

THE OPTIMUM POSITION OF WATER HEAT TRANSFER COILS DOWNSTREAM OF A RADIAL SWIRLER IN A 20KW HEATER

Andrews, G.E*. and Kim, M.N.

*Author for correspondence

School of Chemical and Process Engineering

University of Leeds,

Leeds, LS2 9JT, UK.,

E-mail: profgeandrews@hotmail.com

ABSTRACT

A 76mm outlet diameter radial swirler with 8mm vane depth was investigated in a 140mm combustor diameter condensing 20 kW ultra-low NO_x boiler. The aim was to show that small turbulent flames could achieve compact ultra-low NO_x water heating. Low NO_x was achieved using lean well mixed low flame temperature combustion with a 0.7 equivalence ratio (ϕ). Thermal NO_x formation was also minimised by cooling the flame downstream of the swirler outlet. A water cooled heat transfer coil was traversed into the flame to determine how close to the swirler exit the heat transfer could occur, without a major increase in the combustion inefficiency. This was shown to be 70mm from the radial swirler throat outlet. Rapid fuel and air mixing was achieved using fuel injection through the wall of the 76mm swirler outlet throat, assisted by a 41mm diameter outlet orifice at the exit of the 76mm internal diameter wall fuel injector. This created swirling flow with higher axial velocities and a more concentrated high turbulence region downstream of the orifice outlet. A 4 mb burner pressure loss was used, which is typical of domestic forced draught combustion systems. The air inlet temperature was 400K, which is typical of reverse air flow cooled combustion chambers at domestic water heater conditions. The strong swirling flow interaction with the heat exchanger coil give an 89% thermal efficiency with the front of the coil 70mm from the swirler outlet. The emissions measurements showed that the combustion inefficiency was below 0.1%, the CO/CO₂ ratio <0.001 and the NO_x emissions were 5ppm at 0% oxygen with the heat exchanger at 70mm from the radial swirler outlet. This design easily met the 2018 EU legislation for eco-design of domestic water heaters.

INTRODUCTION

In the UK gas fired heating accounts for about 5% of total NO_x emissions and amounts to 45 kTonnes of NO₂ emissions [1, 2]. In New England, USA, NO_x from residential heating was 7% of total NO_x emissions in 2011 [3]. NO_x emissions from boilers are required to be reduced to ultra-low NO_x levels, in many areas of the World including Europe [4] and the USA [5]. In some areas, operators are required to meet what are called "Ultra-Low NO_x" (ULN) boiler requirements, typically less than 10-12 ppm and as low as 5ppm in some areas [5]. In the Bay Area of California, boiler NO_x emissions for natural gas (NG) are 30ppm for 0.5 – 1.5 MW, 15ppm for 1.5 – 6 MW decreasing to 5ppm for >22 kW boilers [6]. The San Joaquin Valley District of California has water heater NO_x regulation

for residential properties after Oct. 1st 2018 at 14 g/GJ [7] (compare with EU regulations for biomass boilers of 150 g/GJ). This converts to an emission index (EI) of 0.7 g_{NO_x}/kg_{fuel} using a NG CV of 50 MJ/kg_{fuel}. This converts to about 27 ppm of NO_x corrected to 0% oxygen or 58 mg/Nm³.

The EU Ecodesign regulations [4] for small residential gas boilers, require NO_x to be less than 56 mg_{NO_x}/kWh from 26th Sept 2018. For natural gas (NG) with a CV of 50 MJ/kg this is 13.9 g_{NO_x}/GJ and an emission index of 0.78 g_{NO_x}/kg_{fuel} and this converts to 30ppm NO_x at 0% oxygen. This is very similar to the San Joaquin Valley NO_x regulation, but is higher than the Bay Area regulation of 5ppm NO_x for boilers between 22 and 500 kW. The 2018 EU residential boiler NO_x regulations are thus similar to some of the ultra-low NO_x emissions regulations in California. The present work demonstrates a compact residential burner design with ultra-low NO_x emissions of 0.17 EI and 5ppm NO_x at 0% oxygen, well inside the range for ultra-low NO_x emissions in the EU Ecodesign regulations.

ULTRA LOW NO_x RADIAL SWIRLER DESIGN

Normally, domestic water heating boilers use laminar flame combustion system with an air driven fan at 4mb pressure loss. Achieving low NO_x using this design is difficult, as laminar flames do not have sufficient flame stability to achieve low NO_x using very lean combustion. Although, the use of porous tube burners can overcome this problem. Two common methods of NO_x reduction with laminar flame burners are: flue gas recirculation and/or heat extract from the flame using solid copper inserts that conduct heat from the flame to the water cooled heat exchanger.

The present work demonstrates that at 4mb pressure loss, a compact turbulent flame with high swirl can be used to give a compact low residence time low NO_x burner. Combustion noise is proportional to the pressure loss and so at the same pressure loss, excessive noise is not anticipated. However, no combustor noise measurements were made, but there was no significant combustion noise detected by ear in the tests.

The ultra-low NO_x radial swirler design used in the present work, was first developed for industrial gas turbine applications [8], using radial vane passage fuel injection with lean well mixed combustion for gas and liquid fuels [9]. It was also shown that fuel injection through the radial swirler outlet duct wall [10] gave good fuel and air mixing with very low NO_x emissions, which were as low as those for vane passage fuel injection. This mode of fuel injection was used in the present work, as shown in Fig. 1. The design of the radial passage was

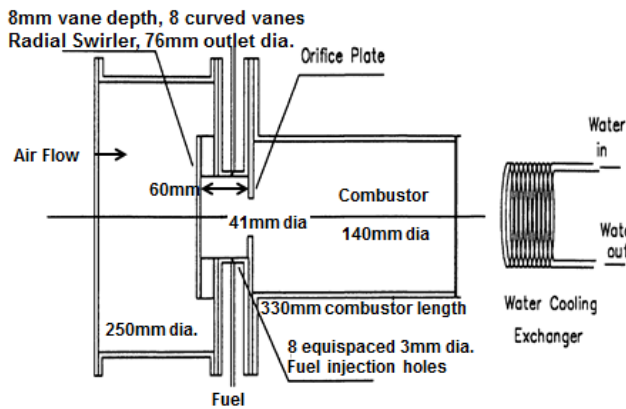


Figure 1: Radial swirler and combustor geometry

shown [11] not to be of critical importance and the original curved passage design had similar NO_x results to rectangular or circular fixed area radial passages.

The first industrial application of this low NO_x radial swirler design [12] used aerodynamic converging passage radial swirlers, which Alkabie et al. [11] showed had similar results to the other vane passage designs. The second application of the radial passage fuel injection design used rectangular radial vane passages, with fuel injection through the passage wall [13]. Since then 6 other industrial gas turbine companies have used radial swirlers with vane passage fuel injection, to achieve ultra-low NO_x emissions. Andrews et al. [14] advocated impingement regenerative cooling of the backside of the combustor with the cooling air flowing into the radial swirler vane passages. The first industrial application of this concept was by Agbonziki et al. [15] using rectangular vane passage fuel injection and impingement backside cooling. This type of impingement regenerative backside cooling has been investigated experimentally and using CFD predictions by El-Jummaih et al. [16].

The ultra-low NO_x radial swirler and combustor design is shown in Fig. 1. It used 8 curved passage radial swirlers [8] with 8mm vane depth. Fuel injection was via 8 equispaced 3mm holes in the wall of the 76mm diameter radial swirler outlet. The low air flow rates used at the low thermal powers of 20kW boilers was utilized to generate the main pressure loss at the radial swirler exit duct outlet, using a 41mm diameter outlet orifice. This has been shown previously [17] for axial swirlers to give beneficial effects on NO_x emissions, as it increased the size of the outer recirculation zone. The distance from the rear wall of the radial swirler and the orifice plate outlet was 60mm and this gave an upstream fuel and air mixing space that led to near premixed fuel and air low NO_x emissions. The flame was ignited using a wall mounted 15J surface discharge ignitor mounted 30mm downstream of the radial swirler orifice outlet plane.

RADIAL SWIRLER AERODYNAMICS

The aerodynamics of the radial swirler were investigated using conventional $k-\epsilon$ turbulence modelling and the FLUENT CFD code. The results for the axial velocity profiles are shown

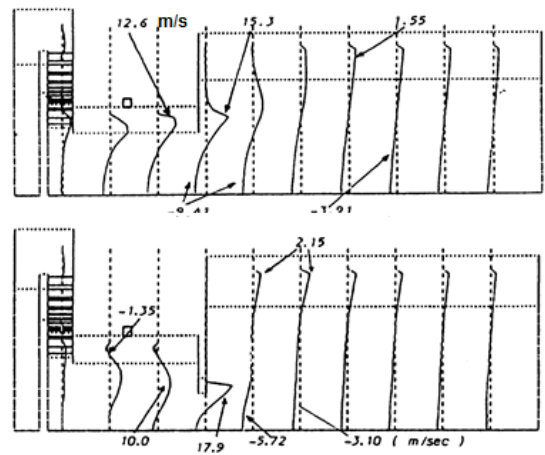


Figure 2: Predicted axial velocity profiles for the radial swirler with and without the 76mm diameter passage exit orifice.

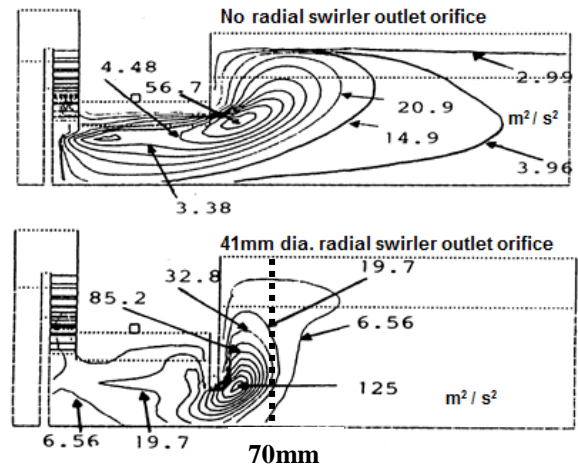


Figure 3: Predicted turbulence kinetic energy, TKE, m^2/s^2 for the radial swirler with and without the 76mm diameter passage exit orifice of 41mm diameter.

in Fig. 2 for the radial swirler with and without the outlet orifice. The predictions were carried out at a test condition of 400K with a pressure loss of 4 mb. This gives a higher flow velocity without the exit orifice and hence the mean velocity was higher. The mean 140mm diameter combustor velocity with the 41mm exit orifice was 0.85 m/s.

The peak velocity at the swirler exit plane was predicted in Fig. 2 to be 18 m/s with the radial swirler outlet orifice and 13 m/s without the orifice plate. The higher shear velocity with the 41mm outlet orifice produced greater shear and higher kinetic energy of turbulence (TKE), as shown in Fig. 3. The TKE was much more spatially concentrated downstream of the radial swirler passage orifice outlet and the peak TKE was double that without the exit orifice plate. This higher TKE produced better fuel and air mixing and lower NO_x emissions.

EXPERIMENTAL TECHNIQUES

The test rig is shown in Fig. 1 and the combustor was 140mm diameter and 330mm long, which was the length used in gas turbine combustor low NO_x research. The determination of the minimum length for small 20 kW burner applications was the objective of the present work. The outside wall of the combustor was uncooled other than by natural convection and radiation heat losses.

The water heat exchanger was made of a spiral of 6.3mm OD steel tubing in the form of a coil with a flat end to the coil and ten loops of the coil on a center-line of 120mm diameter. This placed the water cooled tubes in the outer swirl flow and only the feed-water line passed down the central recirculation zone of the flame. This heat exchanger was mounted so that it could be traversed towards the radial swirler outlet so as to vary the distance of the flame to the quench region. The water flow rate was determined using a variable area flow meter and the temperature rise determined using Type K mineral insulated thermocouples. This enabled the heat transferred to the water to be determined in kW. The tests were carried at 20 kW of heat output.

The emissions were measured using a water cooled gas sampler 330mm downstream of the radial swirler outlet orifice and downstream of the heat exchanger coil. The distance to the heat exchanger quench position was variable by moving the whole heat exchanger coil. The mean emissions were obtained using a 40 hole 'X' sample probe with holes on centers of equal area, 330mm from the radial swirler orifice outlet plane. The exhaust gases were passed to the gas analysis system via a heated sample line, heated filter and pump so that THC and NO_x were measured hot. The gas sample for NDIR CO, CO₂ and O₂ analysis was passed through a 4°C cooler that achieved gas sample water condensation, so that these gases were measured dry. The measured NO_x and THC were converted to a dry basis, as used in small boiler emissions regulations, by correction after the tests. The gas analysis was used to determine the combustion equivalence ratio by carbon balance.

NO_x emissions were measured using a chemiluminescence NO_x analyser. This had a converter to convert NO₂ into NO and this was used to determine the NO₂/NO ratio. CO and CO₂ emissions were determined using NDIR and unburned hydrocarbons were determined using a flame ionization detector (FID).

EXPERIMENTAL TEST CONDITIONS

The combustion air was electrically preheated to 400K, as this is representative of the temperature achieved using backside reverse flow cooling of the combustor. Reverse flow cooling of the combustor was not used in the experiments, but it would be used in applications of this ultra-low NO_x burner concept in a domestic water heating boiler.

The air and fuel flow was set to achieve a 0.7 equivalence ratio with an air flow pressure loss of 4mb across the radial swirler and a fuel thermal input of 23.3 kW. This was a heat release rate of 1.5 MW/m², which is very high for domestic water heaters and as a consequence gives a very compact water heater. The 41mm diameter orifice swirler outlet was designed to achieve 4 mb pressure loss, which is typical of modern fan

driven compact residential water heaters. In related work lower pressure losses were investigated, but not in terms of the minimum distance for the heat exchanger location. The 4mb pressure loss generated high TKE for fuel and air mixing and flame stabilization. At lower pressure loss the NO_x emissions were higher due to lower turbulence for air and fuel mixing, but the increase was not large.

The 0.7 equivalence ratio, ϕ , was chosen to achieve low NO_x for well mixed combustion as the adiabatic flame temperature, at 400K air temperature, was 1970K, which is just above the onset of thermal NO_x formation [18]. The use of lean mixtures (or high excess air – 43% at 0.7 ϕ) for low NO_x used to be associated with a poor thermal efficiency. This was due to the increase in the heat rejected in the flue caused by the higher air flow required for lean combustion, for the same thermal energy input as for 5% excess air combustion. However, with the advent of condensing boilers, which give low temperatures in the flue, the impact of high excess air on thermal efficiency is much lower and most low NO_x burners for domestic central heating condensing boilers use lean low NO_x combustion.

Lower NO_x could be achieved by operating at leaner mixtures with lower adiabatic flame temperatures. Alternatively, power output modulation could be achieved by reducing the fuel flow to reduce the flame temperature. In work for gas turbine applications, using radial swirlers, it has been shown that the lowest NO_x combustors use lean well mixed combustion and have NO_x emissions of 3 ppm corrected to 15% oxygen at 1900k [18]. This converts to 10 ppm at 0% oxygen and the present work will show NO_x emissions of 5ppm at 0% oxygen. The lower NO_x is due to the reduced residence time at the flame temperature, with the water quench at the position just after the completion of combustion. Thus, there is minimal residence time at the flame temperature, as thermal NO_x formation is halted as soon as the water quench reduces the flame temperature.

THERMAL EFFICIENCY

The heat extracted into the stainless steel water cooling coil is shown in Fig. 4 and the thermal efficiency in Fig. 5, as a function of the distance of the coil from the radial swirler. There was a very strong influence of heat transfer on the axial position of the heat exchanger. The peak heat transfer and thermal efficiency was 89% at 70mm from the swirler orifice plate outlet and was very critical to this location, as 20mm either side of this the thermal efficiency fell from 89% to 76-80%. The 89% peak thermal efficiency meets the UK requirement for domestic water heating boilers of a minimum 86% thermal efficiency [20]. The turbulent combustion system is also significantly more compact than current laminar flame type boiler systems with a combustion chamber and heat exchanger that would fit in a 200mm OD cylinder 220mm long complete with 20mm thick external wall insulation.

The 70mm location is marked on the Fluent CFD TKE predictions in Fig. 3. This shows that the peak heat transfer occurs in a region of high turbulence just after the flame development in the region of the peak turbulence, which occurs at 25mm from the radial swirler orifice outlet. The test rig was instrumented with wall thermocouples and in the absence of the

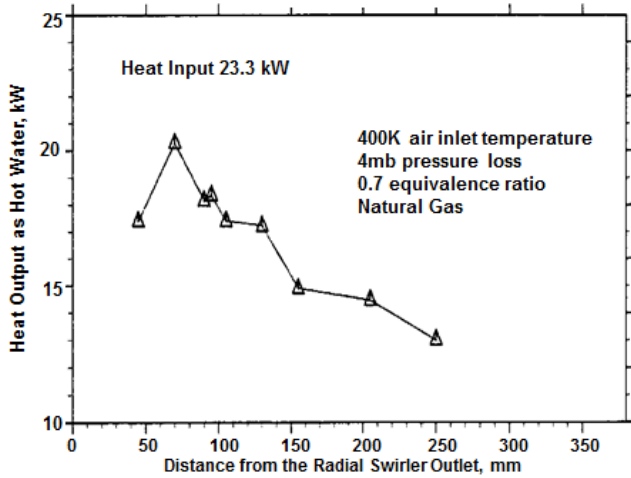


Figure 4: Water coil heat output as a function of the distance of the front of the coil from the 41mm radial swirler outlet orifice.

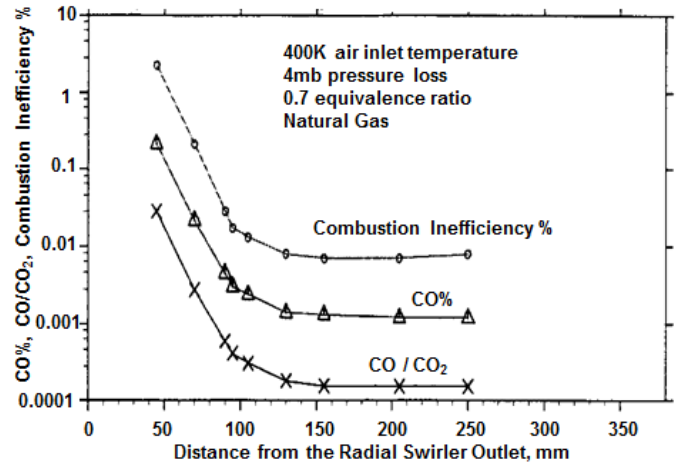


Figure 6: Influence of the distance to the heat exchanger quench zone on CO%, CO/CO₂ and combustion inefficiency%.

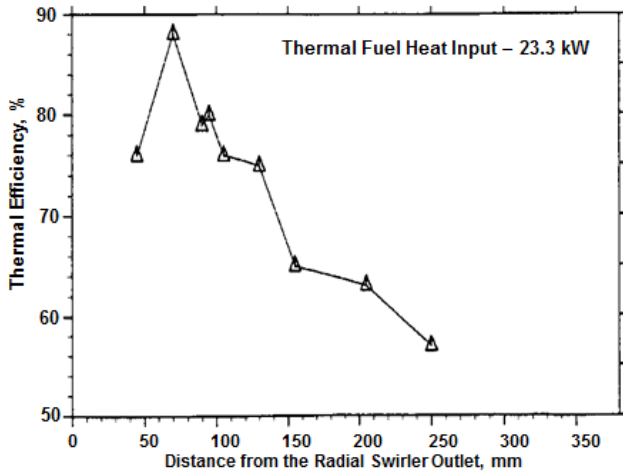


Figure 5: Thermal efficiency of the heat exchanger as a function of the distance from the 41mm orifice radial swirler outlet.

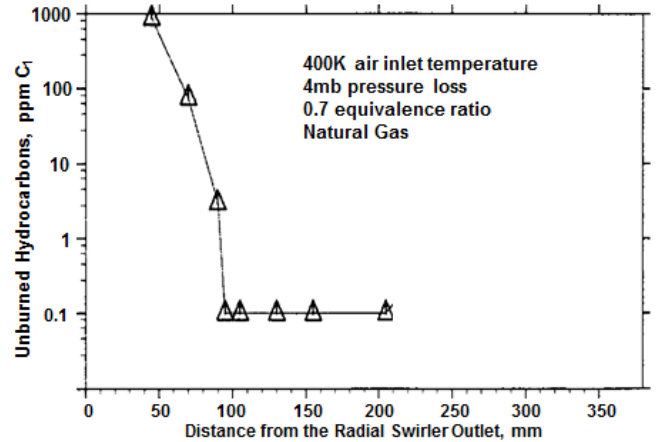


Figure 7: Unburned hydrocarbons (UHC) ppmC₁ as a function of the distance to the heat exchanger quench zone

water cooling coil these showed that the wall temperature 20mm from the radial swirler outlet was 983K and this increased to 1033K at 50mm with 1013K at 70mm and then gradually reduced to 873K at 200mm, due to external heat losses. These wall temperature measurements indicate that the gas side temperature was at its peak at the 50mm position and this zone of high temperature extended to the 70mm position of the maximum heat transfer. It will be shown in the combustion efficiency results below, that combustion was 99.9% complete at the 70mm position. Thus, the peak thermal efficiency occurred for the heat exchanger located where the heat release was complete and the TKE was high. This demonstrates the critical importance of the location of the water heat exchanger for the achievement of a high thermal efficiency.

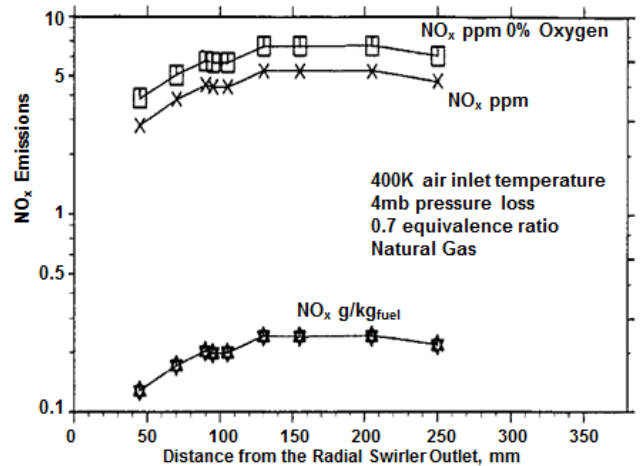


Figure 8: NO_x emissions on a dry basis as a function of the distance to the heat exchanger quench zone

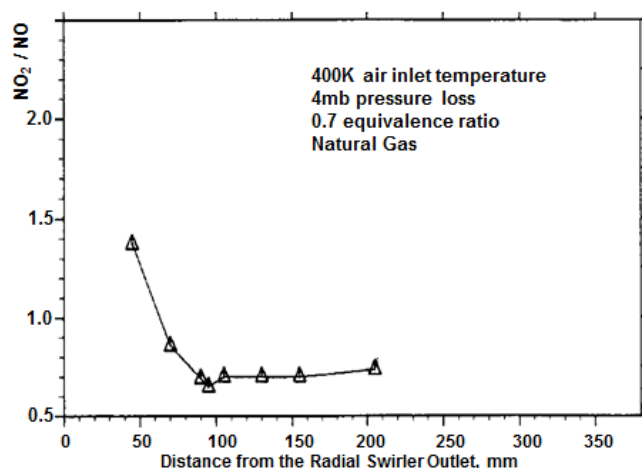


Figure 9: NO₂/NO ratio as a function of the distance to the heat exchanger quench zone.

WEAK EXTINCTION

The weak extinction of the flame at the test conditions of 400K and 4mb pressure loss was determined as 0.44ϕ . This is a very stable flame and this weak extinction is close to the fundamental flammability limits for methane-air, where the critical flame temperature is 1400K – 1500K [19]. At 400K this is $0.4-0.45\phi$ and so the observed weak extinction is close to the fundamental flammability limit. This has been shown [19] to be a key requirement for lean low NO_x combustion for gas turbine applications and this work shows that it is also a key requirement for domestic water heating low NO_x combustion.

The good weak extinction shows that thermal power could be modulated from 0.7ϕ to 0.5ϕ or the thermal output reduced from 20 kW to 14 kW purely on the fuel flow. Further reduction in power could be made by reducing the air flow, which would also reduce the pressure loss. In associated work, the reduction in the air flow has been shown to occur without a major combustion or NO_x problem for pressure losses down to 1mb. This pressure loss change is an air flow reduction of 50%, which would take the minimum thermal output to 7 kW. Thus the design could act as a modulated heat output water heater, with no change in the flame stabilizer geometry.

EMISSIONS AS A FUNCTION OF DISTANCE TO THE HEAT EXCHANGER QUENCH ZONE

The CO emissions are shown in Fig. 6 as a function of the distance to the water cooled heat exchanger. This shows that at the 70mm position, for the optimum thermal efficiency, the CO was 0.02% and the CO/CO₂ ratio was 0.002. These are both well inside the UK recommended values [21] of 0.1% CO and 0.02 CO/CO₂. Fig. 6 shows that CO and CO/CO₂ levels 1/10 of those at the 70mm position could be achieved if a greater distance to the water quench zone was used. This is undesirable due to the reduction in thermal efficiency that occurs when the heat exchanger is moved out of the high turbulence zone.

The THC emissions are shown as a function of the distance to the heat exchanger in Fig. 7. This shows that at the 70mm position the THC was 80ppm. There are no THC emissions

regulations for domestic water boilers, apart from those of thermal efficiency. The THC emissions are an energy loss of unreacted fuel and hence influence the thermal efficiency. The combined CO and THC energy loss is shown in Fig. 6 as the combustion inefficiency. At the 70mm burner to heat exchanger separation distance this 0.15%, which is a negligible contributor to the loss in thermal efficiency. As shown in Figs. 6 and 7, much lower CO and THC emissions could be achieved for a greater separation distance.

The NO_x emissions are shown in Fig. 8 as a function of the radial swirler outlet to heat exchange separation. This shows that at the 70mm optimum position of the heat exchanger the NO_x emissions were 5ppm at 0% oxygen and 0.17 g/kg. If the distance was increased the NO_x increased to 7ppm at 0% oxygen or 0.25 g/kg. These are incredibly low NO_x emissions and are below equivalent gas turbine combustor conditions. The lowest published NO_x for 1900K flame temperature at gas turbine combustor conditions are about 10ppm at 0% oxygen or 3ppm at 15% oxygen [18]. The present 5ppm NO_x emissions are 1/5 of the EU requirement for domestic water heating from 2018 [4] and 1/4 of the San Joaquin Valley District of California NO_x regulations for 2018 [7].

A key reason for these ultra-low NO_x emissions is the cooling of the flame as soon as the heat release was complete, so that there was minimal residence time at the 1970K adiabatic flame temperature. The importance of this is shown by the emissions results for the combustor with no heat extraction and a 330mm combustor length. For the present geometry, 4mb pressure loss, 400K inlet temperature and 0.7ϕ the NO_x emissions were 25ppm at 0% oxygen or 0.6 g/kg. These are still low NO_x emissions but now only just meet the 2018 EU [4] and San Joaquin Valley District of California NO_x regulations [7] of 0.7 g/kg and 27ppm at 0% oxygen. The increase in NO_x was due to the thermal NO_x generation over the 260mm distance at close to the adiabatic temperature. The location of the heat exchanger relative to the radial swirler orifice plate exit plane is thus crucial in the achievement of the present ultra-low NO_x emissions combined with low CO and UHC emissions.

The major problem with NO_x emissions is that they react in the atmosphere to form NO₂ which is harmful to humans as it reduces the lung function. Emissions of NO₂ directly from combustion systems is thus undesirable. The NO₂ emissions as a fraction of the NO emissions are shown in Fig. 9, as a function of the distance from the radial swirler orifice plate outlet plane. At the 70mm optimum location of the water heat exchanger the NO₂/NO ratio was 0.87, which is $0.47 \text{ NO}_2/\text{NO}_x$. The direct conversion of NO to NO₂ occurs in flame reaction zones through the $\text{NO} + \text{HO}_2 = \text{NO}_2 + \text{OH}$ reaction. The HO₂ radical occurs in the flame reaction zone at relatively low flame temperatures of around 1500K [22] and this occurs upstream of the 70mm optimum heat exchanger location. This is why the NO₂ conversion increases at the 50mm heat exchanger location in Fig. 9. NO₂ decreases as the heat exchanger is moved downstream as the reaction zone remains hotter for longer and HO₂ formation of NO₂ from NO declines.

CONCLUSIONS

The 2018 EU and San Joaquin Valley District of California NO_x regulations for domestic gas fired water heaters, of 56 mg NO_x/kWh and 14 g/GJ respectively, convert to 0.7 g/kg and 27 ppm NO_x at 0% Oxygen. These NO_x emissions were shown to be easily achieved with the proposed radial swirler flame stabiliser design and a water heat exchanger positioned 70mm from the radial swirler outlet plane. These results were for 400K inlet temperature at 0.7Ø, where the adiabatic flame temperature was 1970K and thermal NO_x formation was significant. The NO_x formation was limited by the heat exchanger cooling the combustion and lowering the flame temperature, just after the heat release was completed. This gave a minimal residence time at the 1970K temperature. The CO and THC emissions were also very low at this low NO_x condition.

A 76mm outlet diameter radial swirler with 8mm vane depth was used with fuel injection through the swirler outlet duct wall using eight 3mm equispaced fuel injection holes, one for each radial swirler vane passage. The swirler outlet duct was 60mm long from the back face of the radial swirler. At the outlet of the 76mm duct a 41mm diameter orifice was placed that gave a pressure loss of 4mb. The expansion of the swirler flow from the 41mm outlet diameter to the 140mm combustor diameter was predicted to give high turbulence close to the 41mm orifice plate outlet. The 60mm long 76mm diameter cavity upstream of the 41mm diameter outlet orifice gave good mixing of the cavity wall injected fuel with the swirling air flow. The geometry gave very stable quiet combustion with a very short flame development length of 70mm.

The positioning of the 6.3mm OD stainless steel water heating coil at the 70mm axial position was shown to give an 89% thermal efficiency and a water thermal heat load of 20 kW. The positioning of the heat exchanger was critical, as it was located in the highly swirling flow in a region of high turbulence. Further downstream the thermal efficiency was much lower, as the gas side swirl velocities and turbulence levels were lower. Overall the design gave a very compact combustion system with ultra-low NO_x emissions.

ACKNOWLEDGEMENTS

We would like to thank that Korean Gas Corporation for a partial studentship and the British Council for a research scholarship. The test equipment was provided by several UK EPSRC research grants.

REFERENCES

- [1] UK Defra, National Statistics Release: Emissions of air pollutants in the UK, 1970 to 2015, STATISTICAL RELEASE: 21 April 2016.
- [2] Abbott, J., Conolly, C. Cooke, S., Passant, N., Stewart, R., Wagner, A., Assessment of the effectiveness of measures under the Clean Air Act 1993. AEA/R/ED46626/3289, 2012.
- [3] US EPA, NO_x Control Regulations, 2017.
- [4] EU COMMISSION, REGULATION No 813/2013 Implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters. 2 August 2013
- [5] Anonymous, Gas Technology Boiler NO_x Emissions, *Plant Engineering*, 7.20.2010.
- [6] Elliot, J., Boiler NO_x Emissions and Energy Efficiency. *CPAU Facilities Managers Meeting*, Bay Area Air Quality Management District, Mar. 10, 2011
- [7] San Joaquin Valley Air Pollution Control District. Rule 4905 Natural gas-fired, fan-type central furnaces, 2015.
- [8] Alkabie, H.S., Andrews, G.E. and Ahmad, N.T. Lean low NO_x primary zones using radial swirlers. ASME Paper 88-GT-245, *Proceedings of the ASME 1988 International Gas Turbine Conference, Amsterdam* (1988).
- [9] Alkabie, H.S. and Andrews, G.E. Ultra-low NO_x emissions for gas and liquid fuel using radial swirlers. ASME Paper 89-GT-322. *Proceedings of the ASME International Gas Turbine Congress, Toronto* (1989).
- [10] Alkabie, H.S. and Andrews, G.E. Radial swirlers with peripheral fuel injection for ultra-low NO_x emissions. *Proc. of the ASME IGTI Int. Gas Turbine Congress, Brussels*. ASME Paper 90-GT-102 (1990).
- [11] Andrews, G.E., N. Escott, and M.C. Mkpadi, 2008. "Radial swirler designs for ultra low NO_x gas turbine combustion". *Proceedings of the ASME Turbo Expo 2008, Hamburg, Germany*. ASME Paper GT2008-50406.
- [12] Willis, J.D., Toon, I.J., Schweiger, T. and Owen, D.A., 1993. "Industrial RB211 Dry Low Emission Combustor". *Proc. of the ASME International Gas Turbine Congress, 1993*. ASME Paper 93-GT-391.
- [13] Boyns, M.B. and Patel, R., 1997. "The Application of DLN Technology to the Tornado and Tempest Industrial Gas Turbines". *ASME Turbo Expo 1997, Orlando, USA*. ASME Paper 970GT-059.
- [14] Andrews, G.E., Alkabie, H.S., Abdul Hussain, U.S. and Abdul Aziz, M., 1993. "Ultra low NO_x ultra lean gas turbine primary zones with liquid fuels". *AGARD 81st Symposium of the Propulsion and Energetics Panel of Fuels and Combustion Technology for Advanced Aircraft Engines. AGARD Conference Proceedings* pp.24.1-24.14.
- [15] Agbonzikilo, F.E., Owen, I., Sadasivuni, S.K. and Bickerton, R.A., 2015. "Investigation of Flow Aerodynamics for Optimal Fuel Placement and Mixing in the Radial Swirler Slot of a Dry Low Emission Gas Turbine Combustion Chamber". *Proc. ASME Turbo Expo 2015, Montreal, Canada*. ASME Paper GT2015-43716.
- [16] El-Jumaih, A.M., Abdul Hussain, R.A.A., Andrews, G.E. and Staggs, J.E.J. Impingement Heat Transfer: CHT CFD Predictions of the Influence of Reduced Crossflow using Large Gaps. *Proceedings of the 12th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics*, July 2016. Paper HEFAT2016-1570249450. Proceedings p. 1113-1124.
- [17] Andrews, G.E. and Ahmad, N.T. (2012) Axial Swirler Outlet Shroud Influences on Premixed Combustion NO_x Emissions for all the Combustion Air Passing through the Swirler. *Proc. ASME Turbo Expo, Copenhagen*. ASME Paper GT2012-68642.
- [18] G.E. Andrews, Ultra Low Oxides (NO_x) emissions combustion in gas turbine systems. Chapter 16 In: *Modern Gas Turbine Systems, Ed. Peter Jansohn, Woodhead Publishing*, 2013. pp.715-790.
- [19] Andrews, G.E., Ahmed, N.T., Phylaktou, H.N. and King, P., Weak Extinction in Low NO_x Gas Turbine Combustors. *Proceedings of the ASME International Turbo Expo GT2009*, ASME Paper GT2009-59830.
- [20] Energy Savings Trust, Domestic Heating by Gas: Boiler Systems – Guidance for Installation and Specification. Energy Saving Trust CE30, June 2008.
- [21] Cotton, J. and Wilson, M., A Review of Gas Appliance CO Emissions Regulations. *Advantica Report R4162*, Dec. 2000.
- [22] Eggels, R.L.G.M., Modelling of NO_x formation of a Premixed DLE Gas Turbine Combustor. *Proceedings of ASME International Gas Turbine and Aeroengine Congress and Exhibition*, ASME Paper 2001-GT-69.