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Lean Low No_x Primary Zones Using Radial Swirlers

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ABSTRACT

Swirling flow primary zones with between 30% and 60% simulated primary zone air flow were investigated using curved blade radial swirlers. Two radial swirlers were compared with the same open area but different outlet diameters, d, giving different expansion ratios, D/d, from the swirler to the combustor diameter, D. Two combustors were used, 76 mm and 140 mm diameter, the larger one corresponding to the size of several gas turbine can combustors. There was no influence of D/d on the weak extinction. It was demonstrated that an adequate efficiency was not achieved in the weak region until there was a significant outer expansion and associated recirculation zone. It was shown that these systems with central gaseous fuel injection had good flame stability with very low NO_x emissions. Propane and natural gas were compared and the NO_x emissions were 50% lower with natural gas. The optimum NO_x emissions, compatible with a high combustion efficiency, were close to 10 ppm NO_x emissions corrected to 15% oxygen.

NOMENCLATURE

A_1	Combustor pipe cross sectional area, m^2 ($A_1=\pi/\ D^2/4$).
A_2	Swirler minimum open flow area, m^2 (A_2=8Lh).
A_3	Swirler outlet area, m^2 ($A_3\pi d^2/4$).
A/F	Air to fuel ratio by mass.
C_c	Swirler air flow contraction coefficient.
d	Radial swirler vane outlet diameter, m.
d_o	Radial swirler vane inlet diameter, m.
D	Combustor pipe diameter, m.
h	Minimum vane passage width, Fig. 1, m. (h = R_2-R_1 in Fig. 1).
L	Vane passage depth, Fig. 1, m.
M	Mach number, based on the area A_1, the upstream temperature T and the radial swirler mass flow (primary zone Mach number).
n	Number of radial swirler vanes.
$\Delta P/P$	Swirler pressure loss (PL),%.
T	Upstream air temperature, K.
θ	Radial swirler vane angle, Fig. 1.
ϕ	Equivalence Ratio.
μ	Combustion efficiency (based on CO and UHC measurements).
1-μ	Combustion inefficiency.

1. INTRODUCTION

For future higher temperature gas turbines operating with low emissions, more air will be required in the primary zones with better fuel and air mixing than for current conventional combustors (1). For industrial gas turbines, legislation exists for NO_x emissions which currently can only be met by using water injection, with an associated performance penalty. Only a few recent combustor designs have demonstrated low NO_x emissions close to the limits for natural gas, without water injection (2-4). Many combustor designs aimed at low emissions have been proposed (5-7), most of which have involved lean primary zones with many of the proposed designs involving swirl based systems.

There are two approaches to lean well mixed primary zones using swirlers. One is to use a single swirler of large size to pass the extra air and the other is to use multi swirlers with associated multi fuel injection points and the NASA swirl module system is the most well known example of the latter (8). Ahmad et al (9-13) have investigated single axial vane swirlers for lean primary zone applications. They showed for both single and

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counter rotating swirlers that the radial flame propagation was very slow due to the large swirler size relative to the combustor diameter. It was concluded that a larger expansion ratio from the swirler was necessary to ensure rapid flame spread. It was the objective of the present work to investigate the influence on the combustion performance of the swirler expansion ratio, D/d, combustor diameter, D, divided by the swirler outer diameter, d. This is much easier to do with radial swirlers as the swirler diameter can be varied without changing the flow area or swirler pressure loss. This is due to the ability to change the radial passage depth as the swirler diameter is changed so as to maintain the same flow area.

Radial swirlers are common in large scale burners for boiler and furnace applications. They are also common in reverse flow industrial gas turbines and the present swirlers are similar in design to those used in the Ruston Tornado combustor, but with increase flow capacity. Radial swirlers have also been featured in some previous low emission gas turbine investigations (14-16) using the radial vane passage as fuel injection and partial premixing channels. The present systems were designed to have this vane fuel injection capability, but the present work was all for the more conventional central fuel injection.

2. EXPERIMENTAL EQUIPMENT

2.1 Radial Swirler Design

The radial swirlers used a curved blade passage design in attempt to avoid vane passage wall wake effects and to minimise flow separation in the channels. The design features are shown in Fig. 1 which details the vane angle θ. The vane angle θ was the effective radial vane passage jet outlet angle. Water flow visualization investigations showed that the outlet flow was attached to this blade surface. Flow separation occurred at the radial passage inlet on the opposite wall. Flow reattachment occurred within the vane passage but the flow was controlled by the direction of the passage wall with no flow separation. The geometrical details of the three swirlers used are listed in Table 1. Two swirler outlet diameters were studied, 40 and 76 mm, with different vane passage depths to maintain the same swirler flow area. The 40 mm swirler was similar to that used in the Ruston Tornado combustor but with a greater vane depth, L. A second 76 mm swirler was used with a smaller vane depth and this was used to investigate the influence of the pressure loss.

The vane angle used, 45^{o}, was the same as that previously used for axial swirlers (9-12) and was well into the region where the swirl would generate an inner recirculation zone. It was a similar radial swirler design to that currently in use in the Ruston Tornado combustor. The determination of the swirl number from the swirler geometry is somewhat difficult and no simple formulae exist that is equivalent to that of Kerr and Fraser (17) for axial swirlers. Beer and Chigier (18) reported an IFRF formula for vane swirlers which does not produce sensible numbers for practical geometries. In the present work we tentatively suggest a swirl number based on the mean swirler outlet axial velocity (based on the outlet area A_3) for the axial momentum and the tangential vane passage outlet velocity (based on the minimum flow area C_cA_2) for the tangential momentum. This gives a swirl number as in Equation 1, which has been evaluated for the three swirlers in Table 1. Future work will investigate the influence of the vane angle and swirl numbers.

$$S = \frac{\sin\theta}{1 + \tan^{-1}\theta} \; \frac{A_3}{C_cA_2} \qquad (1)$$

where C_c is the swirler contraction coefficient determined from isothermal flow pressure loss measurements.

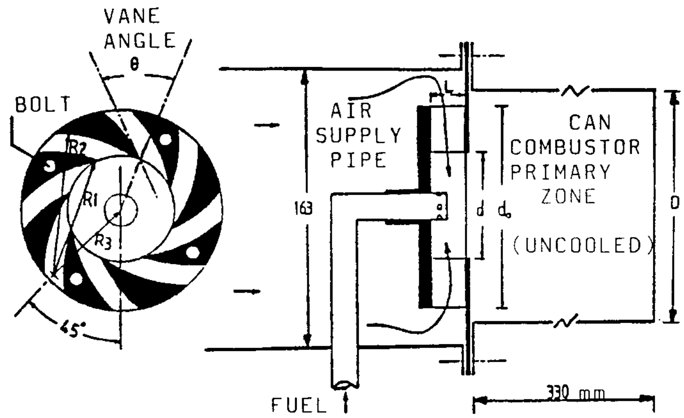

Fig.1 Radial swirler combustor configuration

Table 1: Radial Swirler Design Details.

Swirler Type	A	B	C
Inner radius of curvature of the passage R1.	46.0	78.0	78.0
Outer radius of curvature of the passage R2.	54.0	94.3	94.3
Radius of centre for R1 and R2.	35.0	59.2	59.2
Vane depth(height),L	30.5	15.0	11.5
Minimum passage width, h.	8.0	16.3	16.3
No. of swirl vanes,n	8	8	8
Vane angle.	45	45	45
Swirl No. Eq. 1.	0.54	1.41	1.84
Vane inlet diameter, d_o.	76	127	127
Swirler outlet diameter, d.	40	76	76
Minimum flow area, A_2 mm².	1952	1956	1500
C_c	0.42	0.58	0.58

2.2 Fuel Injectors

The swirlers were tested with central fuel injection using both propane and natural gas fuels. The injectors were placed with their eight radial fuel holes 3 mm downstream of the upstream face of the radial swirlers. The same eight hole 12.7 mm diameter injector was used for each fuel with a hole diameter of 2.5 mm. The different fuel densities resulted in differences in fuel jet velocities for the same fuel mass flow. This may influence the local fuel and air mixing, although the fuel jet momentum was not a major part of the mixing process.

2.3 Combustion Systems

Two simple uncooled can combustors 330 mm long were used. The first was of 76 mm diameter and was used in the previous investigations with axial swirlers (9-13). This 76 mm combustor gave swirler expansion ratios of 1.9 and 1.0 for the 40 and 76 mm diameter swirlers respectively. The 40 mm outlet diameter swirler was also suitable for application to annular combustors and previous work for axial swirlers investigated swirler interaction in a 76 mm deep three swirler simulated sector rig (19).

The second 330 mm long can was a new 140 mm internal diameter combustor. This had an upstream plenum chamber air feed and was connected to the exhaust system with water cooled pipes. The smaller 76 mm combustor had a free discharge into the exhaust. The 140 mm combustor was a similar size to many gas turbine can combustors, for example the Rolls Royce Spey and Tay and the Ruston Gas Turbine Tornado and similar gas turbines by other manufacturers in the same power range. The objective of studying a single large swirler in these configurations was to assess whether multi-swirlers such as those used by NASA (8) were necessary. The two swirler diameters of 40 and 76 mm gave expansion ratios, D/d, of 3.5 and 1.8 respectively in this combustor. Both the 140 and 76 mm combustors had similar expansion ratios of 1.8-1.9 when the 76 and 40 mm swirlers respectively were fitted. This allowed the influence of combustor scaling at the same D/d to be investigated.

One of the complications of testing swirlers in different sized combustors is that new swirlers are required for each combustor for the same combustor Mach number operation at the same pressure loss. This would require four swirlers in the present work and as these are quite complex to manufacture, it was decided to operate the two swirlers in both combustors at approximately the same pressure loss of 2.5% and to accept differences in the Mach number for the two combustor tests, these test conditions are summarised in Table 2. It was previously shown for a non-swirling jet mixing system that Mach number effects on stability, combustion efficiency and NO_x emissions were relatively small provided the pressure loss was maintained constant (4,20). Mach number effects were important, particularly for NO_x, if the same stabiliser was tested at different Mach numbers in the same combustor, thus giving a pressure loss

Table 2: Test Conditions.

Swirler	d mm	D mm	d/D	M	$\frac{\Delta P}{P}$%
A	40	76	1.9	0.03	2.1
A	40	140	3.5	0.014	2.8
B	76	140	1.8	0.014	2.5
B	76	140	1.8	0.020	4.2
C	76	76	1.0	0.03	2.2
C	76	140	1.8	0.014	3.9

and hence turbulent mixing variation with Mach number. It will be demonstrated in the present work that the same conclusion apply for the radial swirlers, thus permitting the comparison of the results from the two combustors at different Mach numbers but the same pressure loss.

Both combustors used the same electrically heated air supply. Two inlet temperatures were used, 400 and 600K, to simulated low and high power operation respectively. Both combustors were instrumented with a line of wall static pressure tappings and Type K mineral insulated thermocouples. For both combustors mean gas samples were obtained from a water cooled 'X' probe mounted at the combustor exit plane. These had twenty 1 mm sample holes on centres of equal area. These were used to determine the overall combustor performance and emissions as a function of the equivalence ratio. Future work will investigate the internal flame structure using single point gas analysis traverses (11). The gas samples were transported along a fully heated sample line and pump system to heated NO_x (chemiluminescence) and UHC (FID) analysers. After passing through a fridge system to dry the sample it was analysed for CO and CO_2 (IRGA) and O_2 (paramagnetic). The equivalence ratio, combustion efficiency and mean temperature were all computed from the gas analysis.

3. WEAK EXTINCTION

The weak extinction was determined at a constant air mass flow rate or Mach number and the fuel flow was gradually reduced until the flame was extinguished. The process was observed directly from the control room through a 100 mm diameter air cooled window in the exhaust. Weak extinction was also easily identified by a sudden increase in the UHC emissions. Weak extinction data were reproducible to within +/- 0.02 of an equivalence ratio. The measured weak extinction results are listed in Table 3 for both propane and natural gas fuels at the 400 and 600K inlet temperatures at atmospheric pressure.

The results in Table 3 show that very similar weak extinction were obtained for both propane and natural gas. Tests at different Mach numbers, not reported here, also showed little influence of the Mach number on weak extinction for both combustors. This was also found by Al Dabbagh and Andrews (21) for premixed grid plate stabilised flames. The effect was attributed to the

Table 3: Weak Extinction for Propane (P) and Natural Gas (NG)

Swirler	D mm	M	D/d	$\frac{\Delta P}{P}$%	T	Fuel	Weak ϕ	Ext. A/F
B (76mm)	140	0.014	1.8	2.5	400	P	0.063	247
					400	NG	0.32	52
					600	P	0.06	252
					600	NG	0.08	215
A (40mm)	140	0.014	3.5	2.8	400	P	0.40	39
					400	NG	0.43	38
					600	P	0.36	43
					600	NG	0.38	46
C (76mm)	76	0.03	1.0	2.2	400	P	0.03	508
					400	NG	-	-
					600	P	0.02	666
					600	NG	-	-
A (40mm)	76	0.03	1.9	2.1	400	P	0.47	33
					400	NG	0.43	38
					600	P	0.38	41
					600	NG	0.39	44

dependence of the stability on the local turbulent burning velocity in the shear layer which varies directly with Mach number due to the dependence of the pressure loss and hence turbulence on Mach number. Similar arguements apply to the present non-premixed work.

There are two important features of the weak extinction results: the similarity in the weak extinction results for the 76 mm and 40 mm swirlers in both combustors and the very great difference in the weak extinction characteristics of the two swirlers. The results in Table 3 clearly show that the swirler expansion ratio has no influence on the weak extinction for the same swirler. This is in direct contrast with the strong effect of the expansion ratio on the combustion efficiency, discussed later. The implication of the result is that the flame stability does not depend on the outer recirculation zone, neither on the presence of the zone nor on its temperature. It is considered that the stability is controlled by the local equivalence ratio within the swirling shear layer between the inner and outer recirculation zones.

The weak extinction results in Table 3 clearly show that the 76 mm swirler has an extremely wide stability, which is as good as most conventional gas turbine combustors. This occurs in both combustors even though, as will be shown later, the combustion efficiency was poor for the no expansion situation. The 40 mm swirler, in both combustors behaves quite close to a premixed situation. For axial swirlers, Ahmed et al. (9) showed that at 600K the premixed weak extinction for propane in the 76 mm combustor was 0.41 equivalence ratio, close to the present central fuel injection weak extinction of 0.38 in the 76 mm combustor. The reason for this diference between the 40 and 76 mm swirlers was the local differences in mixing at the swirler. The 40 mm swirler had a 30.5 mm vane depth which was twice that for the larger swirler. Thus the fuel was injected 27 mm upstream of the swirler outlet and considerable mixing with the swirler vane outlet flows was possible. The large swirler with smaller depth and greater distance of the fuel jets from the vane passage outlets would not have this internal swirler mixing and would inject the fuel into the base of the rotating shear layer with the resultant high stability. These features of the local swirler mixing have recently been confirmed by internal gas sampling. It will be shown that these local mixing differences dominate the combustion performance comparison between the two swirlers. Another difference between the two swirlers is the nominal swirler outlet swirl numbers in Table 1. The lower swirl number for the small swirler may also assist fuel and air mixing inside the swirler.

4. INFLUENCE OF PRIMARY ZONE MACH NUMBER

The primary zone Mach number is based on the combustor cross-sectional area A_1, the total primary zone air mass flow and the upstream temperature. The ratio of the primary zone Mach number to the reference Mach number for the combustor (based on the total combustor air flow and the same area A_1) gives the proportion of the air flow that is being simulated in the primary zone. An associated research programme is investigating the addition of the remaining air in a dilution zone used downstream of the primary zone. In the present work the performance of the primary zone alone is being studied. As discussed above, it was necessary to test the two combustor sizes at different Mach numbers if the same swirlers were to be used in each combustor with a similar pressure loss. Consequently, the Mach number influence was investigated first.

All three swirlers in Table 1 were tested with propane fuel in the 140 mm rig at the Mach number of 0.014 and at 600K. The 76 mm swirler (B) with the lowest blockage was also tested at M = 0.02 where it had a similar

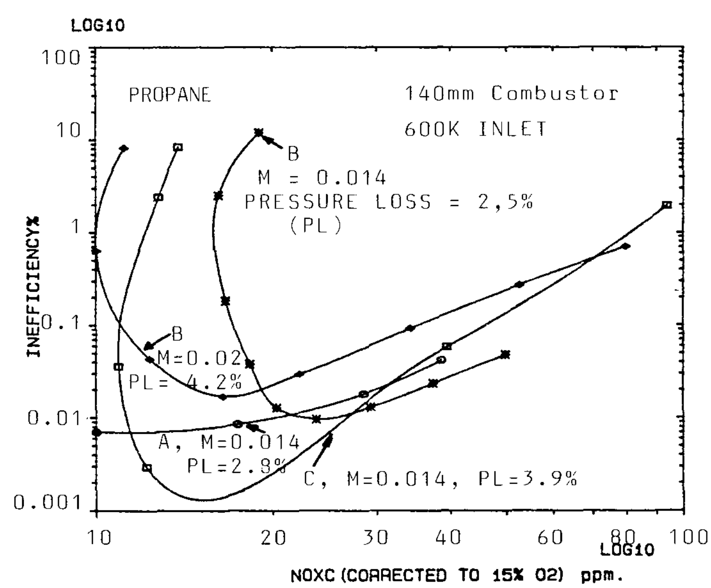

Fig.2 NOx Emissions corrected to 15% oxygen as a function of combustion inefficiency for the three radial swirlers at Mach numbers of 0.014 and 0.02.

pressure loss (4%) to the higher blockage swirler (C) at M = 0.014 as detailed in Table 2. The variation of NO_x emissions corrected to 15% oxygen is shown as a function of the combustion inefficiency in Fig. 2. The influence of Mach number for the lower blockage 76 mm swirler (B) had little influence on the inefficiency as both were less than 0.1% inefficient. However, there was a significant increase in the NO_x emissions at the lower Mach number. This was not necessarily due to the increased residence time but may have been due to the decreased fuel and air mixing at the lower pressure loss and hence turbulence levels at the lower Mach number. The test of the higher blockage 76 mm swirler (C) at the 0.014 Mach number was carried out to assess this. The NO_x emissions, shown in Fig. 2 , were found to be somewhat lower than the lower blockage swirler (B) at the 0.02 Mach number in spite of the higher residence time. It may thus be concluded that it is valid to compare the swirlers at different Mach numbers, provided the pressure loss is maintained at similar levels.

5. COMPARISON OF PROPANE AND NATURAL GAS

Four swirler configurations were investigated, consisting of the 40 and 76 mm swirlers tested in both of the 76 and 140 mm combustors. For the reasons discussed above, the pressure loss was approximately the same with a variation between 2 and 2.8% but the 76 mm combustor was tested at a Mach number of 0.03 and the 140 mm combustor at 0.014 as summarised in Table 2. For the 140 mm combustor, the pressure loss was kept constant by using different swirler blockages, as shown in Table 1, in the two combustors with the lower blockage in the 76 mm combustor.

5.1 Combustion Inefficiency at 600K

The combustion inefficiency as a function of the primary zone equivalence ratio for a 600K inlet temperature are shown in Figs. 3 and 4 for propane and natural gas respectively. The results for a zero expansion ratio for the 76 mm swirler in the 76 mm combustor were relatively poor using propane, with a maximum efficiency at 600K of only 98% compared with better than 99.9% for the other configurations. This was considered to be an unacceptable performance and no tests were carried out on natural gas as there was no evidence from the other swirler tests that natural gas had a much better efficiency than propane at any test condition.

The 76 mm swirler in the 140 mm combustor with a D/d of 1.8 had a much improved combustion inefficiency with an optimum of 0.01%, which is difficult to achieve even at very high combustor pressures using conventional combustors. Furthermore, an inefficiency better than 0.1% was maintained over a wide range of equivalence ratios from 0.3-0.6, with the apparent inefficiency at higher equivalence ratios being predominantly equilibrium CO. Fig. 4 shows that the combustion inefficiency was similar using natural gas. The main

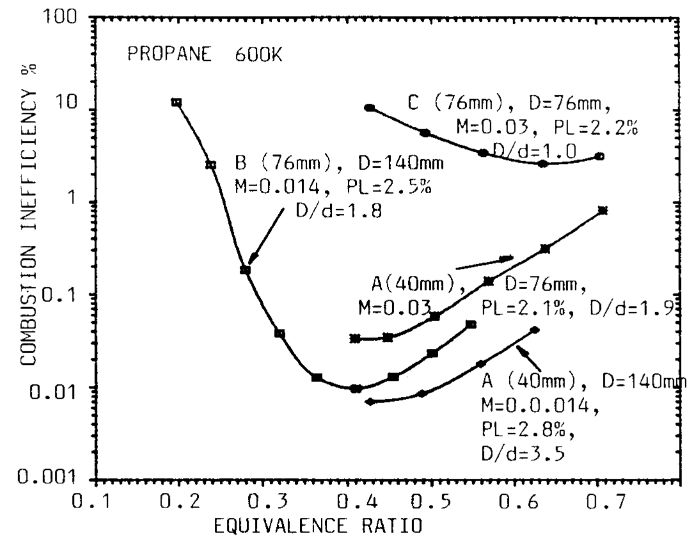

Fig.3 Propane combustion inefficiency at 600K

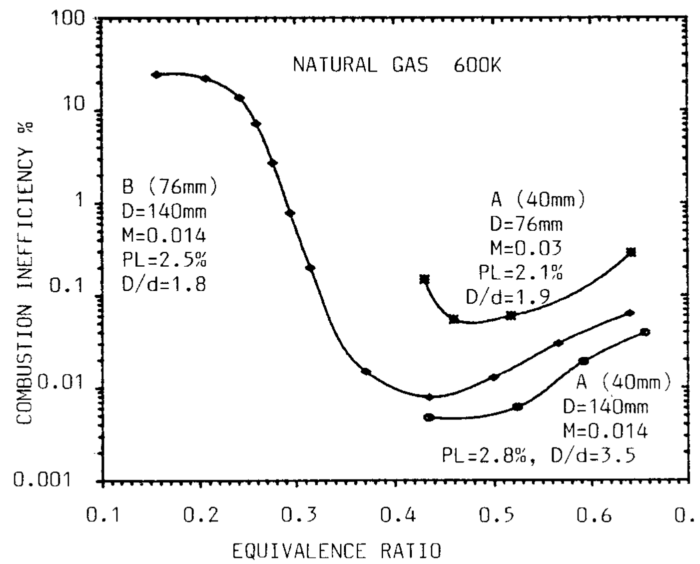

Fig.4 Natural gas combustion inefficiency at 600K

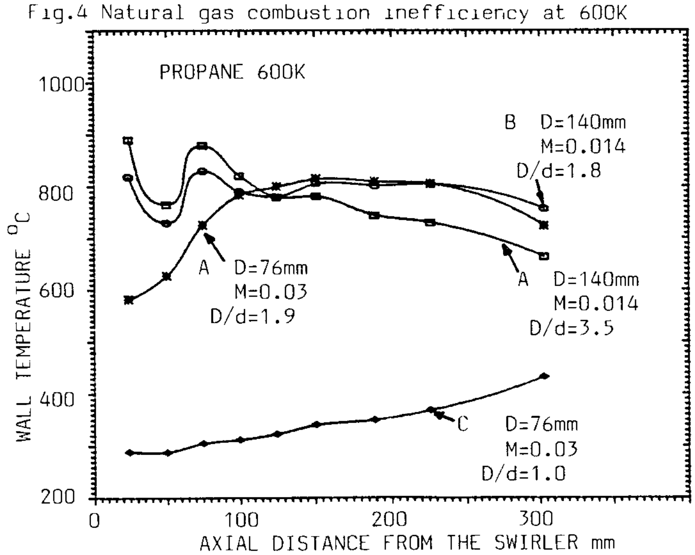

Fig.5 Axial variation of the combustor wall temperature

difference was in the lean burning region, weaker than an equivalence ratio of 0.4, where natural gas had an inferior combustion inefficiency than propane. For example at a 0.28 equivalence ratio, close to the overall equivalence ratio at high power, natural gas had an inefficiency of 1% compared with 0.1 for propane.

The large differences in the combustion inefficiency in the two combustors, for the same swirl number, shows that the poor combustion efficiency with no expansion ratio cannot be due to excessive swirl. The expansion ratio allows the swirl to decrease in the combustor, as well as creating the outer recirculation zone. It has been shown above that the D/d did not influence the weak extinction. The stability was extremely good for the 76 mm outlet swirler in both combustor sizes with the better stability achieved with a D/d of zero. However, Fig. 3 shows that there is a very significant difference in their combustion efficiencies with a much superior performance with a D/d of 1.8. The expansion ratio creates an outer recirculation zone, which although it does not effect the stability, clearly has an important influence on the combustion efficiency.

The wall temperature profiles in Fig. 5 show that with the D/d of 1.8, the flame developed much earlier than with no expansion. This has been confirmed by internal gas sampling and has been observed by others (22,23) in furnace situations with larger expansion ratios than the present. Ahmad et al (9-13) have shown for axial swirlers that a low D/d also gives a poor combustion efficiency, even for the premixed situation. The reason was shown to be due to the difficulty of the flame spreading across the outer high velocity swirling flow. Most of this work was carried out for a range of expansion ratios from 1.2 to 1.6. It was concluded that expansion ratio was a major parameter affecting the swirler performance. Large expansion ratios are difficult to achieve with high air flow axial swirlers, which was the reason for using radial swirlers in the present work. It is likely that the present D/d of 1.8 is close to the minimum acceptable for the flame to spread rapidly downstream of the swirler.

The very low combustion inefficiencies in Figs. 3 and 4, demonstrate that large rich local zones are unlikely in the recirculation zones, as these would generate high CO which would be difficult to completely burn later. It is thus clear that the local rich zone in the rotating shear layer, which give the extremely good stability characteristics, do not form the main heat release region which must occur in a better mixed zone. These arguments are also confirmed by the NO_x emissions discussed later and by comparison with the smaller swirler results which have a poor stability due to better initial mixing.

The smaller swirler results for both combustors in Figs. 3 and 4, show a very similar low inefficiency compared with the larger swirler in the 140 mm combustor. For the D/d of 1.9, the inefficiency was slighty inferior in the 76 mm combustor than for the D/d of 1.8 in the 140 mm combustor. This was probably due to a residence time effect caused by the differences in Mach number, as shown in Fig. 2. This quite small difference between the two combustors indicates that for the same D/d, swirlers will have a similar combustion performance, irrespective of the combustor size. The poor stability of the 40 mm swirler, discussed above and attributed to better internal mixing upstream of the swirler exit phase, prevented any comparison of results in the very lean region. Future work will iynvestigate this swirler with the fuel injector at the swirler exit plane to see if stability characteristics similar to those of the larger swirler can be achieved.

The results for an expansion ratio of 3.5 with the 40 mm swirler in the 140 mm combustor show a small improvement in inefficiency compared with the D/d of 1.9 in the 76 mm combustor and the D/d of 1.8 in the 140 mm combustor. However, it is clear that expansion ratios as high as this are not necessary to achieve either a high combustion efficiency or good stability. The weak extinction results indicate that the 40 mm swirler has considerable premixing upstream of the swirler exit plane. However, comparison with the 76 mm swirler results in the 140 mm combustor shows no major advantage in terms of combustion efficiency in this premixing. The 76 mm swirler results conclusively show that it is possible to achieve a wide stability with good mixing in the main combustion zones. These conclusions are supported by the NO_x results discussed later.

5.2 Combustion Inefficiency at 400K

A combustion system that can only exhibit a low combustion inefficiency at simulated high power conditions is not a viable combustor. However, many low emission combustor designs have this problem of a poor performance at low power conditions. In the present work, an inlet temperature of 400K was used to simulate a low power condition and the combustion inefficiency results are shown as a function of the simulated primary zone equivalence ratio in Figs. 6 and 7 for propane and natural gas respectively.

The results showed very similar trends to those at 600K, with only small differences in the inefficiencies. The large swirler with zero expansion had a very poor inefficiency at 400K with propane. The level of unburnt hydrocarbons (UHC) was so high that it was not safe to continue testing. All the other swirler configurations exhibited inefficiencies much less than 1% over a range of equivalence ratios. Hence it may again be concluded that an expansion ratio of approximately 1.8 is required to achieve an adequate combustion efficiency with an enclosed swirler. It is generally required that an inefficiency of less than 1% is required at low power conditions and Figs. 6 and 7 show that this was easily achieved for both combustor sizes provided the D/d was 1.8 or greater. The influence of the large D/d of 3.5 was somewhat greater than at 600K, but still remained a small effect compared with the difference between a D/d of 1 and 1.8. For the 140 mm combustor, the efficiencies

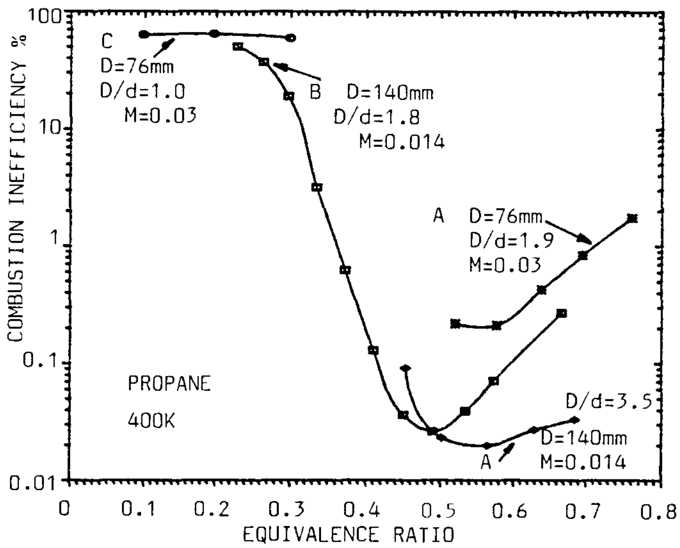

Fig.6 Propane combustion inefficiency at 400K

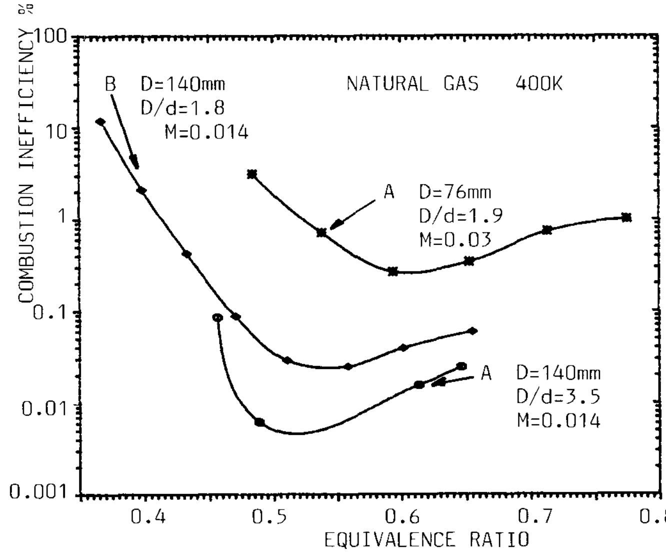

Fig.7 Natural Gas Combustion Inefficiency at 400K

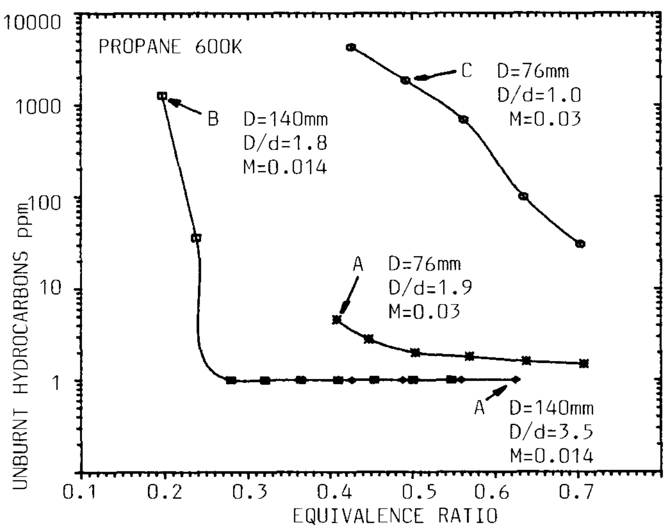

Fig.8 Unburnt hydrocarbon emissions for propane at 600K

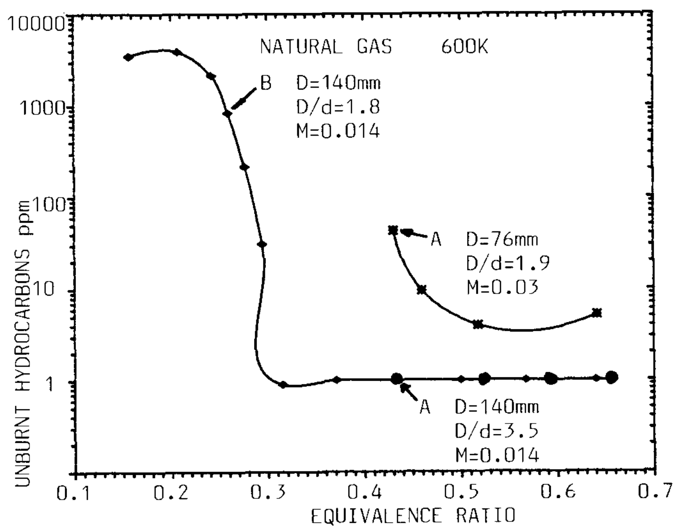

Fig.9 Unburnt hydrocarbon emissions for natural gas at 600K

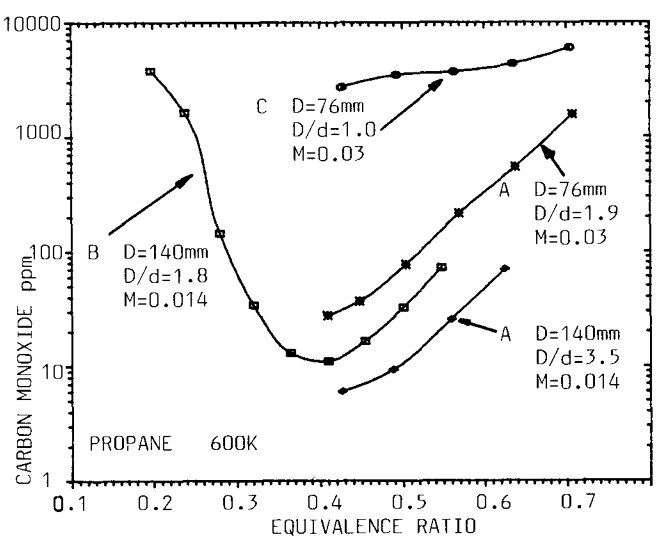

Fig. 10 Carbon monoxide emissions for propane at 600K

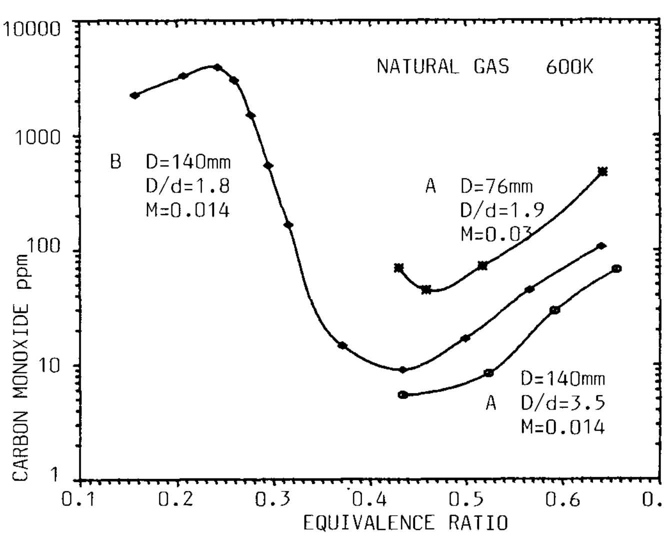

Fig.11 Carbon monoxide emissions for natural gas, 600K

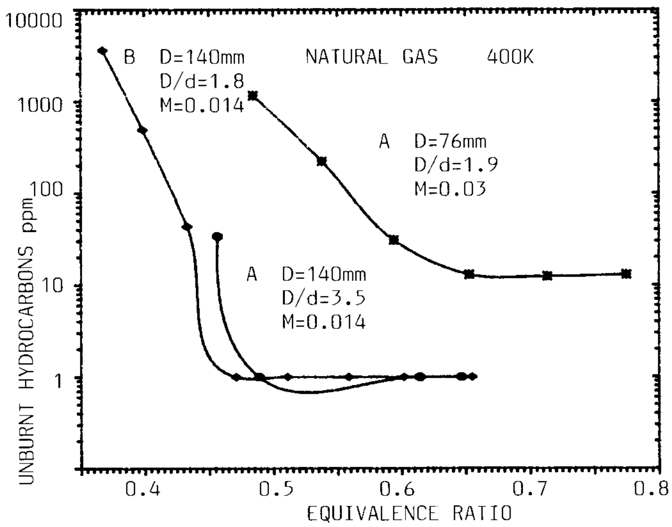

Fig.12 Unburnt hydrocarbon emissions for natural gas at 400K

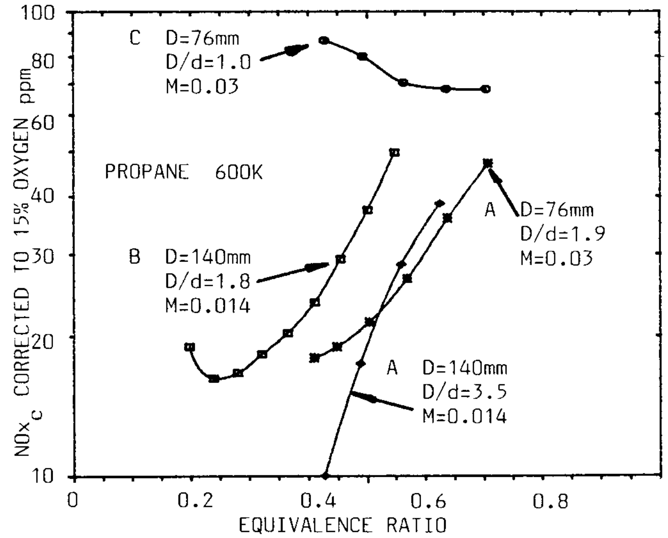

Fig. 14 NOx corrected to 15% oxygen as a function of the equivalence ratio for propane at 600K.

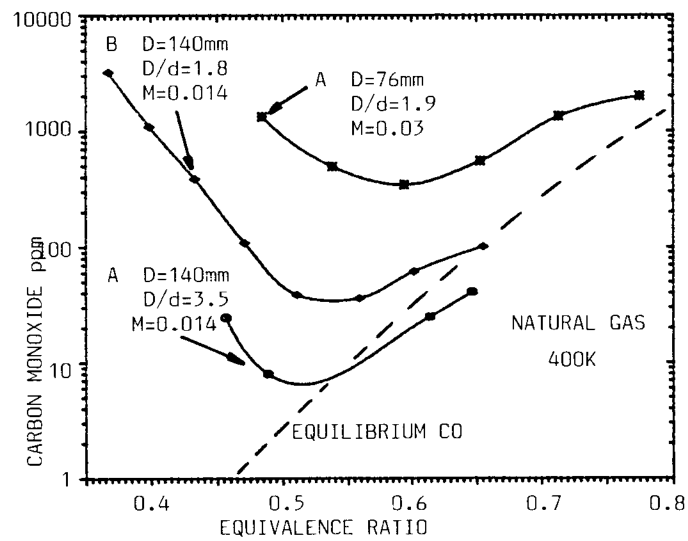

Fig.13 Carbon monoxide emissions for natural gas, 400K

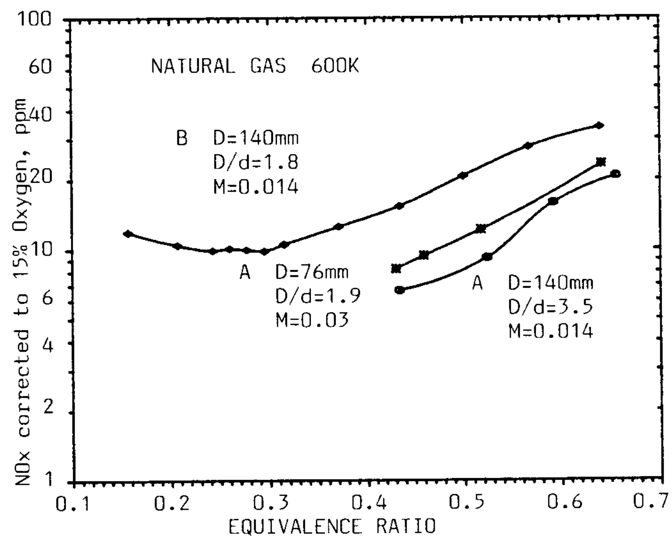

Fig.15 NOx corrected to 15% oxygen as a function of equivalence ratio for natural gas at 600K.

were less than 0.1% over the equivalence ratios 0.45-0.65 for both propane and methane. These are quite remarkable results and can be achieved by few conventional combustor designs.

5.3 Unburnt Hydrocarbon and CO Emissions

The UHC results at 600K as a function of equivalence ratio are shown in Figs. 8 and 9 for propane and natural gas respectively. These show that except for the zero expansion ratio situation, the UHC emissions were negligible at less than 10 ppm for equivalence ratios, much weaker than that at which the inefficiency started to increase in Figs. 4 and 5. Thus the inefficiency must be predominantly due to CO emissions and this is confirmed by the CO emissions in Figs. 10 and 11. These have the same trends as in Figs. 3 and 4, and hence it may be concluded the CO oxidation is the limiting factor controlling the combustion inefficiency. Similar conclusions apply for the simulated low power results at 400K as shown in Figs. 12 and 13 for natural gas.

5.4 NO_x Emissions at 600K

The NO_x emissions were corrected to 15% oxygen and a standard day humidity and are presented as a function of equivalence ratio in Figs. 14 and 15 for propane and natural gas respectively. NO_x regulations for industrial gas turbines are referred to 15% oxygen. The EPA regulations require less than 75 ppm corrected NO_x. The presented results at 1 bar cannot be directly compared with engine related NO_x regulations due to the pressure effect on NO_x. However, if the Zeldovich NO_x kinetics square root pressure correction is used then the NO_x regulations may be converted to the present 1 bar conditions. Industrial gas turbines have pressures at maximum power in the range 10-15 bar, and the corresponding EPA regulation NO_x are 24-20 ppm at 1 bar. The most stringent NO_x proposals involve an inbuilt square root pressure term in the regulations and imply a 10 ppm corrected NO_x at 1 bar. Thus NO_x emissions of less than 20 ppm are indicative of EPA compliance for most industrial gas turbines and 10 ppm levels are indicative of ultra low NO_x systems which will meet any proposed NO_x regulations.

The propane results in Fig. 14 show that for the 76 mm swirler with a zero expansion ratio, not only was the combustion efficiency poor, but also the NO_x emissions were high. Ahmad et al. (9-13) have shown for axial swirlers that this is caused by the generation of rich zones in the central core region with high NO_x as a consequence. The expansion from the swirler was found to be necessary not only for a low combustion inefficiency but also for low NO_x and hence for good fuel and air mixing. Fig. 14 shows similar conclusions may be made for the present radial swirlers. The 76 mm swirler with a 1.8 expansion ratio has corrected NO_x emissions below 20 ppm over a wide range of equivalence ratios (0.2-0.4).

The 40 mm swirler in both combustors had much lower NO_x emissions than for the larger swirler at the same equivalence ratio. This was due to the partial premixing upstream of the swirler exit plane, as discussed above in relation to the weak extinction results. Unfortunately, the poor stability for the 40 mm swirlers results in a very narrow range of equivalence ratios (0.4-0.5) close to the weak extinction over which corrected NO_x emissions less than 20 ppm were achieved.

The natural gas results in Fig. 15 were substantially lower than those for propane in Fig. 14. The reduction was approximately a factor of 2, and this has also been found by Abdul Hussain and Andrews (4) for a non-swirling interacting jet shear layer system. All three swirlers may thus be described as having ultra low NO_x characteristics for natural gas. There is a 40K difference in the peak adiabatic flame temperatures between propane and natural gas and this is often attributed to the cause of the lower NO_x emissions with natural gas. However, although this is a significant factor in the lower NO_x emissions, it is considered that the two fuels have different 'prompt' NO_x mechanisms with possibly lower prompt NO_x for natural gas. The reasons for this are the much larger number of hydrocarbon intermediate compounds for propane combustion. Internal flame gas sample traverses of these enclosed swirl flames have shown the importance of prompt NO_x with an early formation of NO_x close to the swirler.

5.5 Combustion Inefficiency and Corrected NO_x Emissions at 600K.

Low NO_x emissions are of little consequence unless they can be achieved with a low combustion inefficiency. Thus the combustion inefficiency and NO_x correlation is an important method for assessing the viability of low NO_x systems. The present results for propane and natural gas are shown in Figs. 16 and 17 respectively and the

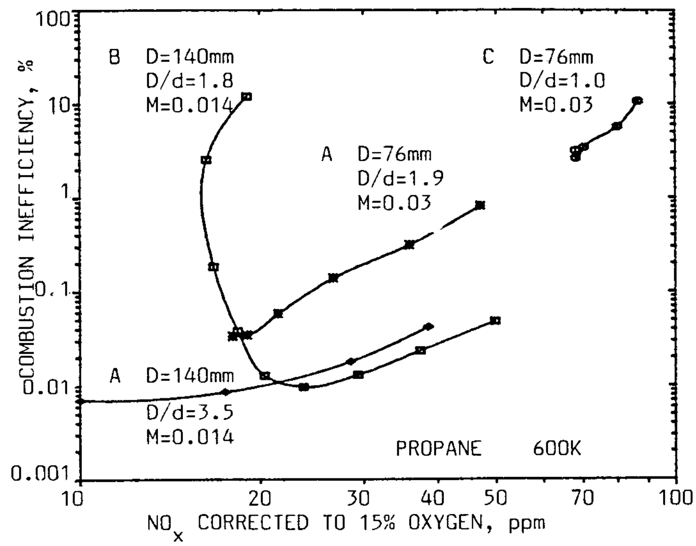

Fig.16 Combustion inefficiency as a function of NOx corrected to 15% oxygen for propane at 600K

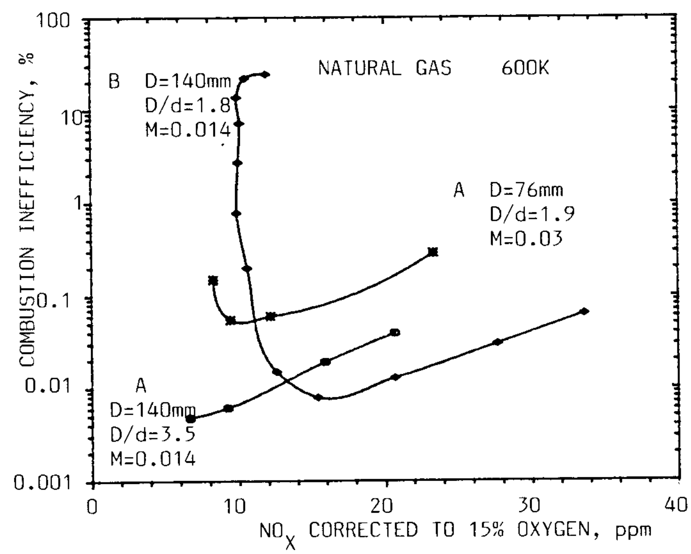

Fig.17 Combustion inefficiency as a function of NOx corrected to 15% oxygen for natural gas at 600K

optimum low NO_x conditions compatible with an acceptable inefficiency are summarised in Table 4. With the exception of the unity expansion ratio situation, all three swirler configurations exhibited corrected NO_x emissions of less than 20 ppm with a combustion inefficiency of below 0.2%. For natural gas Fig. 17 and Table 4 shows that all three swirlers can achieve 10 ppm corrected NO_x with an inefficiency of less than 0.1%. Table 4 also shows the approximately 50% lower optimum NO_x emissions for natural gas compared with propane. It may be concluded that the low NO_x emissions are not achieved with any combustion efficiency penalty. However, the poor stability of both the 40 mm swirler system was a problem as the optimum equivalence ratio low NO_x was too close to the stability limits in Table 3 for viable use in combustors. This is the same situation as most premixed/prevaporised designs, although they often have a combustion efficiency

Table 4: Optimum Primary Zone Conditions at 600K.

Swirler	Fuel	Combustor	d/D	M	Nox_c ppm	$1-\mu$ %	ϕ
A	P	76	1.9	0.03	21	0.06	0.45
	NG	76	1.9	0.03	9	0.06	0.47
A	P	140	3.5	0.014	18	0.01	0.48
	NG	140	3.5	0.014	9	0.01	0.52
B	P	140	1.8	0.014	17	0.2	0.28
	NG	140	1.8	0.014	10	0.2	0.32

problem as well as a stability one. Table 4 shows that the wider stability of the 76 mm (B) swirler allowed optimum NO_x emissions to be achieved that were very close to those for the 40 mm (swirler) which had much lower NO_x emissions for the same equivalence ratio. Also the optimum equivalence ratio for the 76 mm (B) swirler gave a factor of 3 margin on the weak extinction equivalence ratio in Table 3.

6. CONCLUSIONS

6.1 The swirler expansion ratio, D/d, and hence the size of the outer recirculation zone, does not influence the flame stability. This is controlled by mixing in the swirling shear layer.

6.2 The small diameter swirler with a large blade depth has considerable partial premixing of fuel and air upstream of the swirler exit plane. This achieves low NO_x emissions, but with an inadequate stability margin.

6.3 A swirler expansion ratio of approximately 1.8 is required to achieve a high combustion efficiency. A large D/d of 3.5 did not produce any major combustion efficiency improvement, but there was a small NO_x reduction.

6.4 The only significant difference between propane and natural gas operation was in the NO_x emissions, where natural gas had approximately half the NO_x emissions of propane for the same test condition.

6.5 The radial swirler system for natural gas exhibit ultra low NO_x emissions of 10 ppm NO_x corrected to 15% oxygen at 1 bar, with a combustion efficiency of better than 99.9%.

6.6 The upstream mixing inside the small swirler was not a crucial feature of the low NO_x emissions, the large swirler with no upstream mixing achieved ultra low NO_x emissions without the flame stability problem of the better upstream mixing swirlers.

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