EMISSIONS FROM ENCLOSED SWIRL STABILISED PREMIXED FLAMES

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ABSTRACT

The premixed combustion efficiency and emissions characteristics of four axial vane swirlers are compared with a simple grid plate stabilised premixed flame. The four swirlers are designed to investigate the influence of Swirl Number, pressure loss and swirler design. The results show that efficient combustion of weak mixtures at simulated gas turbine combustion conditions is difficult with swirl systems, but relatively easily achieved with grid plate systems. High swirl numbers are shown to have the worst combustion efficiency with a major unburned hydrocarbon problem. NOx emissions are similar for all the stabilisers and they all exhibit a very high proportion of NO2 and NOx emissions for weak mixtures.

INTRODUCTION

Axial swirlers have been a common feature of gas turbine combustion chambers for many years. Generally they have been used to inject swirling air around the fuel injector and coupled with the radial primary zone jets to establish a stable primary recirculation zone (1). Only a small proportion of combustion air passes through the swirler, usually approximately 10%, with the major proportion of the primary zone air entering through the radial jets. In recent years the problem of emissions from gas turbines has led to the use of arrays of swirlers, each with its own fuel injector, as a basis for the primary zone aero-dynamics (2). Alternatively, large single swirler and fuel injector combinations have been proposed, generally for industrial gas turbines, for the primary zone (3,4).

This paper presents the combustion efficiency and emissions results for four enclosed axial swirler designs and compares the results with those of Dabbagh and Andrews (5) for a non-swirling flow grid plate stabilised flame at equivalent test conditions. Albright and Alexander (6) showed that enclosed swirling flames with relatively small flow expansions produce a major extension of flame stability relative to the free flame situation. Thus, this type of swirl stabilised flame may have a leaner operating range than for other types of stabilisers.

Ahmad and Andrews (7) have shown that to achieve a low swirler pressure loss with all the combustion air passing through the swirler, the flow expansion downstream of the swirler must necessarily be small. A maximum flame tube diameter (D) to swirler outer diameter (d) of 1.9 was evaluated for a maximum feasible pressure loss of 8% at a combustor reference Mach number of 0.047. Many investigations of enclosed swirling flows have been carried out with D/d ratios (8-12) much larger than this value. Ahmad and Andrews (7) have identified one of the major influences of D/d to be that due to the centrifugal forces on the spread of combustion. Large D/d ratios enable a large initial corner recirculation zone to be established which acts as an ignition source and centrifugal effects force the light hot gases inwards and the heavier burnt gases outwards and a rapid flame spreading occurs (4,9). However, with small D/d ratios the corner recirculation is small and with high swirl numbers may not exist. Also the outer swirl velocities are high and flame propagation is difficult especially in weak mixtures.

Larger D/d ratios can be utilised if the combustor reference Mach number can be decreased, involving a larger combustor or the proportion of swirler air flow is drastically reduced. Shekleton (4,9) has investigated such a system where D/d a ratio of 2.1 is used, the swirler passes 50% of the combustion air yielding an overall pressure loss of approximately 4%. However, to achieve low NOx such systems have to operate close to a premixed condition. With direct fuel injection the relatively rich mixtures produce much higher NOx emissions and this was found in Shekleton's work (9).

Studies of swirl based low emission gas turbine systems have shown that increasing the proportion of swirler air flow decreases the NOx emissions and only physical swirler size limitations prevented all the combustion air (excluding wall cooling) from being passed through the swirler (13), to achieve the minimum NOx at high power conditions. D/d ratios from 1.3 to 1.7 are investigated in the present work and these are designed to pass all the combustion air.

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AXIAL SWIRL NUMBER

Four different types of eight-bladed axial swirlers have been tested. These were designed to investigate the influence of the type of swirler blade, swirl number and swirler pressure loss. The details of the four swirlers are set out in Table 1 and the test configuration is shown in Fig.1. All the swirlers were of an annular swirler design and were fitted with a central fuel pipe which was not used in the present premixed tests, but will be used in future tests to investigate the influence of fuel and air mixing. The first swirler (SW1) had curved blades with the blade angle increasing gradually from zero to form an annular curved blade passage with a relatively high swirl number. Two swirlers (SW2 and SW3) were designed with straight blades, similar blade angles but with different blockages. These two swirlers enabled the influence of pressure loss at an approximately constant swirl number to be assessed. Swirlers SW1 and SW4 both have similar high Swirl numbers and has been used extensively in the NASA swirl modules (2,16,17).

TABLE 1 Swirler Design Details

<table>
<thead>
<tr>
<th>Type</th>
<th>Blade</th>
<th>(d_1)</th>
<th>(d_d)</th>
<th>D</th>
<th>(S_1)</th>
<th>(S_2)</th>
<th>(\Delta P%)</th>
<th>Blockage %</th>
</tr>
</thead>
<tbody>
<tr>
<td>SW1-C</td>
<td>Curved</td>
<td>58.3</td>
<td>56.6</td>
<td>1.35</td>
<td>1.29</td>
<td>1.02</td>
<td>5.8</td>
<td>65</td>
</tr>
<tr>
<td>SW2-S</td>
<td>Straight</td>
<td>41</td>
<td>20.6</td>
<td>45.7</td>
<td>1.67</td>
<td>0.65</td>
<td>0.47</td>
<td>8.6</td>
</tr>
<tr>
<td>SW3-S</td>
<td>Straight</td>
<td>45</td>
<td>14.8</td>
<td>47.8</td>
<td>1.59</td>
<td>0.71</td>
<td>0.55</td>
<td>4.7</td>
</tr>
<tr>
<td>SW4-T</td>
<td>Twisted</td>
<td>63</td>
<td>19.1</td>
<td>58.3</td>
<td>1.31</td>
<td>1.44</td>
<td>1.59</td>
<td>7.1</td>
</tr>
</tbody>
</table>

SWIRLER PRESSURE LOSS

The swirler pressure loss was measured by passing a metered air flow through the swirler and flame tube and monitoring the static pressure loss upstream of the swirler relative to the atmospheric pressure discharge. The measured pressure loss at a reference Mach number, \(M\), of 0.047 are tabulated in Table 1. The aerodynamic features of swirlers which influence pressure loss will be discussed in a separate paper.

SWIRL NUMBER

Annular vane swirlers of vane angle \(\theta\) (Fig.1) by restricting the flow, cause an increase in the mean annular flow velocity, \(u\), to \(u/cos\theta\) as well as a change in direction (14,18). If the flow in the vane passages is resolved into its axial and tangential components then the Swirl Number, \(S_1\), is basically the ratio of these and is equal to \(Tana\) (14,18). For the present annular vane swirlers the Swirl Number, \(S_1\), has been shown by Kerr and Fraser (18) to be given by Equation 1.

\[
S_1 = \frac{2}{\frac{1-\frac{3}{2}}{1-z^2} Tana} \tag{1}
\]

where \(z = \frac{d_d}{d}\) (Fig.1). The \(Tana\) term is the dominant factor in Equation 1, the constant varies between 0.72 and 0.82 for the different swirlers. Beltagui and Maccullum (19,20) have suggested an alternative definition of swirl number for enclosed flows based on the dynamic axial momentum flux and the furnace diameter. They present results based on the integration of the measured velocity profiles of Beltagui and Maccallum (8,19,20). They split the enclosed flow aerodynamics into four regions, depending on the value of \(S_2\). For a high \(S_2\) \((S_2 > 0.11)\) a central recirculation zone is set-up. For this flow regime their isothermal flow values of \(S_2\) are a function of vane angle and may be calculated by Equation 2.

\[
\log S_2 = 0.463 Tana - 0.723 \tag{2}
\]

The results of Beltagui and Maccallum show no significant difference in the dependence of \(S_2\) on \(Tana\) for different furnace sizes over their experimental range of \(D/d\) of 2.5-5.0. It will be assumed that Equation 2 is also valid for the present \(D/d\) ratios of 1.3-1.7.

Swirl Numbers calculated from Equations 1 and 2 for the present swirlers are tabulated in Table 1. This shows that the two straight vane swirlers (SW2 and SW3) have similar \(S_1\) Swirl numbers. The major difference between these swirlers is the pressure loss. Thus they enable the influence of pressure loss at an approximately constant Swirl number to be assessed. Swirlers SW1 and SW4 both have similar high \(S_1\) Swirl numbers although the \(S_2\) Swirl numbers are somewhat different. The \(S_1\) values are approximately twice those of the SW2 and SW3 swirlers. A comparison of SW1 and SW4 with SW2 should enable the influence of variations of Swirl Number at an approximately constant pressure loss to be assessed. Finally, a comparison of SW1 with SW4 should indicate the significance of differences in swirler design at an approximately constant Swirl Number, mean residence time and pressure loss. Table 1 shows that all the \(S_2\) values lie well above the critical value of 0.11 that Beltagui and Maccallum (20) recommend for the establishment of a central recirculation zone.
Many swirler design parameters cannot be varied independently. In particular the D/d ratio for a fixed enclosure diameter D is related to the swirler pressure loss and the swirl number. Table 1 shows that both the high swirl number swirlers have low D/d ratios. This is because for a fixed blade height, implicit in a constant D/d ratio, the reduction in flow area caused by increasing the blade angle and hence the swirl number, would produce an unacceptable pressure loss increase and so the blade height has to be increased to compensate for this. Table 1 also shows that increasing the pressure loss for similar swirl numbers permits an increase in the D/d ratios as a smaller blade height is required.

It is recognised that the swirl number may not adequately represent the overall nature of the flow downstream of the swirler geometry with no enclosure effects. Although the \( S_p \) swirl number was based on velocity profile measurements in an enclosed swirling flow, the correlation of \( S_p \) with \( T \) in Equation 2 involves no enclosure parameters. However, the enclosure does influence the aerodynamics and has been discussed in the generation of corner recirculation at large D/d ratios. Also the swirl number involves no direct pressure loss effects and hence no information on turbulence levels. The swirl number has been shown to influence the recirculation zone size and this size is independent of the pressure loss (7). Thus the swirl number may be regarded as a convenient parameter to distinguish different swirlers, but the additional effects of D/d, Z and pressure loss may be significant and not necessarily related to the swirl number.

GRID PLATE COMPARISON FLAME STABILISER

The grid plate data that has been used for comparison is taken from the work of Al Dabbagh and Andrews (5). Grid plate flame stabiliser results have not been obtained at very high pressure losses, but pressure loss has been shown to have significant influence on combustion efficiency and NOx. A simple four hole grid plate has been used for comparison as it is typical of the grid plate results. The four hole grid plate had a blockage of 74.4%, a hole diameter of 19.3 mm and a pressure loss at the 0.047 reference Mach number of 3.9%. This pressure loss is lower than three of the swirlers shown in Table 1, but comparable with SW3.

FIG. 2 COMPARISON OF RIG METERED AND GAS ANALYSIS DERIVED AIR TO FUEL RATIOS

FIG. 3 CARBON MONOXIDE EMISSIONS AT 400K

FIG. 4 CARBON MONOXIDE EMISSIONS AT 600K

COMBUSTION TEST EQUIPMENT AND CONDITIONS

The 76 mm diameter combustion test facility has been described by Al Dabbagh and Andrews (21). It consists of an air supply from a fan, venturi flow metering, electrical preheater, 1.5 m long 76 mm diameter approach pipe, flange mounted swirler, 330 m long, 76 mm diameter combustor and an exhaust system with a flame observation window located a short distance from the combustor exit. The 330 mm combustor length is a typical minimum size for aero gas turbines and is much shorter than most industrial gas turbines. Consequently combustor mean residence times are representative of the minimum practical values. The test facility is basically similar to that used by Mularz et al (17) in the evaluation of the NASA swirl module designs. The major difference, apart from the slightly larger combustor diameter, is the atmospheric pressure operating conditions of the present work. The combustor was uncooled, apart from natural convection effects. The combustion work was carried out with premixed fuel and air. To achieve this the system used by Al Dabbagh and Andrews (21) for grid plate stabilised premixed flames was used. Propane was injected downstream of the electrical preheaters and 1.5 metres upstream of the swirler. The injector had an 'X' configuration with twenty holes on centres of equal area. Calculations showed that the turbulent jet mixing would result in a uniform fuel and air mixture well upstream of the stabiliser. Mean gas samples were
obtained at the combustor exit plan using a stainless steel water cooled 'X' probe with twenty holes on centres of equal area. Fig.2 shows a comparison between the rig metered and gas analysis based air to fuel ratios. This indicates that a reasonable mean gas sample was obtained and the deviations wereless than have been reported in other similar studies (17, 22). The gas analysis system (21) had an accuracy set principally by that of the calibration gases which was ± 2%. The test conditions are summarised in Table 2.

TABLE 2 Test Conditions

Fuel: Industrial propane (93% propane, analysed by gas chromatography)  
Pressure: Atmospheric  
Inlet Temperature: 400K (low power simulation)  
600K (high power simulation)  
Reference Mach Number: .047 (Typical of modern gas turbines)

Low and high power conditions were simulated in terms of inlet temperature and at both temperatures measurements over a range of air to fuel ratios were obtained. The air to fuel ratio range studied was mainly limited to the lean regions where the combustion efficiency was maximised and NOx minimised.

CARBON MONOXIDE EMISSIONS

400K Inlet Temperature - Low Power Simulation

The carbon monoxide emissions are shown in Fig.3 as a function of equivalence ratio. For mixtures richer than an equivalence ratio of 0.75, the SW2, SW3 and SW4 swirlers and the four hole grid plate results were identical. The bulk of CO in this region was equilibrium CO and this is shown for comparison in Fig.3. With a dilution zone downstream of the primary zone simulated in the present work, some of this equilibrium CO would be burnt out. However, the efficiency of this process would depend on the design of the dilution zone and the operating conditions. If the CO is expressed as super equilibrium CO then the shape of the CO-equivalence ratio relationship changes, but the minimum position remains at the same equivalence ratio. This is illustrated in Fig.3 for one swirler, but the conclusions are valid for all the results. For leaner mixtures where minimum emissions occur equilibrium CO is much less than the measured CO and considerable differences were found between the four swirlers and the grid plate. The grid plate exhibited the lowest CO emissions at weak mixtures and only the SW2 high pressure loss swirler approached these values. The lower pressure loss SW3 swirler, with a similar swirl number, had much higher CO emissions at weak mixtures and this is most likely to be a stabilizer pressure loss influence and hence an influence of turbulence.

At weak mixtures both of the high swirl number swirlers exhibited high CO emissions. The SW1 swirler had high emissions at all equivalence ratios but the SW4 swirler showed a transition region to join the CO emission line of the other stabilizers at an equivalence ratio of 0.75. This difference may be associated with the pressure loss differences, but these are not very large. For high swirl it may be concluded that simple twisted blade swirlers are preferable to the more complex curved passage swirlers, from a combustion efficiency viewpoint, as shown in Fig.3 for equivalence ratios richer than 0.65. However, weak burning mixtures can achieve lower CO emissions with lower swirl numbers, for the same pressure loss. None of the swirl systems tested appear to offer any major combustion efficiency advantage over the simple grid plate system at this simulated low power condition.

FIG.5 UHC EMISSIONS AT 400K

FIG.6 UHC EMISSIONS AT 600K
The CO emissions are shown as a function of equivalence ratio in Fig.4. For mixtures richer than an equivalence ratio of 0.65 all five stabilisers exhibit similar CO emission levels and these are close to the equilibrium levels. However, as discussed for the 400K results, the super-equilibrium CO still shows a minimum at the same equivalence ratio as shown in Fig.4. At weaker mixtures differences between the swirlers appear and the relative levels are as found at 400K with the SW1 giving the highest emissions and the SW2 the lowest. The Grid Plate results are a little above those of the swirlers except at very weak mixtures.

UNBURNT HYDROCARBON EMISSIONS

400K Inlet Temperature - Low Power Simulation

The unburnt hydrocarbons (UHC) emissions as methane equivalent are shown in Fig.5 as a function of equivalence ratio. This shows a major influence of equivalence ratio on all the swirlers which was not found for the grid plate stabiliser which exhibited low emissions at all the equivalence ratios tested. Each swirller exhibited an equivalence ratio below which a large increase in UHC emissions occurred and a similar feature was found for the CO emissions as shown in Fig.3. The low swirl number swirlers exhibited weaker equivalence ratios at which this occurred than the high swirl number swirlers. The two low swirl number swirlers showed a significant influence of pressure loss on UHC emissions. The higher pressure loss swirler, SW2, had much lower UHC emissions than the lower pressure loss swirler, SW3, over the equivalence ratio range 0.75 to 0.6, which is the operating regime of interest for low emissions. This situation is basically similar to the CO emissions.

The higher turbulence levels generated by the higher pressure loss will produce more rapid mixing between burnt and unburnt gases and promote a greater rate of flame spread and hence lower UHC emissions. However, it is not only the turbulence levels that are important, but their distribution and this is related to the basic stabiliser aerodynamics. It is considered that with the grid plate stabiliser, where the pressure loss and hence turbulence levels are lower, the turbulence is more uniformly distributed across the flow, permitting the observed wall to wall flame propagation and the UHC are significantly below those of the swirlers in the crucial 0.6 to 0.75 equivalence ratio region. Flame observations, wall static pressure and temperature measurement (7) all indicate that with swirl the flame is confined to the periphery of the recirculation zone core which coincides with the high turbulence zone (7). A large proportion of the gases bypass this zone resulting in high UHC.

600K Inlet Temperature - High Power Simulation

The UHC emissions are much lower than at 400K as expected. However, the same trends observed at 400K are evident as was the case for the CO emissions. Above an equivalence ratio of 0.7 all the swirlers and
For weaker equivalence ratios each swirler exhibits a critical condition at which the UHC emissions increase dramatically, whereas the four hole grid plate has low UHC emissions at all the equivalence ratios tested.

**COMBUSTION EFFICIENCY**

The CO and UHC emissions result in a loss in combustion efficiency. This loss, expressed as a combustion inefficiency, is a convenient way of summarizing the two related emissions. The data of Figs.3-6 are shown in terms of combustion inefficiency as a function of equivalence ratio in Figs.7 and 8. The shape of the curves for the SW2, SW3 and SW4 swirlers are dominated by CO emission effects for mixtures richer than approximately 0.65 and by UHC emissions for weaker mixtures. The SW1 swirler at 400K had a predominantly UHC inefficiency whereas the 4 hole grid plate also has similar NOx characteristics to the swirlers at both inlet temperatures. This indicates that the larger recirculation zones of the swirler do not give rise to higher NOx emissions due to the longer local residence times. The implication of this is that the recirculation zones may not be major contributors to the NOx emissions. Studies of the aerodynamics and wall temperature profiles indicate that the major heat release is downstream of the recirculation zone and NOx formation is generally associated with the heat release zones.

**NOx EMISSIONS**

NOx emissions are extremely sensitive to flame temperature and it is considered that flame temperature is the best parameter to use in correlating the data. Equivalence ratio is somewhat inappropriate in the present situation where there are large variations in combustion efficiency. A mean exhaust temperature was computed from the gas analysis CO, CO and UHC results for each condition tested. The NOx results as a function of flame temperature are shown in Fig.9 for both the inlet temperatures tested. The two sets of results are in good agreement for the same flame temperature, except at temperatures below 1800K where the 400K results are somewhat higher.

The swirlers exhibit very similar NOx characteristics with little difference due to swirl number and swirler pressure loss. This is in marked contrast to the influence of swirler design on CO and UHC emissions. The four hole grid plate also has similar NOx characteristics to the swirlers at both inlet temperatures. This indicates that the larger recirculation zones of the swirler do not give rise to higher NOx emissions due to the longer local residence times. The implication of this is that the recirculation zones may not be major contributors to the NOx emissions. Studies of the aerodynamics and wall temperature profiles indicate that the major heat release is downstream of the recirculation zone and NOx formation is generally associated with the heat release zones.

**NOx EMISSIONS**

Although nitrogen oxide emissions are usually reported as NO2 equivalent it is assumed that they are emitted from the combustion process as NO. However, there is increasing evidence that NO2 can be generated directly in gas turbines, particularly in situations with ample excess air in the primary combustion zone (22-28). There is also evidence that NO2 can be formed within the sample probe (23,29,30). This evidence has been mainly gained from samples from laminar flame reaction zones. Many of the mechanisms proposed are not applicable in the burnt gases well downstream of the reaction zone, as in the present situation. There is good evidence now to support the view that in the present situation the measured NO2 is most likely to be flame formed (17,19,21,22,31,32).

The British Oxygen Company NOx analyser used in the present work had a carbon molybdenum converter operated at 400°C for converting NOx to NO with an efficiency of 97%. NO2 was found to be significant at all conditions tested and to form the majority of the NOx for weak mixtures. The ratio of NO2 to total NOx is shown in Fig.10 for all the results. Fig.10 demonstrates that all the swirlers exhibited almost entirely NO2 emissions at low total...
NO\textsubscript{x} levels. Also there are significant differences between the four swirlers, although the overall trends are similar. There is some evidence, particularly at 400K, that the four hole grid plate may generate lower proportions of NO\textsubscript{2} in weak mixtures, although the total NO\textsubscript{x} emissions are similar for all the stabilisers.

The mechanisms proposed for flame generated NO\textsubscript{2} involve the H\textsubscript{2}O radical which reacts with NO to form NO\textsubscript{2}, a reaction which is fast at low temperatures (25, 26, 33). In premixed turbulent flames the H\textsubscript{2}O reaction could occur in the gas phase in weak low temperatures mixtures. It is these regions of weak combustion that NO\textsubscript{2} has been found in the present work and reported in gas turbine exhausts (22, 24).

NO\textsubscript{x} CORRECTED TO 15% OXYGEN

For industrial gas turbines NO\textsubscript{x} regulations are based on a correction to a 15% oxygen condition at the 150 standard day humidity (34). The present results have been converted on this basis and are shown as a function of super equilibrium combustion inefficiency in Fig. 11. Low NO\textsubscript{x} emissions are of little value, if they are achieved at the expense of a combustion efficiency penalty and this method of presenting the results allows the optimum operating condition for low NO\textsubscript{x} and high combustion efficiency to be determined.

Fig. 11 shows that at 400K there were appreciable differences between the four swirlers. The optimum combustion efficiency was achieved at lower corrected NO\textsubscript{x} levels as the swirl number was decreased and as the pressure loss was increased. The four hold grid plate stabiliser exhibited a higher combustion efficiency with lower NO\textsubscript{x} emissions than any of the swirlers. At 600K the swirlers showed a major decrease in combustion efficiency below a critical NO\textsubscript{x} level. This critical NO\textsubscript{x} level was lower for the low

COMPARISON WITH OTHER RESULTS

Very little previous data has been published on the performance of enclosed premixed swirl stabilised flame systems, most investigations have been concerned with flame aerodynamics (6, 8, 19, 20). The only previous results that could be found were those of Roffe and Venkataramani (35) who used a similar test facility to the present with equivalent mean residence times but operated at 10 bar and an 800K inlet temperature. They investigated two curved short blade annular vane swirlers with vane angles of 40° and 50°, but with different pressure losses. The swirlers had large hub diameters (d\textsubscript{h}/d = 0.73) and a zero swirler outer expansion (D/d = 0).

The results of Roffe and Venkataramani (35) are summarised in Fig. 12. In spite of their operating conditions being much more favourable for low CO and UHC emissions, the trends and magnitudes of the emissions are of the same order as in the present work. A significant feature of the results is the major increase in CO and UHC emissions that occurs for the
higher swirl number stabiliser. This is in accord with the present findings.

The NO\textsubscript{x} results in Fig.12 are not directly comparable with the present due to the high operating pressure. The results have been reduced to a one bar test condition using a 0.05 pressure exponent. The results are compared with the present, on the basis of the same flame temperature, in Fig.9. Both swirlers give higher NO\textsubscript{x} emissions than the present and only the higher swirl results show a similar functional relationship between NO\textsubscript{x} and flame temperature. The reason for the higher NO\textsubscript{x} emissions is difficult to explain, the residence times are similar and the present results indicate that swirl design has little influence on NO\textsubscript{x} emissions. Al Dabbagh and Andrews (21) have shown that the grid plate and cone stabiliser results of Roffe and Venkataramani also have higher NO\textsubscript{x} emissions than reported by other workers for similar stabilisers.

DISCUSSION

The results have shown significant differences in the CO and UHC emissions between the different swirlers and between the swirlers and a grid plate flame stabiliser. These differences are caused by differences in the recirculation zone lengths and widths and the different turbulence levels that result from the pressure loss differences. For the swirlers these factors have been examined in Ref.7 where it is shown that the recirculation zone extends to approximately 175 mm downstream of the swirlers and this recirculation zone length is not influenced by combustion. The recirculation zone has two maximum width locations the axial position of which are unaffected by combustion. The initial zone is stronger than the second and is closer to the swirler for the SW1 and SW4 high swirl number swirlers.

Wall temperature profiles showed that for weak mixtures the flame did not propagate from the central core recirculation region to the wall until the end of the recirculation zone was reached. For richer mixtures the maximum wall temperature moved upstream into the recirculation zone region. This transition coincides with the sudden decrease in unburnt hydrocarbons in Figs.5 and 6. The combustion efficiency problem of the swirler was therefore a function of the ability of the flame to propagate from the central core boundary across the high velocity outer swirl flow, where the main mass flow occurred. This was easier with low swirl numbers as the outer swirl velocities and opposing centrifugal forces (7) were lower. Consequently, for high swirl numbers a richer mixture was required for the flame to propagate across the outer zone and hence to achieve complete combustion. This situation is clearly shown in Figs.3-8. The influence of increased pressure loss on improving the combustion efficiency is similarly explained by the faster flame propagation rates due to the higher turbulence levels. Thus the flame propagates across the swirl flow sooner for higher pressure loss swirlers and weaker mixtures can be burnt efficiently as shown by comparing the SW2 and SW3 swirlers in Figs.3-8.

CONCLUSIONS

1. For enclosed flow high swirl numbers are detrimental to the achievement of a high combustion efficiency for premixed flames at lean equivalence ratios (φ < 0.7).

2. Increasing the swirler pressure loss for the same swirl number improves the combustion efficiency.

3. Simple twisted blade swirlers are preferable to more complex curved blade annular swirlers, from a lean mixture combustion efficiency viewpoint.

4. Simple grid plate stabilised premixed flames have combustion efficiency characteristics at least as good as those of the best swirler system tested.

5. For premixed combustion swirl number and pressure loss do not influence the NO\textsubscript{x} emissions and these NO\textsubscript{x} emissions are similar to those of Grid Plate stabilised flames.

6. There are significant levels of NO\textsubscript{2} formed in the flames. This NO\textsubscript{2} as a proportion of the total NO\textsubscript{x} increases as the equivalence ratio is made leaner.

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