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# THEORETICAL ASPECTS OF GAS TURBINE COMBUSTION PERFORMANCE

by

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# Theoretical aspects of gas turbine combustion performance

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## S U M M A R Y

A correlating parameter for gas turbine combustion performance, based on a 'burning velocity' theory for primary zone combustion is derived using a more direct approach than that originally employed by Greenhough and Levebvre.[1] The various applications of this parameter are discussed and it is shown that the shape of correlated performance curves is directly related to the combustion processes taking place in the various zones of the chamber.

An alternative, more basic, theory is presented in which it is assumed that the low-pressure performance of a spray-type combustor is controlled by a balance between the separate effects of chemical reaction, fuel evaporation and mixing. It is argued that combustion efficiency is a function of $p^x/M$ where $x = 2.0$, 1.7 or 1.0 depending upon whether the rate of heat release is governed by chemical reaction, fuel evaporation or mixing respectively. It is postulated that the amount by which values of x determined experimentally fall below 1.7 provides a useful practical indication of the extent to which mixing is intervening in the overall combustion process. At high pressures the mixing process predominates, $x = 1$, and it is shown that, for any given fuel-air ratio, the rate of heat release depends only on flame-tube geometry and mode of fuel injection, and is independent of chamber size, pressure loss factor and the operating conditions of pressure, temperature and velocity.

The basic principles involved in the design of primary combustion zones for maximum volumetric heat release rate and maximum stability in terms of wide burning range are discussed.

Contents

List of Symbols

| | | |
|---|---|---|
| $S_L$ | = | normal burning velocity, ft/sec. |
| $S_T$ | = | turbulent burning velocity, ft/sec. |
| M | = | chamber air mass flow, lb/sec. |
| P | = | static pressure, p.s.i.a. |
| $P_2$ | = | inlet static pressure, p.s.i.a. |
| T | = | air temperature, °K |
| $T_2$ | = | air inlet temperature, °K |
| L | = | characteristic dimension of combustion chamber |
| $A_f$ | = | flame area, $ft^2$. |
| $A_{ref}$ | = | maximum cross sectional area of chamber, $ins.^2$ |
| $D_{ref}$ | = | maximum diameter or width of chamber, ins. |
| $V_{ref}$ | = | mean chamber velocity at $A_{ref}$, ft/sec. |
| $q_{ref}$ | = | reference dynamic head = $\frac{1}{2}\rho V_{ref}^2$ |
| $\Delta T$ | = | theoretical temperature rise due to combustion, °C |
| H | = | lower calorific value of fuel, Chu/lb. |
| f | = | fuel-air ratio by weight |
| $C_p$ | = | mean specific heat at constant pressure, Chu/lb/°c |
| $\rho$ | = | density, $lb/ft^3$ |
| $\mu$ | = | gas viscosity, lb/ft.sec. |
| $\eta_c$ | = | combustion efficiency |
| Re | = | Reynolds number |
| $\Delta P_{ft}$ | = | pressure drop across flame-tube, $lb/ft^2$ |
| $U_j$ | = | velocity of primary zone air jets, ft/sec. |
| $D_{ft}$ | = | flame-tube diameter, ft. |
| $d_j$ | = | diameter of primary injection holes, ft. |

## Introduction

One of the main requirements of the gas turbine designer is a means of relating combustion performance to the main operating variables of pressure, temperature and mass flow, and to the combustion chamber dimensions. If such a relationship could be firmly established significant progress could be made along the following lines:

(a) test data obtained on any given combustor over a restricted range of operating conditions could be used to predict combustion performance at any other operating condition.

(b) the performance of any new combustion chamber could be predicted over its entire operating range from a limited amount of test data obtained from an existing chamber which is similar in design although not necessarily the same size

(c) the relative merits of various combustor designs could be readily compared even if the relevant test data were obtained at different operating conditions from combustors differing in size.

Unfortunately the various processes taking place in the primary zone of a gas turbine combustion chamber are so complex as to preclude a detailed theoretical treatment. They involve fuel atomization and evaporation, and subsequent mixing with air and combustion products, all occurring simultaneously with chemical reaction and heat transfer. Progress in isolating and evaluating the component processes under representative environmental conditions is of necessity very slow and, until more detailed knowledge is available, suitable parameters relating combustion performance to combustor dimensions and operating conditions can only be achieved using very simplified models to represent the combustion zone.

## Burning Velocity Theory

The combustion zone is envisaged as being similar in structure to the flame brush produced on a Bunsen burner under turbulent flow conditions. Combustion performance is then described as a function of the ratio of turbulent burning velocity to the velocity of the fresh mixture entering the combustion zone. It is assumed that all the fuel that burns does so completely. Combustion inefficiency arises when some of the mixture succeeds in passing through the combustion zone without being entrained by a turbulent flame front.

This model was used by Greenhough and Lefebvre[1] in deriving a parameter which was shown to correlate successfully experimental data on combustion efficiency obtained over a wide range of pressure, temperature and air mass flow on various designs of combustion chamber. The following analysis is essentially the same as that employed by Greenhough and Lefebvre, except for minor modifications which allow the end result to be achieved much more directly in a smaller number of steps.

We have:-

Heat supplied to combustion zone = f.M.H. Chu/sec.

where f = fuel-air ratio

M = air flow rate to combustion zone, lb/sec.

H = LCV of fuel, Chu/lb.

Heat released in combustion $= \rho .\ A_f .\ S_T .\ C_p .\ \Delta T$ Chu/sec.

where $\rho$ = gas density, lb/sec.

$A_f$ = flame area, ft$^2$.

$S_T$ = turbulent burning velocity, ft/sec.

$C_p$ = specific heat of mixture, Chu/lb/°C

$\Delta T$ = theoretical temperature rise due to combustion, °C.

Combustion efficiency is defined by

$$\eta_c = \frac{\text{heat released in combustion}}{\text{heat supplied}}$$

$$= \frac{\rho .\ A_f .\ S_T .\ C_p .\ \Delta T}{f.M.H.}$$

and, since $\frac{C_p .\ \Delta T}{f.H.} = 1$, by definition, and $A_f \propto A_{ref}$, we have

$$\eta_c \propto \frac{S_T}{V_{ref}} \qquad (1)$$

According to Damköhler[2] turbulent burning velocity may be related to laminar flame speed by an equation of the form:

$$S_T \propto S_L .\ Re^a \qquad (2)$$

For flow in the primary zone, $Re = \frac{U_j D_{ft} \rho}{\mu}$ (3)

where $Re$ = Reynolds No.

$U_j$ = flow velocity through primary zone air injection holes, ft/sec.

$D_{ft}$ = flame-tube diameter, ft.

$\mu$ = gas viscosity, lb/ft.sec.

The above expression for Reynolds No. is the same as that used previously,[1,3] except that chamber reference velocity has been replaced by primary zone air injection velocity. This is believed to be an improvement on the original theory since, in practice, the level of turbulence is dictated by the air injection velocity and not the chamber reference velocity.

It can readily be shown that

$$U_j = V_{ref}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5} \qquad (4)$$

where $V_{ref}$ = chamber reference velocity, ft/sec.

$\Delta P_{ft}$ = flame-tube pressure drop, $lb/ft^2$

$q_{ref}$ = dynamic head based on $V_{ref}$, $lb/ft^2$

$\dfrac{\Delta P_{ft}}{q_{ref}}$ = flame-tube pressure loss factor.

Substituting (4) into (3) gives

$$Re = \left(\frac{V_{ref}.D_{ft}.\rho}{\mu}\right)\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5} \qquad (5)$$

Substituting (5) into (2) gives

$$S_T = S_L\left(\frac{V_{ref}.D_{ft}.\rho}{\mu}\right)^{a}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5a} \qquad (6)$$

Laminar burning velocity varies with both pressure and temperature. Its variation with pressure may be expressed as[4,5]

$$S_L \propto P^{n-2/2}$$

where n is the order of reaction.

The variation of $S_L$ with temperature is more complicated since it depends on the type of fuel and the fuel/air ratio. If temperature terms are neglected then, for a bimolecular reaction, substitution of (6) into (2) and then (2) into (1) yields:

$$\eta_c = f\left[\frac{\left(P\ D_{ft}\right)^{\frac{a}{1-a}}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{\frac{0.5a}{1-a}}}{V_{ref}}\right] \qquad (7)$$

Now $V_{ref} \propto \frac{M}{P_2 A_{ref}}$ and $D_{ft} \propto D_{ref}$

where $P_2$ = chamber inlet pressure, p.s.i.a.

Also, let $m = \frac{a}{1-a}$

Substitution into (7) gives

$$\eta_c = f\left[\frac{P_2 A_{ref}(P_2 D_{ref})^m \left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5m}}{M}\right] \qquad (8)$$

Lefebvre and Halls[3] demonstrated that combustion efficiency data obtained during low pressure tests on several types of combustion chamber could be satisfactorily correlated by assuming a = 0.43 (m = 0.75). (This compares very favourably with the value of 0.4 obtained in separate experiments on turbulent flames by Damkohler[2] and Delbourg[6]). Their correlating parameter, which also included a temperature term derived from reaction rate theory[1], was of the form:

$$\eta_c = f\left[\frac{P_2^{1.75} A_{ref} D_{ref}^{0.75} \exp\frac{T_2}{b}}{M}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.4}\right] \qquad (9)$$

Due to the relatively small variation in inlet temperature experienced by most combustion chambers, (9) is not very sensitive to the value attributed to b. Suggested 'ideal' values[1] are 300 and 150 for overall air-fuel ratios of 60 and 100 respectively, but with spray-type combustors satisfactory correlation of data is obtained using a constant value for b of 300.

Experimental data to justify the inclusion of a pressure loss term in (9) is sparse and conflicting. Thus whereas Lefebvre and Murray[7] were able to correlate combustion efficiency data obtained on an aircraft combustion chamber against $\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5}$, in close agreement with equation (9), Clarke, Harrison and Odgers[8] and Clarke, Odgers and Ryan[9] found that pressure loss had very little effect on the combustion efficiency of a Longwell-type spherical reactor. The situation is further complicated by the knowledge that flame-tube pressure loss factor not only influences the turbulence level of the injected-air jets and their rate of mixing with burned products, but also the actual quantity of burned products entrained in the recirculation zone.[10] In view of the inconclusive experimental evidence on the influence of pressure loss, and its relatively small variation between different types of chamber, the pressure loss term in equation (9) is usually omitted in practice.

Applications of burning velocity theory

Equation (9) provides a relationship between combustion efficiency and the dimensions, pressure loss and operating conditions of a combustion chamber. For any given combustion chamber, dimensions and pressure loss are constant and (9) may be abbreviated to

$$\eta_c = f\left|\frac{P_2^{1.75}\exp^{\frac{T_2}{300}}}{M}\right| \tag{10}$$

Equation (10) has been applied with considerable success to the correlation of experimental data on combustion efficiency, and has proved very useful in reducing the amount of rig testing required to evaluate new combustor designs. As shown in figure 1 only a few test points are needed to establish the complete performance curve for a chamber. Furthermore it is possible to predict combustion efficiencies with reasonable accuracy at flow conditions which lie outside the capacity of the rig testing facility. The relationship will, of course, break down at conditions where the effects of atomization and heat losses become significant; fortunately such conditions lie outside the range of normal engine experience.

The main advantage of equation (9) is that it provides a method of scaling combustor dimensions and operating conditions to common values so that any differences in performance which remain can be attributed directly to design. This is a tremendous asset when attempting to select a design for a new combustion chamber from a choice of several existing designs, none of which is of the required size or has been tested at the required operating conditions.

The manner in which the parameter is used can be demonstrated by reference to figure 2 from reference 3. This figure illustrates performance curves for three different combustor designs. Clearly design A is superior to design C because for any given value of combustion efficiency the $\theta$ parameter has a lower value. This means that at any given operating conditions of M, P, and T, design A can equal the performance of design C and yet be made smaller in size. In some instances the optimum design is not so clearly defined. Comparison of the curves for chambers A and B shows that, although chamber A has the better performance over part of the range, chamber B is superior at conditions which on an actual engine would correspond to operation at very high altitudes.

The starting point for any new design of chamber must be based to a large extent on previous experience. The most useful way in which past experience can be summarized is by the use of graphs or charts in which performance data from all known system are correlated against all the important variables. Such a chart is shown in figure 3 in which combustion efficiency is shown plotted against $\theta$, $\theta$ being essentially the right hand side of equation (9) with the pressure loss term omitted. In this figure the hatched areas embrace experimental data obtained from a large number of current designs of tubular, tubo-annular and annular chambers. Figure 3 may be used to determine the size of chamber required to meet any stipulated performance requirement. The most arduous operating conditions are those at which the inlet pressure, $P_2$, is a minimum. Fortunately, this invariably corresponds to a high altitude 'off-design' condition at which a low level of combustion efficiency, say 80 percent, may be tolerated. $A_{ref}$ and $D_{ref}$ are then obtained from figure 3 by reading off $\theta$ at a point within the hatched area corresponding to 80 percent combustion efficiency, and substituting into it the appropriate values of $P_2$, $T_2$ and M. The actual point chosen will represent a balance between the conflicting needs of high performance and low development cost.

With any new design, once the main dimensions have been established the next important decision to be made is in regard to the distribution of air to the various zones of the chamber. In particular, the proportion of air allocated to the primary zone has a decisive effect on every important aspect of combustion performance, as illustrated in Table 1 in which the relative advantages and drawbacks of stoichiometric, weak and over-rich primary zones are summarized.

In practice it is often difficult to ensure that a primary zone is operating at the design mixture strength. Fortunately one can learn a great deal about the actual primary zone mixture strength, and other important features of the combustion process, from inspection of the $\theta$ curves. This may be explained by reference to figure 4 which shows that any given $\theta$ curve actually represents the combined performance of the primary and secondary zones, (and also the dilution zone at very low pressures). The broken line emanating from the origin represents primary zone performance. The higher the burning velocity the steeper is the slope of the line. The upper limit to this line is reached when all the fuel has been consumed. At this point the effective combustion efficiency is less than 100 percent owing to losses by chemical dissociation of the combustion products. The function of the secondary zone is to recover this dissociation loss by discreet addition of more air. Its performance is represented by the broken line drawn from an origin corresponding to the maximum combustion efficiency attained in the primary zone. The full line in figure 4 describes the combined performance of primary and secondary zones and indicates the level of combustion efficiency actually measured in practice.

As the pressure is reduced it should be possible in theory to maintain a high level of combustion efficiency by reducing the mass flow. However, with falling gas pressure atomization quality deteriorates and heat losses become more significant, and thus there is a minimum pressure below which combustion is impossible. The effects of poor atomization and heat losses are denoted in figure 4 by the manner in which the lower portion of the overall performance curve falls away from the broken line to intercept the abscissa at a finite value of $\theta$.

The influence of primary zone mixture strength on the shape of the $\theta$ curve is illustrated in figure 5, which shows three typical curves for stoichiometric, weak and over-rich primary zones. With stoichiometric mixtures both burning velocity and chemical dissociation are at a maximum. Thus, as shown in figure 5, the left hand portion of the $\theta$ curve has a steep slope which begins to flatten out at about 80 percent combustion efficiency. With a weak primary zone the left hand portion of the curve starts at a higher value of $\theta$ than the stoichiometric mixture and has a shallower slope due to its lower burning velocity. It rises to a higher level of efficiency, which is consistent with the lower dissociation loss, before terminating in a fairly pronounced 'knee' at about 85 percent combustion efficiency.

With rich mixtures the $\theta$ curve starts closer to the origin and again has a low initial slope due to the relatively low burning velocity. Also, since an appreciable proportion of the total heat release is now taking place in the secondary zone, there is less distinction between the roles of the primary and secondary zones, and this is reflected in the shape of the $\theta$ curve which flattens out gradually with increase in $\theta$, with little or no suggestion of a 'knee'.

The ability of the right hand portion of a $\theta$ curve to attain 100 percent combustion efficiency depends partly on the length of the flame-tube but to a greater extent on the amount of air employed in film-cooling the walls. Unburned or partially burned fuel can become entrained in this air and be conveyed along the flame tube from one layer to the next before being discharged from the chamber. Since the cooling-air temperature is low, chemical reaction rates are also low, and thus, once any unburned mixture has become entrained there is little likelihood of any further reactions taking place. (In general, the lower the amount of air employed in film cooling, particularly in the primary zone, the better is the prospect of achieving 100 percent combustion efficiency at the chamber outlet.)

Influence of pressure

As stated earlier, for most practical purposes it is satisfactory and convenient to assume a constant value for 'a' of 0.43. Since $m = \frac{a}{a-1}$ this corresponds to a value for m in the $\theta$ parameter of 0.75. However, it should be recognized that 'a' is not a constant but actually varies with pressure. Evidence on the effect of pressure has been provided by Childs and Graves[11] who conducted a series of tests using propane fuel and plotted, for various fuel-air ratios, combustor air-flow rate against pressure with parametric lines of combustion efficiency. At low levels of pressure the slope of their curves is approximately 2, which corresponds to a value of m in equation (8) of 1.0 and to a value of 'a' in equation (2) of 0.5. This agrees with the value for 'a' derived theoretically by Damkohler[2] for low Reynolds Numbers. Equation (8) thus becomes

$$\eta_c = f\left[\frac{P_2{}^2 A_{ref} D_{ref}}{M}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5}\right] \qquad (11)$$

At higher levels of pressure the slope of the curves is 1.3, corresponding to values for 'a' and 'm' of 0.23 and 0.3 respectively. Thus 'a' and 'm' decrease with increasing pressure and it is reasonable to assume that at high pressures they are close to zero. Inserting m = 0 into (8) gives

$$\eta_c = f\left[\frac{P_2 A_{ref}}{M}\right] \tag{12}$$

## Heat release criteria

On the basis of equations (11) and (12) it is clear that the performances of two different chambers cannot be compared if the relevant experimental data were obtained at low pressures in one case and at high pressures in the other. Appropriate heat release criteria for the two extreme levels of pressure are tabulated below:

| Pressure Level | Performance Parameter | Heat Release Criterion |
|---|---|---|
| Low (Sub-atmospheric) | $\eta_c = f\left[\frac{P_2{}^2 A_{ref} D_{ref}}{M}\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5}\right]$ | $Chu/hr/ft^3/atm^2/\left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{.5}$ |
| High (Say, above 3 atmospheres) | $\eta_c = f\left[\frac{P_2 A_{ref}}{M}\right]$ | $Chu/hr/ft^2/atm.$ |

## Alternative Theory

The burning velocity model embodies two important assumptions, namely that all the fuel is completely mixed with the incoming air prior to combustion, and that fuel evaporation time is zero. With well-designed combustion chambers operating at sub-atmospheric pressures these assumptions are reasonable and justified. However, a more general and more accurate description of combustion performance should be based on considerations of the basic processes involved, without specifying any particular model to represent the combustion zone. Let us assume that

$$\eta_c = f\,(\text{air-flow rate})^{-1}\left(\frac{1}{\text{evaporation rate}} + \frac{1}{\text{mixing rate}} + \frac{1}{\text{reaction rate}}\right)^{-1} \tag{13}$$

If the evaporation and mixing rates are both infinite then (13) becomes virtually identical to (1) and leads directly to the $\theta$ parameter. It is, however, of interest to examine how combustion performance is affected by finite values of evaporation and mixing rates.

Evaporation rates

It has been shown elsewhere[12] that the lifetime of an evaporating fuel drop is given by the expression:

$$t \propto d\left(\frac{d}{U_{rel}P}\right)^{0.5} \qquad (14)$$

where $t$ = drop lifetime

$d$ = drop diameter

$U_{rel}$ = relative velocity between fuel and gas

$P$ = gas pressure

From considerations of the effect of pressure on $d$ and $U_{rel}$ it was concluded that for any given engine

$$E \propto P^{1.7} \qquad (15)$$

where E is defined as an 'evaporation performance number' and is the ratio of time available for evaporation to the time required for evaporation. Equation (15) shows that for any given engine the problem of fuel evaporation diminishes with increasing pressure. In other words evaporation rates are most significant at the lowest operating pressure of the engine. A further implication is that, provided evaporation performance is satisfactory at low pressures, it should be more than adequate at higher pressures. Since chemical reaction rates and evaporation performance both increase with pressure it follows that any observed deterioration in combustion efficiency with increase in pressure can only be due to inadequate mixing.

Mixing rates

The rate of mixing between a turbulent air jet and the surrounding gas is given by the product of the eddy diffusivity, the mixing area and the density gradient. If it is assumed that the eddy diffusivity is proportional to the product of a mixing length, L, and the turbulent velocity in the air jet, then

$$\text{mixing rate} = (\text{eddy diffusivity})(\text{mixing area})(\text{density gradient})$$

$$\text{mixing rate} \propto (L\ U_J)\ (L^2)\ (\rho/L)$$

$$\text{mixing rate} \propto \rho\ U_J\ L^2 \qquad (16)$$

Substituting in (16) for $U_J \propto \left(\frac{\Delta P_{ft}}{\rho}\right)^{.5}$ yields

$$\text{mixing rate} \propto \frac{P_2 L^2}{T^{.5}}\left(\frac{\Delta P_{ft}}{P_2}\right)^{0.5}$$

Under conditions where mixing is limiting to performance, combustion efficiency will depend on the ratio of the mixing rate to the air-flow rate, and the appropriate correlating parameter is

$$\eta_c = f\left[\frac{P_2 D^2_{ref}}{MT^{.5}}\left(\frac{\Delta P_{ft}}{P_2}\right)^{0.5}\right] \tag{17}$$

Within the primary zone the mixing length may be assumed proportional to $d_J$, the diameter of the primary injection holes. Substitution in (16) gives

$$\text{mixing rate} \propto U_J d_J^{\,2}$$

$$\text{and mixing performance} = f\left[\frac{\rho U_j d_j^{\,2}}{M}\right]$$

$$= f\left[\frac{U_j d_j^{\,2}}{V_{ref} D_{ref}^{\,2}}\right] \tag{18}$$

Now it can readily be shown that

$$\frac{U_J}{V_{ref}} = \left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{0.5} \tag{19}$$

and

$$\frac{d_J^{\,2}}{D_{ref}^{\,2}} = \left(\frac{\Delta P_{ft}}{q_{ref}}\right)^{-0.5} \tag{20}$$

Substituting (19) and (20) into (18) gives the result that $\eta_c = f$ (unity) i.e. mixing performance is independent of operating conditions, combustor dimensions and flame-tube pressure loss factor. Although at first sight this result may seem surprising it may be readily understood by considering the physical processes involved. For example, for any given combustor any increase in mixing requirement arising through an increase in $V_{ref}$ is automatically accommodated by an increase in mixing rate through the corresponding increase in $U_J$. Similarly any increase in mixing area obtained, say, by increasing $d_J$, will be exactly offset by the corresponding reduction in $U_J$. However, these arguments should not be taken to imply that the mixing performance of all combustion chambers is exactly alike. Equation (18) applies to circular injection holes only. In practice the mixing performance of a combustion chamber will depend critically on the shape of the air injection holes and on their location in the flame tube. The relevant correlating parameter for combustion efficiency data obtained under conditions where the rate of heat release is limited by mixing is given by equation (17).

Another important consideration is that whereas for any given total injection hole area the mixing area is independent of hole diameter, the core volume, which is wasted volume as far as the combustion process is concerned, is proportional to hole diameter. Thus from a purely mixing viewpoint the best utilisation of the total available injection area is in the form of a large number of small holes. However, the stability of the system is proportional to the scale of the recirculatory eddies created by the air jets which is in turn proportional to the diameter of the air jets. Thus in practice a certain minimum size of air jet must be stipulated in order to meet the stability requirement.

Analysis

We have $\eta_c = f(\text{air-flow rate})^{-1}\left(\frac{1}{\text{evaporation rate}} + \frac{1}{\text{mixing rate}} + \frac{1}{\text{reaction rate}}\right)^{-1}$

Considering only the pressure effect, and assuming a second order reaction process, we have

$$\eta_c = f\, M^{-1}\, (C_1 P^{-1.7} + C_2 P^{-1} + C_3 P^{-2})^{-1} \qquad (21)$$

The implication of (21) is that in any given combustion chamber operating at low pressures the heat release is governed to varying degrees, depending upon various factors such as the flame-tube size and configuration and the type of fuel injection employed, upon the rates of fuel evaporation, mixing and reaction. If the combustion efficiency data correlate best with pressure exponent of nearly 2 then either evaporation rates or reaction rates or both are controlling the rate of heat release. If the pressure exponent is less than 1.7 then the amount by which it is less is a measure of the extent to which mixing is intervening in the overall combustion process. With increase in pressure the rates of evaporation and chemical reaction increase more rapidly than the mixing rate and thus the performance of the combustion chamber becomes increasingly dependent on the mixing process. At the highest operating pressures it is solely dependent on mixing, and the pressure exponent is close to unity. This decrease in pressure exponent with increase in pressure is fully consistent with the burning velocity theory and is in close agreement with experimental observations.[11]

The relative importance of these various rate processes at low pressure is demonstrated in figure 6. This figure is based on data obtained by Graves[13, 14], using a J33 tubular combustor. Two series of tests were carried out. In one the fuel employed was iso-octane, in the other gaseous propane. With the latter fuel the evaporation time is, of course, zero, and it would also be expected that the rate of mixing is appreciably higher than with liquid fuel. The combined effect of these two factors in improving the low pressure performance of the chamber is clearly demonstrated in the figure.

## Stability

In addition to its main role as the major heat release zone of the chamber, the other important function of the primary zone is to recirculate heat and burnt products to mix with the incoming air and fuel. By this means combustion can be sustained over a wide range of pressures, velocities and fuel-air ratios. Stability is improved by reducing the amount of air flow into the primary zone, since the additional residence time allows the processes of fuel evaporation, mixing and chemical reaction to proceed nearer completion. Unfortunately, a reduction in primary zone air-flow also increases exhaust smoke and so, in practice, the actual amount employed is usually resolved as a compromise between the conflicting requirements of stability and ignition performance on the one hand and exhaust smoke level on the other.

For any given primary-zone air-flow quantity, maximum stability is achieved by injecting the air through a small number of large holes. This is because the size of the eddies created by the jet, and thus the residence time of the material entrained within them, is roughly proportional to the diameter of the injection holes. However, as discussed earlier, large injection holes tend to be wasteful of combustion volume and so again a compromise solution must be sought.

The influence of air-injection hole size on volumetric heat release rates has already been discussed. It is of interest now to examine its effect on stability. Experimental evidence on this point has been provided by Jeffs[15] from studies of the performance of several can-type combustors when supplied with uniform mixtures of kerosine and air. One can featured a large-scale flow reversal at its upstream end supplied with mixture from a single row of large holes. Another, a so-called 'pepperpot' system, employed a large number of small jets in order to try and approach homogeneity throughout the combustion zone. Stability loops obtained with these two cans are shown in the upper diagram of figure 7 which clearly illustrates the superior stability of the single-row can with its large-scale recirculation. However, as pointed out by Jeffs, with the single-row can the level of combustion efficiency at blow-out is very much lower than the corresponding value measured on the pepperpot system. Thus, if the same data were replotted as graphs of equivalence ratio against heat release per unit volume, the stability loops would be of the form illustrated in the lower diagram of figure 7. This shows the pepperpot system in a more favourable light in terms of heat release rate, although still inferior to the single-row can in regard to burning range.

The influence of the fuel injection process on stability characteristics is illustrated in figure 8. This figure contains two stability loops which are typical of those obtained from a single combustor featuring: (a) uniform injection of fuel and air and (b) spray injection. With uniform mixtures the maximum heat release rate is high but the burning range is narrow, its actual width being determined by the level of pressure and the scale of mixing as discussed in connection with figure 7. With spray injection the maximum heat release rate is relatively low. This is because although the equivalence ratio at this point is nominally stoichiometric, because of the relatively poor fuel distribution a large proportion of the total combustion process is taking place at mixture strengths which are either weaker or richer than stoichiometric and reaction rates are in consequence appreciably less. However, with spray systems, although the overall equivalence ratio in the combustion zone may lie outside the limits of inflammability, the poor fuel distribution ensures the existence of regions within the flame in which the local fuel-air ratio is inside the limits of inflammability. Thus spray systems are characterized by wide burning limits, in particular a good weak extinction point (typically 1,000 A.F.R. as compared with 120 A.F.R. for a premixed system). With vaporizer systems, in which all the fuel is well mixed with part of the combustion air, the overall level of fuel-air mixing is superior to the spray chamber but inferior to the premixed system. As a result its stability loop lies somewhere in between the loops for the uniform mixture and the atomized mixture, a typical value for weak extinction being about 400 A.F.R.

From the above discussion it is apparent that for maximum volumetric heat release the primary zone should be designed to provide a large number of small jets of uniform fuel-air mixture. On the other hand, in order to sustain combustion at low pressures and over a wide range of mixture strengths, the primary zone should employ large-scale flow reversals and spray fuel injectors. Most current aircraft combustion chambers are of the latter type, but in the design of new combustors opportunity should be taken whenever possible to move towards the former type and thereby economise in chamber volume. Such an opportunity could arise, for example, in the design of chambers for lift engines which are normally not called upon to operate at very low pressures.

## Conclusions

1. The $\theta$ parameter for combustion performance, based on a burning velocity model for the primary zone, constitutes a very satisfactory means of relating combustion efficiency to the main operating variables of pressure, temperature and mass flow, and to combustion chamber dimensions. $\theta$ curves are of considerable value in the design and development of new combustion chambers.

2. At low pressures, with modern well-designed combustion chambers, neither fuel evaporation nor mixing are limiting to performance, and experimental data on combustion efficiency may be correlated using the $\theta$ parameter

$$\eta_c = f\left[\frac{P_2^{1.75} A_{ref} D_{ref}^{0.75} \exp\frac{T_2}{300}}{M}\right]$$

3. At high pressures mixing is of paramount importance and the relevant parameter for combustion performance is

$$\eta_c = f\left[\frac{P_2 D_{ref}^{\,2}}{M\ T_2^{\,0.5}}\left(\frac{\Delta P_{ft}}{P_2}\right)^{0.5}\right]$$

4. For maximum volumetric heat release primary combustion zones should be designed with a large number of small injection holes and be supplied with uniform mixtures at stoichiometric fuel-air ratio.

5. For maximum stability in terms of wide burning range the primary zone should employ a small number of large holes in conjunction with a fairly poor fuel distribution as obtained, for example, with a conventional spray atomizer.

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TABLE 1

Influence of Primary Zone Mixture Strength on Performance

| Mixture Strength | Advantages | Disadvantages |
|---|---|---|
| Fuel-weak | 1) Clean, non-luminous flame, with no coke deposition or exhaust smoke<br>2) Low flame temperature giving low rates of heat transfer to walls.<br>3) Good exit temperature distribution. | 1) High recirculation velocity adversely affects stability and ignition performance |
| Stoichiometric | 1) Maximum heat release rate<br>2) High combustion efficiency.<br>3) Clean combustion with little or no coke deposition and exhaust smoke. | 1) High flame temperature gives high rate of heat transfer to walls. |
| Fuel-rich | 1) Low recirculation velocity gives good stability and easy ignition | 1) 'Dirty' luminous flame, resulting in coke deposition and exhaust smoke.<br>2) Usually gives poor exit temperature distribution.<br>3) Flame-tube wall temperature very sensitive to fuel type. |

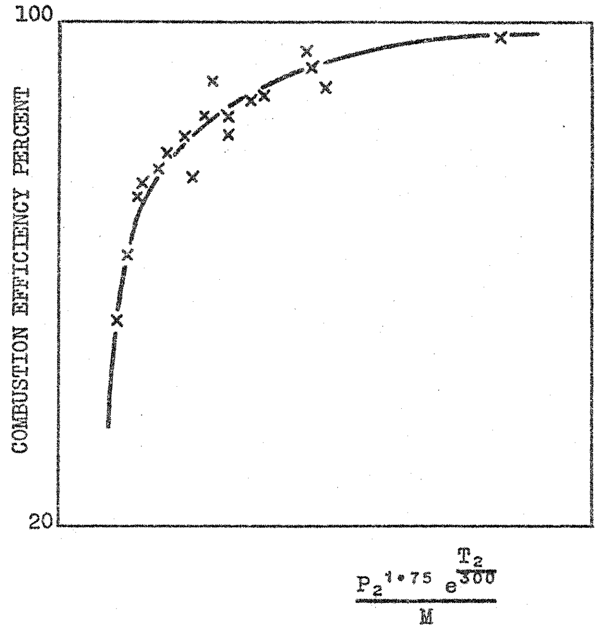


FIG. 1. CORRELATION OF COMBUSTION EFFICIENCY DATA FOR AN AIRCRAFT COMBUSTOR OPERATING AT 60 A.F.R. (ref. 3)

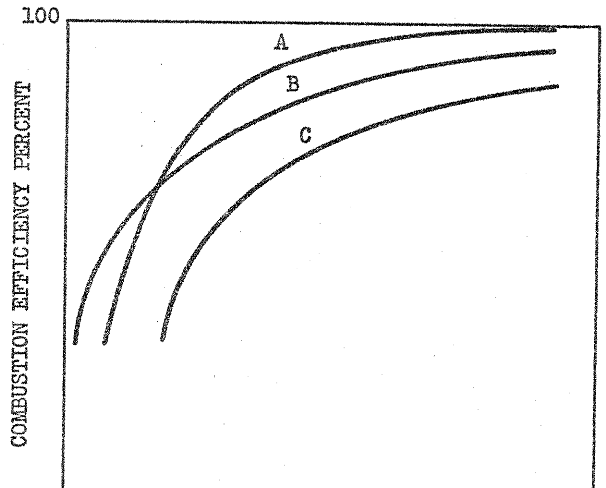


FIG. 2. COMBUSTION PERFORMANCE CURVES FOR THREE DIFFERENT COMBUSTOR DESIGNS. (ref. 3).

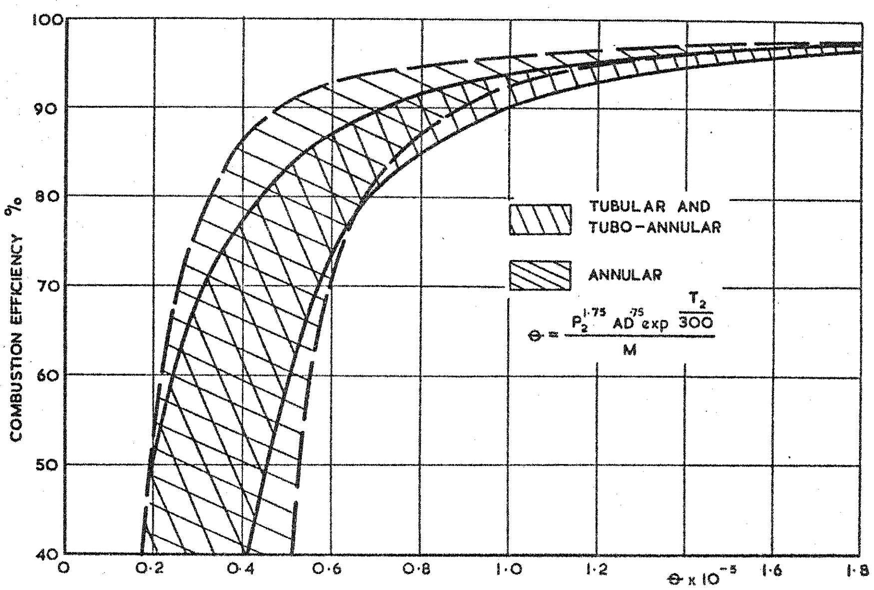


FIG. 3. DESIGN CHART FOR COMBUSTION PERFORMANCE

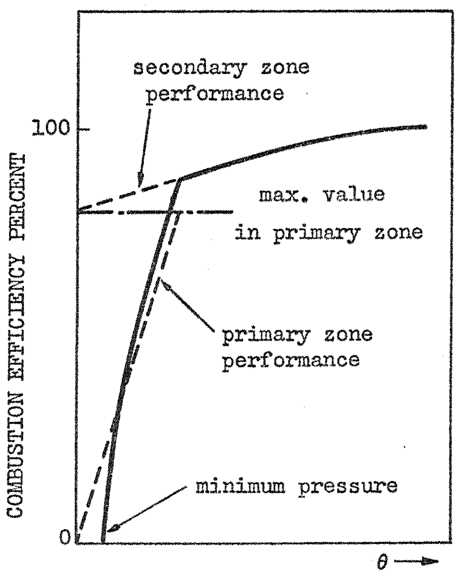


FIG. 4. THEORETICAL CURVE OF COMBUSTION EFFICIENCY (ref. 1)

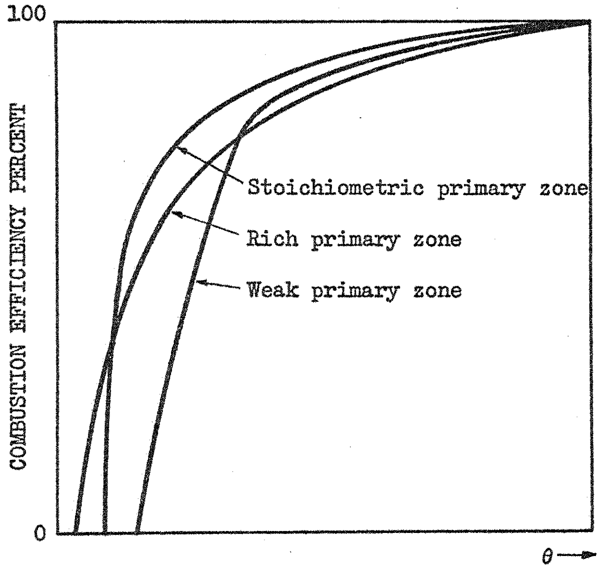


FIG. 5. GRAPHS ILLUSTRATING INFLUENCE OF PRIMARY-ZONE MIXTURE STRENGTH ON SHAPE OF θ CURVES

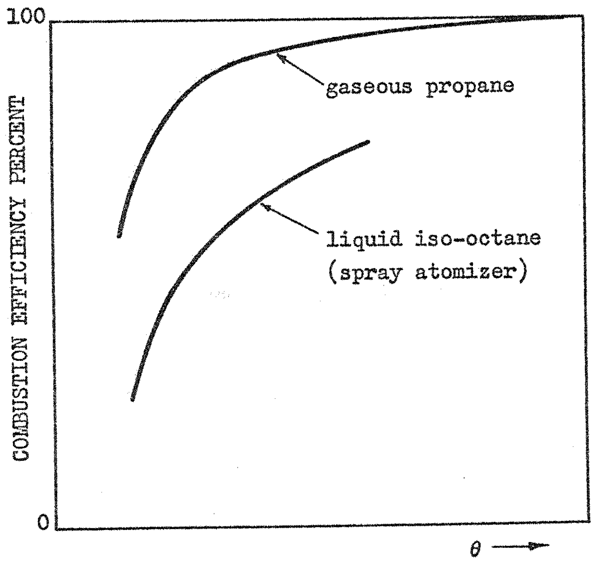


FIG. 6. GRAPHS ILLUSTRATING THE INFLUENCE OF EVAPORATION AND MIXING PROCESSES ON COMBUSTION PERFORMANCE. (Data from refs. 13 and 14)

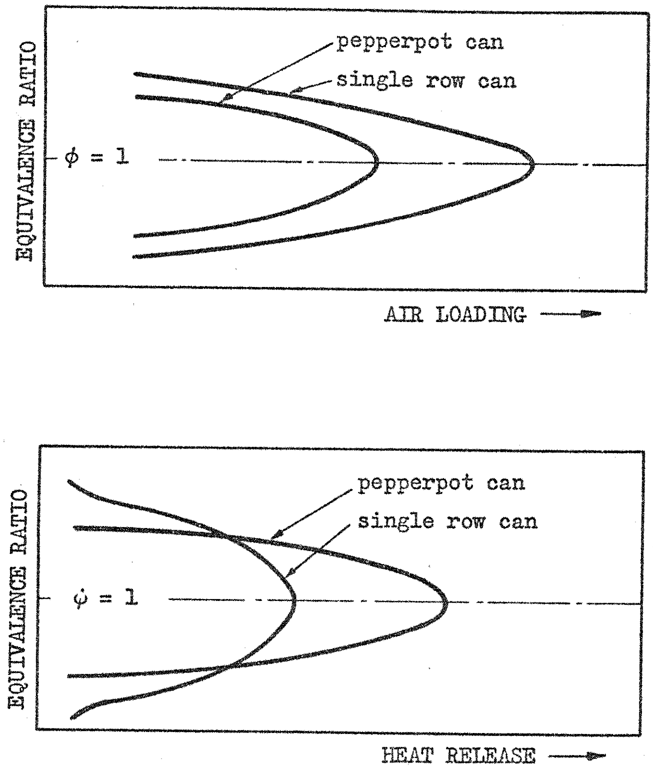


FIG. 7 PERFORMANCE COMPARISON OF 'PEPPERPOT' AND SINGLE-ROW COMBUSTORS.
(Data from ref. 15)

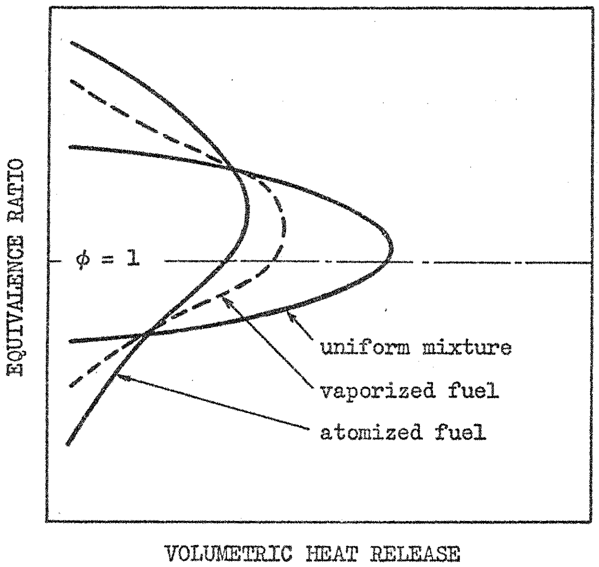


FIG. 8. CURVES ILLUSTRATING THE INFLUENCE OF FUEL INJECTION METHOD ON STABILITY CHARACTERISTICS