JOURNAL OF PROPULSION AND POWER
Vol. 15, No. 2, March–April 1999

Control of Flame Oscillations with Equivalence Ratio Modulation

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A novel technique for controlling oscillating combustion is proposed and demonstrated. For oscillations that exist over a limited range of equivalence ratios, we suggest that periodic modulation around the unstable condition can effectively avoid the oscillating condition, but otherwise produces the desired time-average equivalence ratio. Tests of this concept were carried out in an atmospheric-pressure, swirl-stabilized combustor with a nominal heat input of 30 kW. The fuel was natural gas. We show that it was possible to control a 300-Hz oscillation by modulating the fuel flow at frequencies less than 20 Hz, reducing the observed rms pressure from 2.7 to 0.8 kPa. Limitations on when this technique may produce successful control are also discussed.

Introduction

THE performance requirements of any combustion system include maintaining stable combustion over the entire operating range of fuel/airflow rates. In some instances, the operating range of the combustor is limited by pressure oscillations driven by cyclic variations in heat release. These pressure oscillations can damage the combustor hardware through cycle fatigue, and must therefore be avoided. For example, the magnitude of pressure oscillations in afterburners can increase with thrust, thereby limiting the allowable operating range. As another example, fuel-lean combustor operation is desired for low-NO_x emissions in stationary gas turbines, but lean operating conditions are prone to oscillating combustion.[1] This places a practical limit on the allowable operating range of the combustor. Other examples of oscillating combustion occur in industrial burners,[2,3] rocket engines,[4] and utility boilers.[5,6]

Techniques to eliminate combustion oscillation generally fall into two categories: passive and active. Passive-control techniques typically involve modifying the combustion hardware to eliminate the source of the variation in heat release or to increase the acoustic damping, thereby limiting the magnitude of any pressure oscillations. Putnam[3] discusses these techniques for industrial-burner applications. Harrje and Reardon,[7] and more recently Yang and Anderson,[4] describe passive methods available for stabilizing liquid rocket combustor oscillations. Gutmark et al.[8] investigated passive methods to stabilize combustion in dump combustor (ramjet) applications.

As an alternative to passive control, a number of authors have considered active control to eliminate combustion oscillation. Active control uses some control element to prevent the heat release from coupling to the acoustic pressure. A common proposal for active control is to produce fuel pulses that are out of phase with the variable heat release. The idea is to smooth out the heat release, which otherwise would drive the acoustic oscillation. Langhorne et al.[9] were the first to demonstrate this technique on an experimental afterburner. Similar concepts have been investigated by various authors.[10–13] Aside from modulating the fuel, other attempts to control flame oscillation have used cyclic modulation of combustor boundary conditions[14] or manipulation of mixing-layer vortex dynamics.[15,16] McManus et al.[17] provide an excellent summary of these various active control techniques.

The usual approach to active control is based on the concept that variations in heat release can be stabilized by a repeated control action that occurs at approximately the same frequency as the oscillation. In contrast, some authors have shown that effective control action can occur at frequencies much lower than that of the acoustic oscillation. Brouwer et al.[18] showed that a slow modulation of atomization air (less than 2 Hz) could reduce pollutant emission and combustion oscillation in a liquid-fueled gas-turbine combustor. Richards et al.[19] showed that it was possible to stabilize oscillating natural gas combustion using relatively low-frequency modulation of the fuel. Their technique was shown to stabilize an oscillation even though the fuel modulation frequency was completely unrelated to the natural oscillation. Gemmen et al.[20] reported preliminary findings that the same type of behavior could be observed when modulating a pilot flame. Oscillation control was again achieved by changing the flame conditions at a frequency that was much lower than the acoustic oscillation.

In yet another approach, Knoop et al.[21] have shown that just a few fuel pulses can be used to stabilize oscillations that occur near an operational hysteresis boundary. These authors recognized that some oscillations exhibit a hysteresis behavior, having both a stable and oscillating "branch" at the same operating condition. Such a combustor may oscillate or be dynamically stable, depending upon the history that preceded the particular operating conditions. Within the hysteresis region, Knoop et al.[21] showed that the combustor could be moved from the oscillating branch to the stable branch by using a small number of fuel pulses. By perturbing the system with fuel pulses, it was possible to force a transition from oscillating to steady combustion. These authors reported that the combustor did not spontaneously move back to the oscillating branch when the control pulses were removed. After achieving stability, no additional control action was needed to maintain stable combustion.

Received Aug. 23, 1997; revision received Oct. 5, 1998; accepted for publication Oct. 21, 1998. This paper is declared a work of the U.S. Government and is not subject to copyright protection in the United States.

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In this paper, we report a different approach to producing stable combustion. As in Knoop et al.,[21] this technique is useful near stability boundaries, but it does not require the stability boundary to exhibit hysteresis. The approach is shown conceptually in Fig. 1. The combustor behavior is plotted on a map of airflow and equivalence ratio. The shaded region represents an airflow/equivalence-ratio range in which oscillation occurs. If the desired operating point is inside this region, control will be required to dampen the oscillation. The proposed method to avoid oscillation is to alternate the equivalence ratio between two points outside the oscillation region. By periodically modulating the fuel flow, it may be possible to produce the desired time-average equivalence ratio without operating the combustor within the oscillation region. This type of control is limited to oscillations that are bounded by stable conditions. Examples of this type of instability are reported in this paper, as well as in other studies of premix-turbine combustion,[22] and step-stabilized flames.[23]

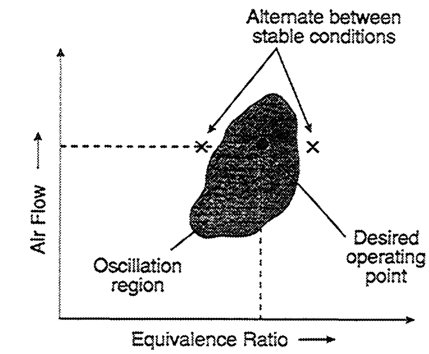

Fig. 1 Proposed method to achieve stable combustion at a desired (but oscillating) operating point by modulating the equivalence ratio.

We experimentally demonstrate the proposed control method in this paper. The subscale test combustor uses a premix fuel nozzle analogous to most stationary gas turbines, employing swirl stabilization around an annular bluff body. We show it is possible to control a (nominal) 300-Hz oscillation by modulating the fuel flow at frequencies less than 20 Hz, reducing the rms pressure from 2.7 to 0.8 kPa.

Experimental Description

The experimental combustor is shown in Fig. 2. This subscale combustor uses a 9.0-cm-i.d. quartz tube as the main combustor body. The quartz tube is contained inside a windowed containment pipe with a 35.6-cm i.d. The large diameter of the containment pipe provides a nearly open acoustic termination at the end of the quartz tube. The dominant oscillation frequency is close to the quarter-wave frequency of the 60-cm tube: ~300 Hz, depending on operating conditions. Dilution air cools the outside of the quartz tube and mixes with the combustion products at the end of the tube. The dilute exhaust products exit through a muffler system outside of the building. Oscillating combustion pressure is measured by a transducer on the upstream end of the combustor body.

Flame chemiluminescence from the OH* radical (315 nm) was recorded with a line filter and photomultiplier mounted 3 m downstream of the combustor. The photomultiplier was positioned to view the entire flame reaction zone. As explained elsewhere,[24] the OH* emission is approximately proportional to the instantaneous value of heat release. Recent studies in-

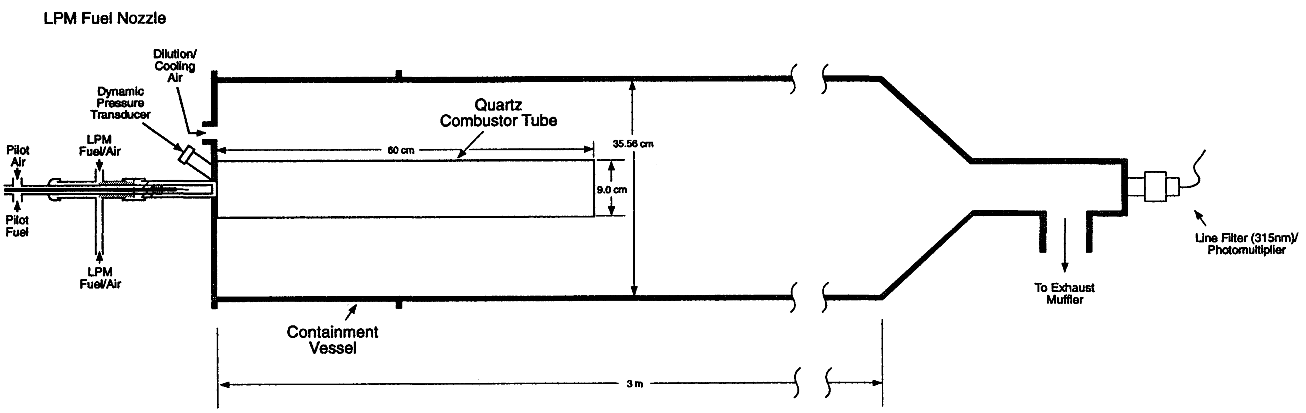

Fig. 2 Schematic of the test combustor.

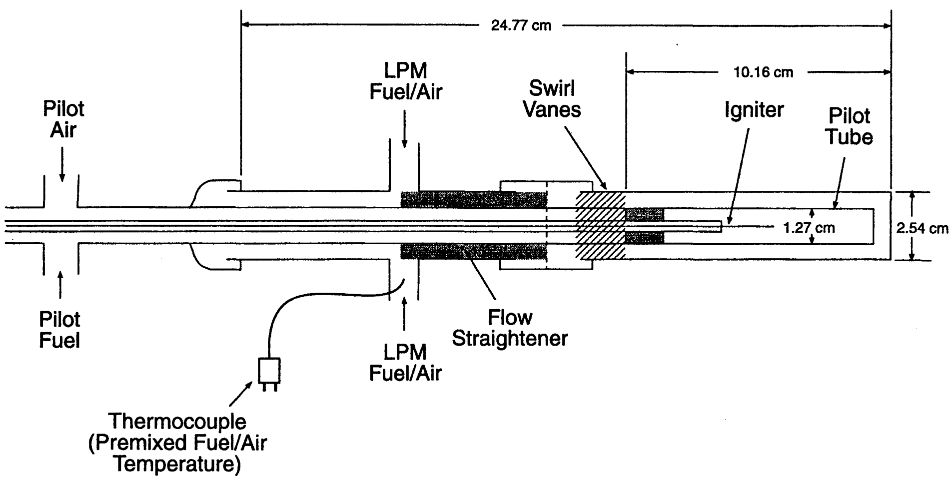

Fig. 3 Schematic of the fuel nozzle. The ignitor and pilot flows were used only during startup.

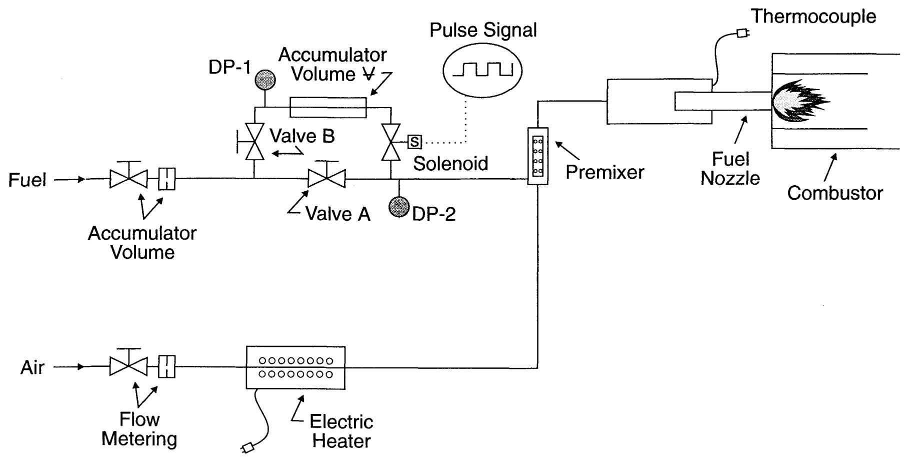

Fig. 4 Fuel and air supply.

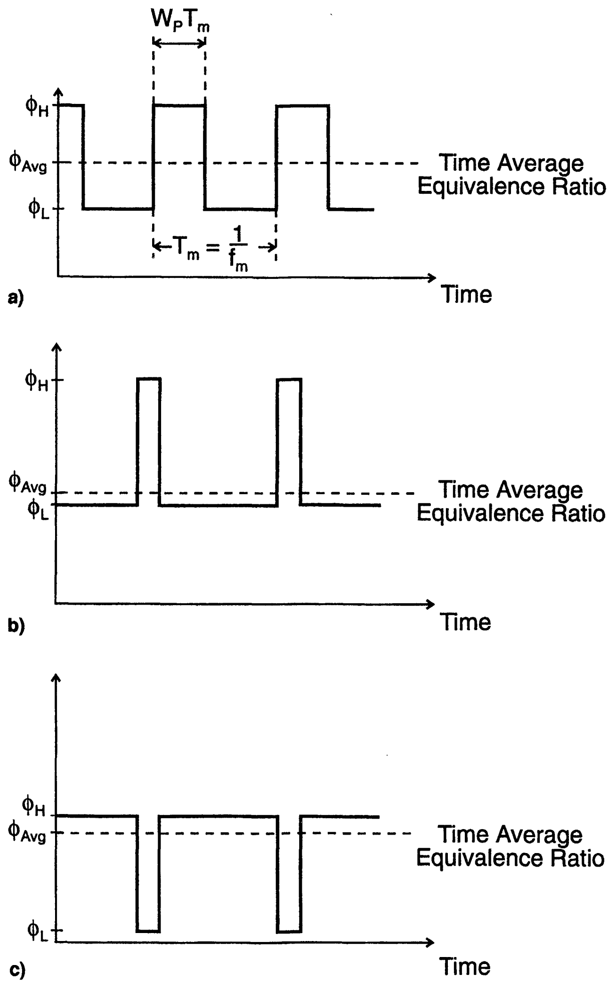

Fig. 5 Definition of fuel pulse parameters: a) medium, b) short, and c) long pulse width.

dicate that the proportionality may be nonlinear[25] and unable to account for all aspects of fuel conversion, thus, the OH* signals should not be viewed as linearly proportional to the reaction rate.

A detailed sketch of the fuel nozzle is shown in Fig. 3. A central 1.27-cm stainless-steel pilot tube supplies premixed fuel to the pilot flame that is on the nozzle axis. The pilot flame is ignited with an internal spark electrode ~5.7 cm upstream of the nozzle exit. For the results presented here, the pilot and ignition spark are used only during startup. The pilot tube is surrounded by the lean-premix (LPM) fuel and air. The LPM fuel and air are blended 100 cm upstream of the nozzle in a concentric tube premixer to ensure thorough mixing. The premixed gases are split into two matched tubes and enter the fuel nozzle through two diametrically opposed ports 21 cm upstream of the nozzle exit. The premixed fuel and air pass through a wire mesh flow straightener and then through a straight-vaned swirler angled at 45 deg to the nozzle axis.

Figure 4 shows the flow loops used to supply the combustor. The fuel supply is of particular interest because of the pulse-generating hardware. After the fuel is metered, the flow branches to valve A and valve B. If the solenoid valve is closed and valve B is open, gas will fill the accumulator volume V until the pressure in the accumulator equals the pressure upstream of valve A. If the solenoid valve is open, the accumulator volume will vent to the pressure downstream of valve A. This will produce a momentary increase in the fuel flow to the premixer. Depending on the setting of valves A and B, it is possible to pulse the solenoid so that a periodic variation in fuel flow is produced without changing its time-average value. We mention that the fuel modulation could, in principle, be produced without the accumulator. However, the accumulator serves to reduce the size of pressure swings that accompany valve opening and closing. In the limit of a very large accumulator, the valve action would produce little change in the instantaneous pressure (although the time-average pressure would decay). This has the benefit of producing a relatively constant pressure drop (and, hence, flow rate) across the solenoid during the open cycle, and it provides little disruption to the upstream flow metering. In addition, the accumulator provides a convenient method to measure the variable flow rate. The magnitude of the fuel flow variation can be estimated from the dynamic pressure transducer, DP-1, as described next.

A control-volume equation for mass conservation in the fuel V is written as the difference between mass flow in and out of the accumulator:

$$\frac{d}{dt}(\rho V) = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \tag{1}$$

Assuming that the pressure in the accumulator follows isentropic compression/expansion, we can relate the pressure P and density ρ in the accumulator as:

$$P\rho^{-\gamma} = \text{const} \tag{2}$$

where γ is the specific heat ratio. Combining Eqs. (1) and (2), and rearranging, we can express the net mass flow into the accumulator as:

$$\frac{V}{c_0^2}\frac{dP}{dt} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \tag{3}$$

In this equation, c_0 is the ideal gas speed of sound at the ambient temperature. Ignoring the small volume of the fuel stored in the connecting lines, the fuel flow [Eq. (3)] can be subtracted from the time-average metered flow to calculate the variation in fuel flow as a function of measured pressure from transducer DP-1. Note we do not need the individual terms on the right side of Eq. (3) to calculate the net change in fuel flow rate. The difference between the inlet and outlet mass flow represents the fuel that is stored in the accumulator, and therefore, not sent to the combustor. Thus, to use Eq. (3), we simply record the pressure signal from DP-1, and calculate the net flow into the accumulator from the slope of the pressure history. This approach was used to estimate the equivalence-ratio shifts described later.

The fuel control solenoid was designed for natural gas reciprocating engines, and could open (or close) in 3 ms. The actual time needed to change the equivalence ratio was much longer because gas stored in the stagnant regions of the tubing and fuel nozzle must be completely purged for every control cycle. Fuel/air premixing was achieved in the annulus formed from two concentric tubes (6.4-mm-annular i.d., 11-mm-annular o.d., 300-mm length). Fuel was injected through 12 holes (1.6-mm diameter) from the inner tube into the airflow moving through the surrounding annulus, and mixed along the length of the premixer. For tests conducted here, the nominal air velocity in the narrow annular gap was 100 m/s. This high gas velocity meant that changes in fuel flow produced abrupt, well-defined changes in the equivalence ratio at the exit of the premixer. However, the stagnant region at the left end of the premix nozzle (Fig. 3) introduces some mixing between the subsequent equivalence ratios. Thus, fuel pulses generated in the premixer are smeared by the stagnant flow regions in the nozzle. If we treat the stagnant region as a stirred-tank, the output fuel concentration will respond to an input step-change according to $[1 - \exp(-t/\tau)]$, where t is the time measured from the step-change, and τ is the residence time in the tank.[26] Using the flow rates and geometry where control was tested, this residence time was ~7 ms. Thus, to achieve 90% of the desired step-change in equivalence ratio requires $(1 - 0.9) = [1 - \exp(-t/7)]$, or $t = 16$ ms. Adding the valve actuation time (3 ms), and assuming the flame can adjust to the new equivalence ratio in one residence time (3 ms at the tested control conditions), the estimated time required for changing the equivalence ratio is 22 ms. As seen later, the measured transition time was ~25 ms. This time could be shortened by eliminating the stagnant regions in the fuel nozzle, but this was unneeded for the present experiment.

Control tests were carried out by operating the combustor at unstable conditions, and then activating the fuel control solenoid. The added effective area of the cycling solenoid would increase the time average flow, but the flow metering valve (Fig. 4) was set in automatic mode to compensate, and would re-establish the time average flow rate in a few seconds.

For the discussion that follows, we consider the definition of several parameters associated with equivalence-ratio modulation. Referring to Fig. 5a, we present the idealized time history of the equivalence ratio ϕ produced by the solenoid valve opening and closing. The solenoid modulation frequency (f_m) has an associated period T_m. The pulse width W_p is the fraction of the modulation period during which the solenoid is open. We identify three different fuel flow rates associated with three different equivalence ratios:

1) $\dot{m}_{\text{HIGH}}$ = fuel flow with the solenoid open, corresponding to ϕ_H.

2) $\dot{m}_{\text{LOW}}$ = fuel flow with the solenoid closed, corresponding to ϕ_L.

3) $\dot{m}_{\text{AV}}$ = time-average fuel flow, corresponding to ϕ_{AV}.

We can relate these quantities by integrating the instantaneous flow rate over one modulation period. The result is

$$\dot{m}_{\text{HIGH}} W_p + \dot{m}_{\text{LOW}}(1 - W_p) = \dot{m}_{\text{AV}} \tag{4}$$

Dividing by the airflow, normalizing with the stoichiometric fuel/air ratio, and rearranging, we have

$$\phi_H W_p + \phi_L(1 - W_p) = \phi_{\text{AV}} \tag{5}$$

This expression shows the relation between the average equivalence ratio, the high- and low-equivalence ratio, and the pulse width. As depicted in Figs. 5a–5c, the pulse width determines where the high- and low-equivalence ratio exist relative to the average. Short pulse widths (Fig. 5b) place the average near the low-equivalence ratio, and the opposite situation exists at long pulse widths (Fig. 5c). Thus, for a given time-average equivalence ratio, the choice of pulse width can produce an asymmetric modulation of the equivalence ratio.

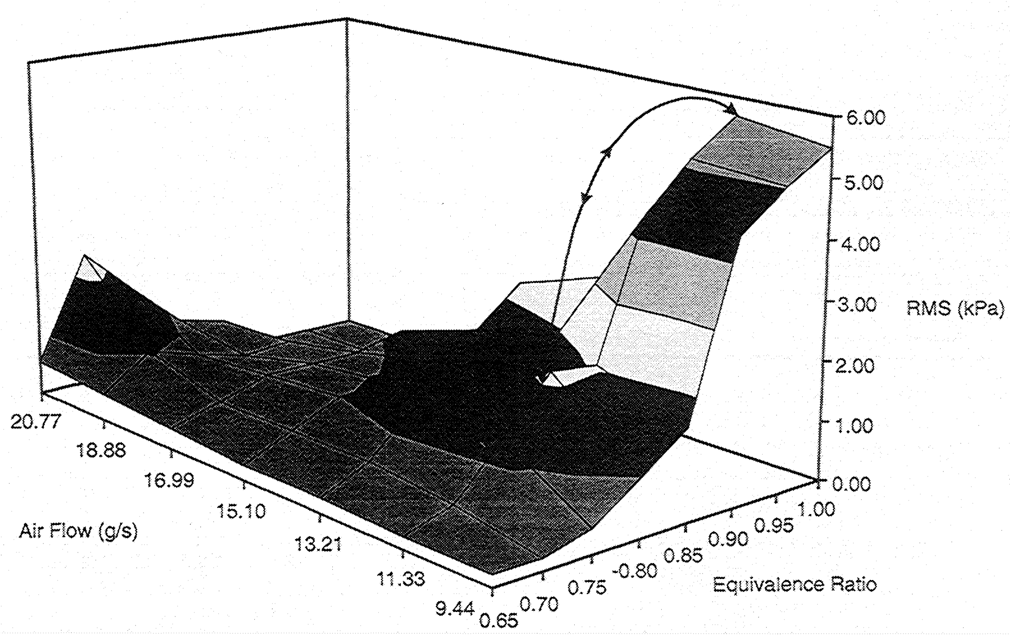

Fig. 6 Oscillating amplitude (rms) measured as a function of the combustor airflow and equivalence ratio, and showing the region of equivalence-ratio modulation.

Results

As described in the preceding text, the intent of this experiment was to test the control technique on an oscillation region which was isolated on a plot of airflow and equivalence ratio. It was necessary to find an isolated oscillation region to test the concept of modulating the equivalence ratio around the instability region. Toward this end, oscillating pressure data were gathered over a range of fuel and airflow rates. Figure 6 is a stability map showing rms pressure vs airflow and equivalence ratio. The frequency of oscillation for all test points was between 270 and 350 Hz, depending on the specific operating condition. A large oscillation region toward the right side of the map occupies approximately one-fourth of the operating map. A second, smaller unstable region is on the left side. Considering the unstable region on the right, the instability is not isolated; we cannot hop over the instability because there are not two stable equivalence ratios bounding the oscillation region. As shown by the curved arrow, we could alternate in and out of the unstable region but cannot avoid it completely.

Although the instability shown on the right side is not amenable to the proposed control method, it is instructive to consider what happens when we attempt to use fuel modulation to move the combustor periodically out of the unstable region. We next present results showing fuel modulation between one operating point that is stable, and one point that is unstable, as shown by the curved arrow. Figures 7a and 7b show plots of the pressure and OH* history for oscillating combustion with steady fuel flow (baseline) and for fuel modulation. The time history of the solenoid is shown below the pressure signal. Note that the time scale is in tenths of seconds; individual oscillations are not discernable at this time scale. The negative time axis was used for convenience in plotting. Two sets of data are shown (baseline and with modulation). These data are not a continuous time series through time zero, but represent conditions before and after modulation, plotted on adjoining portions of the time axis. As explained earlier, activation of control requires a few seconds for the flow metering to reestablish the same mean flow rate before and after control activation; this adjustment period is not shown. The control pulse frequency is 5 Hz, and the pulse width is 50%, i.e., $W_p = 0.5$ in Fig. 5. The data show a clear transition between stable and oscillating conditions as the fuel is modulated. This behavior is representative of all results associated with the instability on the right side of Fig. 6, including tests covering various combinations of airflow, equivalence ratio, control pulse frequency, and control pulse width. For all cases investigated, if fuel modulation merely alternated between a stable and unstable operating condition, the combustor pressure would cycle between oscillating and nearly stable combustion.

With fuel modulation, the OH* signal also alternates between strongly oscillating and low-amplitude noise. Note that the solenoid open and closure do not correspond to immediate changes in the combustion behavior; the response is delayed and requires several oscillation cycles to ramp up (or down) to a limit cycle (or stable combustion). We show this more clearly in Fig. 8, which is a small portion of the OH* data presented in Fig. 7. Note, in particular, that when the solenoid moves to the open position, i.e., high at ~0.3 s, the OH* reaches a new level ~0.025 s later. As explained in the experimental description, this is similar to the expected time required for the fuel nozzle and flame to be completely purged of the previous equivalence ratio. After achieving the new equivalence ratio, note also that an additional 0.04 s pass while the oscillation grows to a limit cycle with frequency of ~300 Hz. More than 10 oscillation cycles occur during the growth to a limit cycle oscillation. If the equivalence ratio could be shifted between two stable conditions, the modest growth rate suggests that the proposed control scheme (Fig. 1) would be effective, even for a relatively slow transition between stable conditions. This is important, because equivalence-ratio modulation requires the combustor to pass through an unstable operating condition. If the limit-cycle growth time is comparable to the time required to change the equivalence ratio, oscillations would emerge during the transition between stable conditions.

We were not able to hop over oscillating conditions shown on the right of Fig. 6 because the dominant oscillation was not an isolated instability. However, we recognized that we could transition across the smaller instability on the left of Fig. 6. Based on the results of other studies conducted in this combustor, we recognized that this small oscillation region could be enlarged by operating at a different inlet air temperature.[27] Figure 9 is a plot of the oscillating region over the same flow rate range as in Fig. 6, but with an inlet air temperature of 367 K (200°F). For reasons discussed by Janus et al.,[27] the increase in inlet air temperature has (almost) shifted the main instability off the map, but enlarged the oscillation shown on the right.

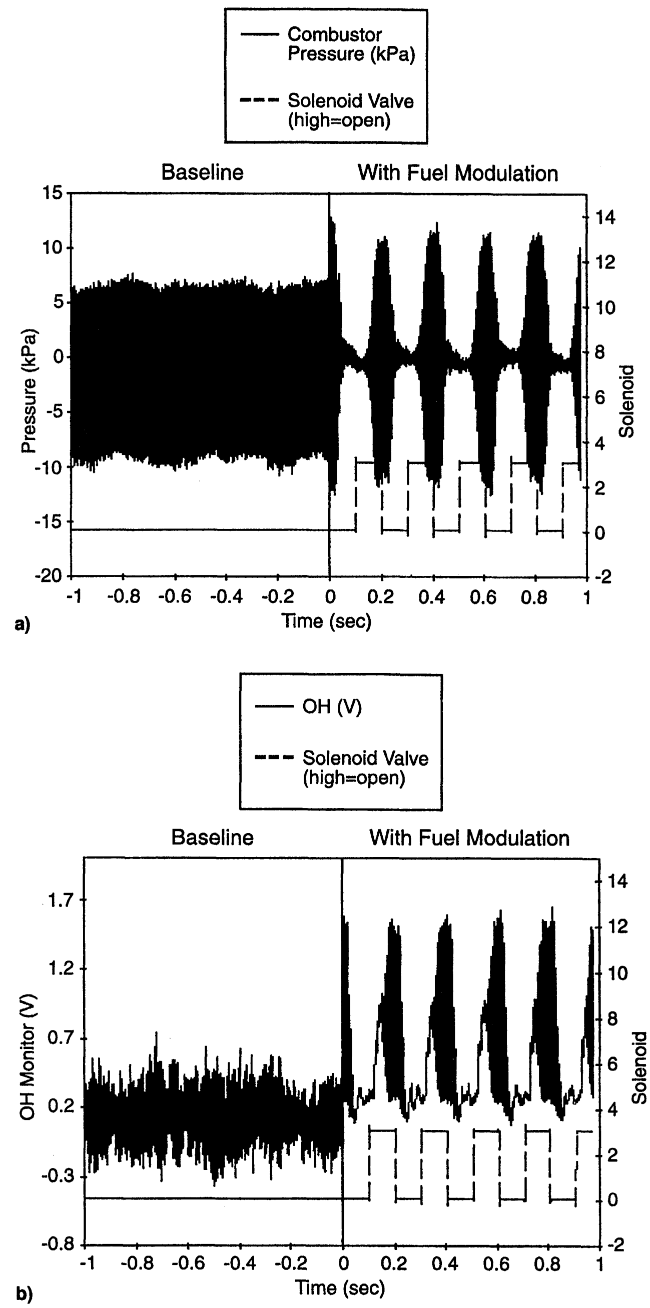

Fig. 7 Combustor pressure and OH* signals for steady fuel flow (baseline) and with fuel modulation (right, 5 Hz, 0.5 pulse width) as shown schematically in Fig. 6.

Notice that this oscillation region has the desired shape; stable regions bound the peak of the oscillations. This shape allows us to modulate the equivalence ratio around the oscillating conditions, and thus, provides a useful test case for the proposed control mechanism.

Figures 10a and 10b are plots of the pressure and OH* history corresponding to the time-average operating conditions indicated on Fig. 9. The baseline oscillation is shown along with fuel modulation. The solenoid position is again shown below the pressure history. The figures show that effective control is achieved. The rms pressure is reduced from 3.2 to 0.8 kPa, a reduction of 12 dB. As expected, the OH* signal also shows that the heat-release variation is reduced with fuel modulation. A closeup of the heat-release time history is presented in Figs. 11a and 11b. Note that the baseline case (Fig. 11a) shows a clearly organized variation in heat release at ~320 Hz. However, with fuel modulation (Fig. 11b), the variation in heat release includes the imposed modulation, high-frequency noise, and some small-amplitude cycles that reflect the baseline oscillation at 320 Hz (see time 0.38 s). Although the equivalence-ratio modulation would ideally avoid the unstable state, these data indicate that smaller-amplitude oscillations intermittently grow during the modulation cycle. As already explained, cyclic modulation of fuel does not produce an immediate, sharp transition between the stable equivalence ratios; indeed, the OH* signals in Figs. 10 and 11 demonstrate that the solenoid opening produces a brief overshoot in heat release as the accumulator supplies additional fuel. Subsequent transients in mixing and flow undoubtedly complicate the heat release, and may momentarily provide the conditions needed to begin the growth of oscillations.

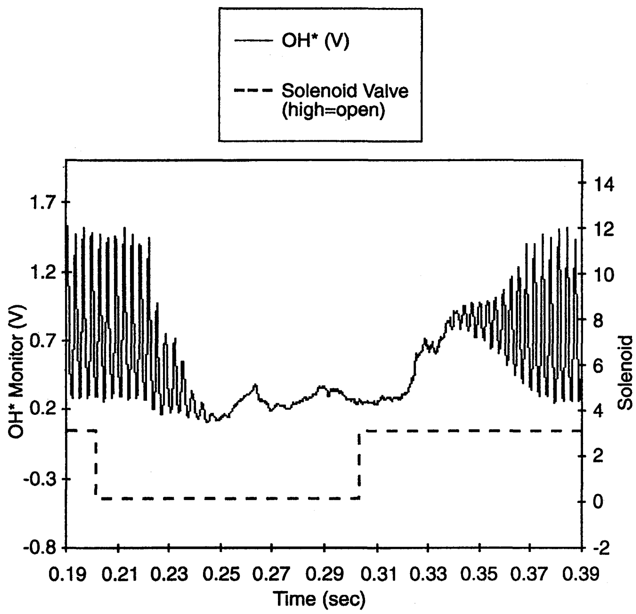

Fig. 8 Closeup of the OH* signal presented in Fig. 7.

As stated earlier, oscillation control requires that the oscillation growth time be significantly smaller than the time required for transition between equivalence ratio states. We point out that the time required for equivalence ratio transition is not simply a function of the solenoid open–close time, but also includes the time needed to bring the entire fuel nozzle and reaction region to the new equivalence ratio. This transition time sets an upper boundary on the frequency of fuel modulation. From Fig. 11b, we can estimate the transition time for this combustor by simply noting that the imposed equivalence ratio shift requires ~0.025 s for increasing or decreasing the heat release rate to a new value. Again, this is similar to the estimated transition time presented in the experiment description. Thus, a complete modulation cycle requires a minimum of 0.025 + 0.025 = 0.05 s, corresponding to a maximum modulation frequency of 1/0.05 = 20 Hz. For modulation frequencies above 20 Hz, we expect a smaller stabilizing effect because the actual change in flame conditions is reduced. In effect, the control action is smeared because of the finite time required to establish the stable equivalence ratios.

We demonstrated this limitation on the modulation control frequency by conducting tests over a range of modulation frequencies, and with a range of pulse widths. Figure 12 is a plot of the observed rms pressure for a range of pulse widths and modulation frequencies. The time average operating conditions are the same as those of Figs. 10 and 11; only the modulation parameters are changed. Focusing on the case of 50% pulse width, note that the rms pressure begins to increase for modulation frequencies above 20 Hz, as expected. For frequencies near 50 Hz, the effect of fuel modulation is very modest. This behavior is consistent with the observed time scales needed to restructure the flame at a new equivalence ratio (Fig. 11b). We emphasize that the frequency limitation is particular to this combustor/control valve assembly. Other combustor configurations or operation at different flow rates may require more or less time for the equivalence ratio transition; this would produce a different range of modulation frequencies for effective oscillation control.

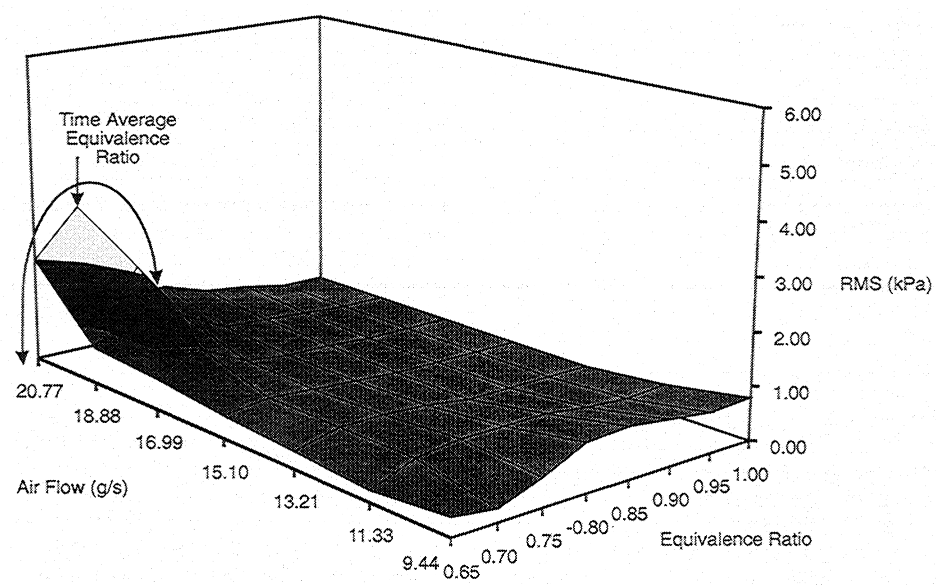

Fig. 9 Oscillating amplitude (rms) with an inlet air temperature of 367 K (200°F), and showing equivalence ratio modulation.

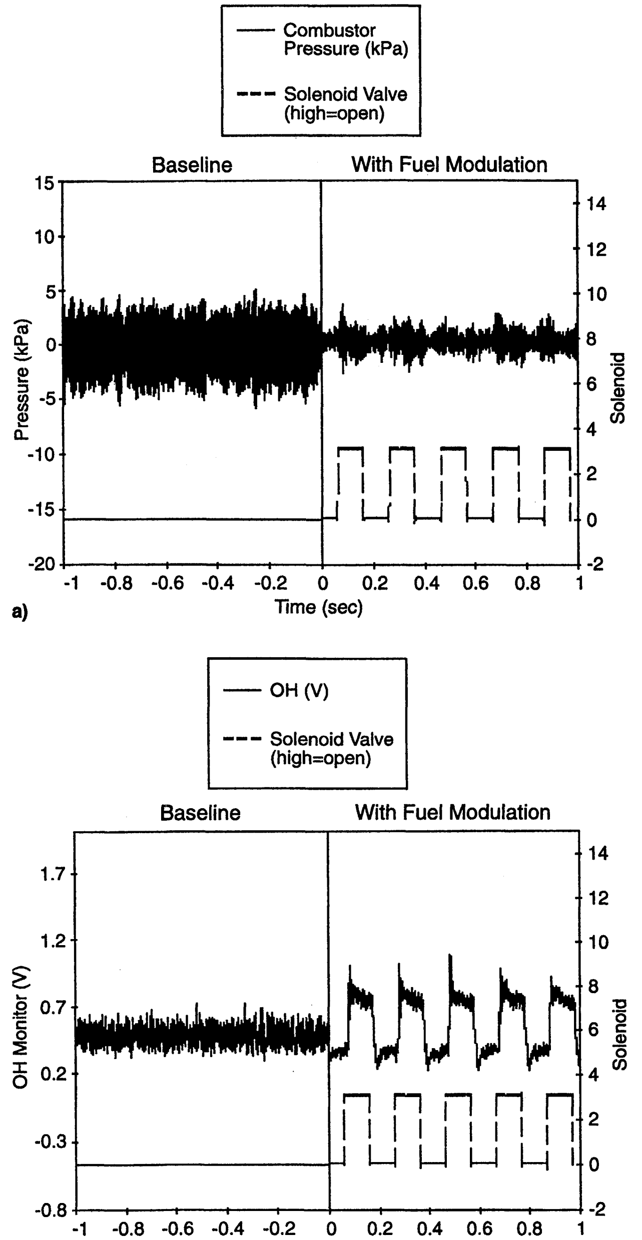

Fig. 10 Combustor pressure and OH* signals for steady fuel flow baseline (left) and with fuel modulation (right) corresponding to Figure 9.

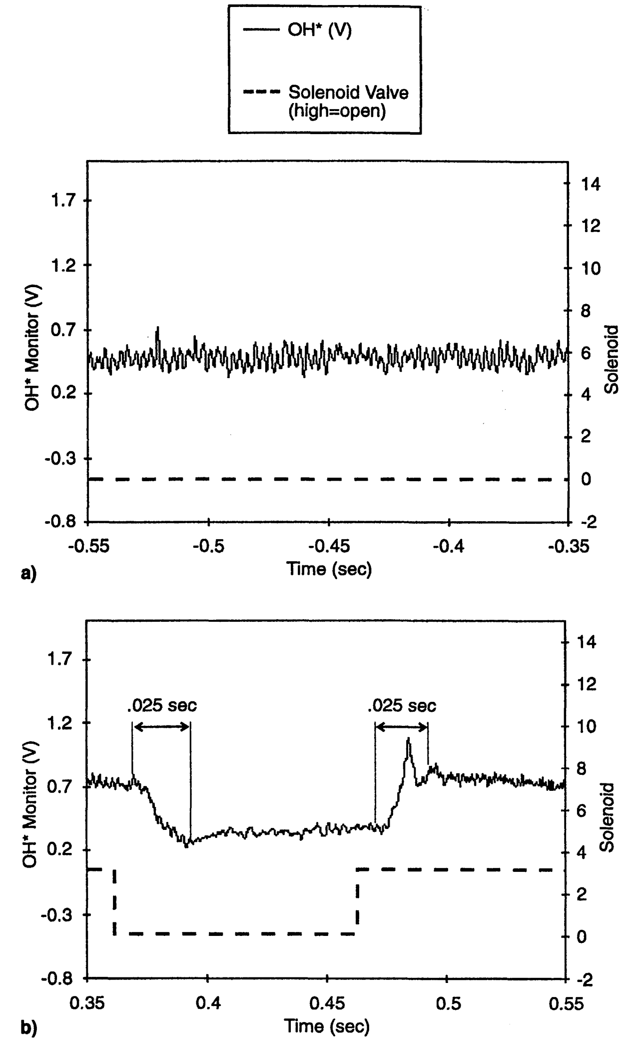

Fig. 11 Closeup of OH* signal for the steady and modulated fuel flow from Fig. 10.

Figure 12 also shows the effect of the pulse width. As explained earlier, pulse width values other than 50% will produce an asymmetric shift in equivalence ratio about the time-average value. For the operating conditions tested, the time-average equivalence ratio was approximately centered in the unstable region of the operating map (see Fig. 9). Thus, ideal control behavior should have occurred at 50% pulse width, as observed in Fig. 12. In the limiting case of very long or very short pulse widths, the solenoid operation produces a short blip in equivalence ratio; the combustor is otherwise operating very close to the (unstable) time-average equivalence ratio established by the fuel-metering valve (Fig. 4), so that fuel modulation has little effect.

Before concluding, we emphasize that the results presented here do not guarantee that an arbitrary instability can be controlled by equivalence ratio modulation. As described, the instability needs to be bounded by stable operating conditions on a plot of airflow and equivalence ratio. However, even when this requirement is met, it is possible that the instability growth time may be shorter than the equivalence ratio transition time. This situation would presumably allow oscillations to grow while the transition was occurring. For the case presented here, the measured heat release history showed that the relevant transition time and oscillation growth time allowed effective control at modulation frequencies up to 20 Hz, consistent with experimental observations. For other applications, it is possible to analytically estimate the transition time as explained earlier (see the experiment description). However, estimates of the growth time are quite involved. The growth time depends on the imbalance between the acoustic energy supplied by heat release fluctuations, and the acoustic losses. Evaluating the acoustic driving from the flame requires a precision model for the flame response, which is a subject of current research. Culick[28] presents an integral formula for the linear growth constant, but numeric evaluation requires assuming a model for the dynamic flame response. Darling et al.[29] present a numeric model to calculate the oscillation growth constant from the complex eigenvalues of the linearized conservation equations. Results from their model show that the combustor was stable or unstable, i.e., a positive or negative growth, depending on the choice of flame model. Because the calculation of growth rates is critically linked to specific flame models, no general approach to estimate the growth time is presented here. The approach of Darling et al., or Culick would need to be carried out for the specific case of interest, with a detailed model for the flame response. Where simplified flame models are used, uncertainty in the calculated results is likely. For

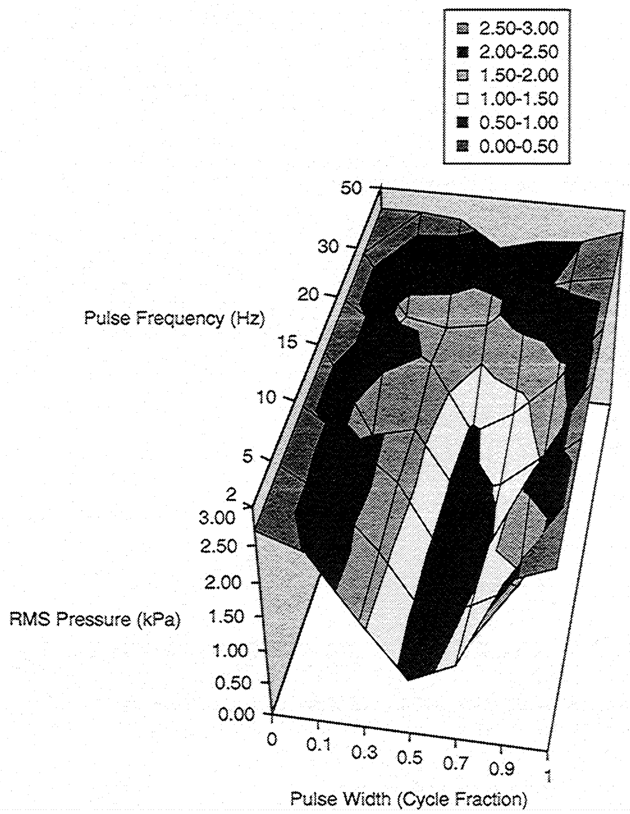

Fig. 12 Effect of modulation frequency and pulse width.

example, Lieuwen and Zinn[30] note that their stability analysis is able to correctly predict unstable operating conditions, but the growth rates are overpredicted because of assumptions about the flame model.

Another important issue not addressed in this paper is pollutant emissions. Pollutant measurements were not made on this lab-scale combustor, but it is reasonable to assume that emissions will be some average of those at the two equivalence ratios, such that some NO_x and CO penalty will occur for large equivalence ratio transitions. Preliminary tests (not reported here) have confirmed that emissions can be estimated from the time average of the two equivalence ratios. Future plans for large-scale tests of this concept will include emissions data. In some applications, such as thrust augmenters, emission levels may not be an important consideration.

Finally, we note that turbine engine application of this technique has another potential drawback: the heat release varies with the equivalence ratio modulation, imposing a cyclic load on the turbine. This is not as serious as it appears because most engines use multiple fuel injectors. Thus, several injectors can be modulated out-of-phase, avoiding low-frequency pulses in overall heat input.

Conclusions

This paper presents a novel open-loop control technique to reduce combustion oscillation. Essentially, the idea is to hop over unstable operating conditions by periodically modulating the fuel flow. In a time-average sense, the combustor operates at the unstable condition, but jumps between stable conditions on an instantaneous basis. The proposed method has been tested successfully on a subscale combustor. We demonstrate that the control technique is limited to a class of instabilities where an oscillating region is bounded by stable regions on a plot of air flow and equivalence ratio. Further, we show that the frequency of fuel modulation for successful control is determined by the time needed to transition between the stable equivalence ratios. For a modulation period shorter than the time needed for transition, the control effect is reduced. In addition, we noted that if the oscillation growth time is larger than the transition time, it is possible that oscillations can emerge during the transition between stable equivalence ratios. Although the transition time can be estimated from a simple analysis of the fuel nozzle, the oscillation growth time is difficult to estimate without a detailed model of the flame response. Evaluation of pollutant emissions during fuel modulation is planned for larger-scale testing.

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