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Effect of Fuel Nozzle Configuration on Premix Combustion Dynamics

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ABSTRACT

Combustion dynamics (or combustion oscillations) have emerged as a significant consideration in the development of low-emission gas turbines. To date, the effect of premix fuel nozzle geometry on combustion dynamics has not been well-documented. This paper presents experimental stability data from several different fuel nozzle geometries (i.e., changing the axial position of fuel injection in the premixer, and considering simultaneous injection from two axial positions). Tests are conducted in a can-style combustor designed specifically to study combustion dynamics. The operating pressure is fixed at 7.5 atmospheres and the inlet air temperature is fixed at 588K (600F). Tests are conducted with a nominal heat input of 1MWth (3MBTUH). Equivalence ratio and nozzle reference velocity are varied over the ranges typical of premix combustor design. The fuel is natural gas. Results show that observed dynamics can be understood from a time-lag model for oscillations, but the presence of multiple acoustic modes in this combustor makes it difficult to achieve stable combustion by simply re-locating the point of fuel injection. In contrast, reduced oscillating pressure amplitude was observed at most test conditions using simultaneous fuel injection from two axial positions.

INTRODUCTION

Lean premix (LPM) combustion has become a standard approach to reducing NOx emissions in stationary gas turbine engines. The advantage of premixing fuel and air upstream of the combustion process is widely recognized. LPM combustion avoids near-stoichiometric combustion and subsequent production of thermal NOx. Careful premixer design and combustor operation can now produce CO and NOx emissions routinely approaching ten parts-per-million concentration. For this reason, development programs from several turbine vendors include various versions of the LPM concept to achieve the lowest possible pollutant emissions (Layne and Hoffman, 1996).

Although lean premix combustion is an attractive alternative for controlling pollutant emissions, premix operational experience has been accompanied by both static and dynamic combustion instabilities. *Static instabilities* occur when operational upsets, or changes in fuel properties, produce unexpected changes in flame anchoring (e.g., flashback, flame extinction near the lean limit, etc.). Static instabilities are not discussed in this paper, although we point out that flame anchoring may be affected by dynamic instabilities. *Dynamic instabilities* occur when minor variations in the air/fuel ratio or mixing processes lead to significant changes in the combustor heat release rate. Subsequent coupling between the heat release rate, and the combustor's acoustic response can result in large pressure oscillations which can damage mechanical components in the turbine. The frequency of these oscillations can range from hundreds to thousands of Hertz, and the amplitudes can be on the order of ten percent of the combustor operating pressure. Keller (1995) discusses the significance of this problem and points out that operation near the lean limit is especially prone to dynamic instabilities.

Depending on the specific combustor application, a large body of literature exists on combustion dynamics. Excellent texts and review articles are available for rocket applications (Yang and Anderson 1995, Harrje and Reardon 1972), industrial burners (Putnam 1971), and afterburners (AGARD proceeding 1989). Miscellaneous review articles by Candel (1992), Schadow and Gutmark (1991), Oefelein and Yang (1993), and Culick (1994) provide further background. Until recently, gas turbine applications have received less attention because traditional diffusion-style combustors are less susceptible to oscillations than contemporary premix combustors. Some articles specific to gas turbine combustion are reported by Mehta et al (1990), Scalzone et al (1990), Shih et al (1996), and Darling et al (1997). None of these earlier articles provide a systematic study of the effect of fuel injector geometry on premix combustion dynamics.

Richards and Janus (1997) characterized the effect of gas turbine operating conditions on dynamic instability. These authors showed that oscillations observed at different operating pressures and inlet temperatures could be described by considering the time-lag required for fuel to move from the point of fuel injection to the flame front. In addition, the time-lag model successfully described changes in the stability boundaries for a modest change in fuel nozzle geometry (i.e., the time lag was increased by about 20%). As stated by Richards and Janus, multiple acoustic modes or changes in flame structure could complicate the time-lag description, but these complications were not observed in the previous work.

In this paper, we consider the effect of a significantly larger changes in fuel nozzle geometry. By moving the point of fuel injection along the

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axis of the premix nozzle, we change the time-lag by more than 130% at a given nozzle velocity. The data show that the time-lag model describes many aspects of the oscillating behavior, including transition between multiple oscillating frequency ranges. Because the combustor can oscillate in several frequency ranges, we demonstrate that it is difficult to achieve stable combustion by simply moving the point of fuel injection along the axis of the nozzle. However, as an alternative method to achieve stability, we also considered the effect of injecting fuel at two positions along the axis of the nozzle. This approach produced enhanced stability in most test cases, and may be a practical option to improve the stability of commercial premix combustors.

BACKGROUND

This paper builds on the work reported by Richards and Janus (1997), and the reader is encouraged to consult that paper for more details. As background to understand the data that follows, we repeat the discussion of the time lag model presented by Richards and Janus.

To explain the time lag model, we consider a hypothetical combustion instability where the combustor pressure fluctuations produce variations in the premixer air flow that are 180 degrees out of phase. Referring to Fig. 1, when the combustor pressure is high, the air flow in the premixer nozzle will be low. For a choked fuel supply, the fluctuations in air flow produce variations in the local fuel/air ratio at the fuel port. These locally rich and lean mixture pockets are transported to the flame with a time lag that can be estimated as the distance between the fuel port and the flame front, divided by the average nozzle velocity. Thus, the time lag, τ, is approximated as

$$Time\ \ Lag\ = \tau = \frac{(L+\acute{L})}{U_{avg}} \tag{1}$$

Subsequent cycles deliver rich and lean pockets of mixture to the flame. If these pockets are "timed" to release heat in-phase with the pressure, the pressure oscillations will grow in accordance with the Rayleigh criterion. In other words, if the richer mixture pockets arrive at the flame when the pressure is again high (i.e., 1, 2, 3, etc. periods later), the heat release, and pressure oscillations can couple to produce dynamic instabilities. The mathematical statement of this criterion is:

$$\text{(time lag)/(acoustic period)} = 1,2,3\ldots$$

or (2)

$$\text{(time lag)} * \text{(frequency)} = \tau f = 1,2,3\ldots$$

The integer values (1,2,3, etc.) are specific to the hypothetical case in which the pressure oscillations produce air flow variations that are 180 degrees out of phase with the pressure, and the fuel supply is choked. If the air flow does not vary 180 degrees out-of-phase (as is likely), or other instability mechanisms exist, the numeric series will not be described by integer values. Putnam (1971) describes various situations producing series analogous to Eq. 2, but having *fractional* values.

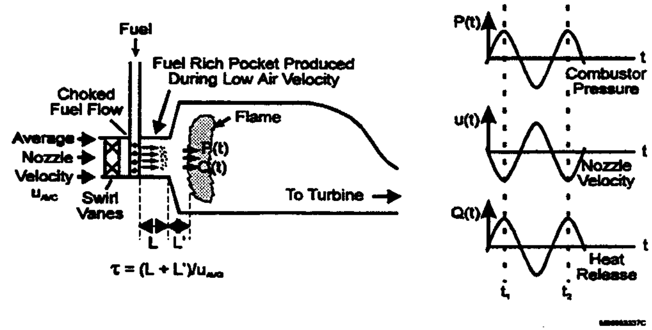

FIGURE 1: Schematic of processes occurring in an example oscillation. The air velocity, U(t), is 180 degrees out of phase with the combustor pressure, and produces a fuel-rich pocket at the fuel port at time, t_1. This pocket arrives at the flame to increase the heat release rate, Q(t) at time, t_2.

Whatever the mechanism, the point to note is that we expect a series of values for τf where oscillations will occur. For convenience we will refer to the numbers in these series as "oscillation indices". The product, τf, is a measure of the phase between the heat release and the pressure. When this product equals one of the oscillation indices, then the heat release and pressure will optimally drive oscillations. Some oscillations may occur for heat release that leads or lags the pressure by as much as one-quarter of the acoustic period (Putnam, 1971). Thus, oscillations may occur for a *range* of plus/minus 0.25 around each oscillation index, but are most significant when the product τf equals one of the oscillation indices. It is significant to note that the time lag determines only the phase between the pressure and the heat release; it does not determine the combustion gain or the acoustic losses. Thus, conditions which produce an oscillation index may or may not oscillate, depending on the gain or loss. However, conditions which produce τf values between the oscillation indices will be stable, regardless of the gain or loss.

Richards and Janus (1997) used a time lag criterion like Eq. 2 to evaluate oscillating data from an experimental gas turbine fuel nozzle. The data showed that appreciable oscillations occurred only for a range of τf values between 0.45 and 0.70 (see Fig. 2). Using the time lag model, it was possible to predict how the stability boundaries would move as the fuel port location was changed. As shown schematically in Fig. 2, increasing the distance, L, between the fuel port and the flame front, should move the stability boundaries to higher values of velocity. Thus, at a desired operating velocity (denoted by * in Fig. 2), stability can be achieved by simply changing the length, L. This concept was demonstrated experimentally by Richards and Janus (1997), and the resulting stability map followed the expected trend (i.e., the stability boundary moved to higher values of velocity).

The results of the previous work were limited to relatively small changes in premix nozzle geometry. As already mentioned, several complications to the time lag model may occur for larger changes in fuel nozzle geometry. In particular:

1) A simple interpretation of the time lag model assumes that a single combustor frequency is activated. In practice, it is necessary to

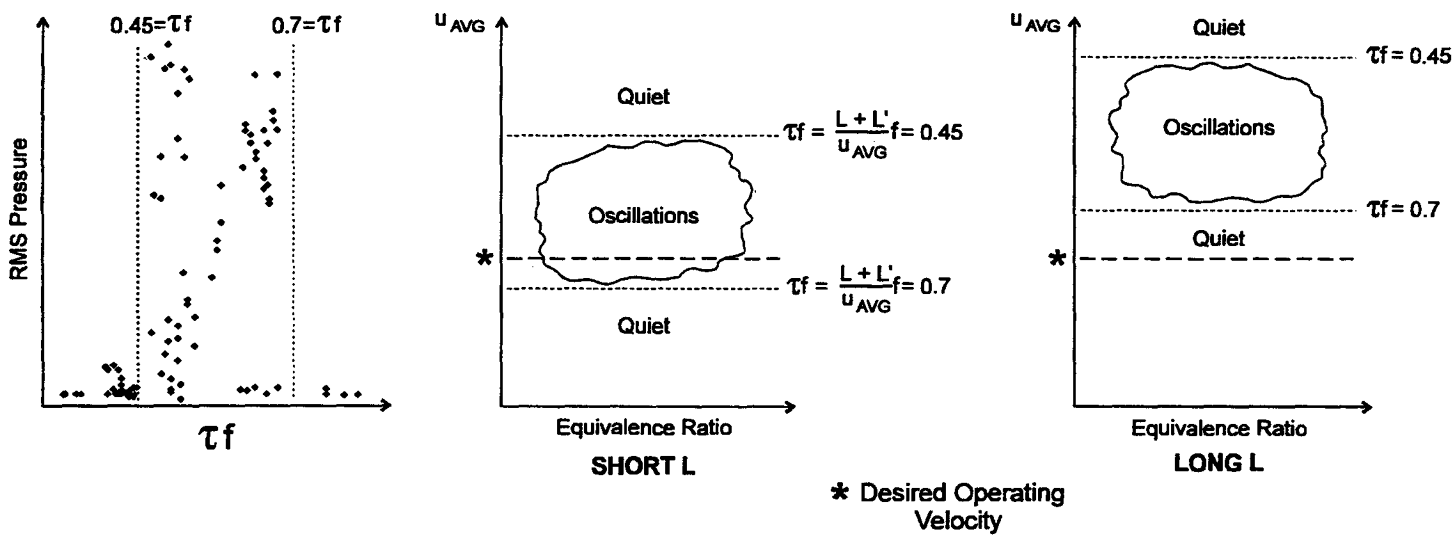

FIGURE 2: Experimental data from Richards and Janus (1997) showing stability boundaries at τf products between 0.45 and 0.7 (refer to Fig. 1 for the definition of L, L' and u_{avg}). Note that as L is increased, the stability boundaries translate to higher nozzle velocities. For a given operating velocity (denoted as "*"), stability can be achieved by increasing L.

account for multiple frequency ranges that are observed in practical combustors. As an example, if a combustor is characterized by two different frequencies f_1 and f_2, dynamic instabilities associated with a given oscillation index may occur at two different values of fuel transport time (i.e., τ_1 and τ_2) so long as the product $\tau_i f_i$ equals an oscillation index. Furthermore, very large changes in τ may produce dynamic instabilities at subsequent oscillation indices (e.g., moving from 1 to 2 to 3, in Eq. 2). Experimental examples of this behavior will be shown in this paper.

2) The flame standoff distance can also change abruptly, invalidating a simple calculation of time lag. Smith and Leonard (1997) showed that a single time-lag does not completely describe the time required for fuel to move from the fuel injector along different streamlines that arrive at the flame surface. Although these limitations are recognized, we assume a constant flame standoff of L' = 2.5 cm (1.0 in) for the purposes of this paper. In spite of these simplifying assumptions, the experimental data was successfully described by the time lag model.

Data presented in this paper are compared to the time lag model described above. Results show that as the fuel nozzle geometry is changed, multiple operating frequencies do indeed complicate the expected translation of stability boundaries as shown in Fig. 2. Changing the axial location of the fuel port either activates lower frequencies, or causes oscillations at different oscillation indices. As a result, dynamic instabilities are not readily silenced in this combustor by changing the location of the fuel port, at least over the range of locations discussed in this paper.

Although stability is not achieved by simply moving a single fuel port, results show that an alternative concept using simultaneous fuel injection from two axial locations along the premix nozzle is a promising approach to achieve stable combustion. The concept is shown schematically in Fig. 3. As pointed out by Keller (1995), injecting fuel along the axis of the nozzle should reduce the variation in air/fuel ratio produced by changes in nozzle velocity. According to Keller, variations in fuel/air ratio occurring at one fuel port can effectively cancel variations produced at the second fuel port, thus providing enhanced stability. Although the underlying mechanism is not well understood, our results confirm that significant reductions in RMS pressure levels can be achieved by injecting fuel at two axial locations verses single port injection.

EXPERIMENTAL DESCRIPTION

A very detailed description of the experimental hardware used in these tests is provided in Richards, Gemmen, and Yip (1997), and Richards and Janus (1997). Information on data acquisition, facility capabilities, and operating approach can be found in these earlier papers. Thus, we outline only the major aspects of the experimental hardware.

A cutaway view of the experimental combustor rig is shown in Fig. 4. The inlet plenum diffuses the preheated combustion air prior to entering the premixer nozzle. The inlet plenum also contains a series of perforated plates to dissipate large scale turbulence and provide a more uniform inlet velocity for the premixer nozzle.

The combustion zone is enclosed by a water cooled liner with 19.8 cm (7.80 inch) inside diameter. A characteristic time analysis, similar to the one presented by Narayanswami and Richards (1996), was used to show that the effect of liner water cooling on dynamics (i.e., heat loss) should be small for a test device of this size, operating at elevated

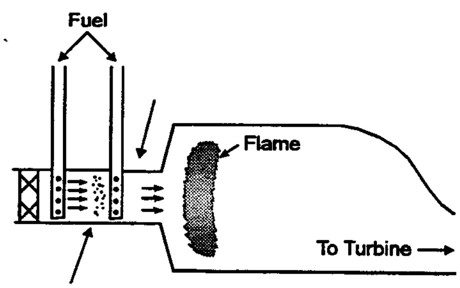

FIGURE 3: Schematic of simultaneous fuel injection from two ports located along the premix nozzle axis.

pressure. In contrast, local heat loss at critical flame anchoring points may influence flame dynamics. In smaller test devices, overall heat loss can significantly influence dynamics (McIntosh and Rylands, 1996). However, in this large test rig, the water-cooled liner is an advantage because it provides significant acoustic feedback with very small acoustic losses. A cylindrical plug rests along the inside diameter of the water cooled liner and forms an exhaust "neck" that approximates a classic Helmholtz resonator. This water cooled plug[1] has an inside diameter of 10.4 cm (4.10 inch) and is 22.9 cm (9.00 inch) long. Immediately downstream of the water cooled plug insert, the exhaust gases are quenched with a water spray prior to exiting the pressure vessel. A pressure control valve in the exhaust line is used to adjust the rig pressure.

The premix nozzle consists of a series of rings that are stacked on three threaded rods. This modular design allows the nozzle configuration to be modified quickly by stacking the spacers and other nozzle components differently. The fuel nozzle configurations tested in this work are shown in Fig. 5. Although there are only two different geometric layouts, the choice of injecting fuel from one or both fuel rings produces a total of five different configurations. That is to say, fuel is injected from the axial positions labeled A, B, or C, individually; and fuel is injected from A&B or A&C, simultaneously. When the fuel is injected simultaneously through two fuel ports, the fuel flow is split evenly between the ports. The effect of varying the fuel flows to each of the fuel ports is not investigated in this paper.

Pilot fuel is injected along the axis of the nozzle center body, and is used during startup only. Previous work has shown that under certain conditions, the presence of a pilot flame can drive combustion oscillations. Therefore, in an effort to simplify the operating strategy and focus on understanding the dynamic instabilities, equivalence ratios of less than 0.59 are not investigated. Noting that this premix nozzle is not optimized to provide uniform mixing, typical NOx concentrations range from 20-60 ppm (corrected to 15% O_2) at an equivalence ratio of 0.57. Lovett and Mick (1995) discuss the NOx and CO emissions from a similar premix nozzle operating at equivalence ratios of less than 0.6.

Premixed fuel is introduced into the nozzle through eight radially oriented spokes. Each of these spokes have six 0.066 cm (0.022 inch) diameter holes which inject fuel tangentially into the nozzle at three different radial locations. These holes are sized to insure that the fuel flow is choked during most operating conditions. Although practical fuel nozzles are typically not choked, it is useful to study dynamic instability mechanisms which isolate the effects of the fuel system acoustics. However, the role of the fuel system acoustics can be investigated in other tests.

Miscellaneous experimental details are summarized below. More detailed information is available in the references noted above. The nominal gas composition during these tests varies from 92-93% CH_4, 4-5% C_2H_6, 1.1-1.2% C_3H_8 and 1-2% of inerts and higher hydrocarbons. The fuel and air flows are metered using standard orifice runs, and the accuracy of these flow loops are within $\pm$3% when compared to flow standards. Dynamic pressure measurements are recorded with an externally mounted pressure transducer using the infinite-coil technique described by Mahan and Karchmer (1991). Mahan and Karchmer have shown that this infinite-coil technique produces a flat frequency response to about 1kHz, but at 2.5kHz the pressure signal may be attenuated by as much as -8dB, or approximately forty percent. It should be noted that oscillating frequencies as high as 2.5 kHz have been observed on this test rig, and are discussed in the following section.

TEST METHOD

Because the effects of inlet air temperature and operating pressure are reported by Richards and Janus (1997), tests are conducted at a fixed pressure of 7.5 atmospheres, and an inlet air temperature of 588K (600F). For the various nozzle configurations, data is gathered over a range of air flows (i.e., average nozzle velocities from 30 to 60 m/s) and equivalence ratios from 0.59 to 0.77. As discussed previously, operation at lower equivalence ratios was not investigated.

Before evaluating the effect of fuel injection configuration, we compare data from the same injector configuration, but recorded on different days. The intent is to demonstrate the variance typical in these types of dynamic instability tests. Figure 6 is a plot of observed RMS pressure and oscillating frequency for tests conducted on consecutive days. To simplify data comparison, the RMS data is shown both as a stability map (upper figures), and as a series of traditional x-y plots. The RMS pressure is reported as a percentage of the average combustor pressure. The data for Test 1 and Test 2 are gathered in reverse order, and the results are shown for nozzle reference velocities of 30, 40, 50 and 60 m/s. Data at 60 m/s could not be obtained for the two highest equivalence ratios (0.71 and 0.77) due to limits on the fuel supply pressure. These points are marked with an 'x' on the stability map.

Except for the very weak oscillation at 60 m/s (discussed below), the observed frequencies are very repeatable in subsequent tests. The RMS pressures show reasonable qualitative agreement between Test 1

[1]Richards and Janus (1997) used a refractory "plug". This was changed to provide longer plug lifetime, and did not produce any notable changes in experiment operation.

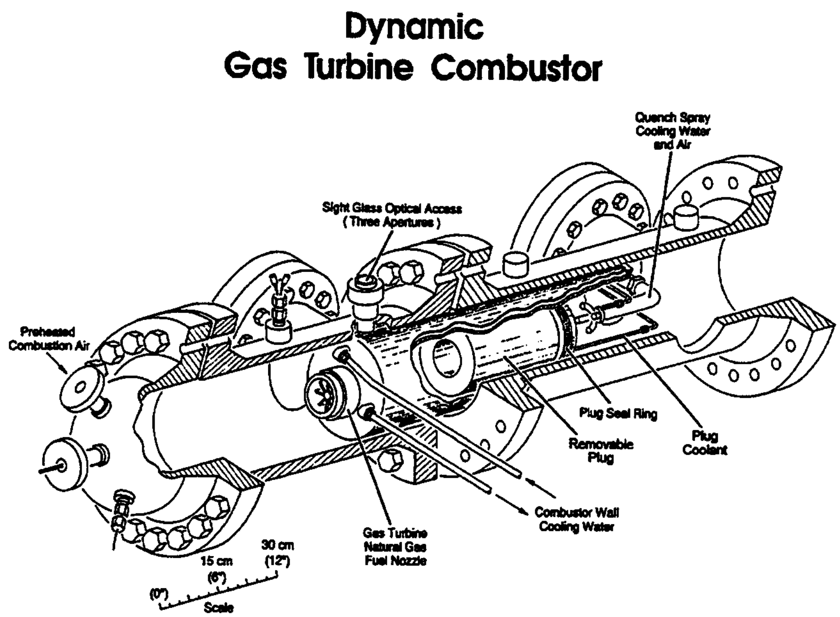

FIGURE 4: Cutaway view of FETC combustor.

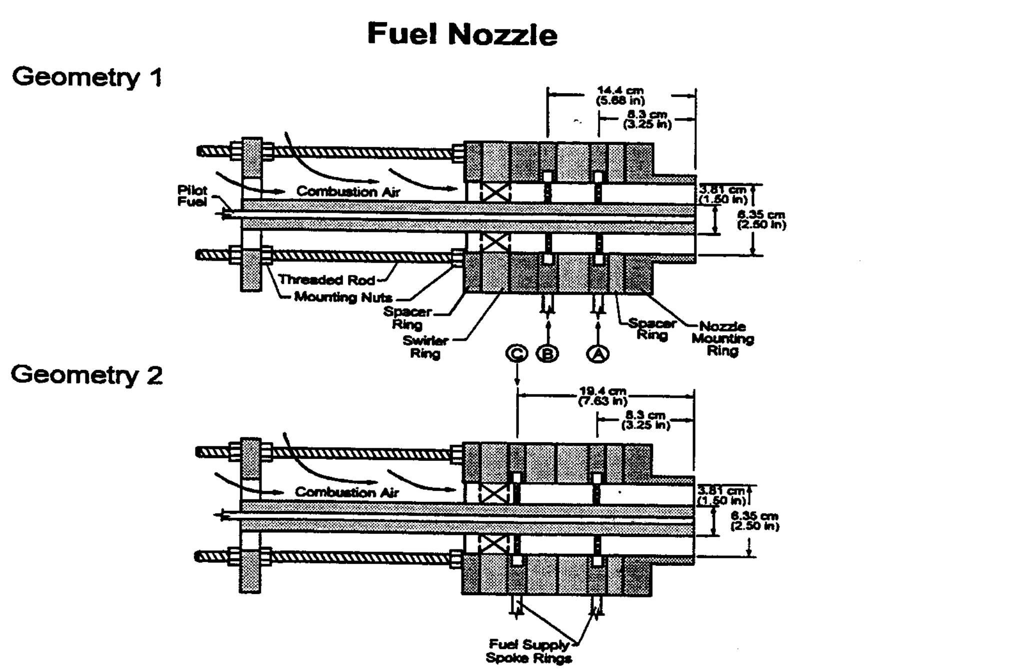

FIGURE 5: Schematic of fuel nozzle layouts. Fuel can be injected at points A, B, or C, or in combinations A&B and A&C.

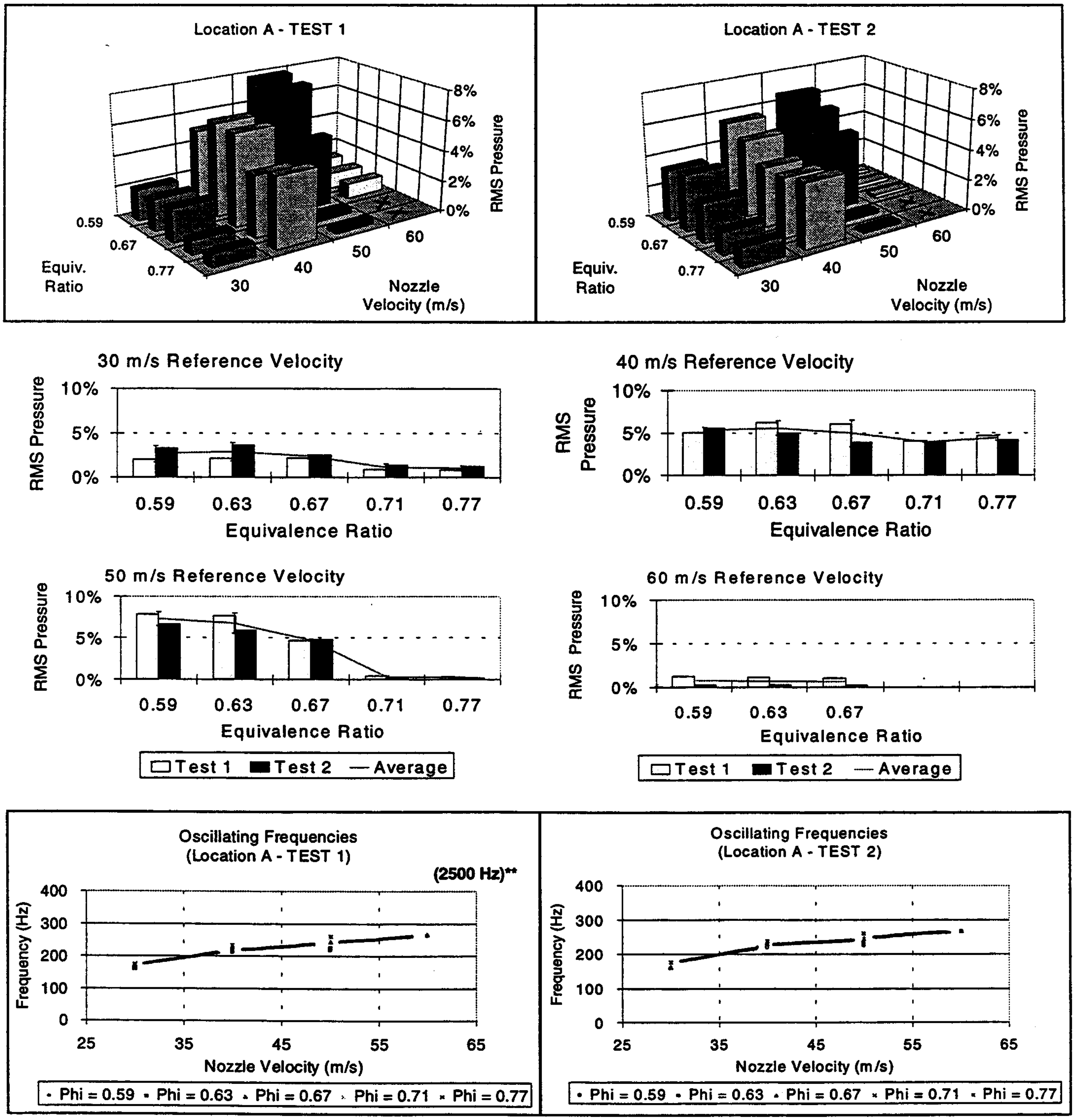

FIGURE 6: Data comparison from two consecutive tests with the same fuel nozzle configuration (Case A). Points marked with asterisks were weak high frequency (2500 Hz) oscillations.

and Test 2, but the quantitative values are appreciably different at select test points. This variability is not directly attributed to failure to replicate test conditions, since flow rates, inlet air temperature, and rig pressure are all controlled to within 3% of target values. Instead, as noted by Richards and Janus (1997), some test points produce RMS pressures that vary over a long period of time (minutes), and hysteresis effects are observed at some operating conditions. In spite of these handicaps, the effects of changing fuel injector geometry are typically large enough that the variability observed in Figure 6 does not complicate the data interpretation. As shown later, the excellent frequency repeatability allows a clear comparison to the time lag model, even if there is some variability in the RMS pressure.

An interesting exception to the frequency repeatability is the 60 m/s case shown in Fig. 6. Note that the frequency data for Test 1 produces a weak 2500Hz oscillation at two conditions. (These points are noted with an asterisk to avoid re-scaling the y-axis). This case is anomalous and the RMS pressure is very low (less than 1% of operating pressure). However, we note that the actual pressure levels may be significantly higher than the measured value, due to the frequency response limitations of the infinite-coil technique, as previously discussed. This high frequency is observed at only two test points, but it has been encountered on this combustor in earlier tests, using a different fuel nozzle (Richards et. al., 1997). These earlier tests confirmed that oscillations are accompanied by high-frequency variations in heat release (i.e., they were not merely acoustic tones). It is not certain that frequencies as high as 2500Hz can be driven by fuel transport mechanisms similar to that shown in Fig. 1, because axial mixing processes could "smear" fuel pockets moving along the nozzle annulus. Because these high frequency oscillations are weak, and observed at only a few test conditions, no definite conclusion on the origin or driving mechanism, for these signals is discussed. We simply note that Test 1 and Test 2 conditions are conducted in opposite order, so it may be possible that these high frequency oscillations are an example of hysteresis between different acoustic frequencies. We denote these anomalous test points with an asterisk (*) in subsequent data presentation.

A clear example of RMS pressure hysteresis, instead of frequency hysteresis, is observed for nozzle configuration C. Figure 7 shows the RMS pressure levels for a nozzle velocity of 50 m/s while carefully decreasing, and then increasing the equivalence ratio. The RMS pressure exhibits a clear hysteresis, depending on the operating history. A detailed investigation of these hysteresis effects is not discussed. However, we have observed in subsequent testing that these hysteresis effects are much less severe when the pilot tube is water cooled. These observations suggest that the heat stored in the structure of the premix nozzle (i.e., pilot tube tip), or other structures in the combustor has a significant effect on these hysteresis events. It should be noted that the pilot tube was not cooled in any of the tests reported in this paper. These observations have been made in tests that will not be reported in this paper.

For the purposes of this paper, we note that hysteresis effects may complicate evaluation of test data, and should be carefully considered. As explained above, interpretation of the test data for the fuel injector geometries studied in this paper is not handicapped by either hysteresis,

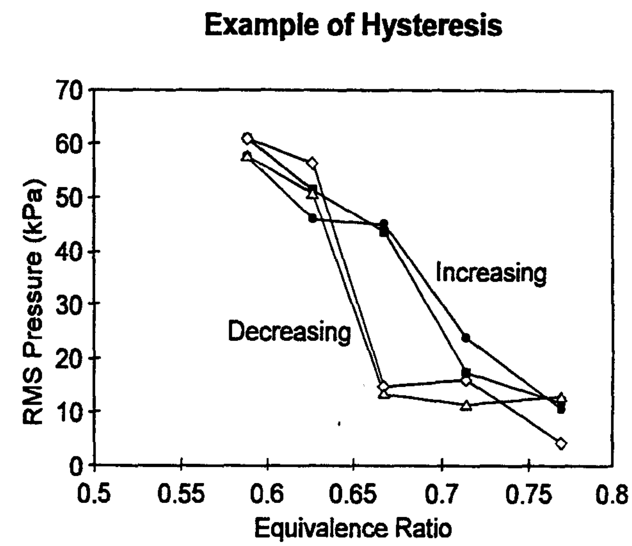

FIGURE 7: Example of Hysteresis. Nozzle configuration C, average nozzle velocity was 50 m/s.

or by the observed variability in RMS pressure. However, we caution this conclusion may not be true in general.

COMPARISON OF FUEL INJECTOR GEOMETRY EFFECTS

The purpose of this work is to investigate the effect of fuel injector location on dynamic stability for a lean premixed combustor. As shown in Fig. 5, we investigate two different premix nozzle geometries, each of which has two identical fuel supply rings. This layout provides the choice of injecting fuel from one, or both, fuel rings producing a total of five different configurations. Fuel is injected from the axial positions labeled A, B, or C, individually; and fuel is injected from A&B or A&C, simultaneously. We will first discuss the results from the individual fuel injector locations.

Single Injector Results

Figure 8 shows the RMS pressure levels and dominant frequencies observed for each of the individual fuel port configurations. Note that the data for Case A represent the average from two tests conducted on different days. These data show that changes to the fuel injection location will produce changes in the RMS pressure at a given operating point. In the discussion that follows, we show that this behavior can be rationalized by a time lag model, but cannot be predicted apriori without a knowledge of the oscillating frequencies.

In Case A, we note that a stability boundary is evident between 50 and 60 m/s. As noted earlier, data at the two highest equivalence ratios at 60 m/s could not be reached due to limits on the fuel supply pressure, and are marked with an 'x' on the stability map. Following the time lag model, if we increase the length from the fuel port to the exit of the premix nozzle, the stability boundary should move to a higher velocity

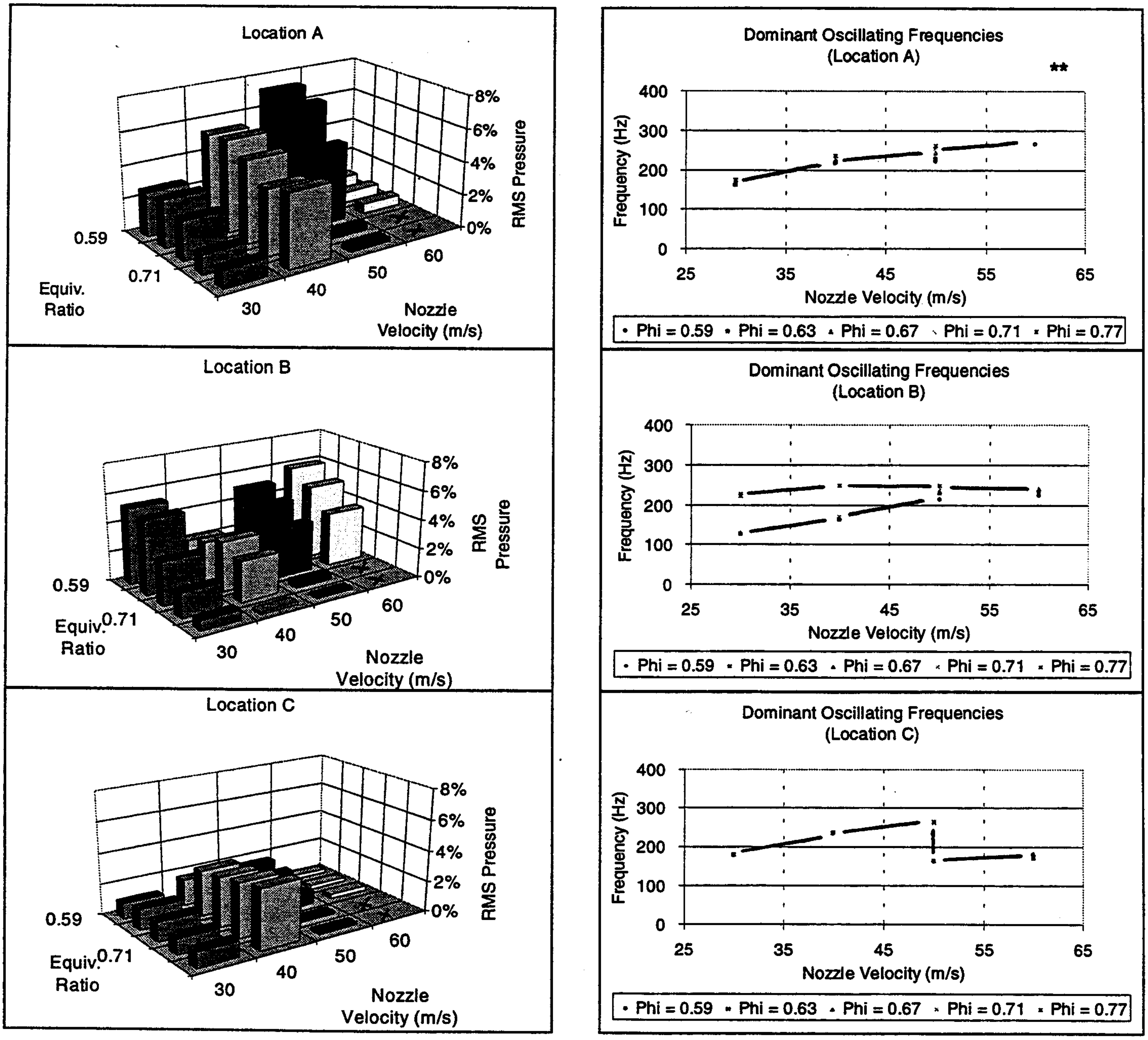

FIGURE 8: Observed RMS pressure and oscillating frequencv for nozzle configurations A, B, and C. The frequency data includes the two anomalous high-frequency cases, denoted by **.

(Fig. 2). Comparing Cases A and B at 50 and 60 m/s, the stability boundary is moved to a higher velocity for Case B as expected. Note that the frequency for Case A at 50 m/s is the same as the frequency for Case B at 60 m/s. In other words, the frequency moves with the stability boundary, and does not change.

We next compare Cases A and B at 30 m/s. From the discussion surrounding Fig. 2, we might expect that a stable region would appear at 30 m/s for Case B. Instead, depending on the equivalence ratio, the combustor oscillates strongly. For the strongly oscillating cases, the frequency has dropped from approximately 175 Hz to 125 Hz. Thus, rather than producing the stable combustion shown in Fig. 2, Case B produces oscillations at a lower frequency. Note also that the highest equivalence ratio (0.77) produces a weak oscillation at a higher frequency for the 30 and 40 m/s cases.

Continuing the comparison, we consider Cases B and C at 30 m/s. Remarkably, for all but the high equivalence ratios, the frequency is now larger for Case C than Case B, and the RMS pressure is lower. For Case C, higher nozzle velocities produce higher frequencies up to 50 m/s, where the observed (weak) oscillations occur at one of two frequencies, depending on the equivalence ratio.

What is the explanation for these observed changes in frequency and RMS pressure? First, we note that the oscillating frequency is strongly dependent on the nozzle velocity, and less dependent on the equivalence ratio. The only exceptions are Case B at the highest equivalence ratio, and Case C at 50 m/s; and neither of these cases exhibit strong oscillations. A further implication of this observation is that the oscillating frequency is largely controlled by the time lag, as expected. However, the time lag alone does not determine the magnitude of the oscillation. For significant oscillations to occur, two conditions are necessary. First, the operating configuration must produce one of the oscillation indices in Eq. 2. Secondly, oscillations must occur at a frequency where the combustor produces significant acoustic feedback.

To further understand this data, Fig. 9 presents all the test data, for configurations A, B, and C, plotting both the RMS versus observed frequency, as well as the RMS versus the time lag* frequency, or the product τf. Ignoring two anomalous data points at 2.5 kHz, significant oscillations occur around three frequency bands at approximately 125, 175 and 225 Hz. These are the experimentally determined frequencies where this combustor produces significant acoustic response. Remarkably, when the same data are plotted against time-lag*frequency (τf), the data collapse to just two banded regions, even though τ has varied from 1.8 ms to 7.3 ms. The implication is that the oscillating frequency adjusts to changes in τ to remain within the τf regions shown in Fig. 9. These regions are the experimentally determined oscillation indices analogous to Eq. 2. The oscillating frequency changes to keep the oscillation near an oscillation index, therefore, as the nozzle velocity is increased, the oscillating frequency increases, consistent with the smaller time lag. However, significant oscillations occur only when the oscillating frequency matches a frequency where the combustor produces appreciable acoustic response.

A very clear example of this behavior was noticed for Case C. Both frequency and " time-lag * frequency" (i.e., τf) data for all equivalence ratios are plotted in Figure 10 as a function of nozzle velocity. Note the oscillating frequency increases with the nozzle velocity from 30 to 40 m/s, but the value of τf is constant at 1.3. Then, a further increase in velocity cannot access a higher frequency with appreciable acoustic feedback. Instead, the operating frequency drops to a lower value, and the value of τf also changes. We note that near this transition region, the amplitude of the combustion oscillations are relatively small (i.e., RMS levels were less than 0.2% of the operating pressure).

The preceding data demonstrate that the time-lag interpretation is helpful for understanding observed data. Attempts to move the combustor into the stable portion of a stability map are hampered by multiple frequencies, or a transition to a different stability indices as discussed in Eq. 2. Although these effects are understandable, they are hardly desirable, and a more practical question is what can be done to stabilize this type of fuel nozzle?

Two Injector Results

The results from the single fuel port configurations showed that the RMS pressure levels change when the point of fuel injection is varied, however, without knowledge of the expected oscillating frequencies, the results cannot be predicted apriori. Although a simple time lag model correlates the experimental data well, using this model to predict movement of stability boundaries can be complicated by the presence of multiple combustor acoustic modes, and transition between the oscillating indices in Eq. 2.

Realizing these limitations, a different approach is taken to control the dynamic instabilities. This approach involves injecting fuel at two axial positions along the premix nozzle. Keller (1995) notes that variations in air/fuel mixture may be reduced by injecting the fuel along the axis of the fuel nozzle. By injecting fuel at two axial locations, it is possible that variations occurring at the upstream fuel port could effectively be canceled at the downstream fuel port, as shown schematically in Fig. 3. With the correct spacing between fuel ports, a pocket of richer fuel/air mixture produced at the upstream port can arrive at the downstream fuel port just as a leaner pocket is produced. The resulting combination should cancel the variation in fuel/air ratio which would otherwise arrive at the flame front and drive combustion oscillations.

In principle, the desired spacing between the fuel ports could be established by noting that cancellation would occur when the time for transport between ports is exactly 1/2 of the acoustic period. However, based on data already presented, the combustor oscillating frequency (i.e., the acoustic period) depends on a number of factors, making it difficult to propose a fuel port spacing capable of producing effective cancellation over the range of operating conditions. Thus, rather than investigate a single fuel port spacing, two different fuel port spacings are investigated as shown in Fig. 5.

The stability maps for the individual fuel ports and both fuel ports are shown in Fig. 11. Note that the stability maps for the individual fuel ports are the same as those shown in Fig. 8, but have been presented together in Fig. 11 to allow direct comparisons between various nozzle configurations. Figure 11 shows that lower RMS pressure levels are indeed observed for a wide range of operating conditions when fuel is injected from two different fuel ports simultaneously. Dramatic reductions in RMS pressure levels (i.e., in excess of 90 percent) were observed at some operating conditions.

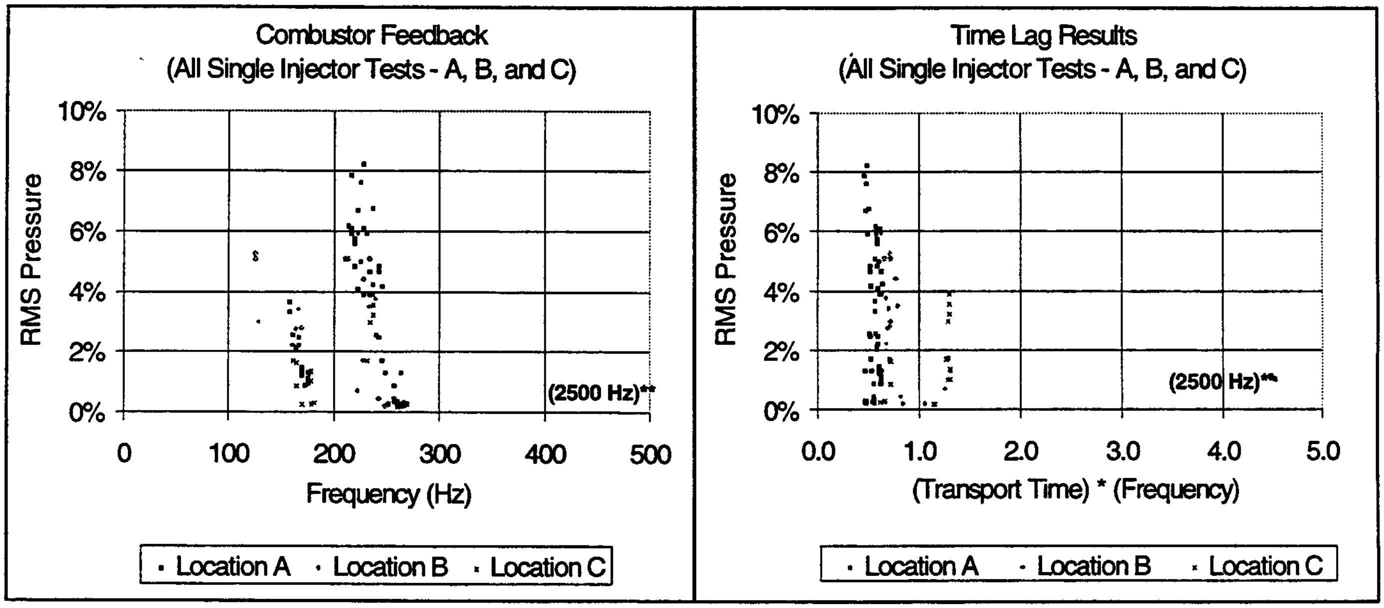

FIGURE 9: RMS pressure versus oscillating frequency (left) and time lag plot (right). All operating points studied for Cases A, B, and C are shown. High frequency (2500 Hz) cases are shown as asterisks.

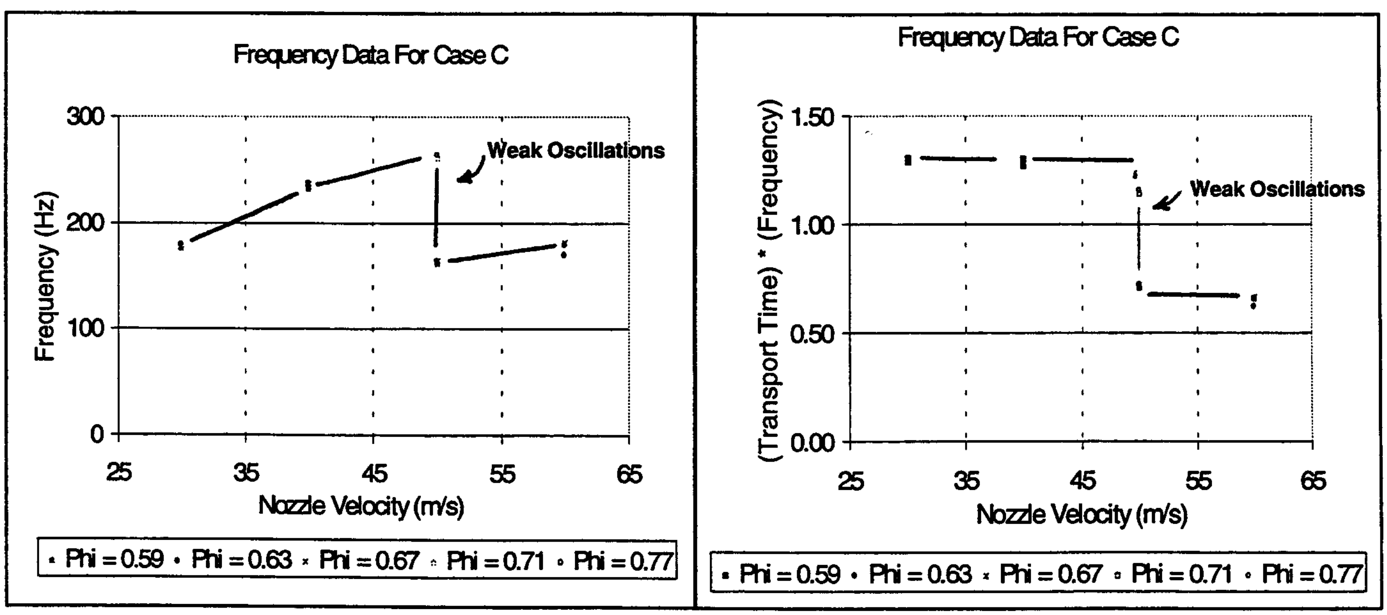

FIGURE 10: Oscillating frequency (left) and time lag (right) as a function of nozzle reference velocity for Case C. All equivalence ratios are shown. Note that weak oscillations are observed near the abrupt transition at 50 m/s.

To provide further insight, the data in Fig. 12 provide direct comparison of both the RMS pressure levels and the dominant frequency modes at individual operating conditions. As in Fig. 11, the RMS pressure data clearly show that both fuel ports work together to achieve low RMS pressure levels in all of the operating conditions, except one (40 m/s, equivalence ratio 0.77). At this same operating velocity, note that the combined injection from locations A&B was effective at reducing the RMS pressure for equivalence ratios 0.59 and 0.63, and yet the dominant frequency for two port injection is different. The fact that significant reduction in the RMS pressure levels is observed at two different frequencies may indicate that cancellation of the fuel/air variation between upstream and downstream ports is not alone responsible for the reduced RMS pressure. Inspecting the frequency data at all the operating velocities, the individual fuel ports (A versus B) typically oscillate at different frequencies. Thus, when combining ports, it is not clear whether the fuel/air variations are canceled, as proposed by Keller, or whether each fuel port competes for different acoustic modes with the net result being a reduction in RMS pressure. Either scenario is speculative, and we suggest that further work is needed to identify the exact mechanism responsible for improved stability from the dual-port injection.

It is also important to note that a high frequency signal is observed for the combined fuel port case in Geometry 1 at 60 m/s and an equivalence ratio of 0.63. This frequency is around 2.5 kHz, and is denoted in Fig. 12 with an asterisk to avoid changing the frequency scale. As discussed previously, the mechanism for driving these high frequency oscillations is not well understood, and is beyond the scope of this paper.

For completeness, Fig. 13 shows the RMS pressure levels and the dominant frequency modes observed for all velocities and all fuel port combinations from Geometry 2. These data show that fuel injection from two fuel ports result in somewhat less dramatic reductions in RMS pressure levels than in Figure 12. However, when both fuel ports were operating together the RMS pressure levels were typically smaller than the RMS pressure levels observed for the single fuel port data. A notable exception occurred at 30 m/s and equivalence ratio of 0.77. In this case, the dual-port injection produced a larger oscillation than either single-port result. We also note that the frequency increased when both ports were active. This was the only test case (out of 38 different test conditions) where the dual-port injection produced a higher RMS pressure than both of the single-port cases. Therefore, dual-port injection will not provide a universal reduction in RMS pressure, but it may be a practical alternative to improve combustion stability for many cases. However, further work is needed to clarify the exact mechanism of stabilization, and provide a rational design approach for enhanced stability.

In closing the discussion of both the single- and dual-port fuel injection, we mention that this paper has not investigated the role of the axial swirler location, and the overall length of the fuel injector. Both of these were constant for the results presented here. It is possible that both lengths could play a role in determining the dynamic response of the air flowing through the nozzle, and would be expected to change the oscillation indices that were observed (Fig. 9). Work is now in progress to define the role of these parameters. For practical studies of fuel injector geometry effects, we therefore suggest that test data be recorded, if possible, by altering the point of fuel injection, L, as an independent parameter, without the confounding effect of changes in overall nozzle length, swirler position, etc.

SUMMARY

This paper presents a study of the effect of fuel injector geometry on dynamics produced by a premix fuel nozzle. The RMS pressure and oscillating frequency were recorded as a function of the axial point of fuel injection, over a range of air velocities from 30 to 60 m/s, and from equivalence ratios of 0.59 to 0.77. Tests were conducted at a combustor pressure of 7.5 atmospheres, and an inlet air temperature of 588 K (600F). The fuel was natural gas.

Preliminary data showed that the measured RMS pressure exhibited hysteresis effects as a function of equivalence ratio at select operating points. A few anomalous conditions produced weak oscillations at very high frequencies (2500 Hz). In spite of these anomalies, most of the data could be readily interpreted using a time-lag model.

Based on earlier work, it was proposed that stable combustion could be achieved by simply re-locating the point of axial fuel injection. This expectation was based on a time-lag model for oscillations that considered just a single oscillating frequency. Test results showed that although the time lag model is valuable to interpret the observed data, the existence of multiple acoustic modes makes it difficult to achieve stable combustion by simply re-locating the point of fuel injection in this combustor.

As an alternative approach to achieve stable combustion, the concept of injecting fuel from two axial fuel ports was investigated, as initially proposed by Keller (1995). Indeed, lower RMS pressure levels were observed at most operating conditions when dual-port fuel injection was used. However, it is not clear whether the fuel/air variations are canceled, as proposed by Keller, or whether each fuel port competes for different acoustic feedback modes in the combustor. In any event, the net result was a reduction in RMS pressure levels. Again, further work is needed to confirm the mechanisms responsible for the observed reduction in RMS pressure levels from dual-port fuel injection.

In closing, we again emphasize that this paper has considered just the effect of changing the axial location of fuel injection. Additional work is planned considering the overall fuel nozzle length, and swirler position.

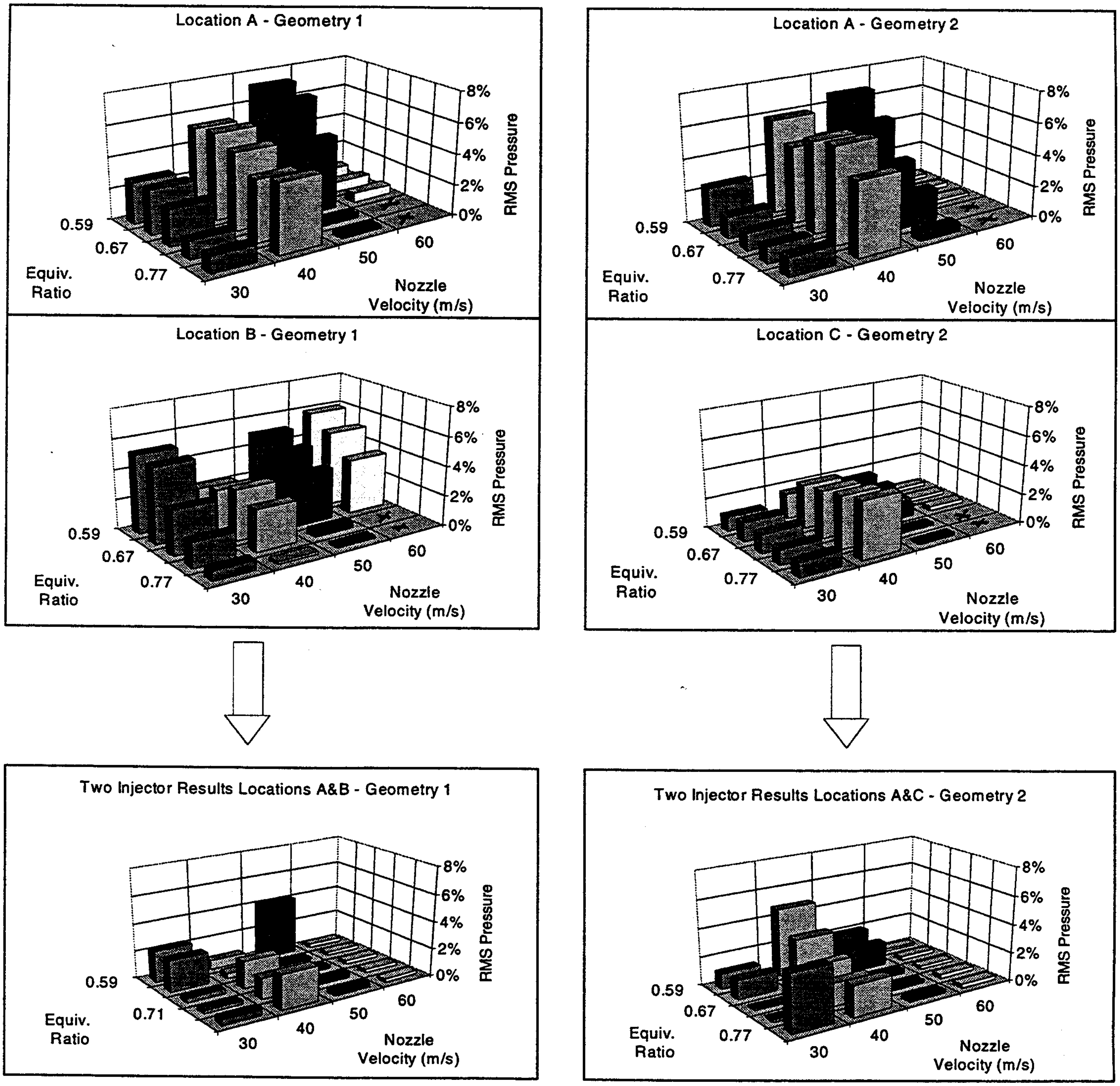

FIGURE 11: Stability maps for individual and simultaneous fuel port injection, Geometry 1 and Geometry 2.

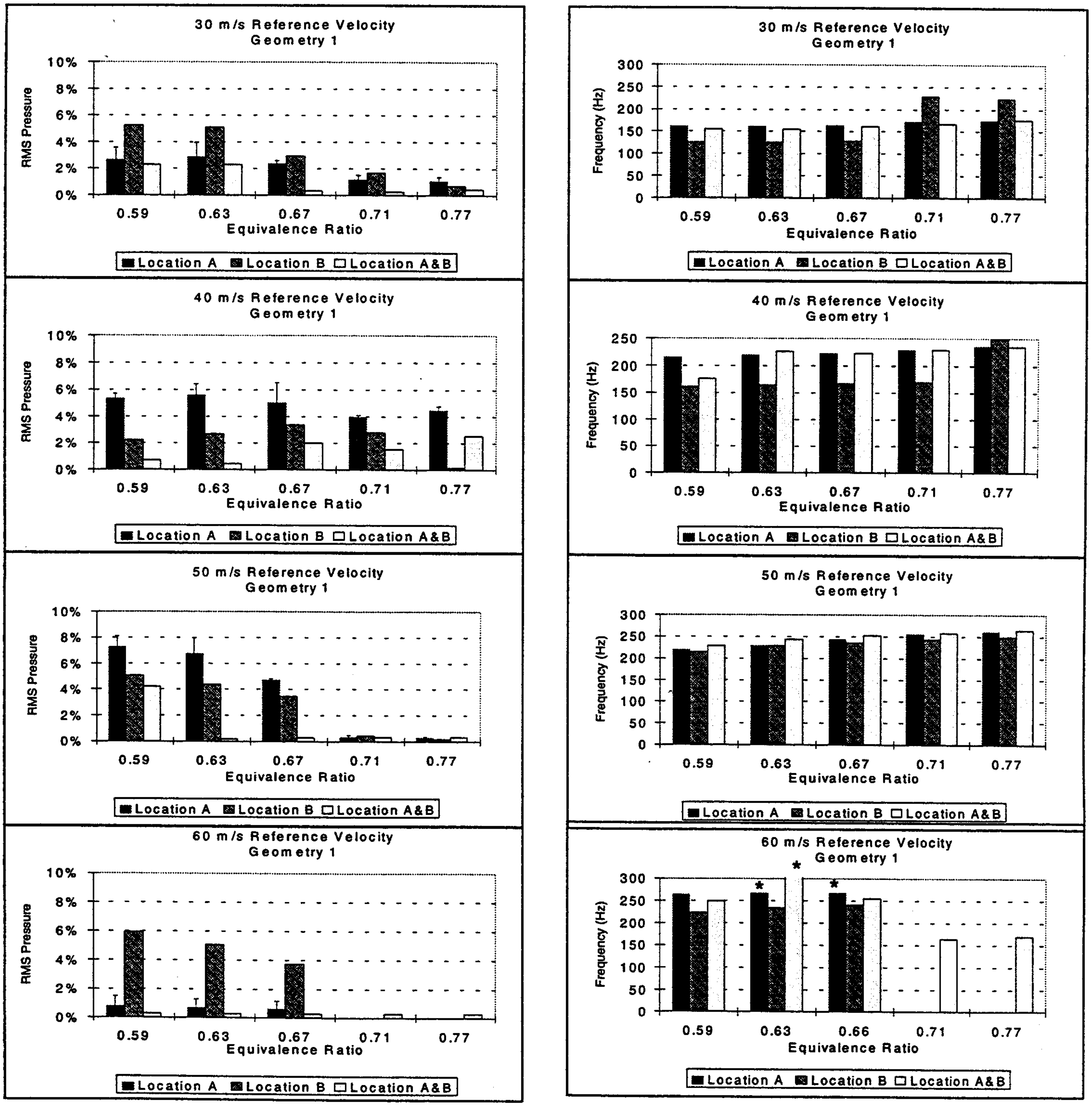

FIGURE 12: RMS pressure and dominant frequency modes observed for all fuel port configurations in Geometry 1. The asterisks (*) denote the high frequency observations. Note that a high frequency mode was observed for dual port fuel injection.

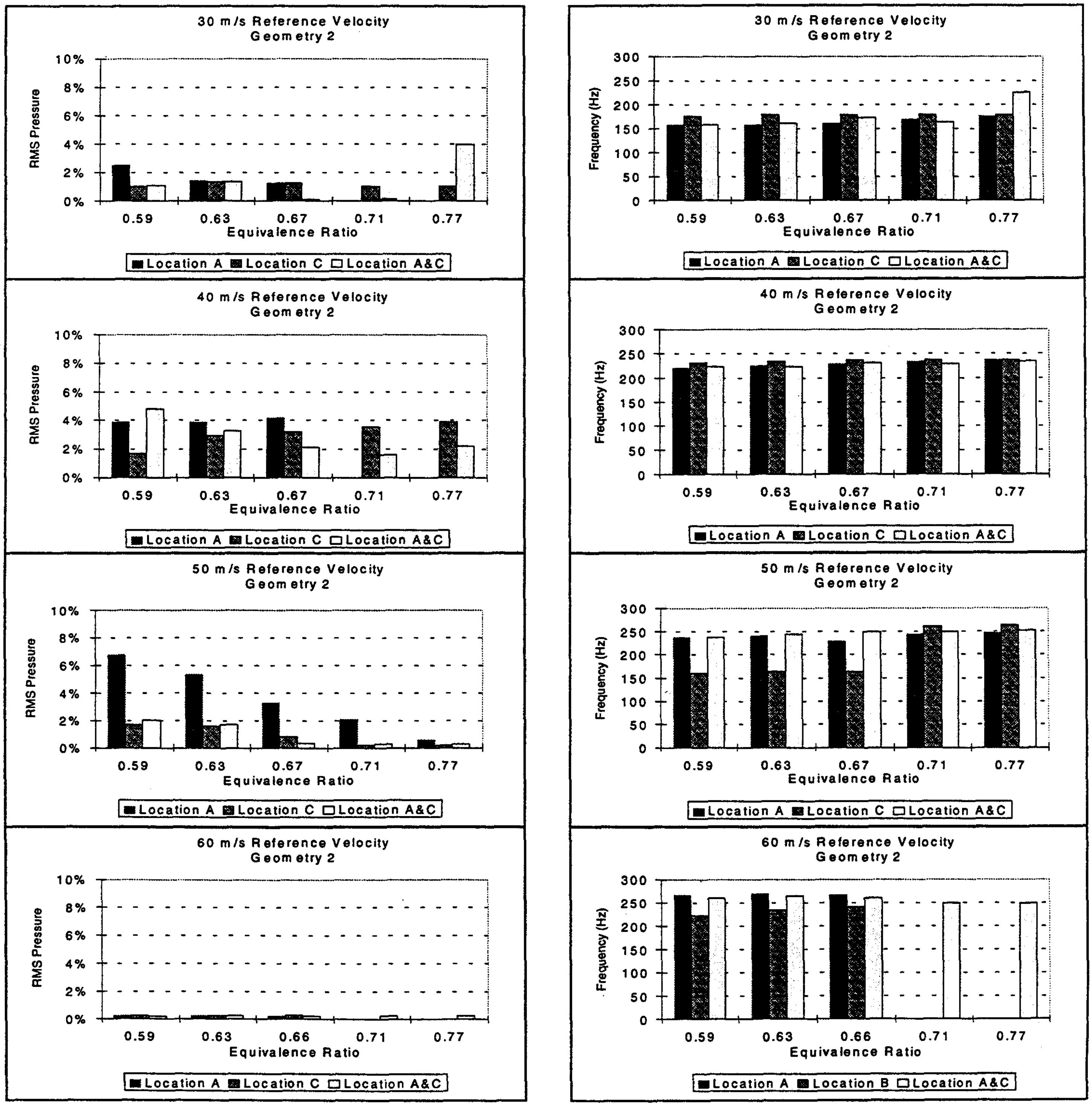

FIGURE 13: RMS pressure and dominant frequency modes observed for all fuel port configurations in Geometry 2.

REFERENCES

AGARD Conference Proceeding 1989, Combustion Instabilities in Liquid Fueled Propulsion Systems, AGARD-CP-450, ISBN 92-835-0503-4. Proceedings of the Propulsion and Energetics Panel 72nd B Specialists Meeting, Bath, UK, 6-7 October, 1988.

Candel, S. M., 1992, Combustion Instabilities Coupled by Pressure Waves and Their Active Control. *The Twenty-Fourth International Symposium on Combustion,* Pittsburgh, PA: The Combustion Institute, pp. 1277 - 1269.

Culick, F. E. C., 1994, Some Recent Results for Nonlinear Acoustics in Combustion Chambers. AIAA Journal, Vol. 32, No. 1, pp 146 - 169.

Darling, D., Radhakrishnan, K., and Oyediran, A., 1997, Combustion Noise at Elevated Pressures in a Liquid-Fueled Premix Combustor, ASME paper 97-GT-308, Presented at the 1997 ASME Turbo-Expo Meeting, June 2-5, Orlando Florida.

Harrje, D. T., and Reardon, F. H., 1972, *Liquid Propellant Rocket Combustion Instability*, NASA SP-194.

Keller, J. J, 1995, Thermoacoustic Oscillations in Combustion Chambers of Gas Turbines. *AIAA Journal,* Vol. 33, No. 12, pp. 2280-2287.

Layne, A., and Hoffman, P., 1996, *Proceedings of the Advanced Turbine Systems Annual Program Review Meeting,* DOE/OR-2048. Available National Technical Information Service, Springfield, VA.

Lovett, J. A., and Mick, W. J., 1995, Development of a Swirl and Bluff-Body Stabilized Burner For Low-Nox, Lean-Premixed Combustion, ASME paper 95-GT-166, Presented at the International Gas Turbine and Aeroengine Congress and Exposition

Mahan, J. R., and Karchmer, A., 1991, Combustion Core Noise, in *Aeroacoustics of Flight Vehicles: Theroy and Practice. Volume 1: Noise Sources, NASA Langley Research Center,* Available from AIAA Technical Library.

McIntosh, A.C. and Rylands, S., 1996, A Model of Heat Transfer in Rijke Tube Burners, *Combustion Sci. Tech.,* Vols. 113-114, pp 273-289.

Mehta, J. M., Mungur, P., Dodds, W., and Bahr, D. , 1990, Thermoacoustics of Unsteady Combustion, AIAA-90-3928, Presented at the AIAA 13th Aeroacoustics Conference, October 22-24, 1990, Tallahassee, FL.

Oefelein, J. C., and Yang, V., 1993, Comprehensive Review of Liquid-Propellant Combustion Instabilities in F-1 Engines, Vol. 9, No. 5, pp. 657 - 677.

Putnam, A. A., 1971. *Combustion Driven Oscillations in Industry*, American Elsevier Publishers, New York, NY.

Richards, G. A., Gemmen, R. S., and Yip, M. J., 1997, A Test Device for Premixed Combustion Oscillations, ASME Journal of Engineering for Gas Turbines and Power, To appear, October 1997.

Richards, G. A., and Yip, M., J., 1995, Oscillating Combustion from a Premix Fuel Nozzle. Presented at the Combustion Institute/American Flame Research Committee Meeting, April 23-26, San Antonio, TX.

Richards, G. A. , and Janus, M. C., 1997, Characterization of Oscillations during Premix Gas Turbine Combustion, ASME Paper No. 97-GT-244, Presented at the 1997 ASME Turbo-Expo Meeting, June 2-5, Orlando Florida.

Scalzo, A. J., Sharkey, W. T., and Emmerling, W. C., 1990, Solution of Combustor Noise in a Coal Gasification Cogeneration Application of 100-MW-Class Combustion Turbines, ASME Journal of Engineering for Gas Turbines and Power, Vol. 112, Jan. 1990, pp. 38-43.

Shih, W.-P., Lee, J., and Santivicca, D. A., 1996, Stability and Emissions Characteristics of a Lean Premix Gas Turbine Combustor, *The Twenty-Fourth International Symposium on Combustion,* Pittsburgh, PA: The Combustion Institute, pp. 2771-2778.

Smith, C. E., and Leonard, A. D., 1997, CFD Modeling of Combustion Instability in Premixed Symetric Combustors, ASME paper 97-GT-305. Presented at the 1997 ASME Turbo-Expo Meeting, June 2-5, Orlando Florida.

Yang, V., and Anderson, W., eds.,1995. *Liquid Rocket Combustion Instability*, Published by the American Institute of Aeronautics and Astronautics, Cambridge, MA.