# Gas Turbine Combustor Stabilization by Heat Recirculation

A. J. Ganji,* J. Short,* M. C. Branch,† and A. K. Oppenheim‡

*University of California, Berkeley, Calif.*

The feasibility of heat recirculation for stabilization of lean mixtures and emission reduction has been studied in detail for a typical aircraft gas turbine combustor. Thermodynamic calculations have indicated temperature and heat recirculation rates for operation of the combustor over a range of combustion zone equivalence ratios and for varying modes of desired engine operation. Calculations indicate the feasibility of stabilizing the combustion zone at equivalence ratios as low as 0.2 with achievable heat recirculation rates. Detailed chemical kinetic calculations suggest that combustor heat release is maintained with reaction completion substantially before the NO forming reactions, even though CO is rapidly oxidized in this same region.

## I. Introduction

AS part of its continuing program to improve ambient air quality, the U.S. Environmental Protection Agency has published maximum aircraft emission standards for carbon monoxide, oxides of nitrogen, and unburned hydrocarbons scheduled to go into effect in 1979. In an effort to generate technology and demonstrate hardware required to meet these emissions standards the NASA Lewis Research Center is administering the Experimental Clean Combustor Program.[1] The work reported here is part of a fundamental study of stabilization of lean combustion in gas turbine combustors undertaken, in part, to supplement the NASA program.

Pollutant emissions from gas turbine engines are greatest under two extremes of operation: idle or low power, and takeoff or high power.[2] High power operation usually is characterized by high combustion efficiency, high combustor temperature, and hence increased NO emissions. When richness of the primary zone of the combustor is increased at high power, smoke emissions also increase. Emissions at low power or taxi/idle operation are derived from low combustor efficiency at the low combustor inlet pressure and temperature characteristic of this mode. Consequently carbon monoxide and hydrocarbon emission are high, whereas nitric oxide is reduced. Aldehyde emissions reflect a similar dependence on engine power to that of hydrocarbons.

The condition most consistent with simultaneous reduction of all emissions is lean burning with high intensity, stable combustion. In conventional gas turbine combustors, nitric oxide is produced predominately in the primary zone in which near stoichiometric mixtures are burned so that the high final flame temperature can maintain stable combustion through the system. As long as these high temperatures remain, the principal variable controlling combustion intensity, the production of NO, remains tied to combustion stability.

A means of avoiding stoichiometric combustion and yet providing stable combustion for lean mixtures is based on the use of heat recirculation rather than flow recirculation. Experimental burners based on local heat recirculation developed by Weinberg[3] have been shown to be capable of operating continuously on methane-air mixtures containing less than half the proportion of methane corresponding to the

normal limit of flammability. The present study examines the feasibility of the use of this method for combustor stability control and emissions reduction. By recirculating heat from the combustion products back to reactants, the temperature of combustion can be increased for a lean mixture and controlled for combustor stability. The peak combustion temperature then can be greater than the adiabatic temperature for the lean mixture without preheat, but lower than that for a stoichiometric mixture. The combustor exit temperature with heat recirculation at a given constant equivalence ratio would be similar to that for a combustor with stoichiometric primary zone and secondary air addition to the same overall equivalence ratio as the heat-recirculation combustor. Since the chemical history of the mixture in the combustors differs, the exhaust composition will not be the same.

The feasibility of the heat recirculation burner for stabilization of lean mixtures and emission reduction has been studied in detail for a typical aircraft gas turbine combustor. Thermodynamic calculations indicate temperature and heat recirculation rates for operation of the combustor over a range of combustion zone equivalence ratios and for varying modes of desired engine operation. Calculations indicate the feasibility of stabilizing the combustion zone at equivalence ratios as low as 0.2 with achievable heat recirculation rates.

## II. Thermodynamic Analysis

The basic components of a combustor with heat recirculation include a heat exchanger that transfers heat from the products of combustion to the reactants, considered to be premixed or partially premixed, and the combustion zone. Combustion occurs in the form of a premixed flame within the combustion zone after preheating.

The overall combustion process can be described by the model in Fig. 1. In this schematic (a + c) is the heat exchanger that recirculates heat from the products of combustion at (c) to the reactants (or some components of the reactants) at (a). Region (b) is the adiabatic combustion zone, which allows the reactants to reach their adiabatic flame temperature and corresponding equilibrium composition. The purely conceptual region (d) is to allow the cooled combustion products to reach their equilibrium composition by an adiabatic process. At locations 1, 3, and 4', the components are assumed to be in thermal equilibrium, whereas the components are in thermal and chemical equilibrium. Thus, the processes from 1 to 2 and 3 to 4' take place with constant or frozen composition.

The significant difference between the preceding system and a conventional combustor (in the case of equilibrium products) is the different temperature and concentration distribution inside the combustor. The effect of this difference on combustor stability and exhaust species is of interest in this study.
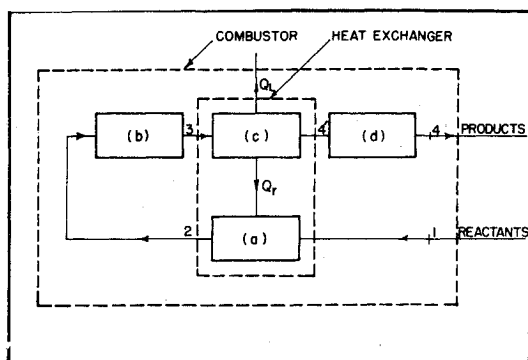
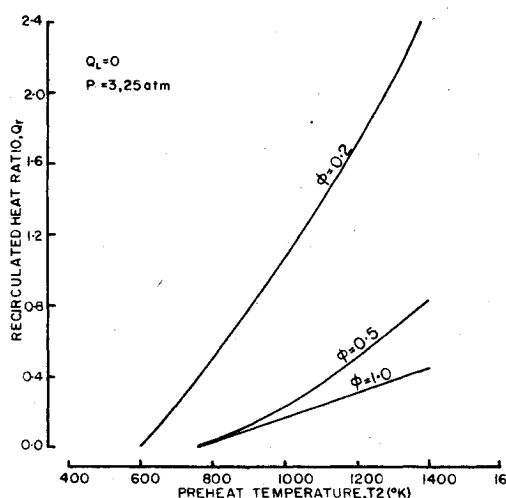Fig. 1 Schematic of constant pressure combustor with heat recirculation for stabilization of lean mixtures.



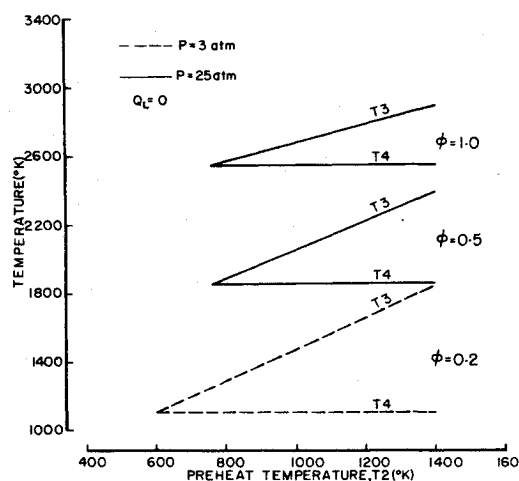Fig. 2 Recirculated heat ratio $Q_r$ required to achieve a given preheat temperature.



Fig. 3 Variation of maximum combustor temperature $(T_3)$ with heat recirculation preheat temperature for several mixture ratios $(\phi)$. The temperature without heat recirculation is constant so that the combustor outlet temperature $(T_4)$ remains constant at each $\phi$.



Fig. 4 Maximum combustion temperature achieved for different combustor equivalence ratios, $(\phi)$ for two pressures.



Fig. 5 Concentration of NO and CO at combustion zone and combustor exit for constant preheat temperature.

The main parameters in the combustor are preheat temperature $(T_2)$, maximum combustor temperature $(T_3)$ and amount of recirculated heat $(Q_r)$. These are also the design parameters for construction of the system. In order to describe the interrelation of the previous parameters and pollutant concentrations in the system, chemical equilibrium and heat-transfer calculations have been made for the model of the combustor. Conditions representative of cruise and takeoff for the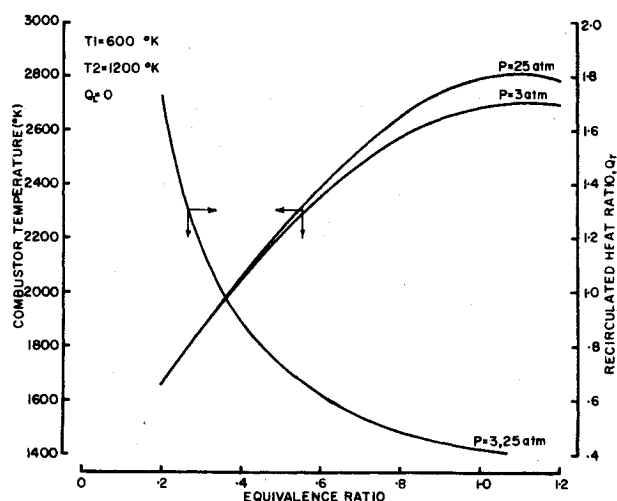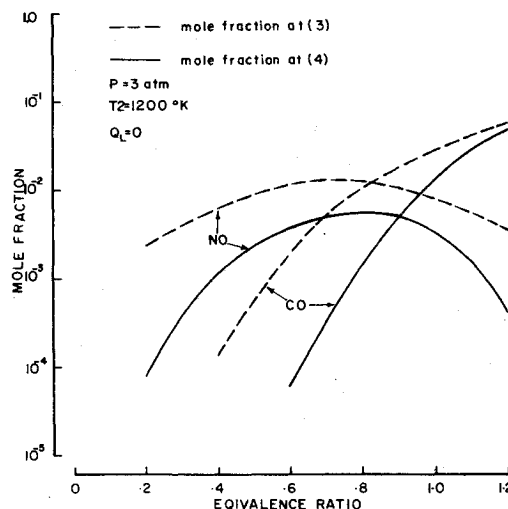 JT9D engine were considered. Both methane and JP-4 were considered as fuels, but since the calculations were so similar, only the results for methane are given here. Methane additionally has the most thoroughly understood kinetics of hydrocarbon fuels.

Typical calculations are presented in Fig. 2-5 for a combustor operation with heat recirculation. The required recirculated heat to achieve a given preheat temperature is given in Fig. 2 for different combustion zone equivalence ratios. Since the recirculated heat is normalized with respect to the combustion enthalpy of the fuel, the ratio $Q_r$ for a given preheat temperature increases as $\phi$ decreases.

The maximum combustor temperature $(T_3)$ and combustor exit temperature $(T_4)$ are given in Fig. 3. Since the fuel/air mixture is preheated at constant mixture ratio, the maximum combustor temperature increases, even though the combustor exit temperature remains constant. The effect of heat recirculation, then, is to preheat the reactants and increase their reaction rate to allow stabilization of leaner mixtures. A mixture at equivalence ratio of 0.4 preheated to 1600 K has the same combustion temperatures as a stoichiometric mixture burning with no preheat. The heat required to preheat the lean mixture is extracted from the higher combustion temperature (i.e., the excess of $T_3$ over $T_4$).

Figure 4 describes the variation of maximum combustor temperature and recirculated heat ratio with mixture equivalence ratio for constant preheat temperature. The effect of combustor pressure on the process is minor.
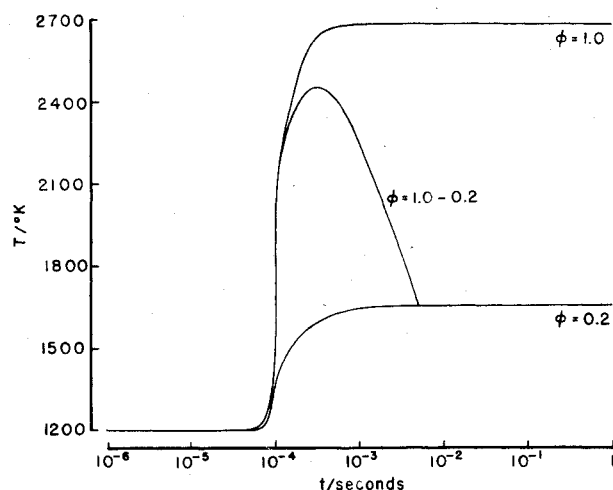
Fig. 6  Temperature history calculated from $CH_4$/air detailed kinetics for stoichiometric combustion, stoichiometric combustion with dilution and lean ($\phi = 0.2$) combustion. Preheat temperature ($T_2$) is constant at 1200K.
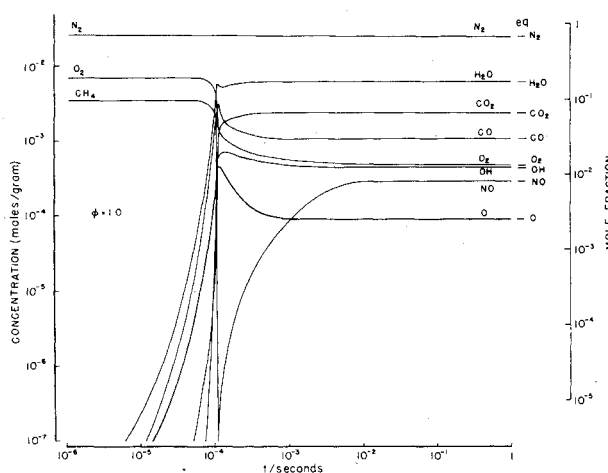


Fig. 7  Species concentration profiles corresponding to stoichiometric combustion of $CH_4$/air, 1200K preheat.



Fig. 8  Species concentration profiles corresponding to lean combustion of $CH_4$/air ($\phi = 0.2$), 1200K preheat.



Fig. 9  Species concentration profiles corresponding to stoichiometric combustion of $CH_4$/air with dilution, 1200K preheat.

Chemical equilibrium species concentrations are presented in Fig. 5 for two locations in the combustor. The concentrations at location 3 correspond to the maximum combustor temperature for varying equivalence ratio. Preheat temperature remains at 1200K. The combustion products then transfer heat to preheat reactants and the equilibrium composition becomes that of location 4, the combustor exit. If equilibrium is maintained, CO and NO emissions can be reduced from their peak combustion values by nearly an order of magnitude over the range of mixture ratio considered. The composition that actually is achieved, however, will be influenced by chemical kinetic effects in the combustion and heat recirculation zones.

## III.  Chemical Kinetic Analysis

In order to appreciate more fully the benefits of lean combustion, it is desirable to study, numerically and experimentally, the kinetics of combustion. The kinetic mechanism for jet fuels is not well-known. However, many of the carbon atoms pass through methane during the reaction. Since methane is one of the few hydrocarbons whose chemical kinetic mechanism is understood, calculations are done using methane as a fuel.

The flow through the combustor is associated with only a slight pressure loss, and heat losses are relatively low. Hence, a good appr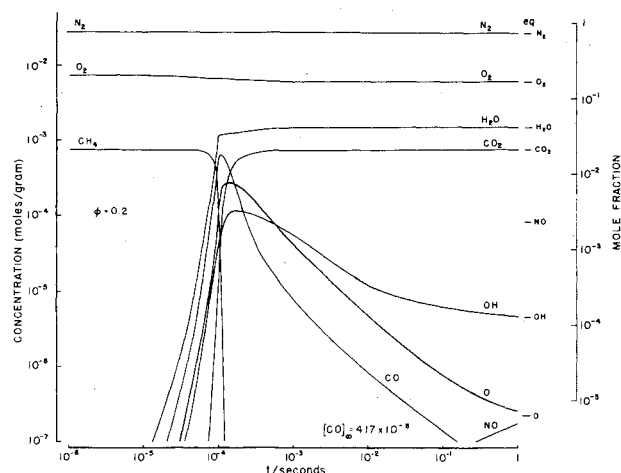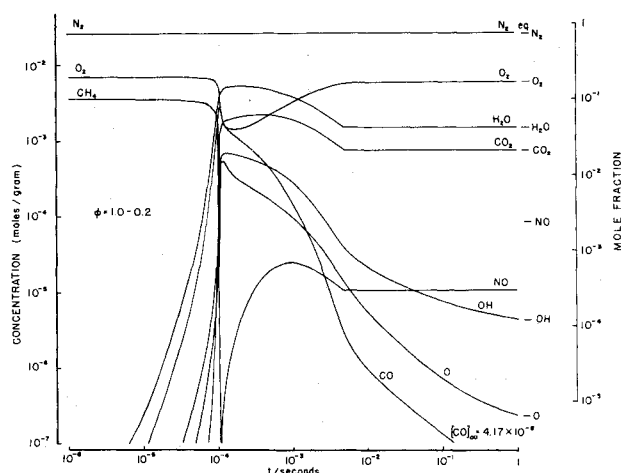oximation to actual conditions is obtained by treating the flow as a constant pressure adiabatic process. This, in effect, decouples the chemistry from fluid mechanics, yielding a zero dimensional model for the combustor. The algorithm used for the analysis of such a process is described in Ref. 4.

The chemical kinetic scheme used for methane oxidation was that of Seery and Bowman,[5] with reverse rates computed to satisfy the relation $K_c = k + /k -$, assuring proper approach to final equilibrium.[6] The scheme was modified to include the "extended Zel'dovich mechanism" for nitrogen oxidation. For the reactions $N_2 + O = NO + N$ and $N + O_2 = NO + O$, the rate constants of Baulch, Drysdale, Horne, and Lloyd[7] were used, whereas for $N + OH = NO + H$, the rate constant $k = 3.3E13$ was employed, as suggested to the authors by Bowman. Three separate cases were considered; in each, the combustor inlet mixture was assumed to be a homogeneous gaseous mixture at 1200K and 3 atm. First, stoichiometric methane-air combustion was analyzed. Second, a fuel lean mixture with an equivalence ratio of 0.2 was calculated. Finally, a stoichiometric mixture was treated under the postulate that it was diluted with air at 1200K and 3 atm linearly, with time between 0 and 5 msec, at which time it reached an equivalence ratio of 0.2. The last two cases simulated the combustion process with heat recirculation and secondary air mixing, respectively.

The temporal temperature profiles for the three cases are depicted in Fig. 6. This illustrates the significant difference between the maximum temperature in the latter two cases, although both have the same end state.

Figure 7 shows the species concentrations for stoichiometric combustion. The reaction very nearly has reached equilibrium within 1 msec. The equilibrium levels are indicated on the right-hand side of the figure. Only the formation of NO lags in this time scale. The level of such pollutants as NO and CO are on the order of $10^4$ ppm. Of course the preheating of a stoichiometric mixture results in a reaction temperature much too high to be of practical interest, and the results for this case have been obtained only to provide proper basis for comparison with other results.

Figure 8 depicts the concentration histories of important species for the fuel lean mixture (equivalence ratio = 0.2). The equilibrium levels for this mixture may be compared directly with those of Fig. 7. The equilibrium value of CO has been reduced to about 5 ppm. The equilibrium value of NO has been reduced to about 2000 ppm. However, the time for combustion has been increased for the lean mixture to about 2 msec. At the time, CO is still decreasing quite rapidly to below 100 ppm, and NO is well below 1 ppm. If the residence time of the reacting particles in the combustion chamber can be made sufficiently long while avoiding preignition problems, extremely attractive air pollution properties can be attained. The final concentration of NO is more than three orders of magnitude below equilibrium.

In the case of the secondary air mixing presented in Fig. 9, the concentration histories start out exactly as the stoichiometric mixture reaction. As significant amounts of air are added, the reaction departs from stoichiometric. The concentrations of various species (except for $N_2$ and $O_2$) begin to reduce rapidly because of the addition of $N_2$ faster than their kinetic rate of production). At 5 msec no more air is added, and most of the concentration profiles look very much like the fuel lean results. The NO concentration history is a notable exception. The so-called "prompt NO" is formed during the time that so little air is added, and the reaction proceeds as a stoichiometric process. When the addition of excess air is completed, the NO level is about 300 ppm, considerably higher than the totally lean combustion process, but still an order of magnitude below the equilibrium level. The NO departs from the "prompt NO" for two reasons. First, as more mass is added, the percentage of NO decreases because of the increase of $N_2$ and $O_2$ as air is added. Secondly, as can be seen from Fig. 9, the temperature for mass addition de-

parts from the stoichiometric temperature only after reaching rather large values. As additional mass is added at 1200K, the mixture cools rapidly, until mass is no longer added, at which time the temperature follows closely the profile for lean combustion. The rapid reduction in temperature leaves the NO formation rate greatly reduced, essentially frozen below final equilibrium. The reduction in temperature changes the NO production rate from quite large to practically negligible.

## IV. Conclusions

These results indicate the feasibility of heat recirculation for stabilization of fuel lean mixtures. Thermodynamic and chemical kinetic calculations at an equivalence ratio as low as 0.2 indicate combustor heat release rates equivalent to $15 \times 10^7$ Btu/hr $\cdot$ ft$^3$ $\cdot$ atm. Combustor heat release is maintained, with reaction completion, substantially before the NO forming reactions, even though CO is rapidly oxidized in this same region.

## References

[1] Niedzwiecki, R. W. and Jones, R. E., "The Experimental Clean Combustor Program-Description and Status," NASA Lewis Research Center, Cleveland, TMX-71547, May 1974.

[2] Jones, R. E., "Advanced Technology for Reducing Aircraft Engine Pollution," American Society of Mechanical Engineers, New York, Paper 73-WA Aero-2, 1973.

[3] Hardesty, D. R. and Weinberg, F. J., "Burners Producing Large Excess Enthalpies," Combustion Science and Technology, Vol. 8, 1974, pp. 201-214.

[4] Cohen, L. M., Short, J. M., Oppenheim, A. K., "A Computational Technique for the Evaluation of Dynamic Effects of Exothermic Reactions," Combustion and Flame, Vol. 24, 1975, pp. 319-334

[5] Seery, D. J. and Bowman, C. T., "An Experimental and Analytical Study of Methane Oxidation Behind Shock Waves," Combustion and Flame, Vol. 14, 1970, pp. 37-48.

[6] Oppenheim, A. K., Cohen, L. M., Short, J. M., Cheng, R. D., and Hom, K., "Shock Tube Studies of Exothermic Processes in Combustion," presented at the 10th International Shock Tube Symposium, Kyoto, Japan, July 1975; also to be published in the Proceedings of the meeting.

[7] Baulch, D. L., Drysdale, D. D., Horne, D. G., and Lloyd, A. C., Evaluated Kinetic Data for High Temperature Reactions, Vol. 2, Buttersworth CRC Press, Cleveland, Ohio, 1973.