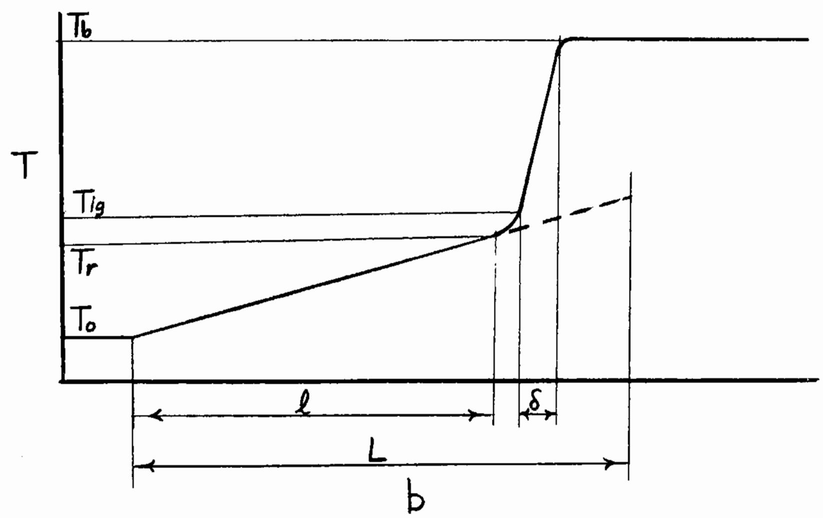


Fig. 3. Proposed duct.

whence
$$v_o = \frac{k}{\rho_o c}\,\frac{1}{\delta}\,\frac{T_b - T_{ig}}{T_{ig} - \dfrac{T_o + \lambda\mu T_b}{1+\lambda\mu}} \,. \qquad (2.5)$$

We have assumed that λ and μ are such that the flame is stationary. This need not be so, in which case the flame speed will have some other value v_f, given by

$$v_f = \frac{k}{\rho_o c}\,\frac{1}{\delta}\,\frac{T_b - T_{ig}}{T_{ig} - \dfrac{T_o + \lambda\mu T_b}{1+\lambda\mu}} \,. \qquad (2.5a)$$

Since $\frac{T_b}{T_o}$ is greater than one, $\frac{T_o + \lambda\mu T_b}{1+\lambda\mu} = T_o \frac{1 + \frac{T_b}{T_o}\lambda\mu}{1+\lambda\mu}$ increases as $\lambda\mu$ increases. The denominator in (2.5a) is therefore reduced, and the flame speed increases, as expected.

For the flame to be stationary, $\lambda\mu$ must have a fixed value μ^* determined by (2.5). (In particular, if conduction is neglected, i.e. $k=0$, $\mu^* = \frac{T_{ig} - T_0}{T_b - T_{ig}}$.) For any given recirculation, $\lambda = \frac{\mu^*}{\mu}$, i.e. the location of the flame is determined. As μ is decreased, λ must increase. However, λ is limited.

(a) λ cannot be greater than 1. Therefore if μ is decreased below μ^*, the flame will blow out.

(b) The space available may be less than L. If the flame moves downstream of the recirculating duct entrance the recirculation of heat stops, or is sharply reduced, since the temperature is lower. Also L is a function of v_0, T_0, T_b, at least. Then when $\mu < \frac{\mu^*}{\lambda_{max}}$ the flame will blow out.

Suppose the flame to be ignited upstream. The flame speed v_f is less than the mixture speed v_0, since $\lambda\mu$ is small or zero. Therefore the flame moves downstream until $\lambda\mu$ reaches the value μ^* corresponding to v_0, and v_f becomes equal to v_0. If the required value of μ^* is less than 0, v_f is greater than v_0 and the flame blows back.

If $\mu^* = 0$, (2.5a) becomes

$$v_f = \frac{k}{\rho_0 c} \frac{1}{\delta} \frac{T_b - T_{ig}}{T_{ig} - T_0} , \tag{2.6}$$

Mallard and Le Chatelier's equation for the flame speed.

The chief faults in this theory are 1. uniform heating of the mixture is assumed; 2. the ignition temperature is not a constant; and 3. the flame thickness δ is unknown. Nevertheless, it does give a qualitative picture.

Ignition temperature and flame thickness can be avoided by the introduction of the reaction rate. The rate of heat release in unit volume can be written

$$q_r' = C_r \rho^n \sqrt{T} \left(\frac{E}{R_o T}\right)^{\frac{1}{2}n-1} \exp\left(-\frac{E}{R_o T}\right). \qquad (2.7)$$

C_r is a factor dependent on the fuel/air ratio, the heating value of the fuel, the composition of air, etc. The frequency of collision between suitable particles is proportional to $\rho^n \sqrt{T}$: the density ρ is a measure of the concentrations of particles, n is the number of energy terms associated with the reaction (called the "order" of the reaction), and $\sqrt{T}$ allows for the average velocities of the particles. Of these collisions, only a certain fraction will involve a joint energy greater than E, the energy required for reaction. This fraction is $\left(\frac{E}{R_o T}\right)^{\frac{1}{2}n-1} \frac{\exp(-E/R_o T)}{\lfloor \frac{1}{2}n-1}$ (Hinshelwood (3)). The denominator was included in C_r in (2.7).

For a simple bi-molecular reaction, n equals 2 and (2.7) reduces to

$$q_r' = C_r \rho^2 \sqrt{T} \exp\left(-\frac{E}{R_o T}\right). \qquad (2.7a)$$

No allowance has been made here for the consumption of the reacting materials: the concentrations are assumed constant. Thus equation (2.7) applies only to the initial stages of combustion. However it is these initial stages with which we are concerned in studying the factors controlling a flame. Once the initial heating has taken place, the reaction proceeds so rapidly that it occupies relatively little space or time.

Let us consider a mixture of air and gas flowing through a parallel tube of cross sectional area A, and let us suppose that heat is added by recirculation, beginning at a fixed point. Then we may consider an element of gas at a distance l downstream from here, of thickness dl.

Its volume is $A\,d\ell$, and the mass contained in it is $\rho A\,d\ell$. The total rate of heat increase in this volume is the sum of the heat added by recirculation, heat released by reaction, and heat conducted upstream through the gas, less the heat lost to the walls.

i.e. $$q = q_\mu + q_r + q_c - q_L \quad . \qquad (2.8)$$

The temperature rise is given by

$$q = \rho A\,d\ell \cdot c\,\frac{dT}{dt} \quad , \qquad (2.9)$$

which can also be written

$$q = \rho A\,d\ell \cdot c\,\frac{dT}{d\ell}\,\frac{d\ell}{dt} \quad .$$

Since the total mass flow is

$$w = \rho A\,\frac{d\ell}{dt} \quad , \qquad (2.10)$$

this becomes

$$q = wc\,\frac{dT}{d\ell}\,d\ell \quad . \qquad (2.9a)$$

The heat gain by conduction upstream is

$$q_c = kA\,\frac{d^2T}{d\ell^2}\,d\ell \quad . \qquad (2.11)$$

The heat loss to the walls will be mostly by convection. Spiers (10) gives the heat transfer coefficient

$$h \propto c(\rho v)^{0.8} D^{-0.2} \quad ,$$

or, from (2.10) $$h \propto c\left(\frac{w}{A}\right)^{0.8} D^{-0.2} \quad .$$

Therefore $$q_L \propto c\left(\frac{w}{A}\right)^{0.8} D^{-0.2} \cdot \pi D\,d\ell\,(T - T_w) \quad ,$$

where T_w is the wall temperature. D is the hydraulic mean diameter. Introducing a proportionality factor C_L ,

$$q_L = C_L \left(\frac{w}{A}\right)^{0.8} D^{0.8} (T - T_w)\, dl \ . \qquad (2.12)$$

The recirculation is difficult to express. However, let us assume that recirculated gases at the final temperature T_b, which will depend on the mixture being burnt and an estimate of the overall heat loss, are added to the unburnt mixture at a constant rate, and that a total recirculated mass flow w_{ro} will have been added after a time t_o. Defining $\mu_o = \frac{w_{ro}}{w}$ and μ as the amount that has been added at time t,

$$\mu = \frac{t}{t_o}\mu_o \ , \qquad (2.13)$$

and

$$\frac{d\mu}{dt} = \frac{\mu_o}{t_o} \ . \qquad (2.13a)$$

We can make a heat balance for the recirculation dw_r added in time dt:

$$(w + dw_r)(T + dT) = wT + dw_r \cdot T_b \ .$$

Whence, neglecting the product of differentials,

$$w\, dT = (T_b - T)\, dw_r \ ,$$

or

$$\frac{dT}{dt} = (T_b - T)\frac{d\mu}{dt} \ . \qquad (2.14)$$

From (2.13a), this becomes

$$\frac{dT}{dt} = (T_b - T)\frac{\mu_o}{t_o} \ ,$$

and

$$q_\mu = (T_b - T)\frac{\mu_o}{t_o}\, c\, \rho A\, dl \ . \qquad (2.15)$$

Remembering that (2.7) was per unit volume, the whole equation (2.8) becomes

$$w c \frac{dT}{dl} = (T_b - T)\frac{\mu_0}{t_0} c\rho A\, dl + C_r \rho^n \sqrt{T}\left(\frac{E}{R_0 T}\right)^{\frac{1}{2}n-1} \exp\left(-\frac{E}{R_0 T}\right) A\, dl$$
$$+ kA\frac{d^2T}{dl^2}\, dl - C_L\left(\frac{w}{A}\right)^{0.8} D^{0.8}(T - T_w)\, dl \qquad (2.16)$$

If we re-arrange this, divide by $kA\,dl$, and substitute $\rho = \frac{P}{RT}$, we obtain

$$\frac{d^2T}{dl^2} - \frac{w}{A}\frac{c}{k}\cdot\frac{dT}{dl} = \frac{C_L}{kA}\left(\frac{w}{A}\right)^{0.8} D^{0.8}\cdot(T - T_w) - \frac{c}{k}\frac{\mu_0}{t_0}\frac{P}{R}\cdot\frac{T_b - T}{T}$$
$$- \frac{C_r}{k}\left(\frac{P}{R}\right)^n \left(\frac{E}{R_0}\right)^{\frac{1}{2}n-1} \cdot \frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}(n-1)}} \qquad (2.17)$$

For a given duct, gases, mixture, and pressure level, new constants can be introduced:

$$\frac{d^2T}{dl^2} - K\frac{dT}{dl} = L(T - T_w) - M\frac{T_b - T}{T} - R\,\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}(n-1)}}, \qquad (2.18)$$

or
$$\frac{d^2T}{dl^2} - K\frac{dT}{dl} = \Phi(T)\,. \qquad (2.19)$$

This type of equation is solved by introducing a new variable $p = \frac{dT}{dl}$, plotting p against T for various values of $\frac{dp}{dT}$, and tracing out a line through the grid formed by the p versus T lines and the slopes $\frac{dp}{dT}$ which are marked on them. When an actual example is tried, it is found that unless the initial temperature is very high the proportions of the graph are such that this cannot be done.

When appreciable heat is added by other means, for example by recirculation, the effect of conduction can usually be neglected. If we put $k = 0$ in (2.16), divide by $wc\,dl$ and substitute as before for ρ, we obtain

$$\frac{dT}{dl} = \frac{A}{w}\frac{\mu_0}{t_0}\frac{P}{R}\cdot\frac{T_b - T}{T} + \frac{C_r}{c}\frac{A}{w}\left(\frac{P}{R}\right)^n\left(\frac{E}{R_0}\right)^{\frac{1}{2}n-1}\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}(n-1)}}$$
$$- \frac{C_L}{cw}\left(\frac{w}{A}\right)^{0.8} D^{0.8}(T - T_w) \;. \qquad (2.20)$$

Introducing constants,

$$\frac{dT}{dl} = M'\frac{T_b - T}{T} + R'\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}(n-1)}} - L'(T - T_w) \,, \qquad (2.21)$$

or

$$\frac{dT}{dl} = \Phi'(T) \;. \qquad (2.22)$$

The solution of this is

$$l = \int_{T_0}^{T}\frac{dT}{\Phi'(T)} \;, \qquad (2.23)$$

and the flame position is

$$l' = \int_{T_0}^{T'}\frac{dT}{\Phi'(T)} \;, \qquad (2.24)$$

where T' is defined as that temperature beyond which further increase in l may be neglected.

In some cases it is preferable to have the temperature as a function of time. Thus if we use (2.9) instead of (2.9a), still ignoring conduction, (2.16) becomes

$$\rho A\, dl\, c\frac{dT}{dt} = (T_b - T)\frac{\mu_0}{t_0}c\rho A\, dl + C_r\rho^n V\overline{T}\left(\frac{E}{R_0 T}\right)^{\frac{1}{2}n-1}\exp\left(-\frac{E}{R_0 T}\right)A\, dl$$
$$- C_L\left(\frac{w}{A}\right)^{0.8} D^{0.8}(T - T_w)\, dl \;. \qquad (2.25)$$

Dividing by $\rho A c\, dl$ and substituting for ρ,

$$\frac{dT}{dt} = \frac{\mu_0}{t_0}\cdot(T_b - T) + \frac{C_r}{c}\left(\frac{P}{R}\right)^{n-1}\left(\frac{E}{R_0}\right)^{\frac{1}{2}n-1}\cdot\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}n-\frac{5}{2}}}$$
$$- \frac{C_L}{cA}\left(\frac{w}{A}\right)^{0.8} D^{0.8}\frac{R}{P}\cdot T(T - T_w) \;. \qquad (2.26)$$

With new constants,

$$\frac{dT}{dt} = M''(T_i - T) + R''\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}n-\frac{5}{2}}} - L'' T(T - T_w) , \qquad (2.27)$$

or

$$\frac{dT}{dt} = \Phi''(T) . \qquad (2.28)$$

The solution is

$$t = \int_{T_0}^{T} \frac{dT}{\Phi''(T)} . \qquad (2.29)$$

Reference has been made to spontaneous ignition delay data. In these experiments, fuel is injected into a stream of hot air. The reaction proceeds rapidly enough for the heat production to exceed the rate of loss, and therefore the mixture becomes hotter and the reaction accelerates. No recirculation is involved: the flame is located at a distance from the fuel injector dependent on the delay time and the flow speed. The heating by reaction is large enough so that the heat gained by conduction can be neglected. If we also neglect the losses (it may be noted that conduction and losses have opposite effects, and both increase as the temperature rises), equation (2.26) becomes

$$\frac{dT}{dt} = \frac{C_r}{c}\left(\frac{P}{R}\right)^{n-1}\left(\frac{E}{R_0}\right)^{\frac{1}{2}n-1}\frac{\exp\left(-\frac{E}{R_0 T}\right)}{T^{\frac{3}{2}n-\frac{5}{2}}} , \qquad (2.30)$$

and (2.29) gives

$$\tau = \frac{c}{C_r}\left(\frac{R_0}{E}\right)^{\frac{1}{2}n-1}\left(\frac{R}{P}\right)^{n-1}\int_{T_0}^{T'} T^{\frac{3}{2}n-\frac{5}{2}}\exp\left(\frac{E}{R_0 T}\right) dT , \qquad (2.31)$$

where τ is written for the delay time.

In actual practice, the delay distance is measured, and the time calculated from an estimated velocity. The delay distance could be calculated directly, using (2.20) instead of (2.26).

$$\frac{dT}{dl} = \frac{C_r}{c}\,\frac{A}{w}\left(\frac{E}{R_o}\right)^{\frac{1}{2}n-1}\left(\frac{P}{R}\right)^{n}\frac{\exp\left(-\frac{E}{R_oT}\right)}{T^{\frac{1}{2}(n-1)}} \quad . \tag{2.30a}$$

$$\int_0^{l_T} A\, dl = \frac{wc}{C_r}\left(\frac{R_o}{E}\right)^{\frac{1}{2}n-1}\left(\frac{R}{P}\right)^{n}\int_{T_o}^{T'} T^{\frac{1}{2}(n-1)}\exp\left(\frac{E}{R_oT}\right) dT \, . \tag{2.31a}$$

Equation (2.31a) allows for variable area.

The power of T in the integral makes little difference, being swamped by the exponential. Therefore the order n of the reaction can be determined from the relation between delay time and pressure. For example, Mullins (7) found that the delay time for kerosene was inversely proportional to the pressure, at the temperature level used in these experiments (around 1000 K). Therefore $n = 2$, and (2.31) becomes

$$\tau = \frac{c}{C_r}\,\frac{R}{P}\int_{T_o}^{T'} T^{\frac{1}{2}}\exp\left(\frac{E}{R_oT}\right) dT \, . \tag{2.32}$$

This presents a method of measuring the quantities C_r and $\frac{E}{R_o}$.

Let us define a function

$$f(x_o) = \int_{x_o}^{x'} x^{\frac{1}{2}}\exp\left(\frac{1}{x}\right) dx \quad . \tag{2.33}$$

Then letting $x_o = \frac{R_oT_o}{E}$, (2.34)

$$\tau = \frac{c}{C_r}\,\frac{R}{P}\left(\frac{E}{R_o}\right)^{\frac{3}{2}} f(x_o) \quad . \tag{2.35}$$

Two points are chosen on the experimental delay time versus initial temperature curve, (T_{o1} , τ_1) and (T_{o2} , τ_2). Now we choose an x_{o1} . From (2.34), we know that $x_{o2} = \frac{T_{o2}}{T_{o1}}\, x_{o1}$. (2.36)

From (2.33) which can be plotted for an appropriate range, we then know $f(x_{o1})$ and $f(x_{o2})$. These must satisfy the relation

$$\frac{f(\chi_{02})}{f(\chi_{01})} = \frac{T_2}{T_1} \tag{2.37}$$

which follows from (2.35). We plot the ratio $\frac{f(\chi_{02})}{f(\chi_{01})}$ against χ_{01}, and obtain the value of χ_{01} for which (2.37) is satisfied.

Then
$$\frac{E}{R_o} = \frac{T_{01}}{\chi_{01}} \tag{2.38}$$

and
$$C_r = c\,\frac{R}{P}\left(\frac{E}{R_o}\right)^{\frac{3}{2}} \frac{f(\chi_{01})}{\tau_1}, \tag{2.39}$$

from (2.34) and (2.35) respectively.

This calculation was carried out for the following pair of points

	T_o	τ
1	1100 K	0.0166 sec
2	1200	0.0031

taken from Mullins' curves for kerosene at atmospheric pressure (7). The results were

$$\frac{E}{R_o} = 25,800 \text{ K} \tag{2.40}$$

and
$$C_r = c\,\frac{R}{P} \times (1.086 \times 10^{15} \text{ deg}^{\frac{3}{2}} \text{ per sec}). \tag{2.41}$$

Having obtained these quantities we may carry out a solution of equation (2.26) to obtain a curve of temperature against time. (A curve of temperature versus distance could be obtained from equation (2.20).)

Since $n = 2$, the reaction constant R'' becomes

$$R'' = \frac{C_r}{c}\,\frac{P}{R} = 1.086 \times 10^{15} \text{ deg}^{\frac{3}{2}} \text{ per sec}. \tag{2.42}$$

If we take the mass flow w to be 0.01 kg per sec, the cross sectional area A 0.12×10^{-4} m^2, and hydraulic mean diameter D 0.035 m, values typical of the size of stabilization duct to be built, the loss factor L'' is

$$L'' = 0.0084 \text{ per deg sec} . \tag{2.43}$$

The recirculation coefficient M'' is simply $\frac{\mu_0}{t_0}$. If we assume that an average pre-combustion velocity of 7 metres per second will give a mixing distance of 0.1 metre, the time t_0 until the recirculated gases are all added is 14.3 milliseconds. Let us choose a recirculation parameter $\mu_0 = 0.50$. Then the ratio of recirculation to total flow is

$$\mu_0' = \frac{\mu_0}{1+\mu_0} = 0.33 .$$

The coefficient is

$$M'' = \frac{\mu_0}{t_0} = \frac{0.50}{0.0143} = 35 \text{ per sec} . \tag{2.44}$$

If the wall temperature T_w is taken to be 300 K and the post-combustion temperature T_b 2000 K,

$$\Phi''(T) = 35(2000-T) + 1.086\times10^{15}\,\frac{\exp(-\frac{25,800}{T})}{T^{\frac{1}{2}}}$$

$$- 0.0084\,T(T-300) . \tag{2.45}$$

The reciprocal of this, $1/\Phi''(T)$, is plotted against T in broken lines in Fig. 4. Time is given by the area below this curve - equation (2.29) - and is plotted as a solid line. If Fig. 4 is viewed from the left, it becomes a plot of temperature against time. When $t = 14.3$ msec the recirculated gases are used up. From then on the recirculation term must be removed from $\Phi''(T)$, which becomes negative: that is to say, heat is lost to the walls faster than it is produced by reaction. To keep time increasing, dT must become negative, and the temperature drops. There is insufficient recirculation to stabilize the flame. In fact, since combustion does not proceed, the recirculated gases are cool and the incoming mixture is not heated even this much: the whole process dies out.

Let us choose a larger recirculation: $\mu_o = 0.80$, $\mu_o' = 0.44$.

Then

$$M'' = 56 \text{ per sec} . \qquad (2.44a)$$

The same process is carried out, Fig. 5, with the mixture heating more quickly - time increasing less rapidly with temperature. When 14.3 msec have passed and $\Phi''(T)$ is modified it is still positive, and the temperature continues to rise, though slowly. Eventually the reaction accelerates, so that the flame occurs when t equals about 22 milliseconds.

If the recirculation is still larger, say $\mu_o = 1.00$, $\mu_o' = 0.50$, $M'' = 70$, the flame appears before the recirculation is used up, Fig. 6, and there is no break in the curves.

The rate of temperature rise decreases at first as the losses become larger, but later increases rapidly as reaction becomes significant.

The heat conducted upstream at any point is $kA \frac{dT}{dl}$. This could heat the incoming flow w an amount ΔT given by $wc\Delta T = kA \frac{dT}{dl}$. Now

$$\frac{dT}{dl} = \frac{dT}{dt}\frac{dt}{dl} = \frac{dT}{dt}\frac{A\rho}{w} = \frac{dT}{dt}\frac{A}{w}\frac{P}{R}\frac{1}{T},$$

so

$$\Delta T = \frac{k}{c}\left(\frac{A}{w}\right)^2 \frac{P}{R} \frac{\Phi''(T)}{T} . \qquad (2.46)$$

Even at 1600 K, this works out to less than 0.2 Kelvin degree, so the effect of conduction is certainly negligible.

It is evident from Figs. 4 and 5 that some intermediate degree

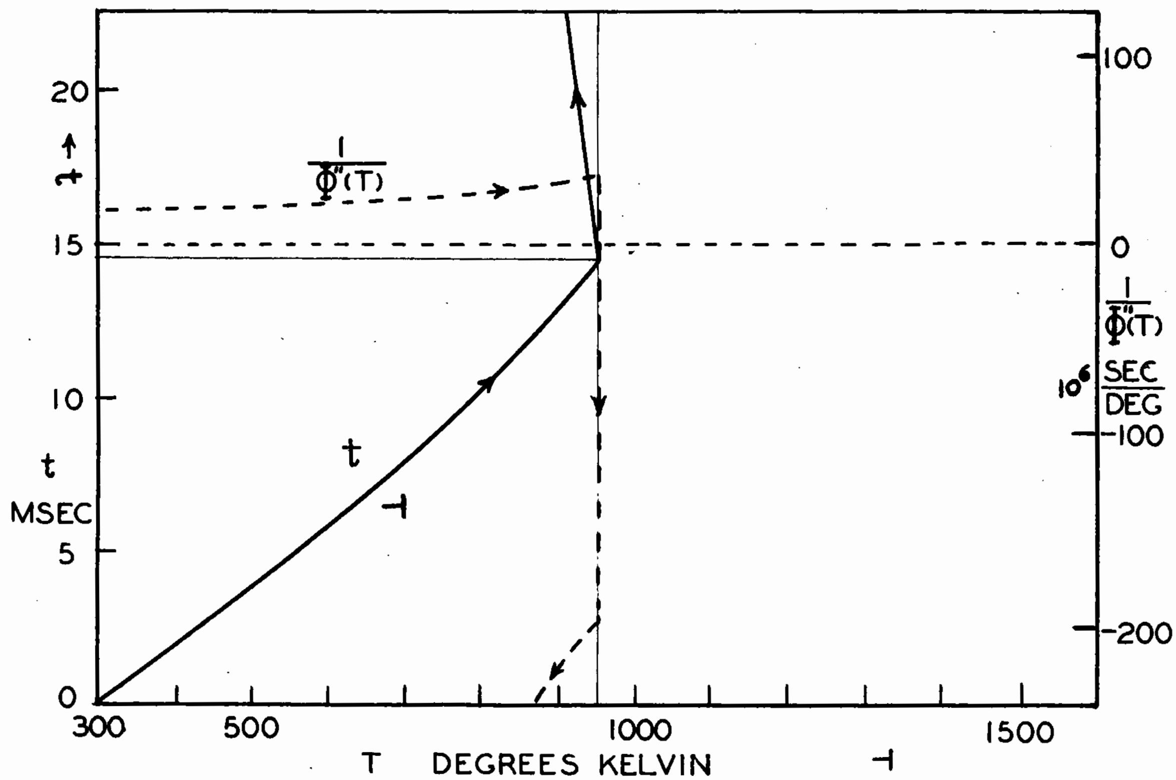


FIG. 4. TIME VERSUS TEMPERATURE-LOW RECIRCULATION

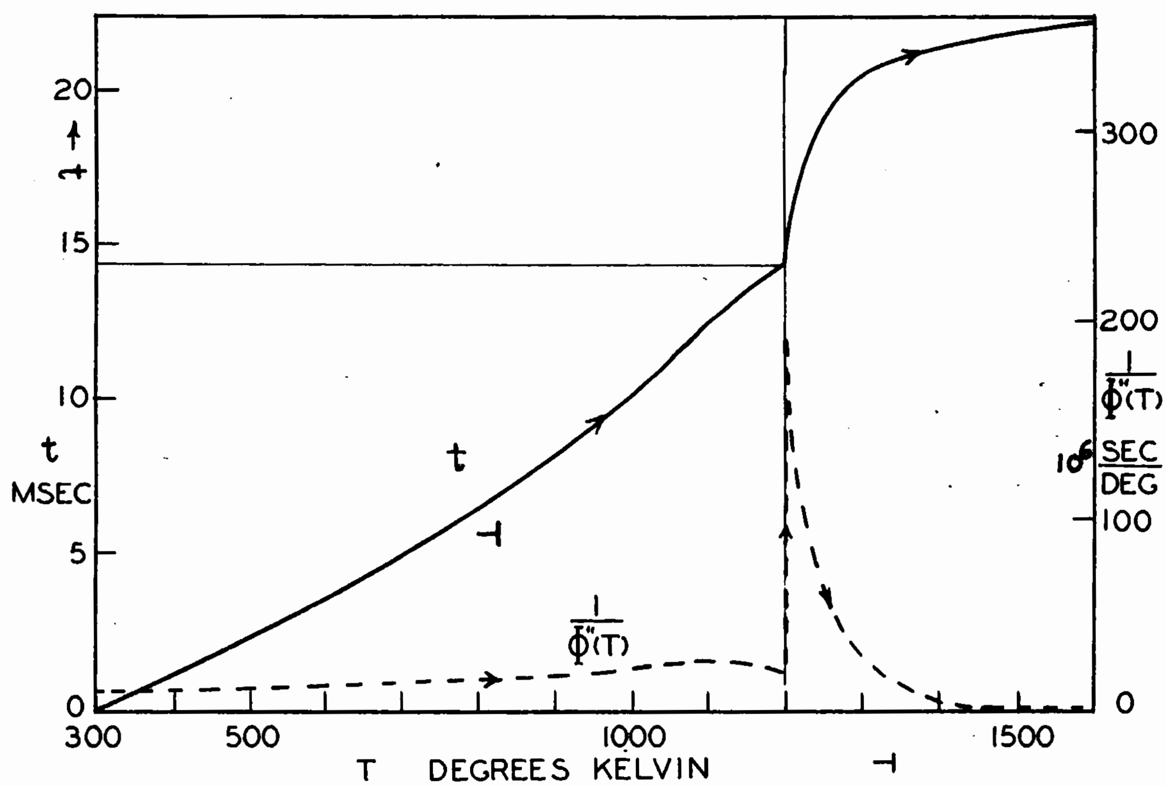


FIG. 5. TIME VERSUS TEMPERATURE-HIGH RECIRCULATION

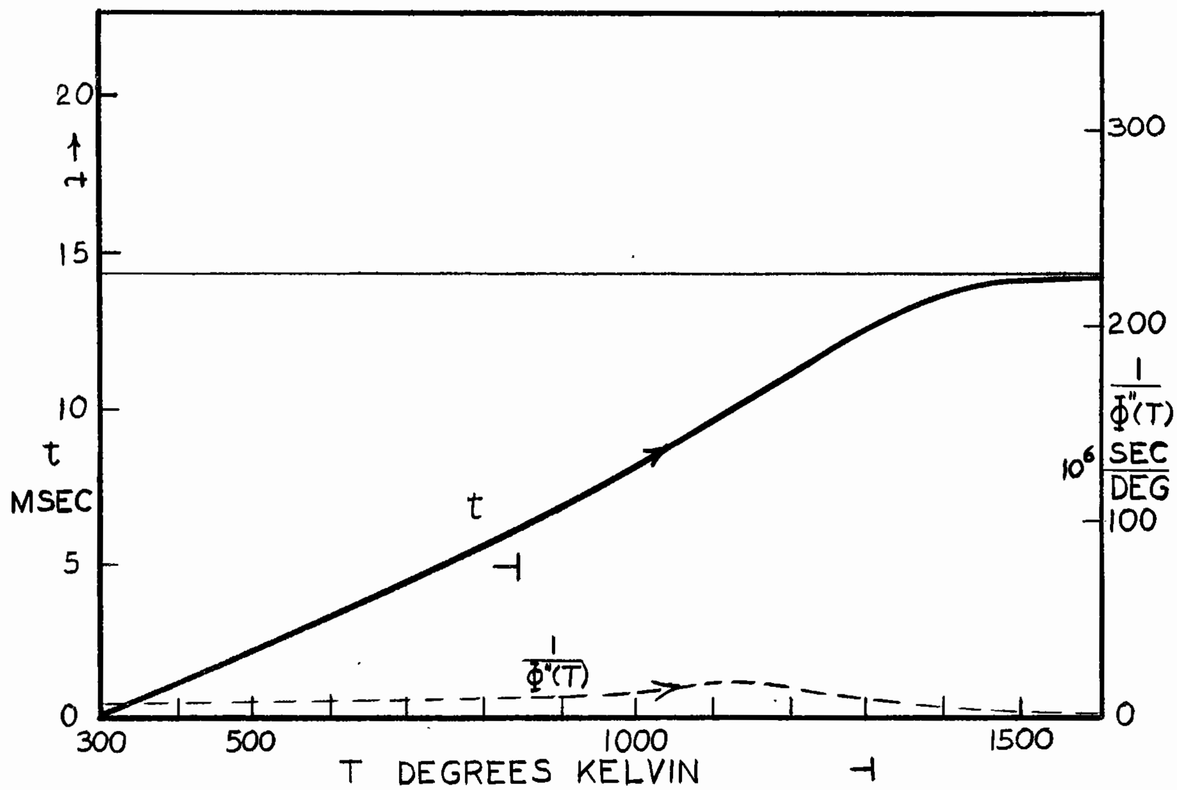


Fig. 6. Time versus temperature - very high recirculation

of recirculation would bring the gases to a critical temperature T_c at which the heat produced by reaction just balances the heat lost to the walls.

i.e.

$$R'' \frac{\exp\left(-\frac{E}{R_o T_c}\right)}{T_c^{\frac{1}{2}}} - L'' T_c (T_c - T_w) = 0 \quad . \qquad (2.47)$$

When this equation is solved graphically, T_c is found to be 1170 K for this particular set of conditions. If the recirculation is sufficient to heat the mixture above this temperature, combustion will proceed. If not, the gases will cool down again.

Since the left hand side of (2.47) is small for temperatures below T_c , we may write equation (2.27) approximately as

$$\frac{dT}{dt} = \frac{\mu_o}{t_o} (T_b - T) \quad . \qquad (2.48)$$

Therefore

$$\frac{\mu_0}{t_0}\int_0^{t_0} dt = \int_{T_0}^{T_c} \frac{dT}{T_b - T} , \qquad (2.49)$$

and

$$\mu_0 = 2.303 \log \frac{T_b - T_0}{T_b - T_c} , \qquad (2.50)$$

which is independent of t_0, since we have neglected losses and reaction. Thus for our case,

$$\mu_0 = 2.303 \log \frac{2000 - 300}{2000 - 1170} = 0.72 , \qquad (2.51)$$

$$\mu_0' = \frac{0.72}{1.72} = 0.42 . \qquad (2.52)$$

This figure seems high: the theory still assumes that the fresh mixture is heated uniformly by the recirculated gases. Still more recirculation would be required if the recirculated gases were cooled during their circuit, lowering the effective value of T_b.

So far it has been assumed that the combustion chamber is long enough for combustion to take place before the point where the recirculated gases are withdrawn. If this is not so, a further limitation on the stability exists. The distance to the flame can be determined roughly by assuming a mean temperature for the pre-combustion region:

$$l' = \frac{w}{\rho A} t' = \frac{w}{A}\frac{R}{P} T_m \cdot t' , \qquad (2.53)$$

or else T can be obtained as a function of l in the first place.

The reaction constant C_r and the final temperature T_b depend on the fuel/air ratio, so it would be possible to obtain curves of required recirculation against fuel/air ratio for constant inlet mass flows, for a given combustion chamber.

3. STABILIZATION BY AN AIR JET

An effort was made to obtain further data on the stabilization produced by a jet of air directed upstream. The apparatus used, Fig. 7, was similar to that of Rogers (8) except that the air and gas, in this case propane, were mixed before entering the Vycor glass outer tube, and there was no extraneous recirculation caused by enlargement of section. The jets were made from hypodermic needle tubing, curved so that the silver-soldered joints were out of the hot gas stream. A heavier surrounding tube gave support and some protection from the flame; however, combustion could only be allowed to continue for about 15 seconds at a time. The jet tubes reached temperatures around 1450 K, according to the optical pyrometer.

The Vycor tube had an inside diameter of 34 mm, and 2 mm walls. The jet gauges and inside diameters, measured with a vernier microscope, were as follows:

16 gauge	1.20 mm
18	0.87
20	0.59
22	0.42

It must be noted that because of expansion, distortion, and formation of scale the diameters were far from constant.

A three dimensional traversing gear permitted centering of the jet and removal for maintenance but during combustion the tip often lifted because of the jet tube being heated more on the upstream side. A pair of pointers straddling the glass tube and slidable on the traversing carriage permitted the distance of the flame apex from the jet tip to be measured just before extinction, which was brought about by weakening the mixture. Rich extinctions could not be done because the excess

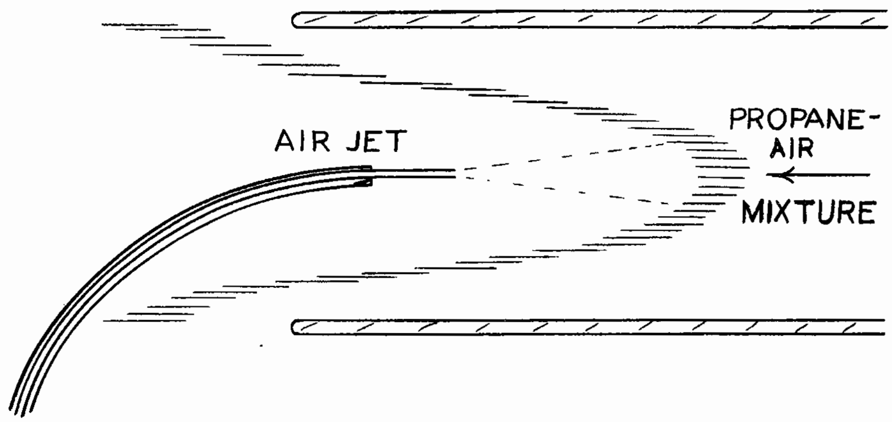


Fig. 7. Stabilizing jet within outer tube

fuel would burn outside the end of the glass tube, endangering the fume hood which collected the exhaust gases. In any case they would not be valid (unless the jet were of propane) because the jet air would promote combustion itself.

In making the observations, the jet flow varied greatly from one moment to the next; in particular it dropped rapidly during combustion. Therefore the main air flow was held at fixed values, as nearly as possible, and in Fig. 8 the jet mass flow w_j is plotted against fuel/air ratio w_f / w_a for constant inlet velocities v, with a separate set of curves for each jet of diameter d. The mean inlet velocities used, with their standard errors, were

v	Standard error
3.79 metres per sec	.08
4.90	.08
6.13	.11
8.34	.24
10.50	.23

FIG. 8

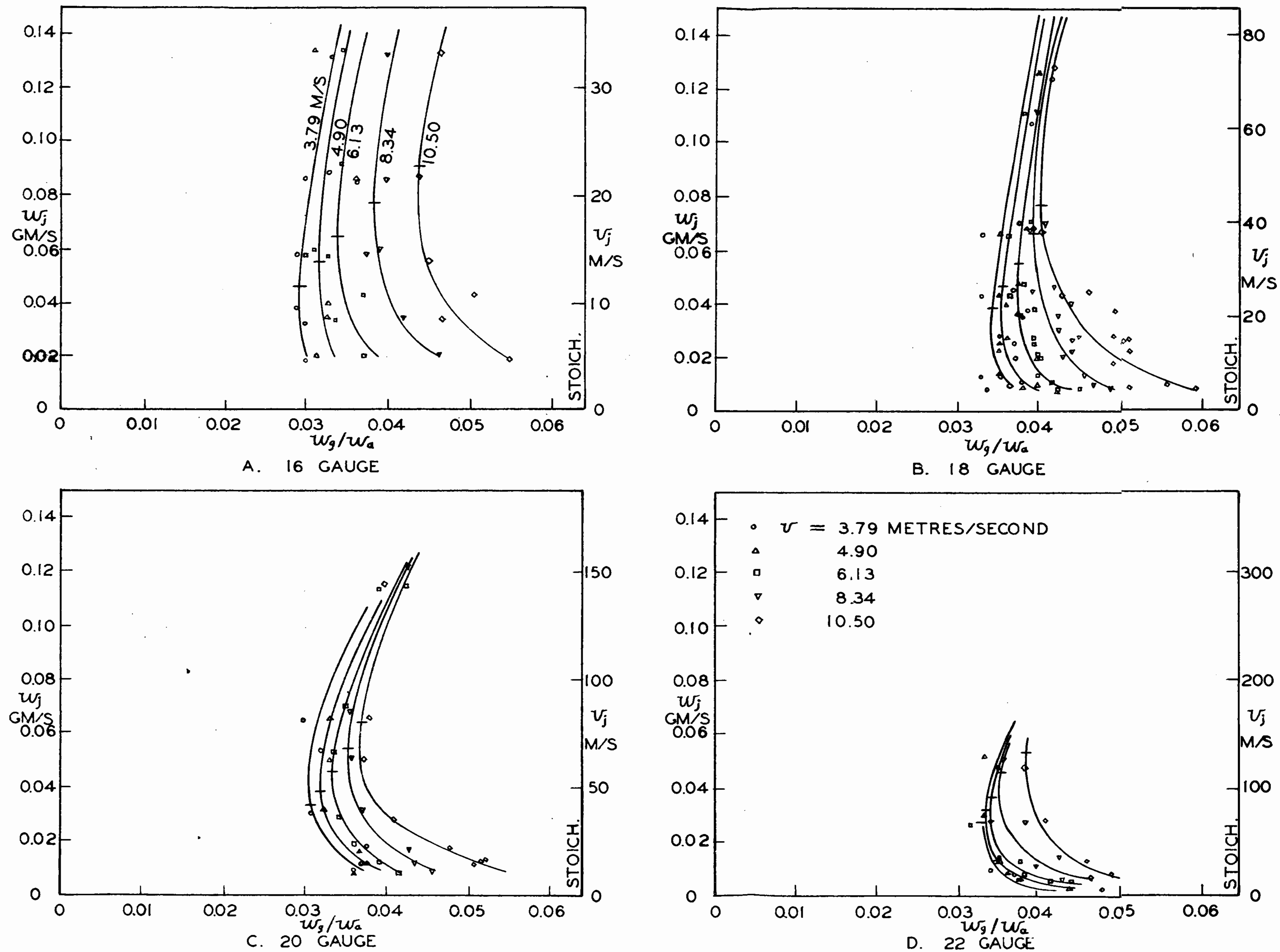


FIG. 8. JET MASS FLOW VERSUS FUEL/AIR RATIO

FIG. 9

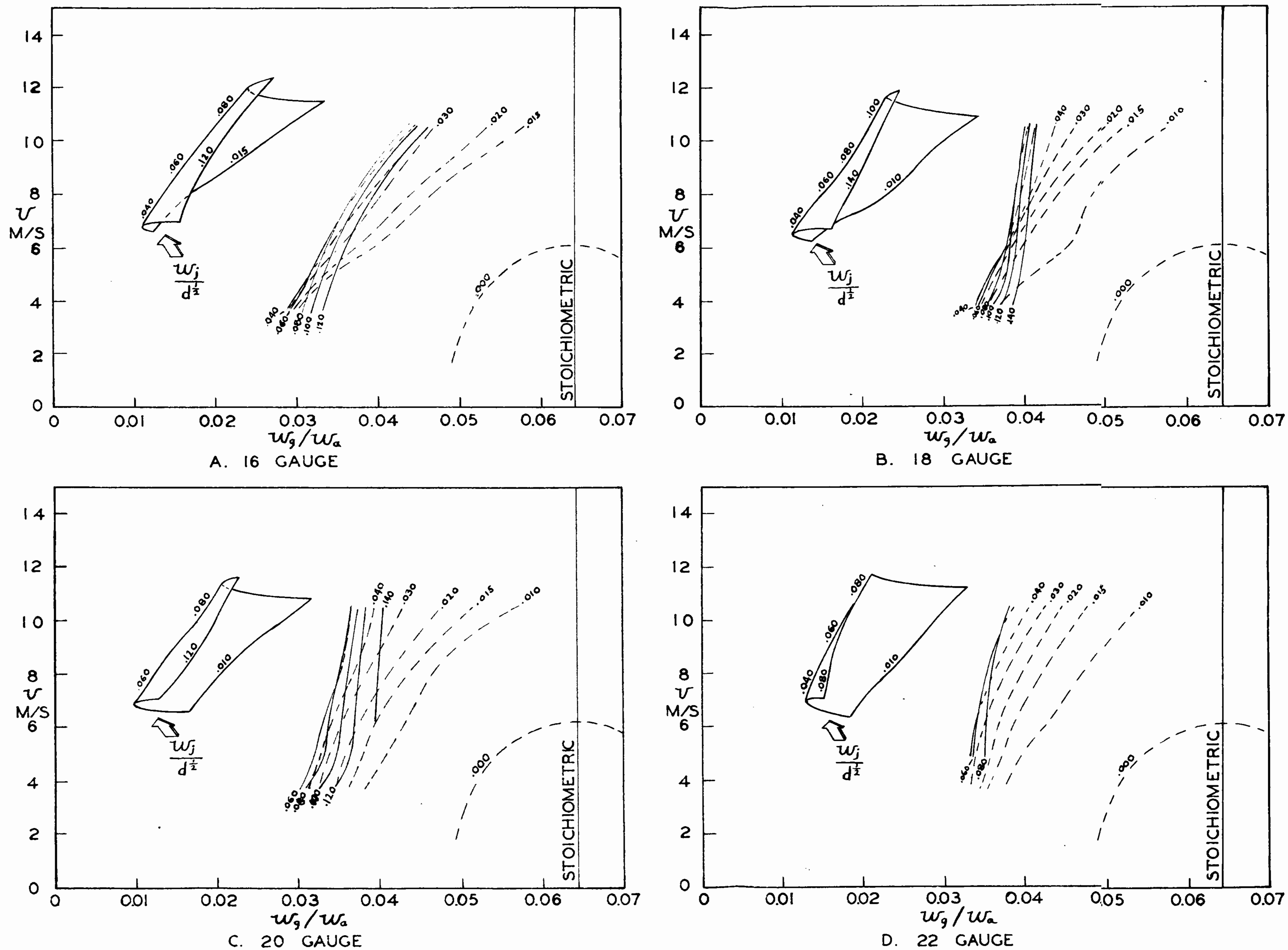


FIG. 9. MAIN STREAM VELOCITY VERSUS FUEL/AIR RATIO

The approximate jet velocities v_j are shown on the right-hand side of each graph.

It is seen that while increasing jet flow initially extends the range of stability, a maximum is soon reached after which the system becomes less stable. This may be accounted for by spreading of the relatively stagnant region where the jet flow meets the main flow, so that incoming mixture has to go through a smaller space near the outside of the glass tube, and is therefore accelerated. The jet mass flow for maximum stability is greater for higher inlet velocities.

As the jet diameter is reduced, the points of maximum stability occur at lower jet mass flows, but higher jet velocities. The effect of inlet velocity on range of stability varies markedly from one jet to another, but not very systematically. It is greatest for the 16 gauge tube, which gives the lowest jet velocities.

It was found that the jet mass flow for maximum stability, w_j^* (gm per sec), could be correlated with jet diameter d (mm) and inlet velocity v (metres per sec), as follows:

$$\frac{w_j^*}{d^{1/2}} = 0.0173\, v^{2/3}. \qquad (3.1)$$

The value of w_j^* obtained from this equation is shown on each constant velocity curve in Fig. 8 .

From each graph in Fig. 8 a plot was made of inlet velocity against fuel/air ratio for instant values of $w_j / d^{1/2}$. These are presented in Fig. 9 . The curve for no jet flow gives the velocities at which a flame, stabilized on a jet, blows back upstream. The lines of constant $w_j/d^{1/2}$ lie on a surface, as sketched on each graph. The twisting of the lines relative to the fold follows from the dependence of $w_j^*/d^{\frac{1}{2}}$ upon the inlet velocity v. The range of maximum stability does not vary greatly with jet diameter.

When a jet is directed upstream against a flow of air emerging from a large tube, a toroidal vortex is formed which recirculates hot combustion products to ignite the fresh mixture - see Fig. 10. It had been hoped that this would also be the case when the jet was within the large tube and the flow paths could be determined or estimated. Then it might be possible to calculate the amount of entrainment of combustion products, by a method similar to that of Goff and Coogan (2), based on Tollmein's theory (11). However, this was not the case. When confined within the tube the flame assumed a sharp paraboloid shape, as in Fig. 7, the flame appearing to start in the stagnation region at the end of the jet penetration.

The distance x from jet tip to flame apex is plotted against jet flow w_j for constant inlet velocities in Fig. 11. As the jet flow increases the jet penetrates farther at first, then the penetration reaches a maximum and decreases again, finally turning to increase slowly. This may be accounted for by the blocking of the large tube referred to earlier, with consequent speeding up of the fresh mixture. Also the way in which the jet spreads may change. At high jet flows the flame front becomes much flatter, almost forming a transverse disc across the tube. This suggests that the mechanism of mixing has changed, with turbulence becoming important.

The jet flow for which the penetration is a maximum is practically independent of main stream velocity for any given jet size. It can be related to jet diameter as follows:

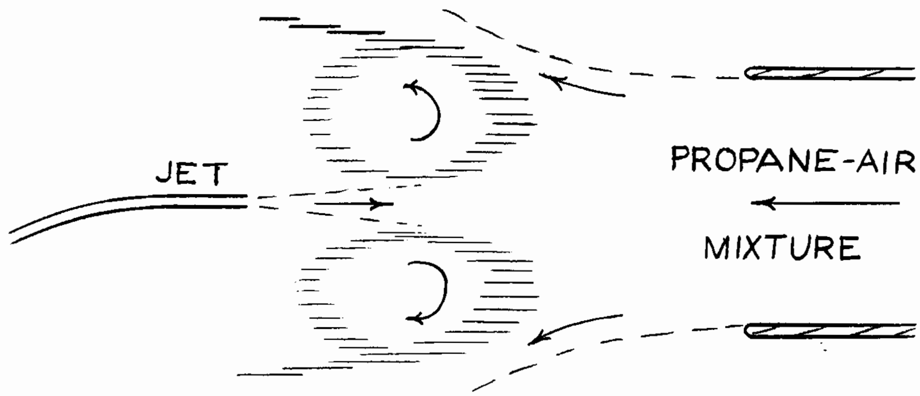


Fig. 10. Stabilizing jet outside

$$\frac{w_j^*}{d} = 0.031 \quad . \tag{3.2}$$

Likewise the maximum penetration x^* (cm) is related to the jet diameter d (mm) and the inlet velocity v (metres per sec) by

$$\frac{x^*}{d^{\frac{1}{5}}} = \frac{18.3}{v^{\frac{1}{2}}} \tag{3.3}$$

This being so, $\frac{w_j}{d}$ was plotted against $\frac{x\,v^{\frac{1}{2}}}{d^{\frac{1}{5}}}$ for all the observed extinctions, Fig. 12. No attempt is made here to identify the points: on the original plot colours were used to distinguish 20 symbols. There was no apparent systematic trend in the scatter about the curve.

The ratio $\frac{w_j^*}{d} = 0.031$ or $\left(\frac{w_j^*}{d}\right)^2 = 9.6 \times 10^{-4}$ for maximum penetration corresponds to a constant momentum flux

$$w_j^* v_j^* = \frac{w_j^{*2}}{\rho A} = \frac{4}{\pi\rho}\left(\frac{w_j^*}{d}\right)^2 \tag{3.4}$$

of 4.1×10^{-3} kg metres per sec per sec, which is equivalent to a force of 4.1×10^{-3} newtons (0.42 gm force).

An attempt was made to make cold air flow observations with the aid of smoke produced by mixing ammonium hydroxide and hydrochloric acid fumes. When both were introduced into the jet air supply the salt col-

FIG. 11

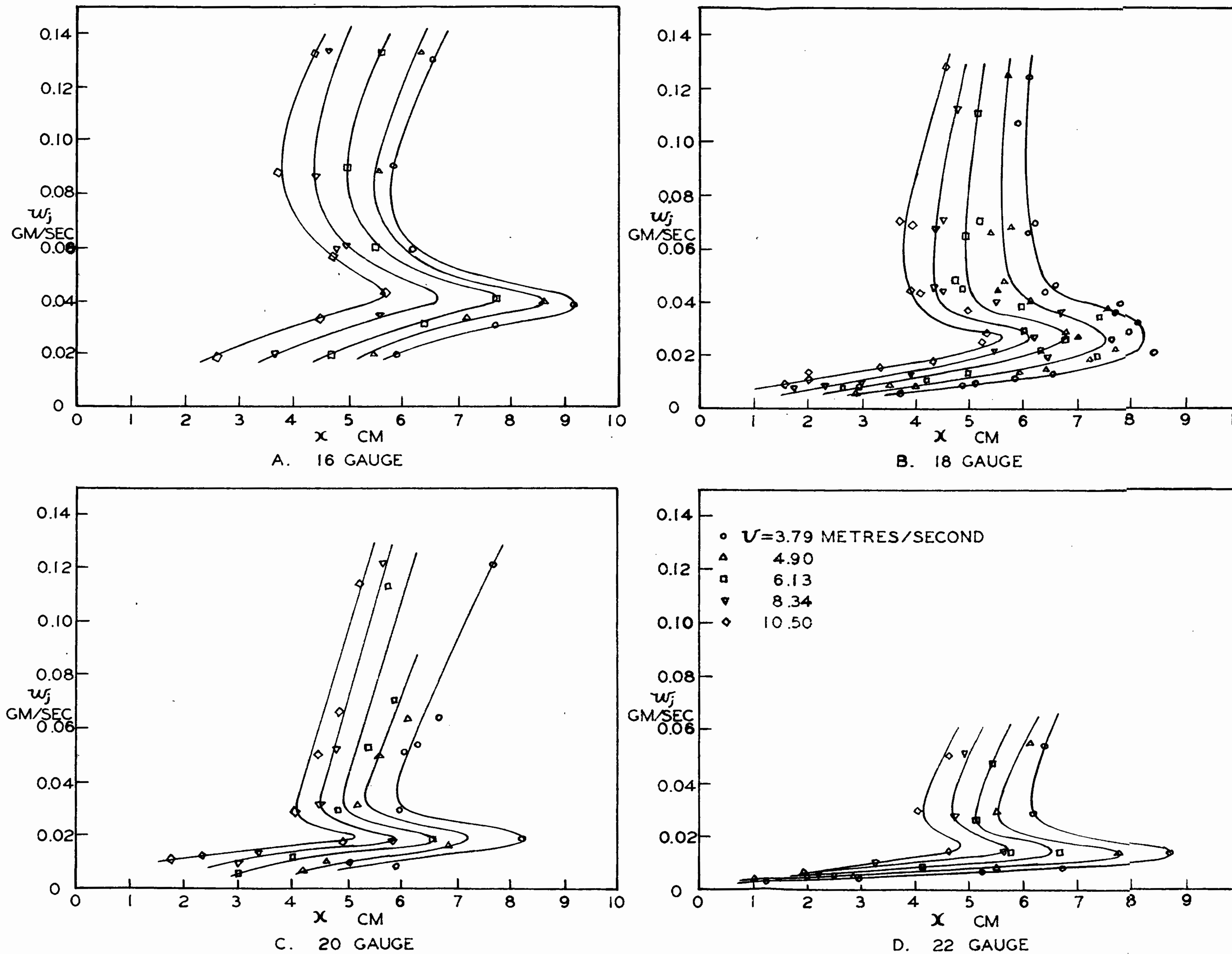


FIG. 11. JET MASS FLOW VERSUS PENETRATION

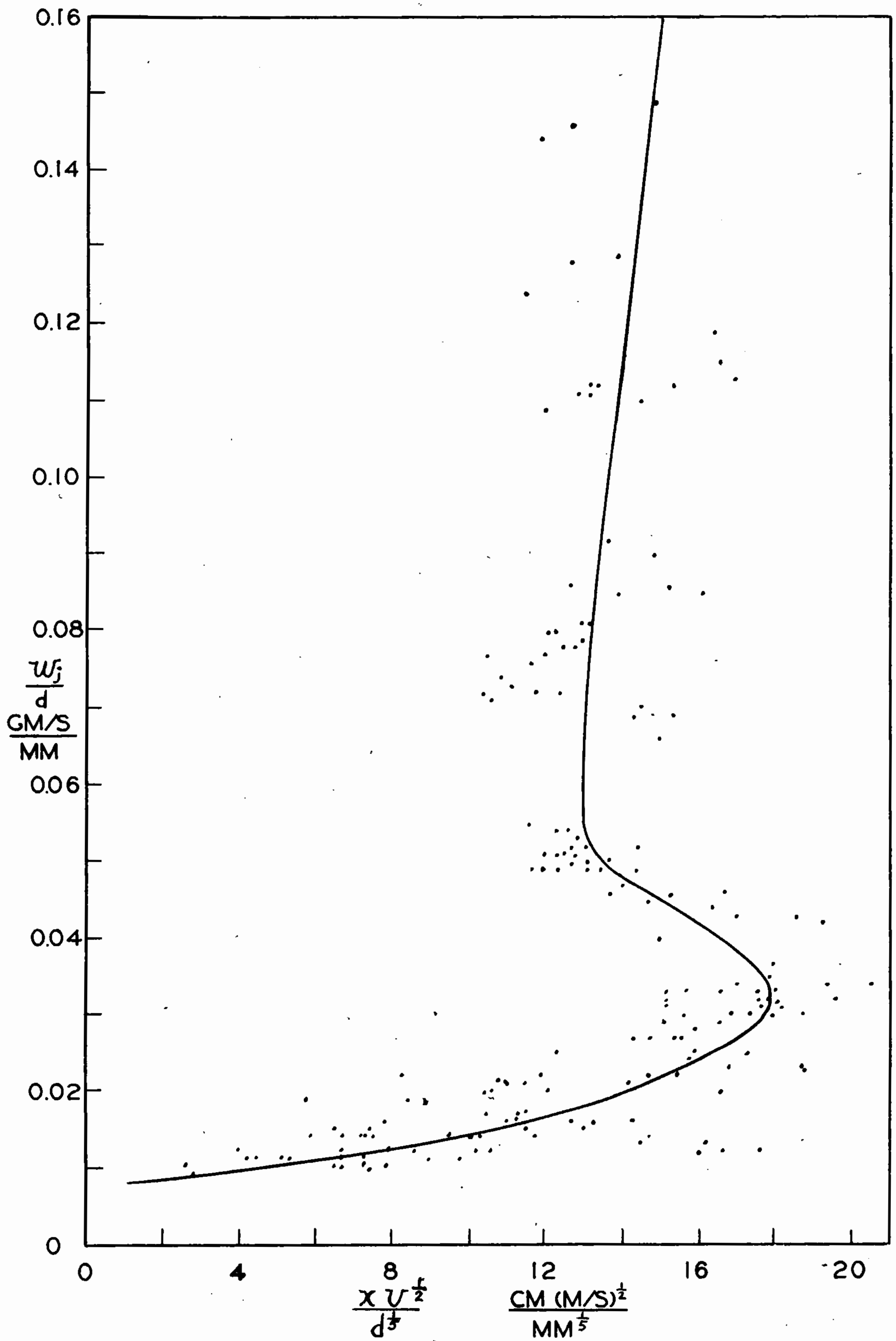


FIG. 12. $\frac{w_j}{d}$ VERSUS $\frac{x\,U^{\frac{1}{2}}}{d^{\frac{1}{3}}}$

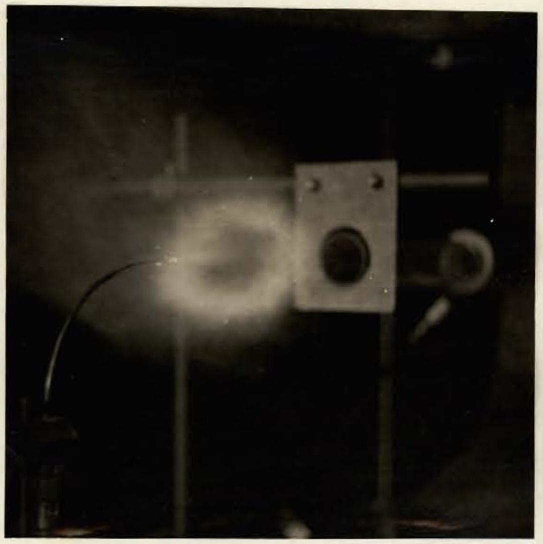


Fig. 13. Flame stabilized outside large tube

Fig. 14. Flame stabilized inside large tube

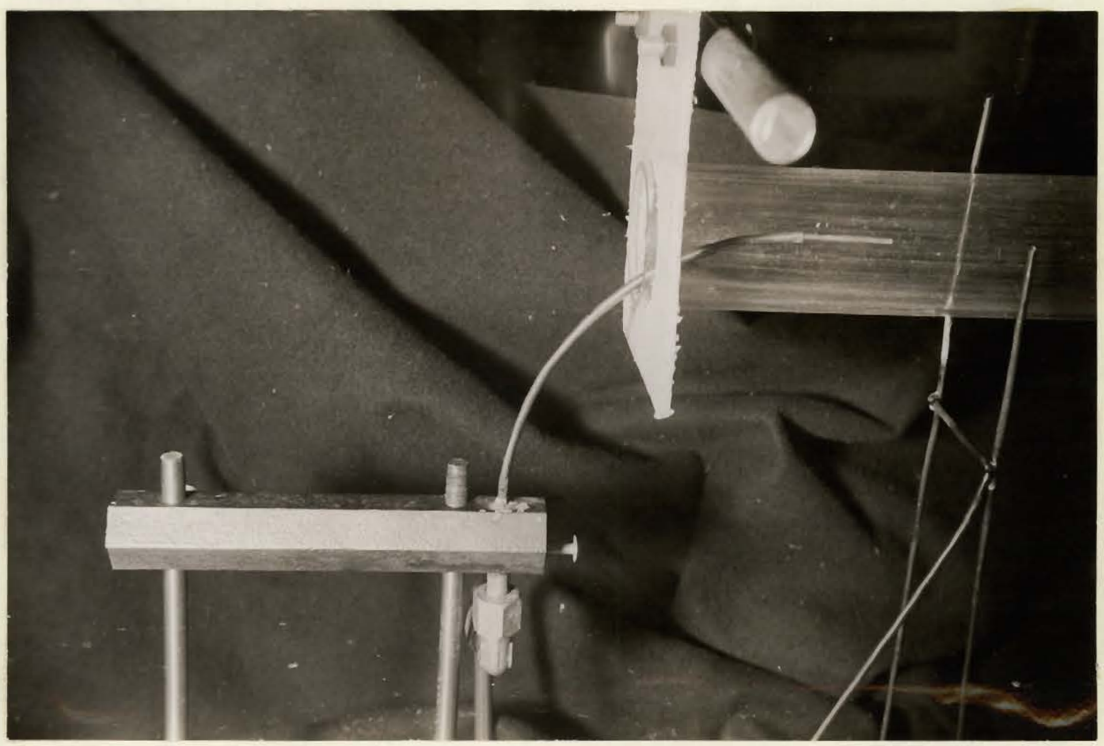


Fig. 15. Apparatus for upstream jet measurements

lected in the joints of the connections, blocking the air flow and clogging the jet tube. When one was introduced into the jet air and the other into the main air supply, smoke was formed at the meeting surface, but only at extremely low flows. At higher flows the smoke did not have time to appear until the vapours had left the mixing zone. This was therefore abandoned.

A very large enclosing tube might reduce the wall effects and permit the type of stabilization shown in Fig. 10 to be obtained. A large air supply would be required.

4. STABILIZATION DUCT I: DESIGN AND CONSTRUCTION

Most of the research was taken up with the investigation of the type of recirculating combustion tube referred to earlier, in which a portion of the combustion products is led back through a separate duct to join the incoming fresh mixture. As a preliminary, it was decided to build a very simple unit to try out the use of castable refractory for construction and the principle of forcing the recirculation by means of an ejector. Stabilization Duct I was therefore designed and built.

Fig. 16 illustrates the paths of the gas-air mixture and combustion products. A mixture of commercial propane gas and air entered the combustion duct through an entry section 2.5 cm square. It was joined by hot combustion products to initiate combustion through heating and introduction of active radicals. Combustion took place in a section 2.5 by 4.0 cm, about 15 cm long.

Part of the combustion products entered the recirculating loop while the remainder was discharged into the room. The recirculated portion was joined by a jet of air whose ejector action increased the recirculatory flow and provided some control over it. These hot gases then joined the incoming fresh mixture near the entrance.

A retractile stainless steel baffle plate was provided at the entrance to stabilize the flame when starting up. A spark plug excited by an oil burner ignition transformer formed the ignitor.

Small holes (visible in Figs. 17 and 19) in the duct walls permitted a limited view of the flame and the opposite walls. Four platinum, platinum - 10% rhodium thermocouples were inserted into four of these holes so that the junctions were flush with the inner surfaces

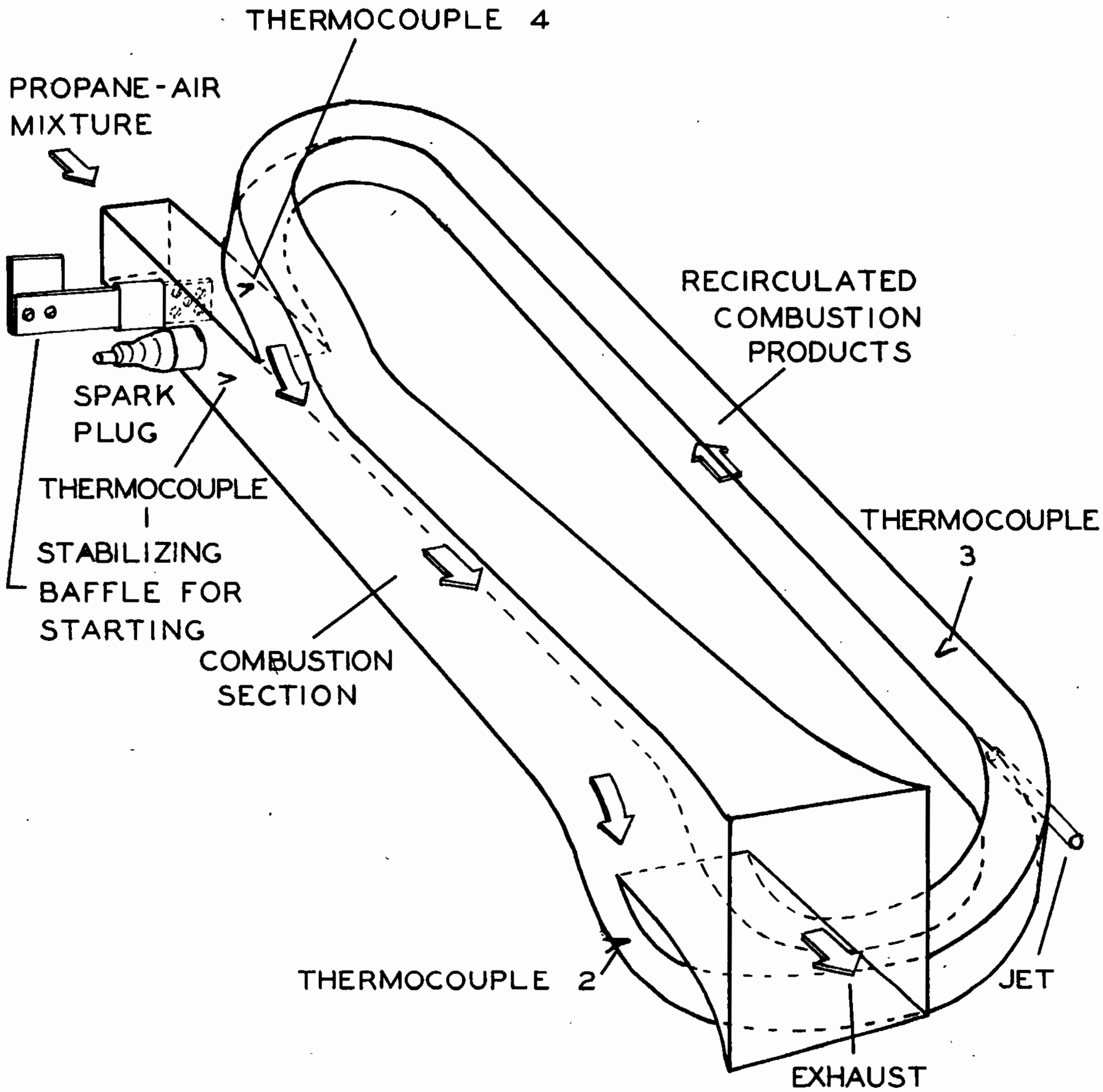


FIG.16.

FLOW PATH IN STABILIZATION DUCT I

of the walls, at the locations indicated in Fig. 16. These measured the wall temperatures, which were only a rough indication of the adjacent gas temperatures.

The recirculated gases were taken from the bottom of the duct and returned at the top to reduce the likelihood of the same gases circulating all the time. The inclination of the main and recirculating sections permitted the duct to be cast in only four parts (Figs. 17 and 18). (The smaller pieces were removable plugs to fit into the recirculating tube, so that cold air could be substituted for the combustion products if desired.)

The duct was cast of Sternson Chromepour, a chrome aggregate with self-bonding agent which sets at room temperature. It is recommended for temperatures as high as 2100K (3300F). The sections were cemented together with Sternson No. 60 Silachrome cement, recommended for temperatures up to 2150K (3400F). Molds were built up of balsa wood directly on the blueprints, and then varnished and greased before pouring the Chromepour. No difficulty was encountered in casting these sections, presumably because of their large size and simple design.

The duct was covered with Bird-Archer Bacite insulating cement, recommended for temperatures as high as 1370K (2000F). At higher temperatures it hardens and blackens. Some of the Bacite next to the combustion tube walls fused, forming a glassy residue; it had probably reached temperatures around 1600K.

The stabilization duct was connected by a metal tube and a length of heavy rubber tubing to a glass tube used for demonstrating various types of stabilization. Originally the air was metered by the pressure drop across an 11.10 mm diameter thin-plate orifice. A 2.48 mm orifice was used for the propane gas and a 4.98 mm orifice for the air supply to the

Fig. 17. Stab. Duct I: main section castings

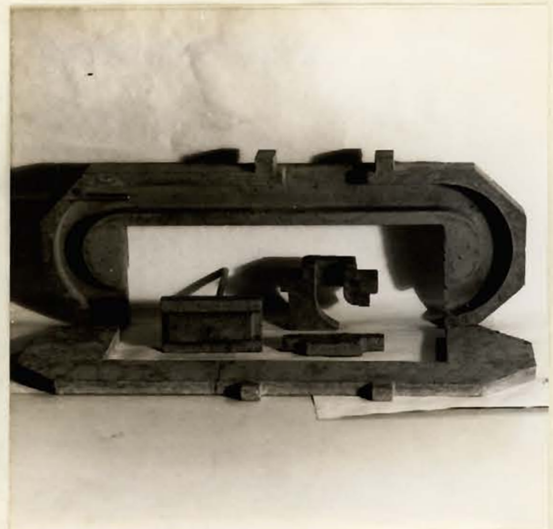


Fig. 18. Stab. Duct I: recirculating section castings

Fig. 19. Stab. Duct I, assembled

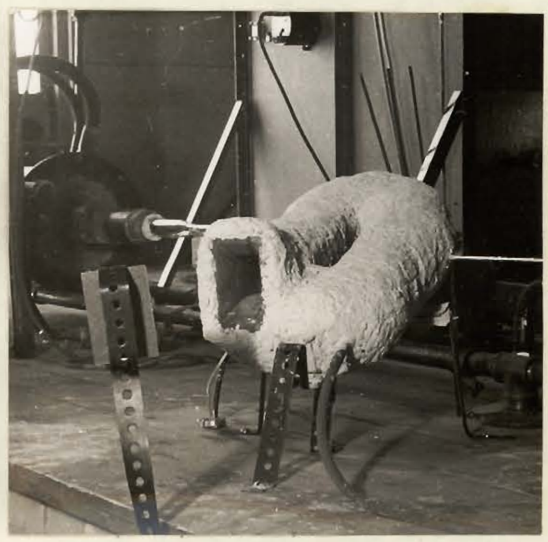


Fig. 20. Stab. Duct I, installed

ejector jet. These were later enlarged to 19.74 mm, 4.36 mm, and 8.80 mm respectively. The main air and propane were mixed in a mixing chamber visible in Fig. 20. Pressure differentials were read on water manometers, and upstream pressures on mercury manometers.

The thermocouples were connected to a millivoltmeter pyrometer. Compensating leads and selector switch were arranged so that the cold junction was at the terminals of the indicator.

It was found necessary to cool the metal entry tube with air jets because, while the incoming gas-air mixture kept it cool when running, upon shutting down heat was conducted back from the duct to the rubber tubing. (The air was turned off immediately combustion was ended because too rapid cooling might lead to cracking of the refractory.)

A mirror was located in line with the main air path through the duct about 80 cm from the exhaust end and was oriented so that the operator could see straight up the combustion tube from the control position. This mirror was cooled on both surfaces by air jets. (Fig. 20 shows an earlier arrangement in which the mirror was closer to the exhaust and uncooled. The backing and silvering disappeared very quickly.)

5. STABILIZATION DUCT I: RESULTS

AIR FLOW TESTS

To get some idea of the air flow characteristics of the duct, total and static pressure measurements were taken for various main and jet air flows at room temperature. Because of the low velocities (less than 30 metres per second), the velocity distributions were poor, and mass flows calculated from them did not agree well with each other and with metered flows. However it was considered that detailed velocity traverses would not be justified because flow conditions would be much different when the duct was running with combustion.

As had been expected it was found that the main air flow induced a considerable recirculation even when there was no jet flow. The jet influenced the recirculation more than did the main air flow, and a given increase of main air flow increased the recirculation less at high recirculatory flows than at low ones.

For example, with no jet flow the recirculated air flow was 3.0 gm per sec compared with a combustion section flow of 13.5 gm per sec. With a jet flow of 1.3 gm per sec the recirculated flow increased to 6.6 gm per sec. With no main flow the same jet strength induced a recirculation of 5.5 gm per sec.

TEMPERATURE TESTS

A number of tests were made to find out how well a duct of this construction would withstand operation at high temperatures. It was found that the stabilizing baffle was unnecessary for starting - the flame would stabilize from the spark-plug. After a few seconds, while the wall next to the flame became red, the flame would stabilize without the spark-plug.

The main section of the duct gradually became red hot and eventually orange.

However, the recirculating duct lagged far behind. In the initial tests it only reached a temperature of 940K after 30 minutes running. This was one reason for enlarging the metering orifices and increasing the flows. After this had been done, a temperature of 1500K was reached. The temperatures of the four thermocouples during a typical run are plotted against time in Fig. 21. The temperature at Thermocouple 1 dropped when the main flow was increased from 10.7 to 15.9 gm per sec, presumably because the flame moved farther downstream, so that the thermocouple was cooled by the incoming mixture. When the jet was reduced from 5.0 to 2.0 gm per sec, T_3 rose, because less cold air was being introduced. Likewise, T_1 rose slightly.

If, from cold air flow measurements, we assume that a jet flow of 5 gm per sec. gives a recirculating total flow (including jet air) of 20 gm per sec, a rough heat balance can be made to find the gas temperature T before the introduction of the jet air:

$$20\ T_3 = 5T_0 + 15T \qquad (5.1)$$

where T_0 is the jet temperature ($\doteqdot$ 300K). Thus when T_3 is 1300K,

$$T = \frac{20 \times 1300 - 5 \times 300}{15} = 1630\ K, \qquad (5.2)$$

which agrees well with T_2 (1610K). A more efficient ejector, with smaller primary flow, would reduce the cooling.

The fuel/air ratios used in all these runs were within 10% of stoichiometric. The maximum flow of 15 gm per sec. air plus 0.9 gm per sec gas corresponds to a mean entry velocity of about 20 metres per second (65 ft. per sec). After combustion this would rise to

$$\frac{2000\ K}{300\ K} \times 20\ \text{m per sec} = 130\ \text{m per sec (430 ft per sec)}. \qquad (5.3)$$

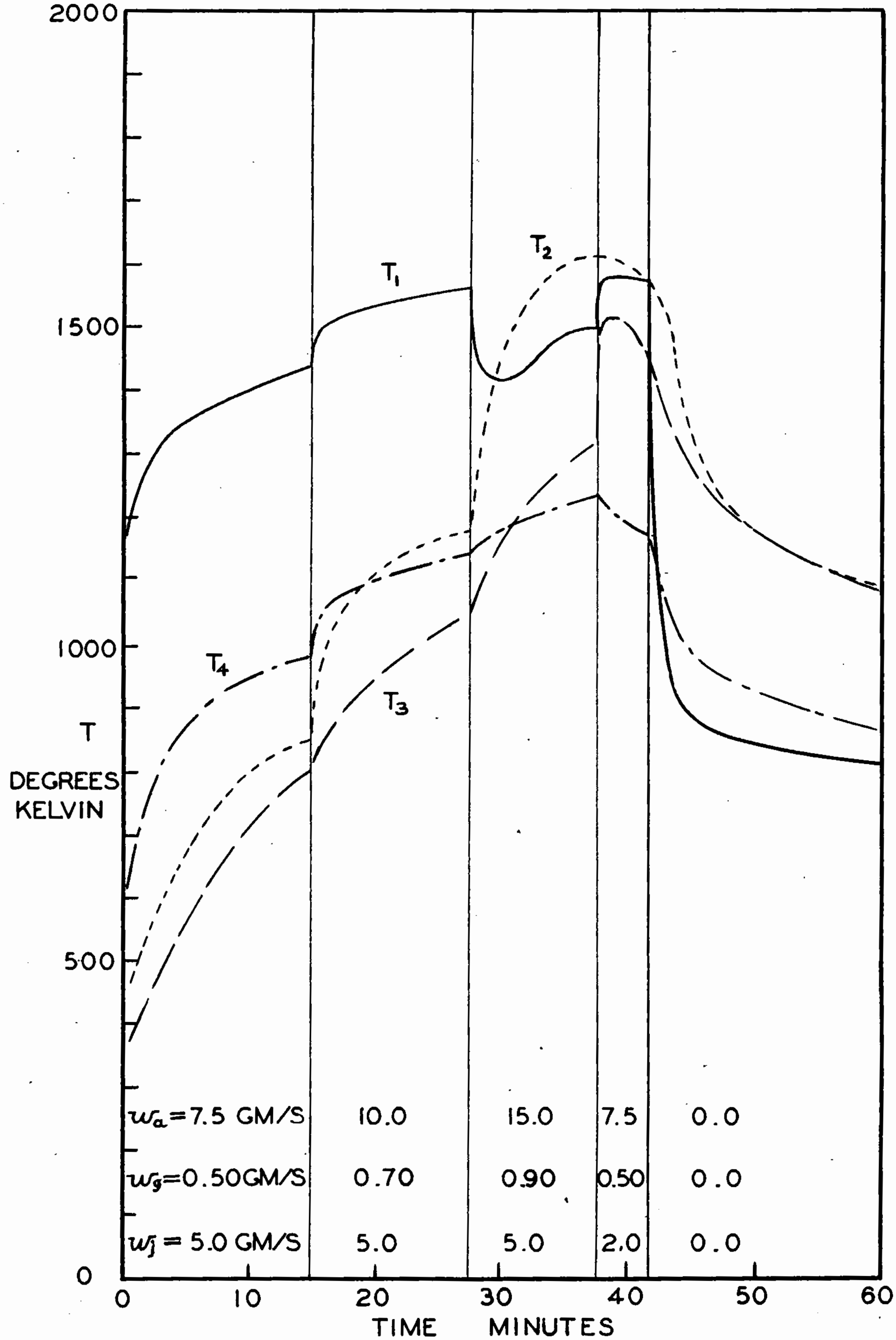


FIG.21. STAB. DUCT I : TEMPERATURE VS TIME

The maximum gas flow of 0.9 gm. per sec. corresponds to a heat release of 45,000 watts, or about 60 horsepower, assuming complete combustion.

After stabilization tests the duct was taken apart for examination. The Chromepour castings had cracked into about a dozen separate pieces which had been held together by the hardened Bacite insulation. In the hottest parts of the duct (particularly the lips at entry and exit of the recirculating loop) these pieces in turn were laced with small irregular cracks which however did not divide them into separate pieces.

The inner 2 to 4 mm of the Chromepour was blackened by vitrification, and in the case of the lips where heat came from both sides this continued right through. There was also some distortion.

The resistance to high temperatures was therefore not really satisfactory, but it was felt that it would have to do. It was thought that the cracking could be reduced by using thinner sections, thus reducing the temperature difference between inside and outside of the walls. Also the reduction in heat capacity would lead to more rapid heating up, and less inertia to temperature changes.

STABILITY TESTS

It had been hoped to find conditions in which the flame would stabilize with recirculation but not without. Actually it was stable under almost all conditions. This may be explained as follows. With no recirculation the lip separating the main entrance from the recirculating duct exit acted as a bluff obstacle, producing turbulence (see Fig. 22a) which meant there was small scale recirculation. Similarly turbulence resulted when the recirculation was high and the main air flow was low. (Fig. 22c). When the velocities matched (Fig. 22b), there was a minimum of disturbance and a minimum of stability.

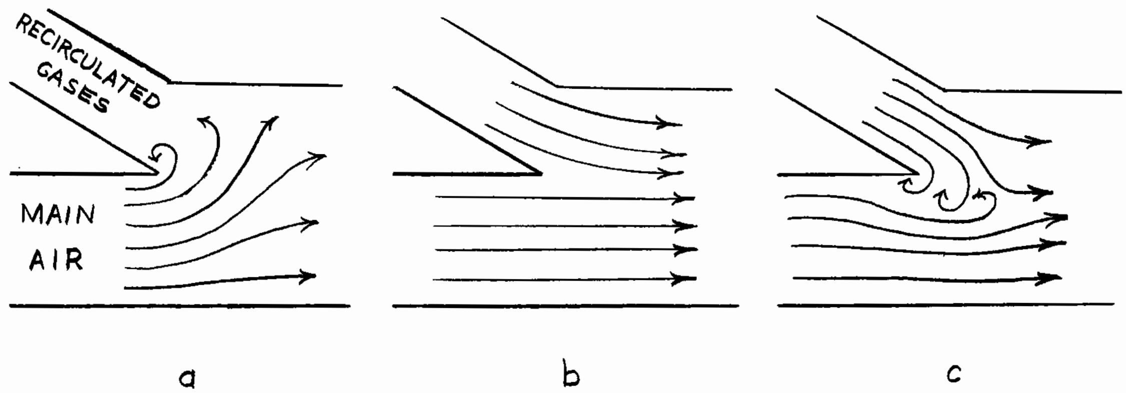


Fig. 22. Turbulence caused by velocity mis-matching

Thus as the recirculation was increased from zero the stability (measured by the range of combustible fuel/air ratios for a given velocity) would decrease, reach a minimum and increase again. This effect would be superposed on the stabilizing effects of the walls and the recirculated gases.

The recirculation would only be effective after the walls of the recirculating duct had become hot enough to allow the combustion products to pass through without being cooled. Thus it was decided to determine stability limits for the duct while cold (no effective recirculation) and again when hot (with recirculation), to see if there were appreciable differences. The hot wall effects would increase also, but would not depend in the same way on the recirculatory flow.

Fig. 23 shows the results of the cold measurements, obtained by running the duct just long enough to stabilize the flame at approximately stoichiometric ratio, and then altering the mixture until extinction occurred, plotted in the usual way: velocity (at

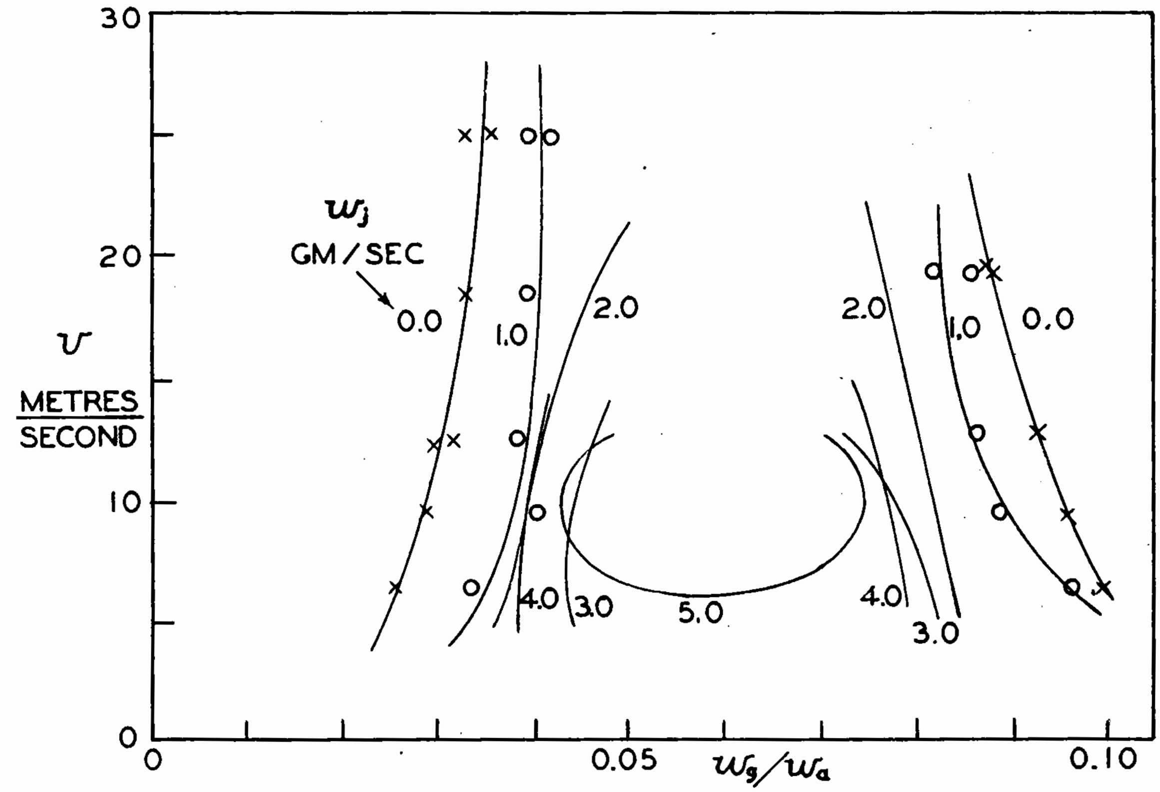


FIG. 23. STABILIZATION DUCT I: STABILITY DIAGRAM

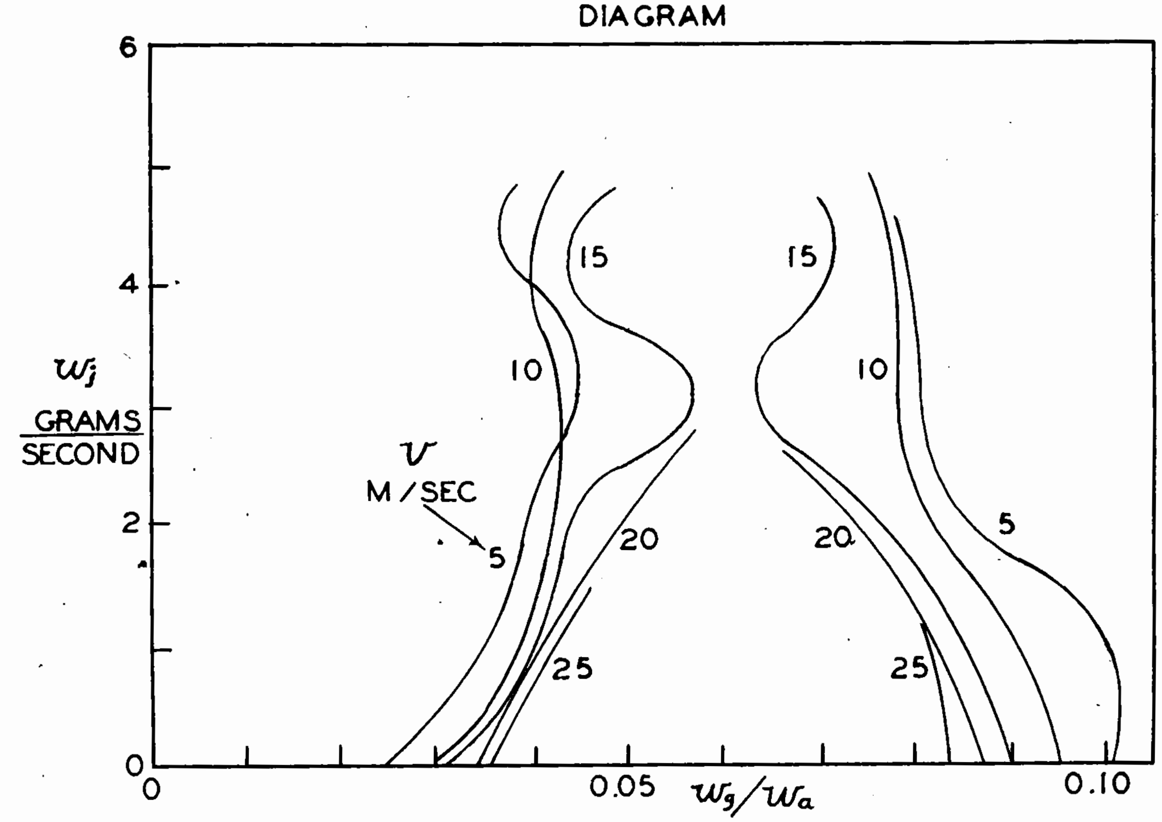


FIG. 24. CROSS-PLOT OF PREVIOUS FIGURE

entry, from mass flow and area) versus fuel/air ratio (on a mass flow basis). Points are shown for jet flows of 0 and 1.0 gm per sec to indicate degree of scatter.

Fig. 24 was obtained by cross-plotting Fig. 23, giving jet flow versus fuel/air ratio. Here the minimum of range of stability referred to above may be seen. As the jet was strengthened further the stability decreased again because of the diluting effect of the air from the jet, weakening the mixture. This effect was responsible for the requirement of a minimum entry velocity for stabilization at high jet flows, as seen in Fig. 23.

Two attempts were made to obtain extinction limits with the duct hot; however the stabilizing effect of the hot refractory caused the flame to persist, though becoming thinner, at weak and rich mixtures, making it impossible to assign extinction limits.

By this time the duct was badly cracked and leaking, and throwing out glowing particles of cement, and it was decided that no further results could be obtained with it. The designing of a new rig was begun.

6. STABILIZATION DUCT II: DESIGN

Stabilization Duct II was designed to avoid as far as possible the difficulties encountered with its predecessor. Thin sections were used where possible to reduce thermal stresses and inertia, a more effective design of ejector was incorporated, and provision was made for measuring the recirculated flow and for matching the recirculated and main flow speeds at their meeting point.

The latter was to be accomplished by projecting parallel light through the mixing region onto a frosted screen, so that the recirculated flow or the entry velocity could be adjusted to give a minimum of turbulence, as judged by the shadow pattern on the screen. Since this would impose limitations on the ratio of recirculated to main flow, it was proposed to use interchangeable sections with varied exit widths as part of the recirculating duct.

A flame would be considered stabilized when combustion began just after the incoming mixture had been joined by the recirculated gases. A flame stabilized anywhere else would be regarded as extinguished as far as recirculation was concerned.

Fig. 25 is a flow diagram of Stabilization Duct II. Metered and pre-mixed propane and air were supplied to the main entrance which was rectangular in cross section: 4.0 by 3.0 cm. There was a combustion section, 4.0 by 3.2 cm, about 12 cm long, followed by an exhaust section.

The entrance to the recirculating loop was perpendicular to the main tube wall at the end of the combustion section. There followed a constriction in which a pair of small high-pressure jets

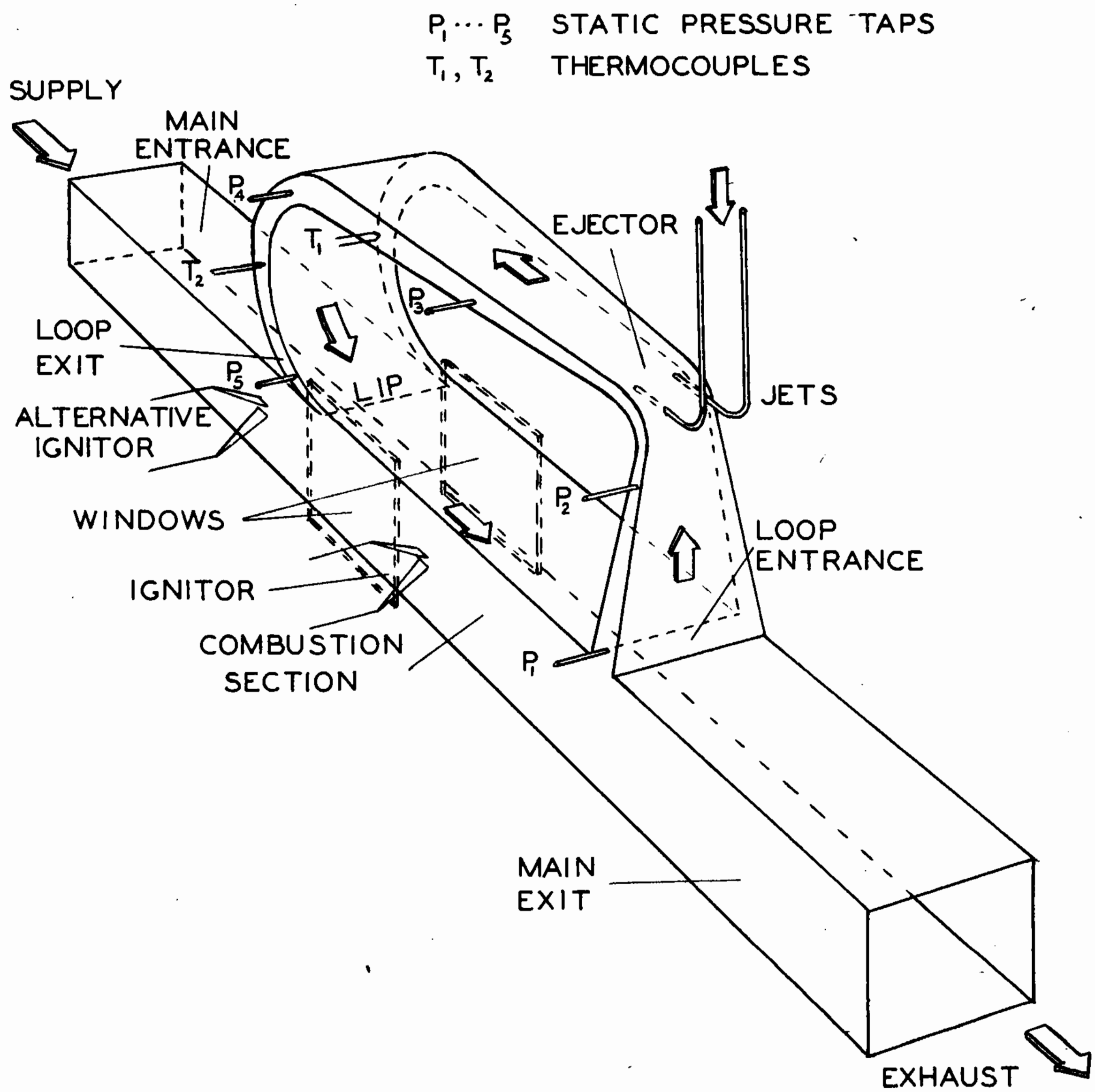


FIG. 25.

STABILIZATION DUCT II : FLOW DIAGRAM

provided the primary flow of an ejector to induce and control the recirculation. From here the gases passed through a diffusing section leading to a final contraction, to improve the velocity profile, during which they were turned through 180 degrees to join the fresh mixture nearly tangentially at the beginning of the combustion section.

The recirculated gases left and rejoined the main stream on the same side, instead of opposite sides as in Stabilization Duct I, because the cross-over would have greatly complicated the design. It was felt that if there were a tendency for the same gases to be recirculated repeatedly this might not be too serious because it could also happen in the stabilizing eddies in an actual combustion chamber.

Five pressure taps ($P_{1,2,3,4,5}$) were provided as shown in Fig. 25, giving three contracting or expanding sections in which flow measurements could be made after calibration. Thermocouples were provided in the wall at two points to obtain lower estimates of the gas temperature. In the centre of the combustion section was an ignitor for starting. In case this became inoperative an alternative one was located farther upstream.

A pair of Vycor glass windows was provided at the beginning of the combustion section to permit a light path through the region where the hot recirculated gases were mixing with the fresh mixture. Light from a point source was made parallel by a projector lens, shone through the above windows perpendicularly to the flow, and fell on a viewing screen, so that the recirculating flow could be adjusted to give smooth mixing.

Fig. 26 shows the construction of Stabilization Duct II in more detail. The parts were cast of Chromepour in three main units. The largest casting formed the main flow duct and the outer enclosure of the recirculating loop section. The recirculating loop was made in two parts, which will be referred to as the "ejector" and the "U". These fitted into the large casting, and were removable for alterations; in particular it was intended to employ U's of differing outlet widths to accommodate varying recirculatory flows, since these had to be matched to the main stream velocities. The walls of these two castings were made as thin as was feasible to reduce the temperature difference between inside and outside, and thus the danger of cracking, and to reduce thermal inertia. It was found that they could be cast satisfactorily about 4 mm thick. (See section on casting.)

The loop was surrounded by Fiberfrax insulation, a blown aluminum-silicate fibre made by The Carborundum Company. Cooling air was passed through the centre of the loop to reduce the temperature of the refractory to a safe level.

It was important to avoid, as far as possible, heat transfer between the gases in the U and the fresh mixture passing along the wall near the lip, since this could lead to premature stabilization. Therefore a cooling passage was provided as shown in Fig. 26 .

Two cover plates were cast of Chromepour to close the large casting after the smaller sections were inserted.

The ejector jets were 16 gauge (1.65 mm o.d.) stainless steel hypodermic needle tubing. It was hoped that they would be cooled

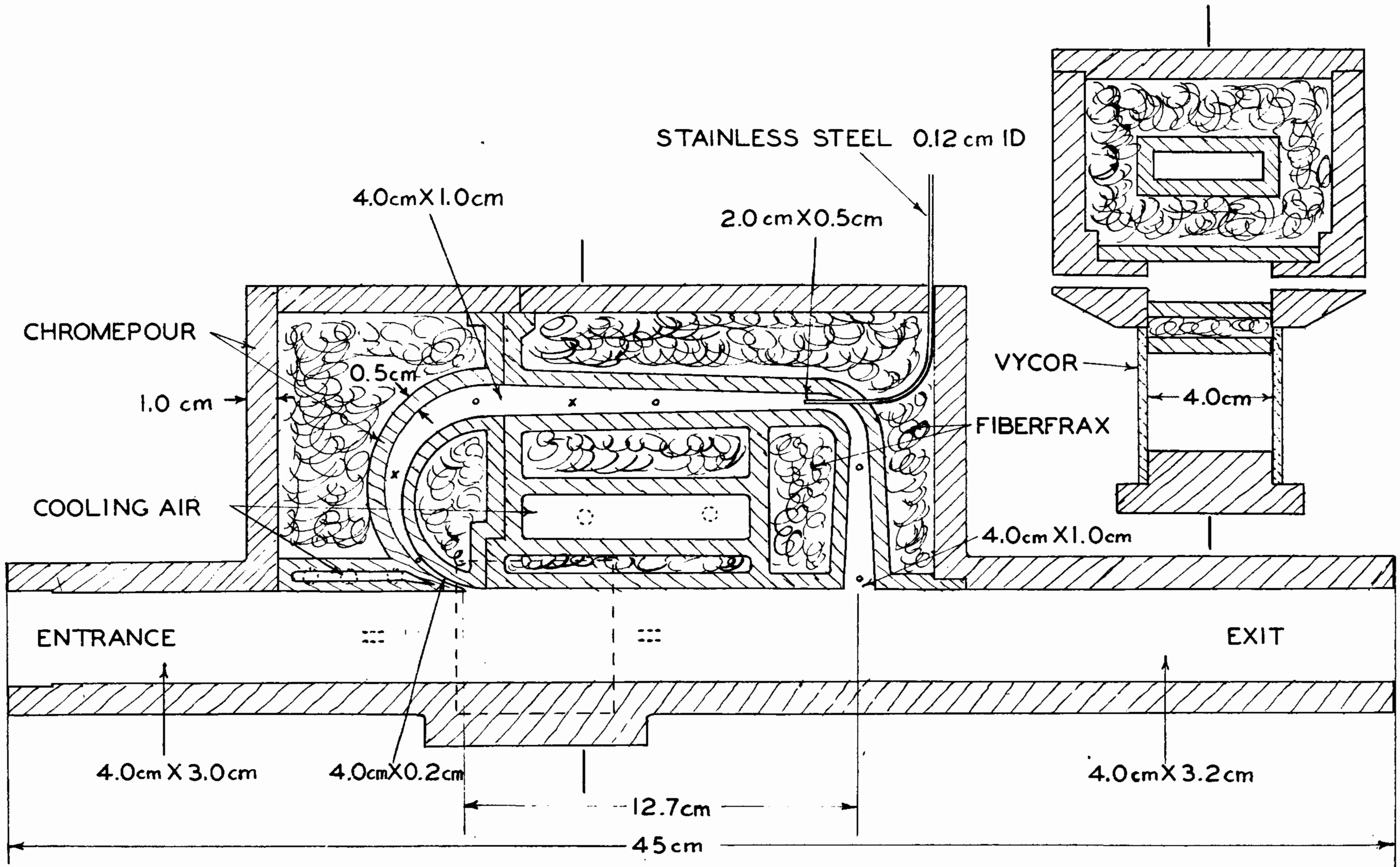


FIG. 26. STABILIZATION DUCT II : SECTIONS. ONE-HALF FULL SIZE

adequately by the jet air passing through them. They were squeezed somewhat oval at their outlet ends to reduce the area, giving a crude nozzle, and to more nearly resemble the narrow rectangular cross section of the ejector casting.

The pressure taps were made by cementing hypodermic tubing into perforated lugs cast on the ejector and U, or into holes cored in the large casting (see Figs. 27, 28, 33). The thermocouple junctions were cemented into holes in similar lugs.

The ignitor consisted of a pair of parallel platinum wires flush with the inner surface of the duct. A 10,000 volt 60 cycle voltage was applied.

Fig. 27. Stab. Duct II:
U casting

Fig. 28. Stab. Duct II:
ejector casting

Fig. 29. Stab. Duct II:
main casting

Fig. 30. Stab. Duct II:
main casting

7. STABILIZATION DUCT II: CASTING AND CALIBRATION

A material suitable for constructing a combustion duct of the design required would need the following properties:

Ability to withstand temperatures as near as possible to the flame temperature (about 2200K, 3500F).

Resistance to thermal shock.

Ease of casting (or shaping in some other manner) to complex shapes with thin walls.

Neutrality to combustion gases.

Reasonable cost. Rare metals like columbium could hardly be considered for the complete duct, although small amounts might be used for critical parts, for example the lip at the recirculating loop exit. They are also difficult to fabricate.

At least enough strength to retain its shape when hot.

Low shrinkage to avoid cracking.

Sternson Chromepour was the first refractory tried, and no better has been found. Attempts were made to cast sections in sillimanite and alumina, but these shrank and cracked upon drying, and lacked strength.

Chromepour is mixed with water to the desired consistency and then poured. It sets within 24 hours in air at room temperature and is quite strong. Surfaces are smooth, though nothing like metal or glass.

Stabilization Duct I was poured in fairly simple molds and no difficulty was encountered. However, the more complex shapes of

Duct II led to many difficulties.

The cement cannot be made too thin, or it will shrink, and the bond may be weakened. Therefore it must be rammed or prodded into the corners of the mold, and even then it is very hard to avoid entrapping air bubbles. About an hour was required to pour each of the castings for Duct II. The cement is very quick-setting, and after 5 to 10 minutes it thickens noticeably, becoming difficult to pour. Water was usually added but this could only be done once or twice, then a new batch of cement had to be mixed.

Molds for Stabilization Duct II were made of balsa wood, shellacked and greased. The first large casting attempted had cracks and large air spaces. After a new mold was designed and build a second attempt was made, and this produced an acceptable casting. There were some small gaps which were filled with a mixture of Chromepour and Sternson Silachrome cement. The Chromepour alone would not stick, while Silachrome sticks but shrinks: a mixture was found to work fairly well.

The recirculating loop U was attempted next. It was found that the thin sections desired here (3 to 5 mm) were much more troublesome. They cracked, and worst of all, the cement crumbled and had no strength. Mixtures of Chromepour and Silachrome were tried with no success, the shrinkage leading to cracking. The manufacturers of the cement were not able to offer much advice - they did not recommend casting sections thinner than one-half inch.

Many small samples were cast in efforts to find out why thin castings were harder to make than thick ones. Eventually it was found that firm castings could be obtained if they were immediately covered up or put in a container when poured, or kept sprinkled with water for several hours after pouring.

Apparently thin sections dry too quickly for a good set. Setting may take time, or perhaps some of the bond is carried off when the water evaporates too quickly.

Thin sections (less than 5 mm) could not be poured from the edge because the small stones in the mixture clogged the mold and large air spaces resulted. If the stones were sifted out, the remaining cement shrank. Therefore the molds had to be designed so that the cement could be applied by plastering, with all parts accessible to a knife or similar implement.

It was originally intended to use wax cores for the recirculating loop and melt them out. It was found that (a) the wax went into the Chromepour when heated, probably weakening it, and (b) the wax expanded, cracking the castings anyway. Therefore cores were made of very thin balsa wood (1/64 inch), shellacked and greased, and were later burnt out. The thin wood ensured that if there were differential expansion effects on heating, the core would not be strong enough to crack the cement.

The fourth attempt at casting the U was successful, the few small holes being patched with Silachrome, which can be filed a few hours after application, before it has completely hardened.

The second casting for the ejector was acceptable, and the very simple cover plates were cast in one attempt.

All the castings were heated to 1000C in the large electric furnace of the Metallurgical Department, by Professor J.P. Ogilvie.

On being heated above about 500C, Chromepour castings became reddish (having been gray before) and ring slightly when tapped. There is also appreciable shrinking and warping. The ejector and U which had fitted easily into the main casting when poured now required trimming. This was accomplished by filing where possible, and cautious grinding where particles of stone were at the surface.

The main casting had many fine hair-cracks - except on the surfaces which had been exposed when in the mold. It may be that all moisture in the casting had to leave by these surfaces, so they took longer to dry and achieved a better set. There were also a few large cracks, particularly near the window openings.

The main casting is shown in Figs. 29 and 30. Warpage is seen as drooping of the ends and concavity of the flat surfaces. The dark patches in Fig. 29 are Silachrome used to patch the larger cracks.

The ejector and U had practically no hair-cracks and were in good condition, except for the fact that the flat surfaces to complete the walls of the combustion section were bowed outward. They are seen in Figs. 27 and 28.

It was evident that further warping when the duct was run would make removal of the loop sections impossible, so it was decided to cement them into place, reducing leakage and making a stronger unit, but preventing substitution of alternative U sections. Fig. 33 shows the duct with Bacite applied and ejector and U cemented in place. The pressure taps, thermocouples, and jet connection are visible, as well as some of the Fiberfrax insulation.

Figs. 31 and 32 show the complete installation. In Fig. 32

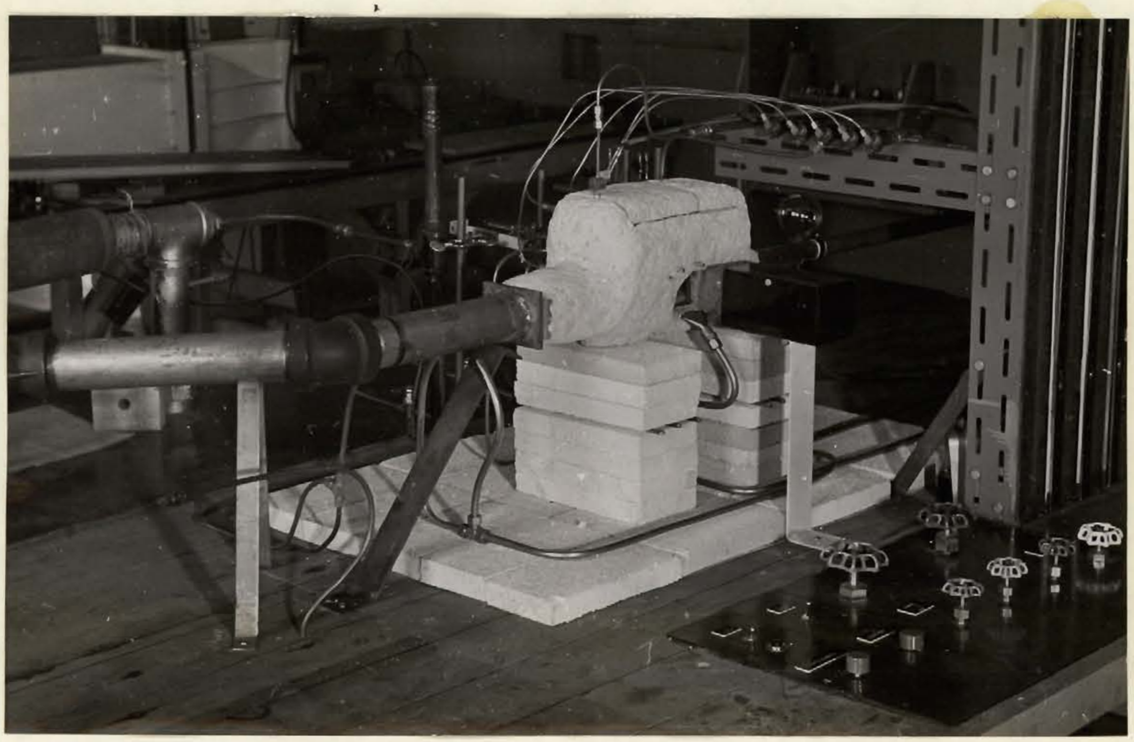


Fig. 31. Duct II: general view

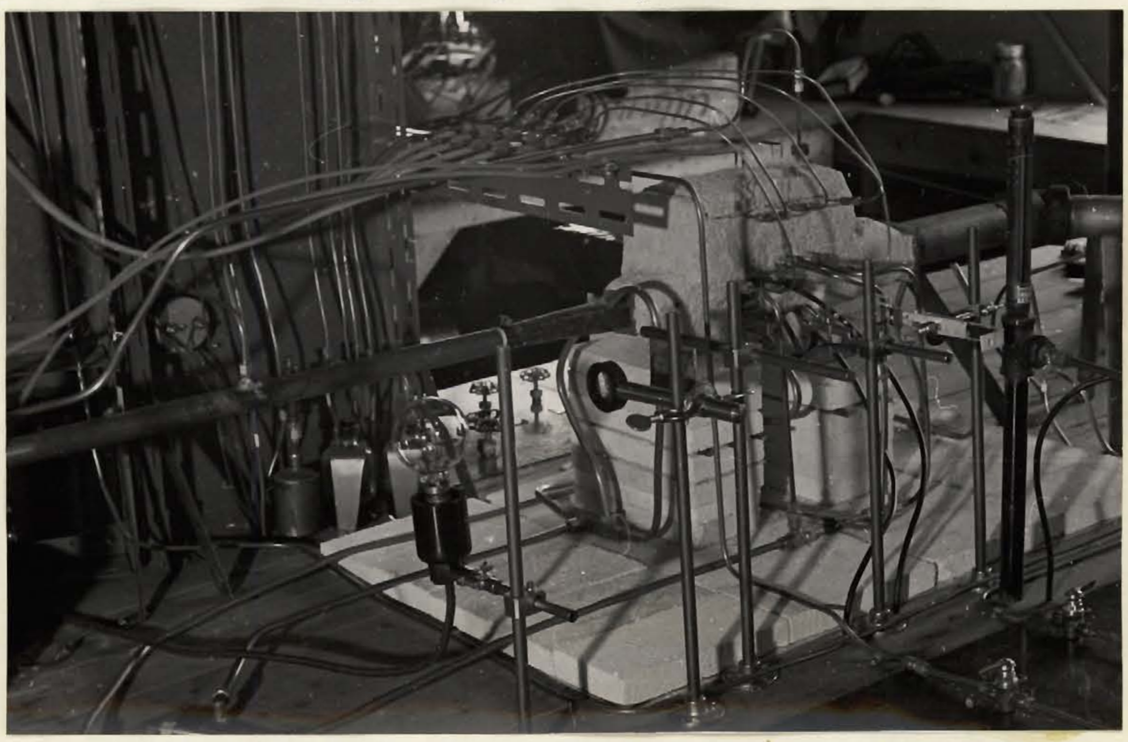


Fig. 32. Stab. Duct II: General view, showing optical system

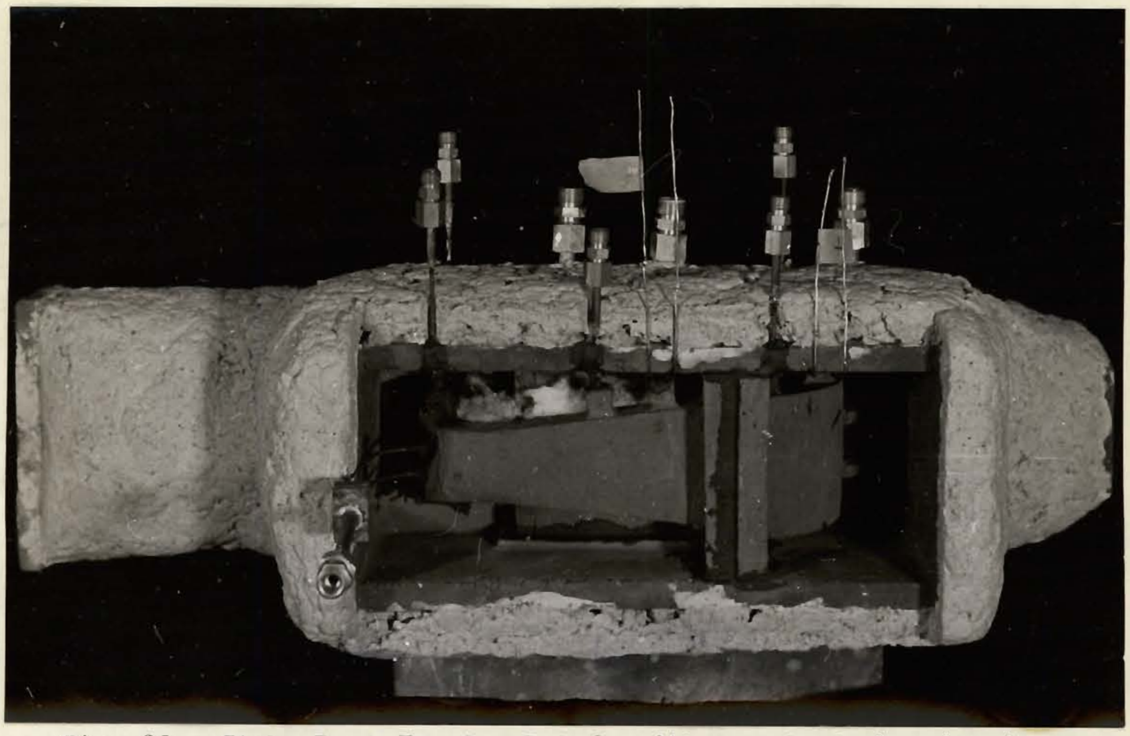


Fig. 33. Stab. Duct II, insulated, with U and ejector in place.

Fig. 34. Hole in exhaust section; Bacite removed.

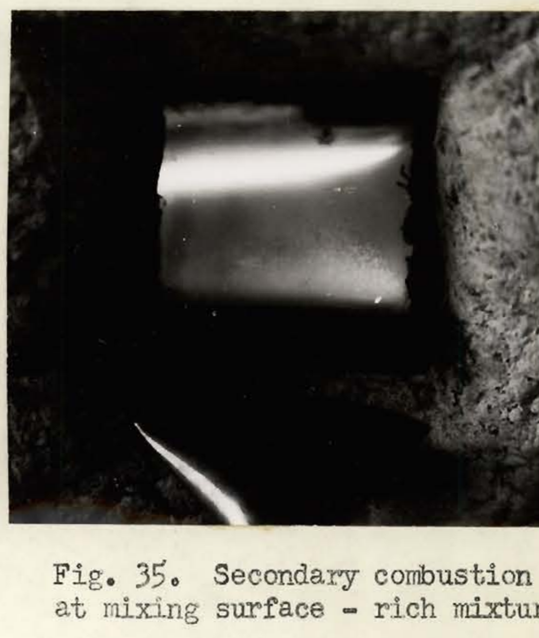


Fig. 35. Secondary combustion at mixing surface - rich mixture

the optical system may be discerned. It consisted of a 150 c.p. Royal "Ediswan" Pointolite lamp, a lens, a pinhole forming a secondary source of small size, a projector lens, and a mirror producing a 90 degree change of direction. A hood surrounding the viewing screen is in front of the window in Fig. 31.

Before cementing the recirculating loop sections into place they were calibrated. A balsa jig was made to hold the ejector and U together in proper relation, and the ejector entrance was connected to a supply of metered air. Measured air flows were then passed through the loop (together with various known jet flows) and the static pressures $P_{1,2,3,4,5}$ were observed. From this it was possible to draw curves of air flow versus the product of pressure difference and density for the pairs of taps P_1 and P_2, P_3 and P_4, P_4 and P_5. The velocity profiles in the loop were much dependent on the jet strength, so a separate curve had to be drawn for each jet flow.

The source of air was then disconnected and the ejector was operated drawing from atmosphere. A baffle prevented air emerging from the U exit lip from affecting the ejector entrance. Flows were calculated from the calibration curves. In Fig. 36 the induced flow w_r and its ratio w_r/w_j to the jet flow are plotted against the jet flow. Because of the poorly developed velocity profiles the flow measurements were lacking in precision and the scatter was large. In particular the measurements at taps P_3, P_4 (close after the jet) were so inconsistent that they were discarded.

Similar curves are obtained from plots of the calibration data for the cases in which P_1, the pressure at ejector inlet, is

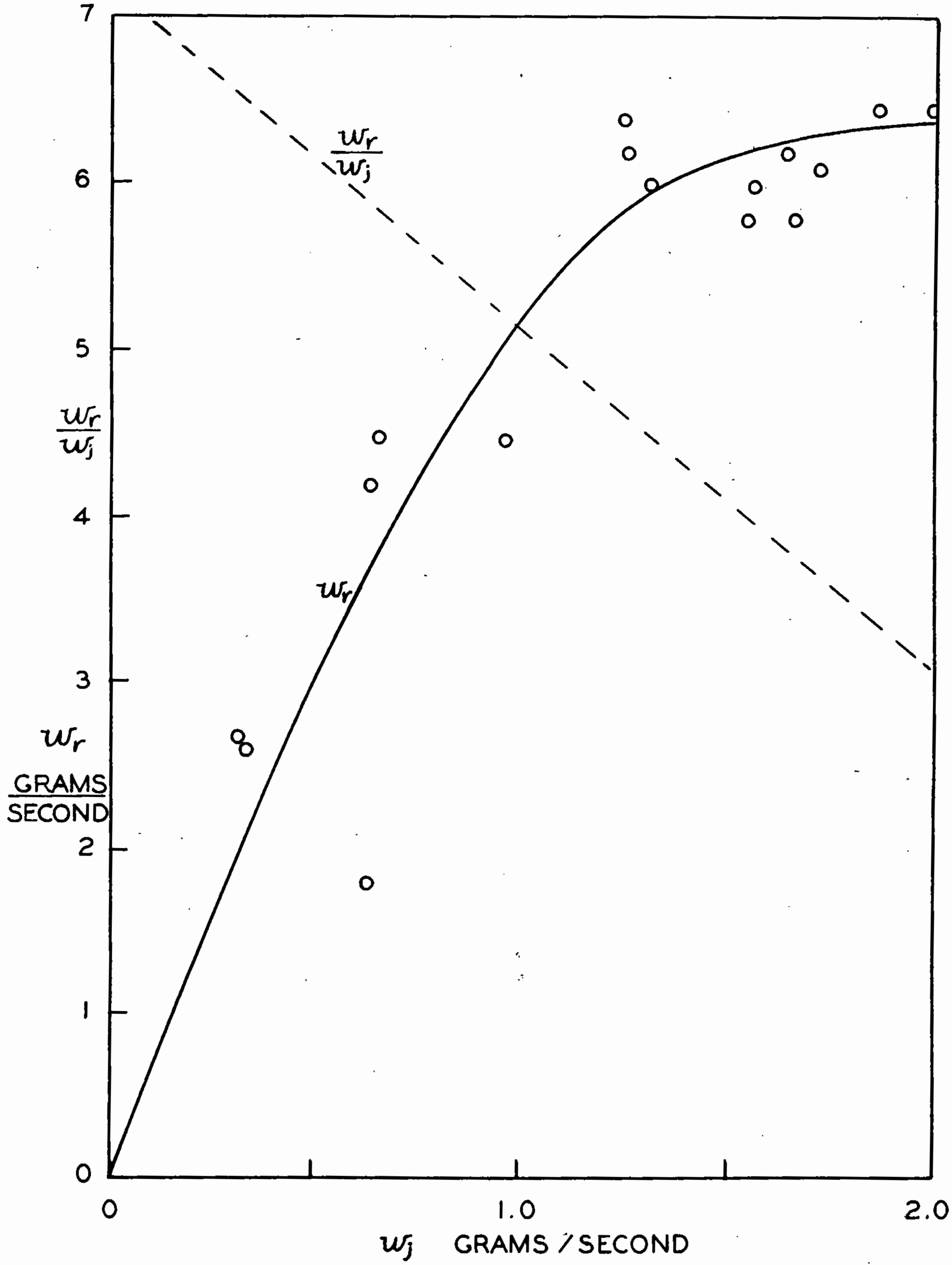


FIG.36. STABILIZATION DUCT II EJECTOR: RECIRCULATED FLOW AND SECONDARY TO PRIMARY RATIO VERSUS JET FLOW

approximately atmospheric.

As the jet flow increases the induced flow increases to a diminishing degree and the ratio of induced to primary flow drops. The latter is undesirable because the smaller this ratio, the greater the cooling and diluting effects of the jet. If the operating point is taken as w_j = 1.0 gm per sec, w_r/w_j is about 5. Taking the temperatures of jet and induced flow to be 300 and 1800K respectively, the resulting temperature would be

$$\frac{300 + 5 \times 1800}{6} = 1550\,K\ ,$$

a drop of 14 percent.

After the ejector and U had been installed in the main casting, a run was made to see if the main air flow itself would induce any recirculation in the loop. Such recirculation, if any, was too small to show on the manometers.

8. STABILIZATION DUCT II: AUXILIARIES

The schematic diagram (Fig. 37) of auxiliary apparatus associated with Stabilization Duct II is largely self-explanatory.

Propane was supplied from 500 lb. pigs at pressures up to 90 psig. Low pressure air came from a compressor supplying 0.4 lb per sec at 90 to 95 psig. Up to 0.012 lb per sec. of high pressure air was available at 130 to 150 psig. This was used for the ejector to reduce the mass flow required. A cross-over valve permitted testing lines on high pressure air.

Metering was by thin-plate orifices. In order to provide for a wide range of flows a tank was designed in which a rotating sector operated by a lever at the control panel brought two smaller orifices over the largest one, giving three ranges for each line. The orifice diameters were approximately in the ratios 5: $\sqrt{5}$: 1 to give flows in the ratios 25:5:1 for a given pressure differential. The orifice diameters, measured with a vernier microscope were as follows:

	Air	Gas	Jet
Small	8.02 mm	1.77	2.52
Medium	18.26	3.89	8.96
Large	40.29	8.96	12.75

Inevitably there was leakage past the sliding orifice plates, so the orifice tanks were calibrated against a set of seven standard orifices (diameters in ratio $\sqrt{2}$) discharging to atmospheric pressure from a large tank. The low and medium gas orifices were also calibrated against a wet-test meter of 0.25 cubic foot capacity.

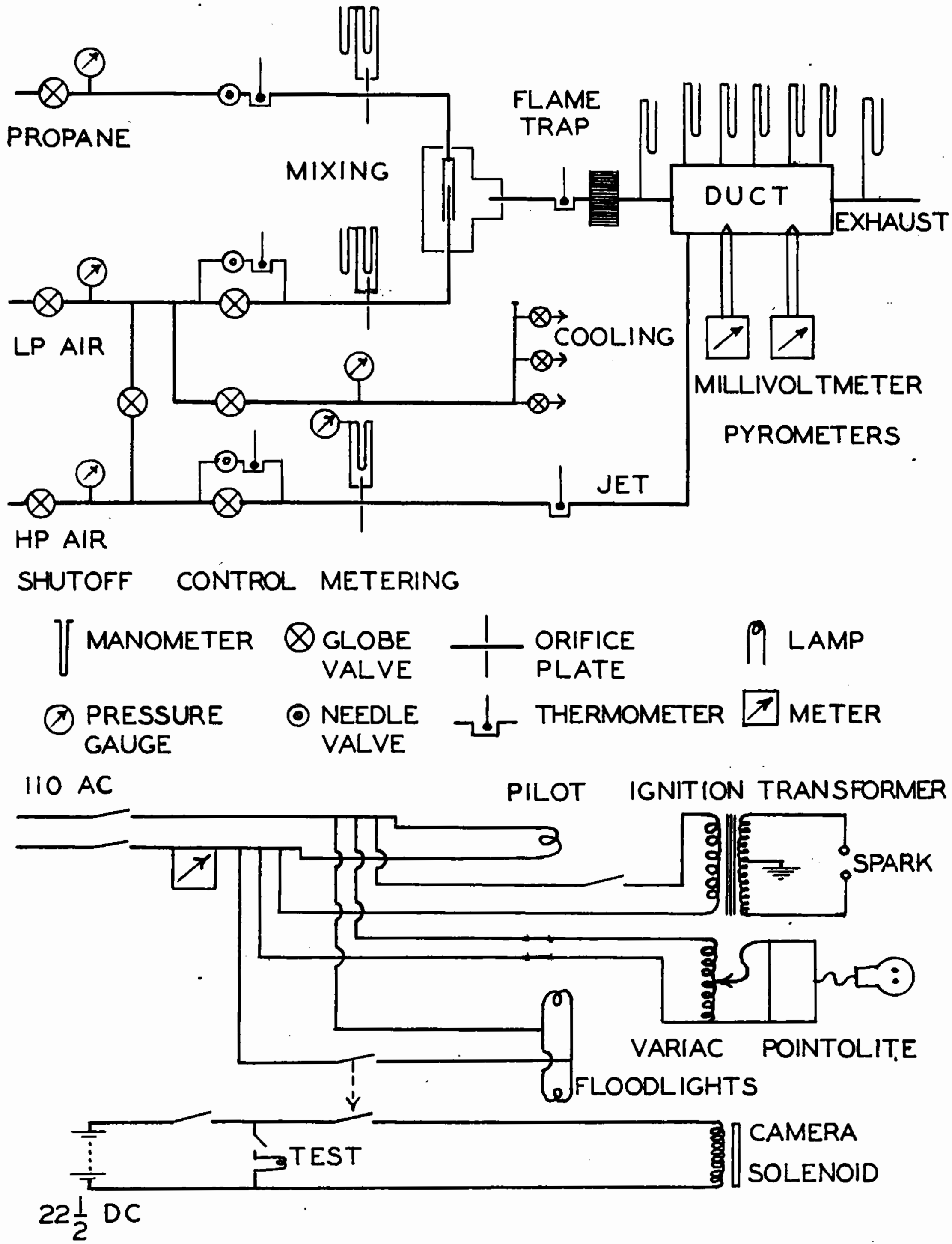


FIG. 37
AUXILIARIES

From the calibration data a curve of discharge coefficient versus differential pressure was plotted for each of the nine orifices. Flows were calculated from the formula

$$w = K\sqrt{\frac{P}{P_o}\,\frac{300}{T}\,\Delta P} \qquad (8.1)$$

where w is the mass flow, P is the absolute pressure upstream of the orifice, P_o is standard atmospheric pressure, T is the temperature of the gas or air when metered, and ΔP is the pressure differential. K is the calibration factor which varies with ΔP (more strictly with the radical in the above formula for w to which the Reynolds number is proportional), and since it was determined for each orifice it includes the square of the diameter, as well as a numerical factor dependent on the units and the standard densities of air and propane.

Pressure differentials were measured by well-type water manometers, whose properties were taken care of when the orifices were calibrated. The jet flow manometer was subject to high pressure, so it was made with Saran tubing and fittings. While Saran is not very transparent, the level of coloured water is clearly visible through it.

The upstream pressures for air and gas were measured with U-type mercury manometers, and a bourdon pressure gauge was used for the jet. Cooling air flow was set by means of a pressure gauge.

In the mixing chamber the 3/8 inch copper tube of the gas line projected into the end of the one inch air tube, which in turn projected into a three inch solder tee, from which one inch copper pipe and soldered fittings led through an aircraft type flame trap and straightener to the stabilization duct, the final connection being made by a stainless steel transition piece.

The pressures $P_{1,2,3,4,5}$ associated with the duct itself were read on a set of water manometers. The pressures in the inlet and exhaust pipes were read on mercury manometers. Each platinum thermocouple was connected to a millivoltmeter pyrometer.

The exhaust from the duct was led through a large cross section pipe so that the pressure in the duct would be at or below atmospheric and leakage (almost sure to exist with a refractory duct) would be small and inward.

Since conditions in the stabilization duct would change immediately upon extinction it was necessary to arrange for instantaneous recording of all data. A Robot II camera with automatic film transport and remote release was mounted so as to photograph the instrument panels. A progressive button at the control station first turned on the floodlights, then triggered the camera. To give good photographic contrast all water manometers were filled, and mercury manometers topped, with water coloured black by the addition of both red and blue vegetable colours.

The 35 mm negatives were later examined under a 9 power microscope and the readings recorded. There was a permanent record, and errors due to incorrect reading were avoided.

9. STABILIZATION DUCT II: RESULTS

The performance of Stabilization Duct II was disappointing.

Ignition was achieved smoothly enough at a low inlet flow, and the flame jumped back and stabilized in the entrance to the duct, possibly due to a rough spot on the refractory or to an imperfect fit between the duct entry and the stainless steel connector from the supply pipe. Since it would not be expected to stabilize downstream in the combustion section while the recirculating loop was cold, the flame was allowed to remain in the entry section while the loop heated up.

In Fig. 38 the temperatures T_1 and T_2 of the two thermocouples in the recirculating loop wall (Fig. 25) are plotted against time. The air and jet mass flows and the calculated recirculating flow are also shown. The latter was calculated from the pressure readings, and for reasons to be discussed later is suspect. The propane flow was such as to give a stoichiometric mixture at the entrance.

Initially, the recirculating loop heated up quite rapidly, but the temperatures soon leveled off at about 900 and 700K, in spite of the main mass flow being increased until the theoretical heat release rate was 20 kilowatts. The low temperatures suggest that the recirculation was much less than was indicated by the pressure measurements. T_1 was higher than T_2. This may have been because the jet had not yet spread as far as the wall.

It was difficult to see the characteristic blue of the flame to determine its position: the orange radiation from the refractory walls was much brighter, and even when looking directly through the

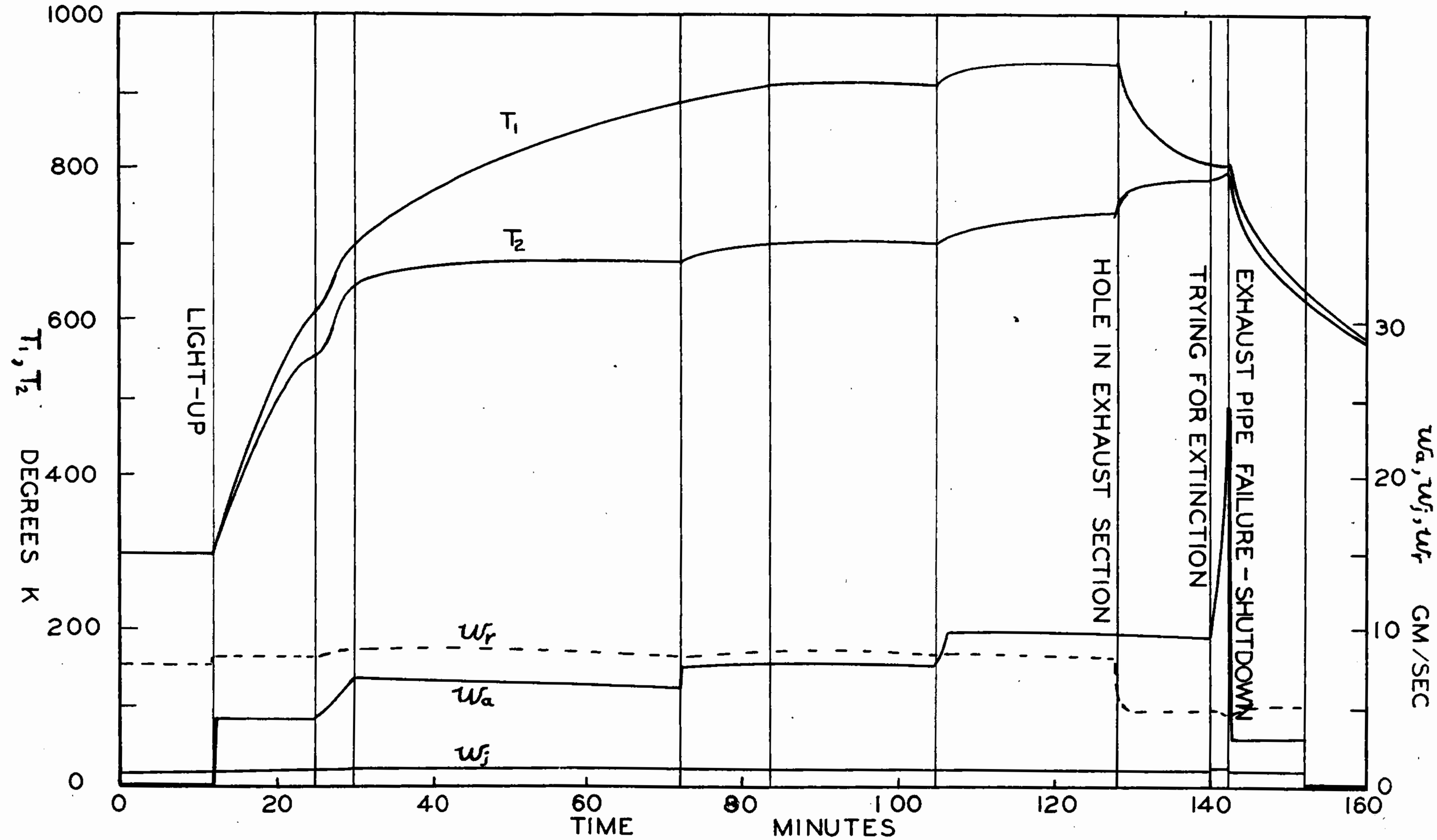


FIG.38. STABILIZATION DUCT II: TEMPERATURES AND FLOWS VERSUS TIME

windows there was strong sodium radiation (Chromepour gives off sodium when hot). This light illuminated the viewing screen making it impossible to see any shadows at all. The optical system was discarded for later runs and the viewing screen was removed, facilitating direct observation.

The calculated values for the recirculated flow w_r show it to be about equal to the main flow w_a. (The ratio of recirculated flow to total combustion section flow is $\frac{w_r}{w_r + w_a + w_g}$, in this case about one half.) The temperatures reached by the recirculating loop walls suggest a much lower recirculation - indeed, less than the jet flow. The high temperature stream flowing past the loop entrance and exit may have disturbed the velocity profiles in the loop, giving false pressure readings.

During the run, gas samples were taken from the exhaust pipe and from pressure tap P_3 in the recirculating loop and were analyzed in an Orsat apparatus, with the assistance of R.E. Chant and J.T. Rogers of the Gas Dynamics Lab. These results were on a dry volume basis. Recalculated on a mass basis and including steam, they are as follows:

	CO_2	O_2	CO	N_2	H_2O
Exhaust	13.6%	4.8	0.0	74.2	7.4
P_3	7.8	11.2	0.2	76.5	4.4

If we assume that the gases w_r entering the ejector have the same composition as the exhaust, and that the jet consists of 23.2% oxygen, 76.6% nitrogen, and 0.2% carbon dioxide, we can write an equation for the oxygen:

$$.048\, w_r + .232\, w_j = .112\, (w_r + w_j) \quad ; \qquad (9.1)$$

and similarly for the other gases, except carbon monoxide, whose quantity was too small to be measured with any accuracy. Each equation can be solved for the ratio w_r/w_j. The nitrogen equation was unsuitable because the measured percentage at P_3 was nearly that of air - it gave a ratio w_r/w_j of 0.04. The other three results were:

Oxygen:	$w_r/w_j =$	1.87
Water:		1.46
Carbon dioxide:		1.31
	mean	1.54

These also indicate that the recirculation was small.

On a subsequent run an effort was made to obtain simultaneous gas samples from pressure taps P_2, P_3, and P_5 by displacement of water in sampling bottles. Large amounts of air were obtained. To investigate this, propane alone was run through the rig. Again air was obtained (20% by volume), even when the lines had been flushed out with propane. It was evident that air was leaking into the pressure tap connections inside the duct. (All external connections were checked.) This being the case, neither the recirculated flow measurements based on the pressure readings nor those based on the gas samples could be trusted. In fact, some of the inter-wall cooling air may have been leaking into the recirculating loop, thus keeping the temperatures down. This could account for T_2 being lower than T_1.

It had been necessary to cement the cover plates in place to prevent them flying off during hard starts, so the inside of the duct could not be reached. In any case, all accessible joints had been sealed during assembly: the leaks must have been in inaccessible places.

After the duct had been running for slightly under two hours, a hole appeared in the back wall of the exhaust section. The hole in the outer surface of the Bacite insulation, which by this time was baked hard, was only about 4 by 2 cm, but the Bacite next to the Chromepour had fused all around the neighbouring region, leaving a gap between the duct itself and the outer surface of the Bacite. Fig. 34 shows the hole after the outer Bacite had been broken away. The hole in the Chromepour was about 5 by 3 cm. A piece of the wall had peeled back and downward. The edges of the hole had every appearance of fusion. Why a hole should have appeared in this particular place is not known. There may have been a region of poor composition in the refractory, although this seems unlikely after the mixing which preceded pouring.

The hole was later patched with Silachrome cement, the cavities beneath the neighbouring Bacite were packed with Fiberfrax, and the outside was covered with fresh Bacite. No more trouble was experienced in the exhaust section.

Immediately the hole appeared the pressure level in the rig dropped and the apparent recirculating flow decreased by about one-half. The temperatures T_1 and T_2 gradually approached one another, which could be accounted for by the drop in recirculation.

Since the rig would soon have to be shut down, an attempt was made to see if the flame could be stabilized downstream of the loop exit lip, i.e. by the recirculation. The flows were increased greatly, but the flame became so tenuous and fluctuating that it was impossible to say just where it was. At this point the exhaust piping began to melt and the rig had to be shut down.

After patching the exhaust section, installing a length of stainless steel pipe in the exhaust ducting, and removing the viewing screen, a second run was made, consisting of quick starts and extinctions. When ignited at a low flow the flame jumped back to the inlet section. To dislodge it the flows had to be increased greatly; then the flame blew right out the exhaust. The blue flame was easily visible before the refractory had time to get hot.

After the spark had been in operation for a few seconds the Chromepour would fuse between the platinum electrodes, giving a white glow instead of the blue spark obtained when cold. This was not enough to ignite the mixture. On cooling, the spark was once more operative.

Part of the dividing wall between the combustion section and the central cooling passage cracked and fell down, finally wedging in the exhaust section. The Fiberfrax so exposed remained in place, resisting both gravity and hot gas flowing past.

When running at rich mixtures, the extra air (from the jet, perhaps also from leaks) emerging from the exit lip of the U caused further combustion along the meeting surface, appearing as a curve when viewed edgewise - see Fig. 35. The lip is just outside the top right-hand corner of the window. This mixing surface produced a permanent marking on the Vycor windows, much more intense than the general devitrification. As the main flow was reduced this surface inclined downward, as would be expected. At all times it was uniform across the duct, showing that the flow in the loop exit was well distributed.

Weak and rich extinctions were taken, on this and later runs, for the flame stabilized in the inlet section and are presented in

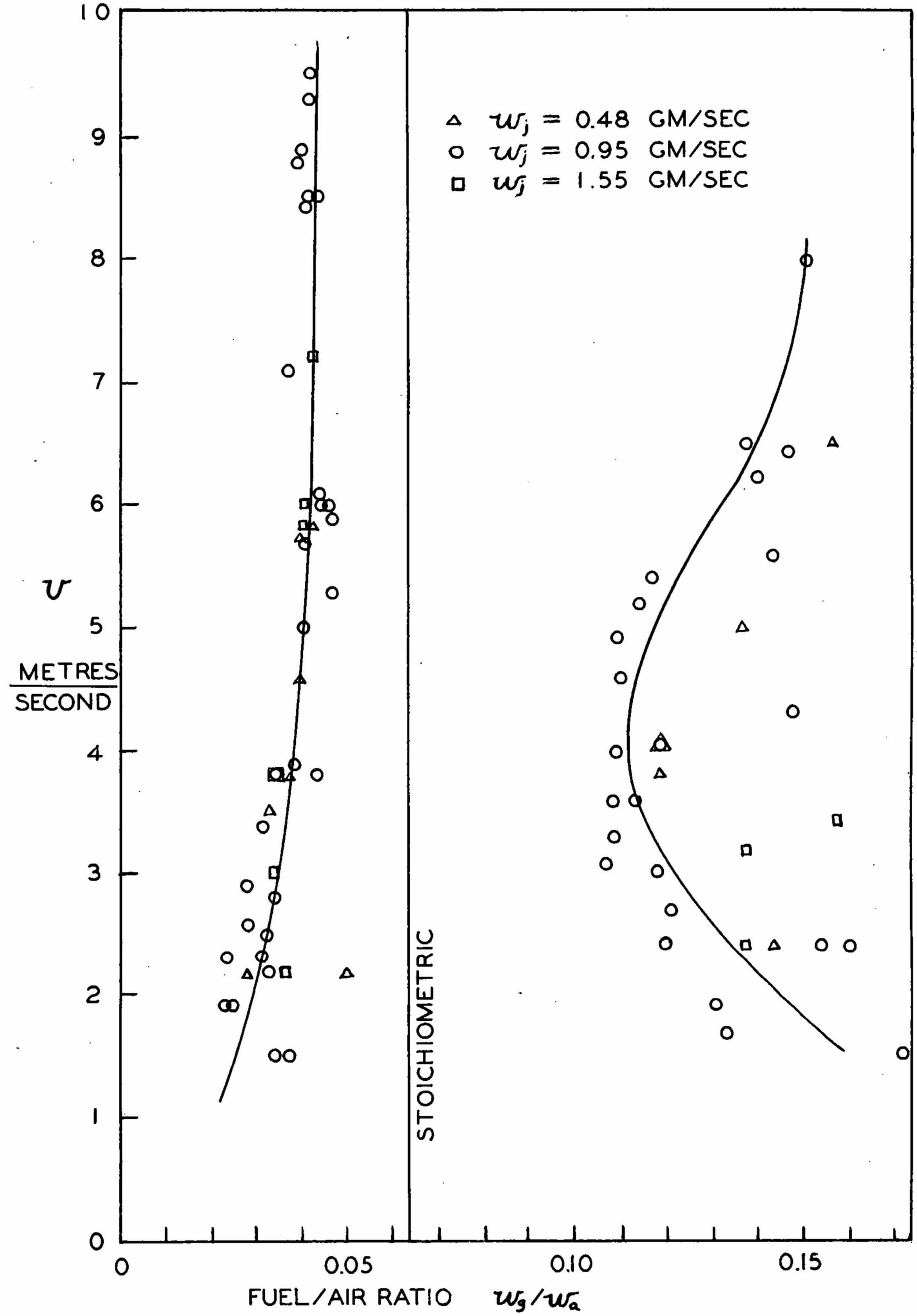


FIG.39. STAB. DUCT II: STABILITY DIAGRAM - ENTRANCE

Fig. 39. There is no systematic effect of jet flow in the weak extinctions; however, in the rich extinctions a jet flow of 0.95 gm per sec seems to give less stability than smaller and larger ones (0.48 and 1.55 gm per sec). This may be just scatter: the jet flow would not be expected to influence the stability of a flame upstream of the U exit lip.

As the entry velocity increases beyond five metres per second, the rich extinction limit is extended, contrary to expectation. The higher the flows, the longer the time taken to open the gas valve and enrich the mixture, and the hotter the duct walls become, thus extending the extinction limit. This explanation also accounts for the weak range ceasing to decrease at high inlet velocities.

With an entry velocity of three metres per second it was possible to get the flame stabilized off the exit lip of the recirculating duct at rich mixtures - fuel/air ratio around 0.13, or double stoichiometric. The position of the flame could be varied by changing the fuel/air ratio. However, this was not stabilization by recirculation: it was stabilization by adding air to a rich mixture.

The inlet section had been peeling open and was leaking badly - easily seen at rich mixtures. Figs. 40 and 41 show it just before it was finally shut down.

Fig. 42 shows the holes in the inlet section. There were large cavities beneath the Bacite surface as in the case of the exhaust hole. The Silachrome which had been used to seal the stainless steel connecting tube to the inlet section had cracked and separated from the connector, allowing leakage (Fig. 40). Fig. 43 shows the entry section with Bacite removed. The protruding mass in the foreground is

Fig. 40. Inlet section leaking

Fig. 41. Inlet section failure, immediately before shut-down.

Fig. 42. Inlet failure

Fig. 43. Inlet failure, Bacite removed

Fig. 44. Main casting broken open, showing ejector

Fig. 45. Inlet section; recirculating loop U

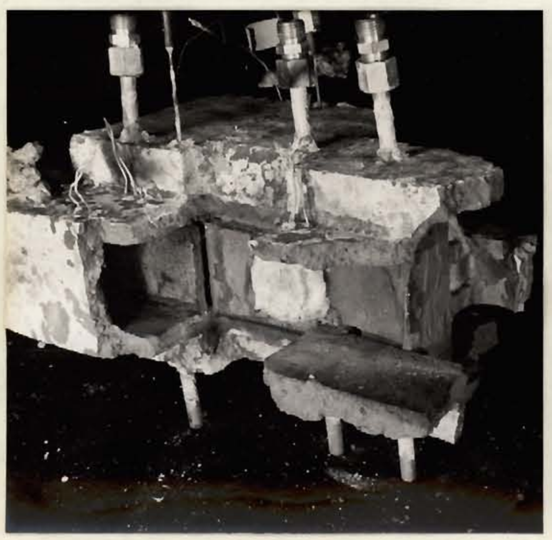


Fig. 46. Combustion section Exposed Fiberfrax. Exit lip of U left of centre

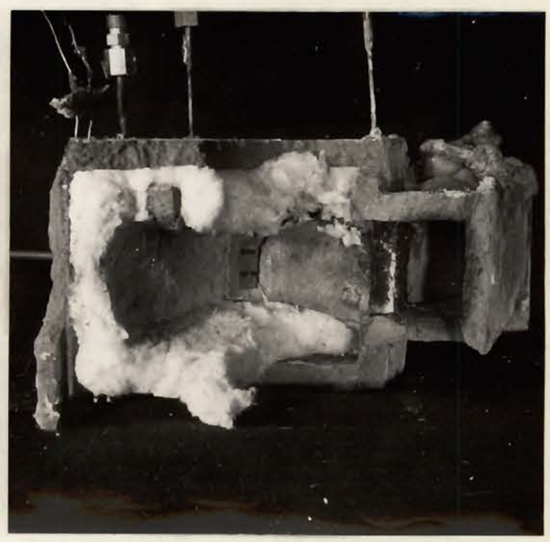


Fig. 47. Exhaust section and ejector housing. Jet tubes at centre

hardened Bacite; the glassy black material at the left is vitrified Silachrome.

The main casting was broken open to permit inspection of the interior. Figs. 44 and 45 show respectively the diffusing portion of the ejector and the U. They are completely unharmed: they did not get very hot.

In Fig. 46 the exit lip of the U may be seen slightly left of centre. The edge had been built up of Silachrome because Chromepour was not strong enough for such a thin section. It is intact. At centre is the Fiberfrax exposed when a piece of the dividing wall broke away and lodged in the exhaust (where it may be seen right of centre in Fig. 47). It was somewhat crisp but otherwise unaffected. The Fiberfrax in the rest of the duct is the same as when installed (Figs. 44 , 47).

In Fig. 47 the ejector entrance is seen immediately right of centre. The depth to which the darkening of the Chromepour extends into the wall of the main section is clearly visible. The irregular shape in the top right corner is the patch applied to the hole in the exhaust section.

Except in the regions where the two holes occurred the Chromepour was little affected: it was yellowish, and the inner wall brown. In places the Silachrome used for sealing joints and cracks had run downward and formed glassy black beads. It appears to be less heat resistant than Chromepour.

The recirculating loop was examined for signs of leakage. The lugs for attaching thermocouple T_1 and pressure tap P_2 were broken off, but this appeared to have happened when the castings were being broken open. There was a serious leak near the outlet of the U, joining the U, the cooling passage, and the interior of the main casting. Cooling air could therefore enter the loop, possibly flowing back along the wall, and perhaps providing enough back pressure to reduce the effectiveness of the ejector. Yet the flow of gases issuing from the U exit appeared uniform in spite of this addition of air from the side.

10. CONCLUSIONS AND RECOMMENDATIONS

THEORY

For stable combustion at the flow rates encountered in gas turbine combustion chambers some fraction of the combustion products must be recirculated to initiate reaction in the fresh mixture as it enters the combustion zone. In general, the greater the amount of recirculation provided the wider will be the range of flows and fuel/air ratios in which combustion is possible. However, the reversals of flow involved are bound to cause pressure losses, reducing the work obtainable from the turbine. Thus it is highly desirable to find out exactly how much recirculation is necessary to ensure reliable operation throughout the required range of flows and mixtures.

The elementary theory given at the beginning of Section 2 illustrates the way in which recirculation controls the location of the flame, and predicts extinction when the amount of recirculation is inadequate. However it does not lend itself to quantitative treatment because of the concepts of ignition temperature and flame thickness which are incorporated. These are avoided in the subsequent reaction rate theory.

Four factors have been considered. They are the heat added by recirculation, by reaction, and by conduction, and the heat lost to the walls. The heat released by reaction is assumed to be proportional to the reaction rate obtained from chemical kinetics. Thus the activation energy and the order of the reaction are introduced. A difficulty here is that both these quantities depend on the conditions under which the reaction occurs, particularly on the temperature. As the reaction proceeds, the actual mechanism alters as different chain branching and breaking processes predominate. In studying flame stabilization, however, it is the initial stages of the reaction which are important, and it has been assumed that the activation energy and the order of the reaction are fixed for a given set of conditions. This must be kept in mind when the values of these quantities are determined from data on spontaneous ignition delay, which are obtained at fairly high temperatures. It would be useful if such observations could be extended to lower temperatures.

The theory neglects consumption of the reacting materials. This is justifiable because, as seen in Figs. 5 and 6, reaction does not become significant (as evidenced by upward curvature of the temperature versus time curve) until the temperature is about 1300K, when the time (or location) of the flame is nearly established.

Heat carried upstream by conduction complicates the theory by introducing a second derivative with respect to distance, and an accurate solution is in practice impossible. Fortunately this heating is usually extremely small, as shown by a numerical example, and can be neglected.

Heat loss to the walls is assumed to be entirely convective. Even when steam and carbon dioxide are present, heat transfer by radiation is small in laboratory scale combustion tubes, and in the pre-combustion region they are found only in the recirculated gases. Various formulae for the heat transfer coefficient could be used: a very simple one was selected for use here.

The recirculation term is the most important one, and the most difficult to handle. It is assumed that the incoming mixture is heated uniformly by addition of hot combustion products at a constant rate until a fixed quantity has been added. This leads to a simple expression. However, it is pointed out that in reality the heating is not uniform. When recirculated gases are added to the fresh mixture, small regions of the latter will be heated much more than others: combustion will begin in these and spread to the rest, no further mixing of hot gases being necessary. Therefore this theory, in common with that of Mordell (6), calls for higher degrees of recirculation than would be actually required. A calculation based on a small combustion tube similar to Stabilization Duct II shows that 42% of the combustion section flow would have to be returned for stability. This would be reduced if the walls were considered to be above room temperature (as, in fact, they were), but increased by cooling of the combustion products in the recirculating loop. Larger combustion chambers would need less recirculation because of relatively smaller cooling by the walls. It is possible that a more accurate representation of heating by recirculation might be formulated along the lines of Semenov's discussion (9) of ignition by a hot wall or wire.

UPSTREAM JETS

Before a theory of recirculation can be evaluated and corrected if necessary, measurements must be made of the degree of recirculation required in experimental combustion tubes. The easiest way of inducing recirculation is to blow a jet of air upstream. In an open space, this causes a toroidal vortex in which combustion products are carried upstream through the inside to meet the fresh mixture, which passes around the outside. It might be possible to calculate the amount of entrainment, but the flow conditions are hard to determine in an open space.

Unfortunately for our purposes, when the jet is surrounded by a confining tube, even of 80 times the jet diameter, the flow system appears to be very different. The flame takes the form of a parabolic shell surrounding the jet, with no visible break at the apex. Probably stabilization is by very small eddies in the surface of the flame. This type of stabilization was investigated briefly, as described in Section 3. The conclusions are:

1. Increasing jet flow initially extends the range of stable fuel/air ratios.

2. There is a certain jet flow for a given entry velocity and jet diameter which gives maximum stability. Larger jet flows reduce stability. This jet flow is given by a relation of the form

$$\frac{w_j^*}{d^{\frac{1}{2}} \; v^{\frac{2}{3}}} = \text{CONSTANT} .$$

   Thus higher jet flows are required with larger diameters or higher inlet velocities.

3. Increasing inlet velocity reduces the range of fuel/air ratios for stable combustion, more so for large jets which have lower jet velocities.

The jet flow required for maximum stability is small, ranging from 1.1% of the main mass flow for the 16 gauge jet and 3.79 metres per second inlet velocity, down to 0.5% for the 22 gauge jet and 10.50 metres per second inlet velocity. Jet flows one-half as large give almost as good stability.

The final decrease in stability with increasing jet flow is attributed to blocking of the main tube by the jet, so that the incoming mixture has to accelerate to pass through the remaining space.

Observations of the distance to which the jet penetrated the main stream led to these further conclusions:

4. Increasing jet flow initially increases the jet penetration.
5. There is a certain jet flow, for a given jet diameter, which gives maximum penetration, and this is independent of entry velocity. This jet flow is given by a relation of the form

$$\frac{w_j^*}{d} = CONSTANT.$$

This corresponds to a constant momentum flux or force. Larger jet flows reduce penetration up to a point, then slowly increase it again.

6. Increasing inlet velocity reduces the jet penetration, according to

$$\frac{x\, v^{\frac{1}{2}}}{d^{\frac{1}{2}}} = CONSTANT.$$

A plot of $\frac{w_j}{d}$ against $\frac{x\, v^{\frac{1}{2}}}{d^{\frac{1}{2}}}$ brings the penetration observations fairly well together, the remaining scatter being random. It should be noted that the uncertainty in determining the location of the apex of a flame near extinction, and usually fluctuating, was up to 0.5 cm, or about 5% of the maximum penetration.

The jet flow for maximum penetration is less than that for maximum stability, especially at high inlet velocities. Both effects may have the same cause, but this seems improbable in the light of their different dependence on main stream velocity. Also, at high jet flows the flame apex is much blunted, suggesting a change in the type of flow.

Because of the uncertainty regarding the nature of the flow, no attempt was made to calculate entrainment of combustion products. However, it had been conclusively shown that increasing recirculation increases stability, up to the point where the walls begin to interfere. Shadow or Schlieren photographs might provide more information. A larger main tube should reduce the effect of blocking by the jet.

RECIRCULATING DUCTS

A more positive method of measuring recirculation was considered essential so two combustion ducts were made in which a fraction of the combustion products could be recirculated through a separate passage, where the flow might be measured by calibrated pressure taps. The first, of simple design, provided experience for the design and construction of the second. The difficulties proved to be larger than anticipated, and no reliable measurements of recirculation were obtained with either duct. However the lessons learnt from them are reported here as a guide to future work.

Because it was essential to avoid cooling the combustion products during recirculation, the walls, at least, of the recirculating loop had to be near the flame temperature, which precluded the use of metals. This meant that a refractory would have to be used, and the complicated shapes required made a cold castable imperative. It was found that Sternson Chromepour would provide acceptable castings in very complex shapes, if a certain amount of ingenuity were used in the design of the molds and the castings themselves. The following recommendations are made for such construction:

1. Sharp corners should be avoided wherever possible: cracks will form in acute angles.

2. Sections should be as thick as possible; this will be limited by considerations of heat capacity, thermal stresses, etc. Sections less than one centimetre thick are difficult to cast.

3. Sections as thin as four millimetres can be made by plastering the cement onto the mold with a knife or similar instrument. Provision for this limits the design of the mold.

4. Molds must be waterproof and strong enough to resist the weight of the wet cement. Well varnished wood is suitable. It should be greased.

5. Cores must be easily removable. It is best to cut or break them into small pieces for removal. Wax cores cannot be melted out without damaging the casting.

6. If at all possible a duct should be cast in one piece. It is impossible to work to very close tolerances, and joints are difficult to seal. Pieces which fit inside one another should be particularly avoided if the castings are to be heated before assembly, otherwise the ensuing warpage will make assembly very difficult.

7. As soon as poured, castings (particularly small ones) should be covered up or placed in a closed container to avoid too rapid drying, which leads to crumbling.

In the design and construction of the duct, great care must be taken to avoid any irregularities which might cause turbulence, thus stabilizing the flame. Similarly, where the two streams of gases meet, their velocities must be carefully matched to avoid extraneous stabilization such as was encountered in Stabilization Duct I. Therefore some means of controlling the velocity of the recirculated gases is essential. In the two ducts used in this work, ejectors were provided for control. These were not very satisfactory. The ejector in Duct I was certainly crude, but that in Duct II was quite carefully designed, and the maximum secondary to primary flow ratio was only about seven. To obtain appreciable secondary flow, the jet flow must be increased; the secondary flow increases, but less rapidly, so the ratio is poorer. This means that the recirculated gases are seriously cooled and diluted by the jet, reducing their effectiveness. Cooling and dilution could be practically eliminated by supplying the jet with a mixture of fuel and air, which would burn after injection.

The two refractory stabilization ducts described here did not withstand high temperatures as well as was expected. The first developed many small cracks and would have fallen apart had it not been held together by the insulation. The second did not crack seriously but failed in two places. The first failure, in which the wall of the exhaust section was melted through, is not explained. In the second, the inlet section walls split at the corners of the rectangular cross section and peeled back, which could be attributed to thermal stresses.

To reduce thermal stresses the refractory walls should be as thin as casting considerations will permit, and effective insulation

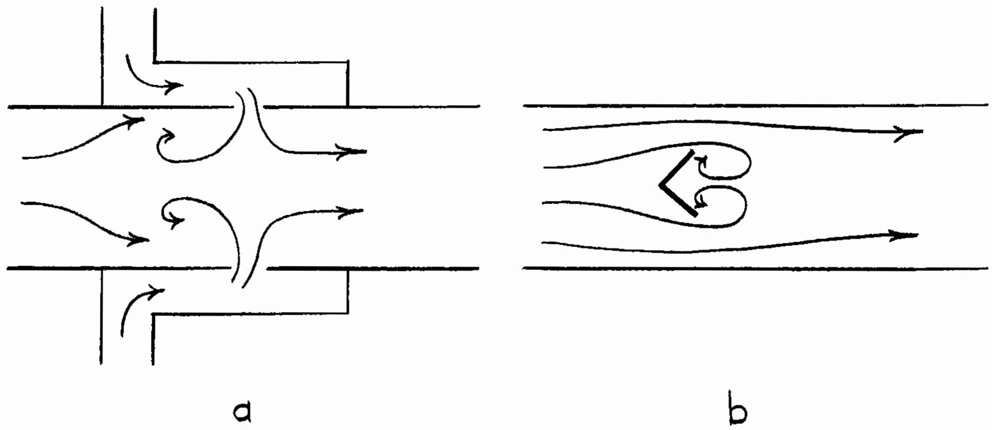


Fig. 48. Suggested ducts

must be applied. Bird-Archer Bacite insulating cement is easy to apply and supplies useful strength, but it does not withstand the temperatures existing near the refractory. The Carborundum Company's Fiberfrax insulation was unharmed by any of the conditions met in Stabilization Duct II. However, it is in the form of a loose fibre and must be supported. Probably a layer of Fiberfrax next to the refractory, covered with a coating of Bacite, would be satisfactory.

SUGGESTIONS FOR FURTHER WORK

With the experience gained from these two ducts it might be possible to construct another which would permit quantitative measurements of recirculation to be made, but the probability of success appears to be somewhat less than one-half. There may be a better chance by resorting to a less direct approach. Fig. 48 shows two suggestions for a two-dimensional recirculating duct. In the first, recirculation is induced by the introduction of air through slots in the walls, which are cooled by the air passing outside them. It is not necessary for the walls to be very hot since the recirculation takes place away from them; therefore they can be made of stainless steel. In the second arrangement, an obstacle is inserted in the stream: it is cooled by the unburnt mixture striking its upstream side. In both cases the other two walls are formed by Vycor windows. The flow patterns are to be observed, perhaps by shadow or Schlieren photographs, or by introducing particles into the stream. From the flow paths an estimate is to be made of the amount of recirculation. The second arrangement duplicates that of Williams (12), but as far as is known no attempt was made to measure the recirculation.

APPENDIX

Symbol	Definition (Defining equations in parentheses)	Units, where used (secondary units in parentheses)
A	Cross sectional area	metres^2 (mm^2)
$\mathcal{A}$	(1.13)	
B	(1.13)	
c	Specific heat	joules per kg deg
C_L	(2.12)	
C_r	(2.7)	
d	Jet tube inside diameter	metres^2 (mm^2)
D	Hydraulic mean diameter	metres
E	Activation energy	
h	Heat transfer coefficient	watts per metre^2 deg
H	Heating value	joules per kg
k	Thermal conductivity	watts per metre deg
K	(8.1)	
$\mathcal{K}$	(2.18)	
ℓ	Length	metres
ℓ'	Flame distance	metres
L	Distance for complete mixing	
$\mathcal{L}$	(2.18)	
$\mathcal{L}'$	(2.21)	
$\mathcal{L}''$	(2.27)	
$\mathcal{M}$	(2.18)	
$\mathcal{M}'$	(2.21)	
$\mathcal{M}''$	(2.27)	

Symbol	Definition (Defining equations in parentheses)	Units, where used (secondary units in parentheses)
n	Order of reaction	
p	$\frac{dT}{dl}$, supplementary variable	
P	Pressure	atmospheres
P_o	Atmospheric pressure	
ΔP	Pressure differential	(in. water, in. mercury)
q	Rate of heat gain	watts
q_c	Rate by conduction	watts
q_L	Rate of heat loss	watts
q_r	Rate of heat gain by reaction	watts
q_r'	Rate per unit volume	watts per metre^3
q_μ	Rate of heat gain by recirculation	watts
R	Gas constant	
R_o	Universal gas constant	
R	(2.18)	
R'	(2.21)	
R''	(2.27)	
t	Time	seconds (minutes)
t'	Flame time	seconds (milliseconds)
t_o	Time for complete mixing	seconds (milliseconds)
T	Temperature	degrees Kelvin
T_b	Final temperature	degrees Kelvin
T_c	Critical temperature	degrees Kelvin
T_{ig}	Ignition Temperature	
T_r	Temperature due to recirculation	
T_w	Wall temperature	degrees Kelvin
T'	Temperature when flame occurs	
T_o	Initial temperature	degrees Kelvin

Symbol	Definition (Defining equations in parentheses)	Units, where used (secondary units in parentheses)
v	Velocity	metres per sec
v_f	Flame speed	
v_j	Jet velocity	metres per sec
v_j^*	Jet velocity at maximum penetration	
v_o	Initial velocity	
V	Volume	metres^3
w	Mixture mass flow	kg per sec (gm per sec)
w_a	Air mass flow	kg per sec
w_g	Fuel mass flow	
w_j	Jet mass flow	kg per sec (gm per sec)
w_j^*	" at maximum penetration	kg per sec (gm per sec)
w_r	Recirculated mass flow	
w_{ro}	Total recirculated mass flow	
χ	$\frac{R_o T}{E}$, supplementary function	
x	Distance from jet tip to flame apex	metres (cm)
x^*	Maximum penetration distance	metres (cm)
α	(1.10)	
β	(1.11)	
δ	Flame thickness	
$\Phi(T)$	(2.19)	
$\Phi'(T)$	(2.22)	
$\Phi''(T)$	(2.27)	
η	Combustion efficiency	
λ	$\frac{l}{L}$	
μ	Recirculation parameter	
μ'	Fraction of total flow recirculated	

Symbol	Definition (Defining equations in parentheses)	Units, where used (secondary units in parentheses)
μ^*	$\lambda\mu$ for stability	
μ_0	Total recirculation	
ρ	Density	kg per metre^3
ρ_0	Initial density	
τ	Delay time	seconds (milliseconds)
ω	Fuel/air ratio	

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