Energy from Biomass and the use of Small Direct Fired Gas Turbine Systems

By

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Acronyms

ACS	Air cooled slag
CCP	Central collector pocket
CE	Cement
CRE	Coal Research establishment (UK)
DNC	Declared Net generating Capacity
Fuel A	Austrian wood powder
Fuel B	British wood powder
Fuel C	Swedish wood powder
GHGE	Greenhouse Gas Emissions
IO	Iron oxide
LDA	Laser Doppler anemometry
MSW	Municipal waste
NFFO	Non Fossil Fuel Obligation
NI-NFFO	Northern Ireland NFFO
PFA	Particle fly ash
ROC	Renewable Obligation Certificate
SRO	Scottish Renewables Obligation
VCP	Vortex collector pocket

Abstract

This paper discusses the context for the use of biomass for electricity generation in the UK and similar markets and evaluates the possibility of using cyclonic gasification coupled to small gas turbine systems. In the UK the Government has strongly pushed for a significant increase in the use of renewable energy for electricity generation with only very modest success, nearly 3% coming from this source at present, predominantly hydro and wind. Subsidy for the early tranches of these systems came from an elevated price for generated electricity, but since attempts at price convergence with that pertaining with conventional fossil fuel generation systems has occurred the number of biomass systems being constructed and their net generating capacity has not increased in line with other technologies. Although utilisation technologies exist, and are well proven technologically in Scandinavia, when translated to markets such as the UK, give generating costs which are not competitive with other forms of renewable energy. Problems have arisen with many systems, being predominantly due to fouling/slagging, the different nature of the fuels, and elevated moisture content.

In this context this paper describes an EU sponsored programme of work to develop a simple cyclone gasifier and combustor which can produce a medium calorific fuel gas for materials such as sawdust, retain up to about 80% of the total ash/residues in the system, and fire simple, low cost gas turbines for power generation. The system is shown to have a very wide operating range and can handle sawdust with significant quantities of material up to 4 mm in size, whilst tolerating significant variation in moisture content and capturing very significant quantities of the ash/particulate matter as well as volatile species.

1. Introduction

The context of this paper is the need to develop efficient, low capital cost and small scale power generation systems that can be directly fired on a range of biomass materials, many of which will be grown as energy crops in Europe in this century. The driver for this is the very clear draft EU directive of May 2000. The key features of this proposal are as follows:

Member Sates will be required to adopt national targets for renewables that are consistent with reaching the Commission's overall target of 12% of energy (and 22.1% of electricity) from renewables by 2010; for the UK this translates to 10% of electricity by that date. There are some ambiguities on what is included (i.e. energy from landfill gas and municipal waste (MSW)). The UK government has set its own targets, from a baseline of 1990, of:

- Reducing Greenhouse Gas Emissions (GHGE) by 20% by 2010
- Increasing renewable electricity to 10% of the total by 2010, in line with the proposed EU directive, provided that it is generated at a competitive price.

Think Tanks in the UK have also discussed 60% reductions in GHGE by the year 2050, in order to help slow down climate change.

Reductions of 10% in GHGE have already been achieved on the 1990 figure by the so-called 'dash for gas' and the closure/reduction to part load for coal fired power stations. A moratorium on the construction of new combined cycle gas fired power stations has recently been lifted and this is likely to lead to reduced generation of electricity from coal in the UK by 2010 (Lees, 2001).

The industrial use of coal in the UK has also been discouraged by a climate change levy, introduced in April 2001, which is 0.25Eurocent/kWh for natural gas, 1.95Eurocent/kWh for coal, 1.6 Eurocent/kWh for LPG and 0.71 Eurocent/kWh of electricity used.

Independent analysis indicates that in the UK the largest contribution to reduced GHGE up to 2010 will result from reduced CO_2 emissions from power stations due to improved new station efficiencies and the continuing 'dash for gas', coupled with a steady increase in the contribution from renewables (Lees, 2001), hopefully expected to increase from just under 3% last year to 10% of total generation by 2010. These trends are likely to be reflected in much of Europe, and infer substantial decreases in coal consumption, initial measures including co-firing of biomass, wastes etc. Long term UK view is that no new coal fired power stations can be built after 2020 without CO_2 sequestration.

Turning to the use of biomass and wastes in power stations there are many problems to be dealt with. Technically there are many solutions as shown in Scandinavia, especially Sweden, Finland and Denmark. However it is clear that when these technologies are translated to other markets capital cost are very high, and they are only economic in 'protected markets'. A good example is provided by the recent straw fired power station in Ely in the UK (Loram, 2001), based on Danish technology. Capital costs for a 36MW(e) station are quoted as Euro 3.1Million per MW, with a contract price for electricity of nearly 10 Eurocents/kWh until the year 2013. The UK government has developed it's so called NFFO, NI-NFFO, SRO and now ROC schemes to encourage the use of renewables for electricity generation over many years; the price guaranteed for electricity was very favourable in the early schemes. However since the price paid for electricity under these schemes has been made to converge to that from other sources, the number of new schemes that have actually come on line to generate power from biomass has not increased. Analysis of data from the UK Government (News Review-DTI, 2000) gives the following data:

Declared net renewable capacity (DNC) as at November 2000 from 343 projects: 897.46 MW

Contributions from various sectors

Landfill gas	386.0 MW	(43.01% of total)
Municipal and industrial waste	166.1 MW	(18.51% of total)
Wind	171.34 MW	(19.09% of total)
Biomass-chicken litter/straw etc.	54.8 MW	(6.11% of total)
Hydro	40.36 MW	(4.5% of total)
Sewage gas	25.1 MW	(2.3% of total)
Other	58.2 MW	(6.48% of total)

Landfill gas, Municipal Waste, Wind, hydro schemes are all approaching the situation where there are difficulties in evolving significant numbers of new schemes which will actually be constructed.

Turning to biomass power generation it is clear that if electricity is to be generated from biomass in a manner which allows a close to market price to be acceptable, significant advances must be made in reducing capital costs whilst maintaining system efficiency. This is clear from the above table where Biomass has only made a small impact in the total renewables market, about 6%. The capital cost of a specialist boiler, grate, steam turbine system is very high as shown by Loram, 2001. Long term capital cost reductions of at least a factor of 2 to say 1.5 Million Euros per MW are needed and for smaller systems, say less than 5 to 10MW direct fired gas turbines are attractive, and when a regenerator is used it appears that cycle efficiency for electricity generation approaching 30% may be possible, the same figure as reported in (2) for a much more expensive system. The European Union has recognised this in its last work programme Framework 6. Some discussion on the perceived EU requirements are given in Pilavachi, 2000.

Direct solid fuelled fired gas turbines are of course not new. There have been many programmes on this subject, starting in the 1950s/60s with very simple gas turbines. These programmes were discontinued in the late 1960's with the cheap price of conventional fossil fuels, but were restarted in the late 1970's/1980's with the various fuel price shocks that were encountered at that time. Two programmes need to be discussed here; the pressurised fluidised bed combustion combined cycle system for coal, developed at CRE in the UK, followed by large-scale tests at Grimethorpe, under a number of programmes. This technically was very successful and led to a number of combined cycle plants being constructed, primarily in Scandinavia, with efficiencies of power generation up to and just past the 40% level. However, the high capital costs, the 'dash for gas' and the slow, but steady improvement in conventional pulverised coal power stations lead to little replication. Few problems are reported from these systems, providing appropriate turbine entry conditions are adhered to (~900°C) as well as maintenance schedules. Typical gas turbine requirements for particulate removal have been quoted as removing all material greater than 5 microns and this was achieved by the use of two large Stairmand high efficiency cyclones connected in series (Reed, 1985). Similarly in the 1980's Allison, then part of General Motors, had a well-publicised programme on coal fired gas turbines for trucks, trains and similar vehicles. These gas turbines ran on micronised coal. Little technical information is available in the literature.

This paper thus analyses the performance of an inverted cyclone gasifier for biomass, designed to produce relatively clean fuel gas to fire a suitably modified gas turbine combustor, with the aim of incorporating this into small gas turbine systems. Cyclonic gasification is well established and can be carried out at high velocity, thus giving very compact pressure vessels and low capital costs. It does require a fairly fine fuel such as sawdust or similar (95% <5 mm), but the costs of producing such

fuels are low compared to the extra running and capital costs from pressurised fluidised bed systems. Fluid bed systems are anyway not particularly well suited to very high volatile yield fuels such as sawdust. The concept is thus to gasify the fuel at low temperature (~900°C) to minimise volatilisation of potassium and sodium salts, whilst the provision of collector pockets at the appropriate place on the system can take advantage of the cyclonic action to separate out up to 90% or more of the ash generated, without the extra pressure drop of a parasitic separate cyclone dust separator. The paper describes a family of tests on an atmospheric version of this device and its potential translation to a small gas turbine system. Also described is an associated cyclonic combustor, with integrated ash removal, capable of accepting a wide range of different low calorific value gases, as well as start up fuel.

2. Hot Gas Clean-Up

An important consideration in biomass fired power plants is hot gas clean up. Removal of particles, alkali and salts from the flow is essential to minimise damage to turbine components and in some cases gas cooling is required to achieve suitable turbine inlet conditions.

2.1 Corrosion, Erosion and Deposition in Gas Turbines

Corrosion, erosion and deposition from fuel gases derived from biomass can cause damage to gas turbine components. Particulates can damage turbine blades and gases can cause deposition. As the field of biomass driven gas turbines is relatively new, little information is available of the effect of the biomass gas on the gas turbine system. Some information however on hot gas clean up requirements is available for IGCC plants that use biomass. Engstrom (1998) gives some acceptable values of particulates and emission values of hot gas that can enter their turbine without causing damage, Table 2.1.

Table 2. 1: Gas clean-up requirements

Environmental:	
NO_X	Less than 50 mg NO_2/MJ
SO_X	Less than 25 mg S/MJ
Dust	Less than 10 mg/Nm^3
Gas Turbine:	
Alkali Metals	Less than 0.1 ppm (w)
Dust	Less than 2.5 ppm (w)

Most large scale plants use ceramic filters for gas cleanup. Ceramic filters have performed well especially at low temperatures and Engstrom (1998) describes the use of ceramic candle filters in IGCC cycles. However the use of ceramic filters in the present programme was to be avoided as they would increase pressure drops to the system, decrease cycle efficiency and cause additional maintenance costs.

Sulphur deposition has appeared to be a problem for some natural gas driven gas turbines even when the inlet gas had low sulphur concentrations, Wilkes (1999). Sulphur deposits were found to form around the throat of the choked control valve and continued downstream. Investigation however showed this deposit could be prevented by moderately increasing the gas temperature upstream of the deposit. This could also remove an existing deposit. This sulphur deposition however should not create a problem for this work as the turbine inlet temperatures are in the region of $800^{o}C$, and biomass gas generally has low sulphur content.

An important area of consideration is thermal barrier coatings. These may allow higher temperatures and hence efficiencies and so play an important role in thermal insulation of the components. Leyens et al. (2000) discusses such thermal coatings. Although the system described in this paper uses relatively low turbine inlet temperatures at the moment, future development will demand the use of more efficient turbines and higher inlet temperatures.

2.2 Corrosion and Erosion Damage

Simms *et al.* (2000) developed a methodology for the measurement of corrosion and erosion damage on gas turbine material. The amount of acceptable corrosion and erosion determines the life of the component. Assessment of the materials performance is made in five stages, and covers pre-exposure contact and post-exposure optical/image analysis. This method also allows determination of metal loss, depths of internal corrosion pit depths, as well as taking into consideration the requirements of the material. Tests taken allowed statistical analysis of the material's degradation to be made and enabled models of material performance to be developed. Tests with a standard gas turbine material (IN738LC) at $700^{o}C$ are shown in Figure 1. They show maximum metal damage with different deposit compositions.
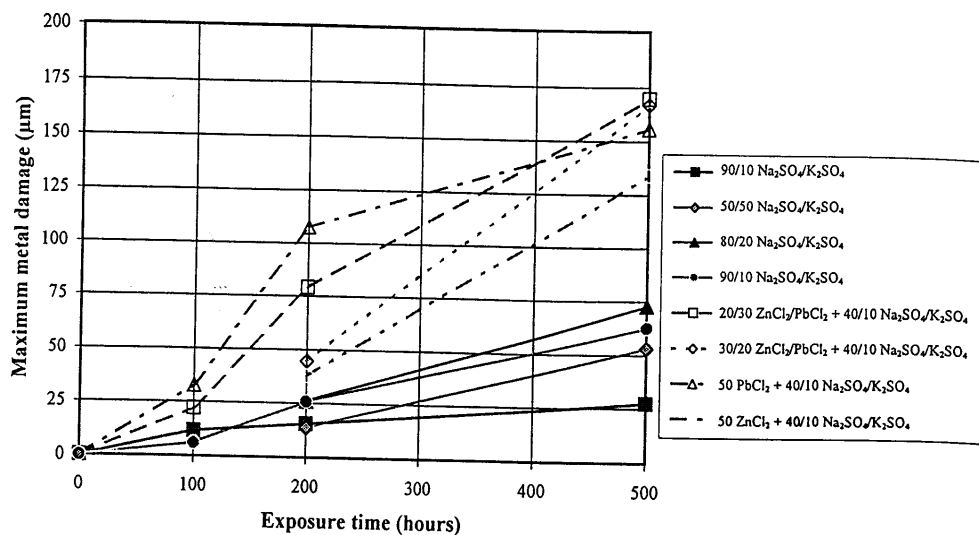

Figure 1: Effect of different deposit composition on the performance of IN738LC at $700^{o}C$, Simms et al. (2000)

Neilson (1998) gave specifications for acceptable contaminant concentrations in the fuel for General Electrics LM2500 gas turbine. Table 2.2 shows that the turbine is not very tolerant of alkali metals, as tests showed premature corrosion of turbine blades running in high alkali environments.

Table 2.2: Acceptable contaminant levels in fuel, *Neilson (1998)*

	Maximum concentrations in combusted mass flow to the turbine ppb	Calculated maximum allowable concentrations in a typical biofuel ppb
Solids		
$d < 10$ μm	600	3000
$10 < d < 13$ μm	6	30
$13 < d$ μm	0.6	3
Lead	20	100
Vanadium	10	50
Na + K + Li	4	20
Calcium	40	200
Sulphur		
Alkali metal sulphates	12	60

2.3 Alkali and Heavy Metal Removal

Alkali and heavy metal removal from gas flow is of particular importance to reduce corrosion and/or erosion of turbine components. Biomass gas contains alkalis formed primarily from Potassium and Sodium in the feedstock. Tolerable concentration of alkali vapours for gas turbine applications is very low. Alkali removal from biomass gas is not well documented. The gasifier exit temperature has a large influence on the alkali concentration. At relatively low gasifier exit temperatures the alkalis appear to condense on particulate matter. Thus the alkalis can be controlled by controlling particulates, Williams and Larson (1996). Fixed bed gasifiers keep exit temperatures between 500°C and 600°C and hence avoid problems of vaporised alkalis and condensed tars. Fluidised bed gasifiers however have problems of gas quality control due to the substantially higher exit temperature. Alkalis are in the vapour phase and could require fuel gas cooling to condense the alkali. Salo and Mojtahedi (1998) measured vapour-phase alkalis after the gas cooler in an (air-blown) Integrated Gasification Combined Cycle system. A pressurised fluidised bed gasifier was used with varying biomass fuels. The alkali content of the product gas was reduced to 0.05-0.1 ppm (wt) when the gas is cooled and filtered at 400-500°C. These values are below the specification for their industrial gas turbine. Temperature of the gas exiting the gasifier should be high enough so tars are in the vapour phase. These tars are then burnt in the secondary combustor, and can boost the heating value of the gas. Many biomass power projects use fluidised bed gasifiers and require expensive gas cleaning techniques. Lulea University of Technology studied biomass powder gasified in a combine cyclone separator and gasifier operated at low temperatures as an attempt to avoid excess amounts of alkali compounds in the producer gas. The conditions are such that the volatilisation of corrosive elements such as sodium and potassium may not occur. These compounds should remain solid in char particles. Fredriksson (1999) indicated that alkali separation in cyclone gasification of wood powder was 40-60%. For fluidised bed gasification inconclusive results for alkali retention were not available in the literature, therefore Gabra *et al.* (2001) made a comparison of alkali retention and separation in fluidised bed and cyclone gasifiers. Bagasse biomass was used for the gasification experiments. For cyclone gasifiers stable wall temperatures were found to be in the range of 650°C to 950°C with a fuel equivalence ratio of 0.25 to 0.21. Experiments were taken for the same type of bagasse and gasified at 850°C. Alkali separation, or

effectively retention by the bed, of approximately 70% of the input alkali was found with an equivalence ratio of 0.25, Figure 2. The fluidised bed showed on average 12% retention after 1.4 hours of operation which dropped to 4% for an operation time of 12 hours. The carry-over of potassium and sodium to the product gas is considerably less for a cyclone gasifier than a fluidised bed gasifier and gives major operational advantages.
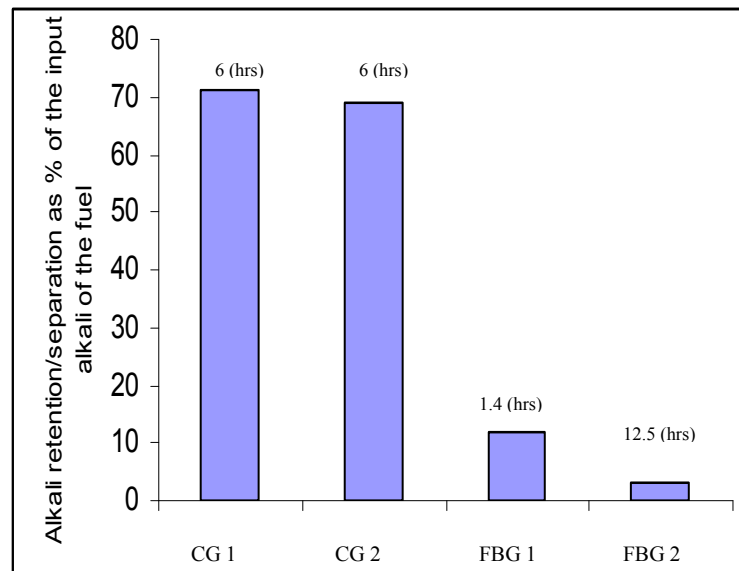

Figure 2: Comparison of alkali retention / separated with char samples from Bagasse gasification in FBG/CG, Gabra (2001)

The difference in the performance between the cyclone and fluidised bed gasifiers is explained as cyclone gasifier char is continuously separated and removed, and the ash included in this char carries significant amounts of the potassium and sodium. The residence time is much lower for cyclone gasifiers, up to 1 second than the minutes in a typical fluidised bed. In the fluidised bed gasifier no solid material is removed from the bed except the particles entrained in the product gas flow. Gabra et al. (2001) also carried out a chemical equilibrium model analysis that showed that the performance of alkali separation and retention can be expected to increase for elevated pressure compared to atmospheric gasification tests. Figure 3 shows the effect of pressure on various alkali compounds.
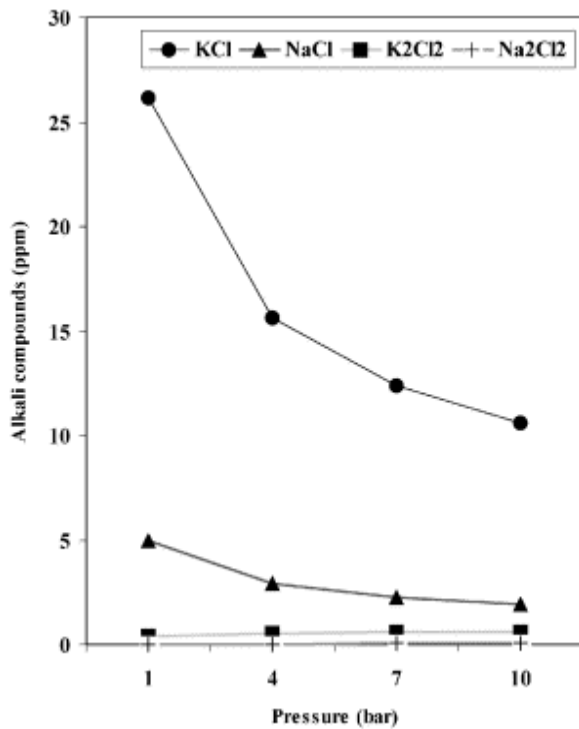
Figure 3: Bagasse Gasification –Alkali Concentration (ppm) with pressure, Gabra et al. (2001)

The equilibrium calculations show that under gasification with equilibrium conditions the dominant alkali gas species are monochlorides KCl (g) and NaCl (g) and, to a lesser extent dichlorides, K_2Cl_2 (g) and Na_2Cl_2 (g). Formation of K and Na and hydroxides (KOH (g) and NaOH (g)) was an order of magnitude less. The calculations showed that the gas alkali phase in the producer gas from bagasse gasification increased with temperature or with a decrease in the equivalence ratio, Fig4.

M. Gabra et al. / Biomass and Bioenergy 21 (2001) 461–476
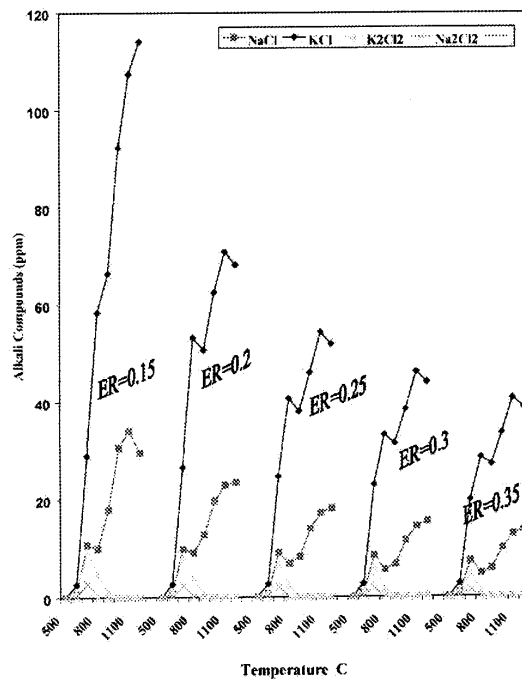
Figure 4: Variation of alkali compounds in the producer gas from gasification of bagasse with temperature and equivalence ratios, Gabra et al. (2001)

Figure 5 shows the variation of the alkali phase with temperature and at different equivalence ratios. At temperatures between 500°C and 600°C the alkali is in the solid phase. For the gasifier operation temperature of 800-900°C the alkali was predominately in the liquid phase. As the temperature is raised the alkali phase increasingly becomes gaseous. Experiments showed that for the range considered the alkali phase is not greatly affected by the equivalence ratio. However equilibrium calculations over a greater range showed that the liquid alkali phase increases with increasing equivalence ratio and the gaseous phase decreases.
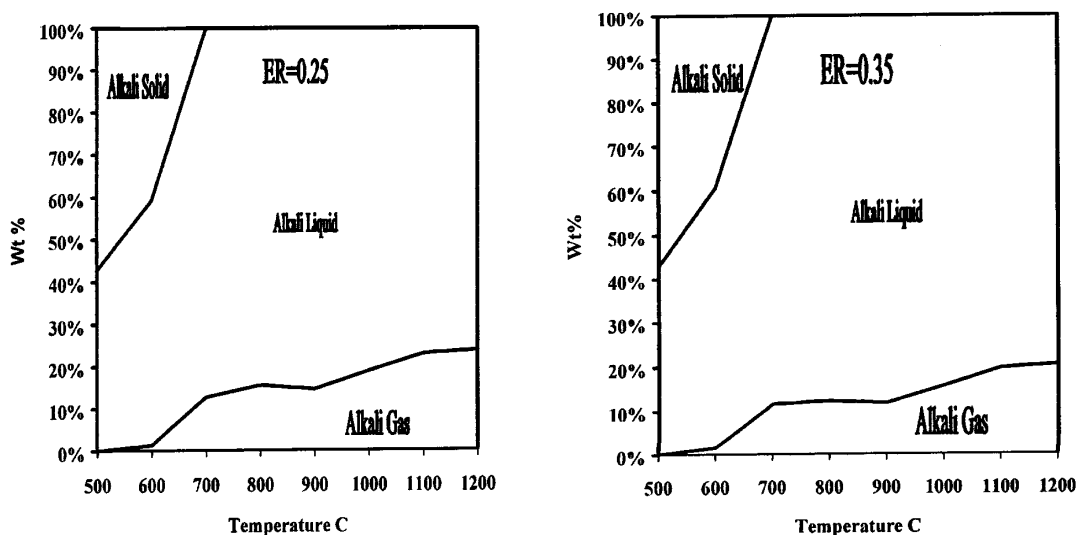

Figure 5: Variation of alkali phase with temperature for bagasse gasification at 1 bar, Gabra et al. (2001)

Gasification trials were carried out at Cardiff University and determined the separation of alkalis through an inverted cyclone gasifier fired on two types of pulverised biomass using Plasma Excited Alkali Resonance Line Spectroscopy (PEARLS) developed at Tampere University of Technology. These results are shown in Section 3. Alkali release was larger for higher airflow rates caused by higher reaction temperatures experienced by fuel and ash particles. The potassium concentrations found in the exhaust gas ranged between 6.7 and 20.6 ppmV (by volume) and the sodium was between 1 to 3.2 ppmV. The retention rates of the alkali's were independent of the fuel type and were identical for most cases, around 50% for both potassium and sodium.

3 Cyclone Gasification and Combustion.

Studies have shown that research into the use of biomass fuels for electricity production is needed. These studies have indicated that direct fired wood powder gas turbines are economically attractive for co-generating power plants up to 20 MW. Such plants incorporate a gas turbine and steam turbine in series to maximise the electricity output (Fredriksson, 1996). However problems are encountered with erosion from particles and deposition of alkali compounds, which could cause damage to the turbine blades as discussed in section 2. These problems can be increased by high inlet temperatures, which are desired to increase cycle efficiency. Direct fuelling of a gas turbine with wood powder (Hamrick, 1991) however showed no erosion or corrosion but severe deposits at inlet temperatures greater than 800°C. A possible solution to this process is the two-stage combustion of wood powder where the first stage is a cyclone gasifier/separator. In the cyclone the volatile and some of the fixed carbon are oxidised at low temperatures. The corrosive ash remains solid in the char particles

and are separated from the flue gas in the cyclone. The wood powder can also be injected into the cyclone with steam as transport media. The exhaust gas can feed directly into the modified secondary cyclone combustor. The cyclone combustor will produce stable strongly swirling flow that will enhance mixing and burnout rates. A schematic of the system is shown later in Fig 7.

Separation and retention of ash is enhanced using a device termed a vortex collector pocket (VCP) and evolved from cyclone separator work (Biffin, 1984, Biffin and Syred, 1985 and Syred et al., 1986). In these devices the main cyclonic flow drives a weak subsidiary flow in a small auxiliary vortex chamber, with no net gas exchange, which slightly overlaps the main cyclone chamber. Particulates are concentrated into a thin boundary layer region. On encountering the VCP, the particulate matter passes directly into the VCP, where it rapidly decelerates due to wall friction and a change of direction where after it drops out into bins. VCP's are used in both the gasifier and combustor.

3.1 Gasifier Design

A novel vertically orientated, inverted cyclone gasifier was designed at Cardiff University to gasify pulverised biomass. The design was based on a combination of principles including those involved in the general design of dust separators, where high angular momentum fluxes are used to separate dust suspensions and those involved in cyclone combustor design via the use of internally generated high centrifugal force fields, secondary flows and well stirred reactor conditions to mix fuel and air which promote good gasification characteristics (Morgan, 1990).
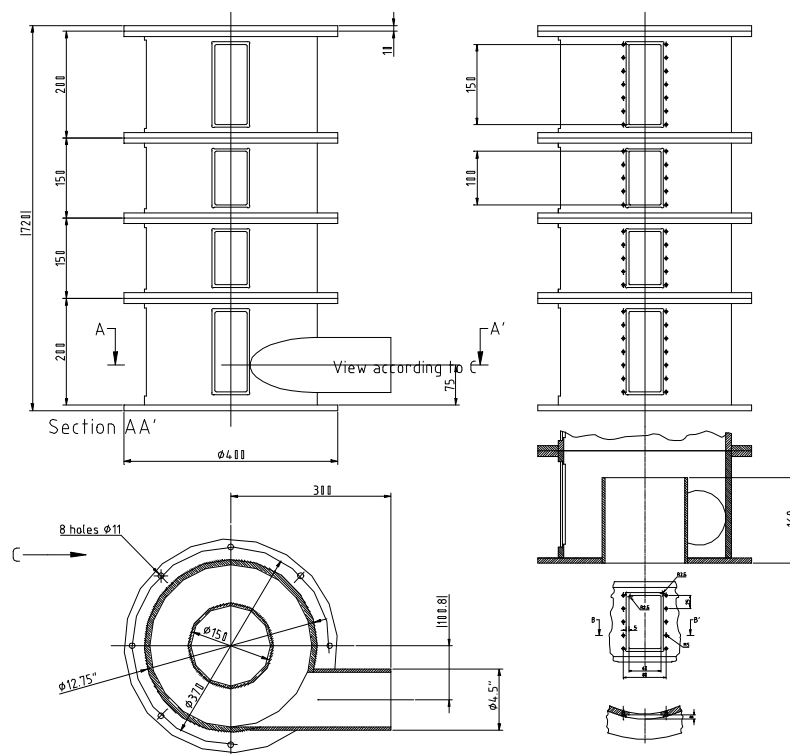

Fig. 3.6 3D View of Cyclonic Gasifier

A 2 dimensional view of the test rig is shown in Figure 6. The main cyclone gasification chamber consists of 4 sections each with 2 rectangular access ports, covered with plates to allow access, for e.g. optical measurement techniques or thermocouples and sample probes. The bottom main section includes the tangential inlet and the round axial outlet to the attached exit section. The geometric swirl number for the present design with no inserts (restrictions in the tangential inlet) is 3. However, with the use of inserts the geometric swirl number can be varied from 3 to 7.5. The exit- and ash bin section are attached to the main chamber as highlighted in Figure 7. Two vortex collector pockets (VCP), as developed in principle by Biffin (1984), and one central collector pocket (CCP) are incorporated to provide the necessary ash and dust removal capability. One vortex collector pocket is attached to the top main section and a second vortex collector pocket is incorporated in the exit section, attached to the central collector pocket leading to the ashbin section. A section attached to the tangential inlet allows the premixing of natural gas before it enters the gasification chamber (Figure 7). This section had a venturi tube before the natural gas inlet and a pressurised air supply. An additional air supply from a fan can be used via a secondary tube attached to the inlet section. The spark ignition system is mounted at the intersection of both inlet tubes to provide an ignition source for premixed combustion conditions independent of the air supply selected. A hopper was attached to the venturi section to enable the input of wood powder for the gasification tests or inert particles for the separation trials. Wood fuel was fed to the system using a hopper attached to a screw feeder driven by an electric motor. Wood was dropped onto a vibrating table that provided a constant and reliable fuel feed rate.

K-type thermocouples were positioned geometrically opposite in the centre of each gasification section and at the exhaust. The position of the thermocouples is denoted using the non-dimensional position z/D_e, where z represents the axial distance from the gasifier outlet/ exit ($z = 0$ is at the gasifier exit rim) and D_e is the gasifier exit diameter. The notation "IN" is for the thermocouples situated at the inlet side of the gasifier and "OUT" are for the thermocouples at the outlet/exhaust side. K and R type thermocouples were used for the exhaust gas temperature for calibration purposes.
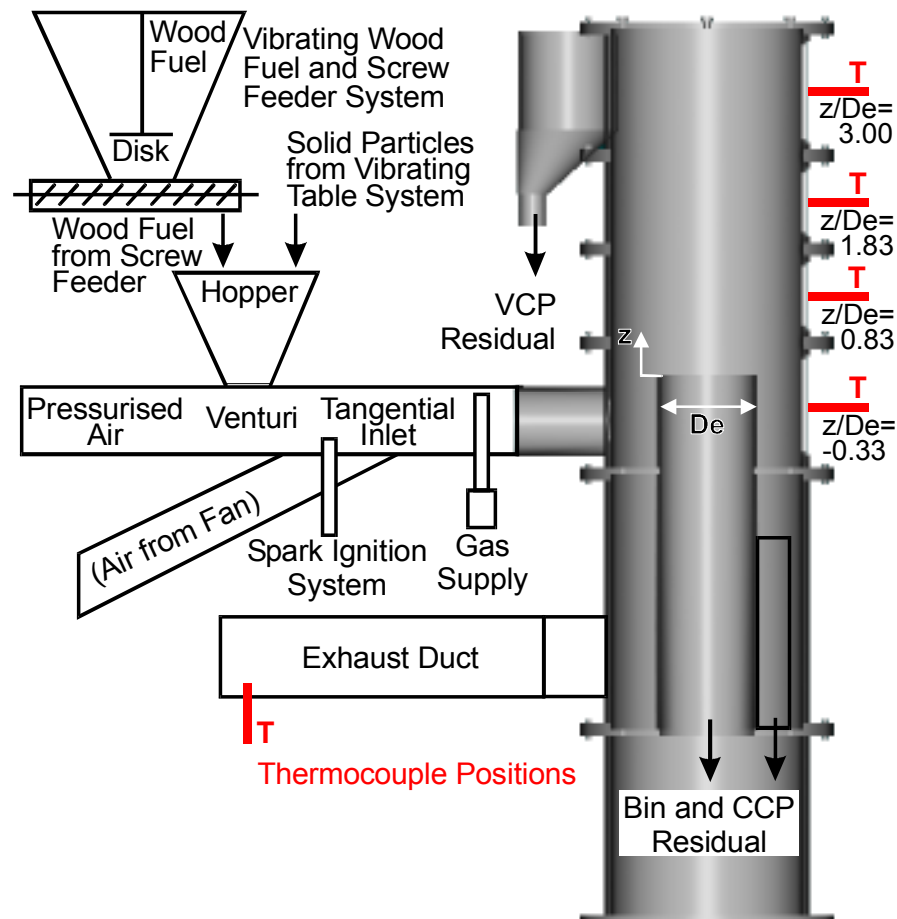
Fig 7: Experimental Set Up of Cyclonic Gasifier for Separation and gasification Trials

Typically 1000 samples or more were recorded at 1 sample per second. Figure 8 thus shows a record of wall temperatures during the warm up phase

Detailed velocity, temperature and pressure measurements were taken to characterise the cyclone gasifier under isothermal and combusting conditions. These results are reported in detail in Fick et al. (1999a and b). Gasification trials with a number of different fuels were undertaken to fully characterise the system, these results are reported in the following section.

3.2 Separation Tests and Gasification Trials

The cyclone gasifier was warmed up using premixed natural gas with a mixture ratio of 1. This enables sufficient temperatures to be reached to allow gasification of the wood. Operating experience showed an exhaust temperature of approximately 600°C is sufficient to start gasification. This occurred after 10 minutes (Figure 8). The natural gas is then turned off, the air flow rate is turned down and the feeding of the wood fuel begins.

Figure 8: Warm-Up of Cyclone Gasification Chamber
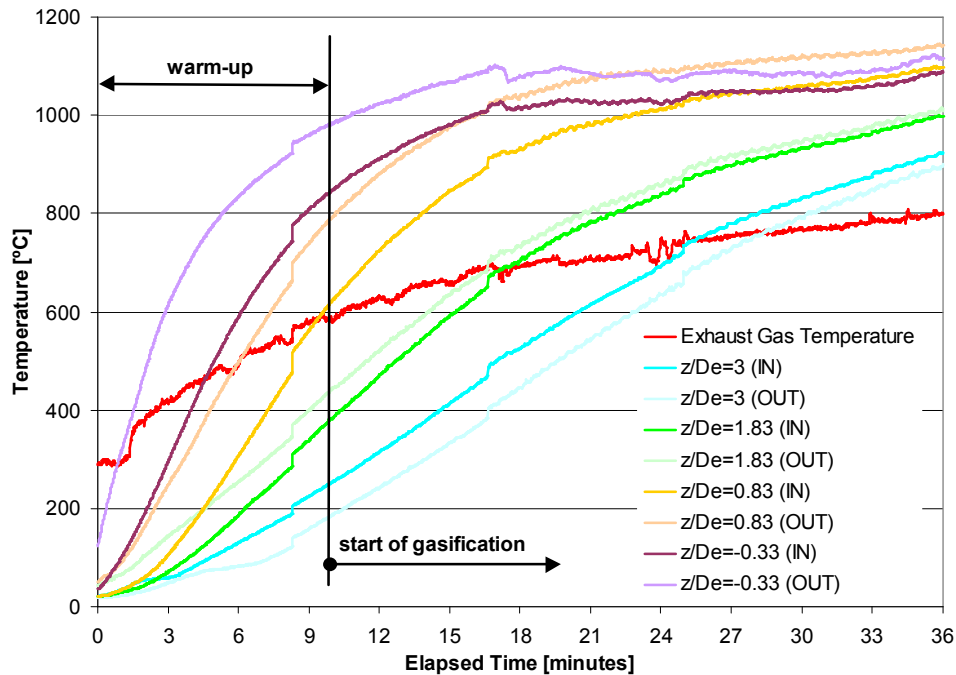
The temperature of the section of the gasifier at the position $z/D_e = -0.33$ is heated to about 900^oC, the section at $z/D_e = 0.83$ to about 700^oC, the section at $z/D_e = 1.83$ to about 400^oC and the section $z/D_e = 3.00$ to about 200^oC.

3.2.1 Separation Trails under isothermal conditions

The separation behaviour of inert particles, chosen to match the conditions for gasification, was studied in isothermal flow. Four types of particles were used, particle fly ash (PFA), cement (CE), air cooled slag (ACS) and Iron oxide (IO). Feed rate, size and absolute density (weight) were chosen to match the conditions for gasification in an isothermal flow. The particles were fed to the cyclone using the vibrating table and hopper. The particles separated by the vortex collector pocket (VCP) and the central separator pocket (CCP) was collected. The collection efficiency as a function of primary flow rate was determined. Three different flow rates were used (Re=15027, 30055 and 45082) at a geometrical swirl number of 5.1 for the four particle types. The size distribution of the particles was obtained from sieve tests. The grade efficiency curves for the inert particles are shown in Figure 9. The largest and heaviest particles are the iron oxide (absolute density ρ_a of 8.3g/ml). The air cooled slag are the second heaviest particles with ρ_a =3.39g/ml. The cement and PFA have a similar size distribution with ρ_a =3.24 and 2.29 respectively and a d_{50} of less that 63μm. The size distribution and absolute density of all inert particles and all particulate matter collected in the VCP and CCP under gasification conditions shows that the inert particles chosen for the isothermal separation trials realistically represent gasification conditions in terms of particle flow.
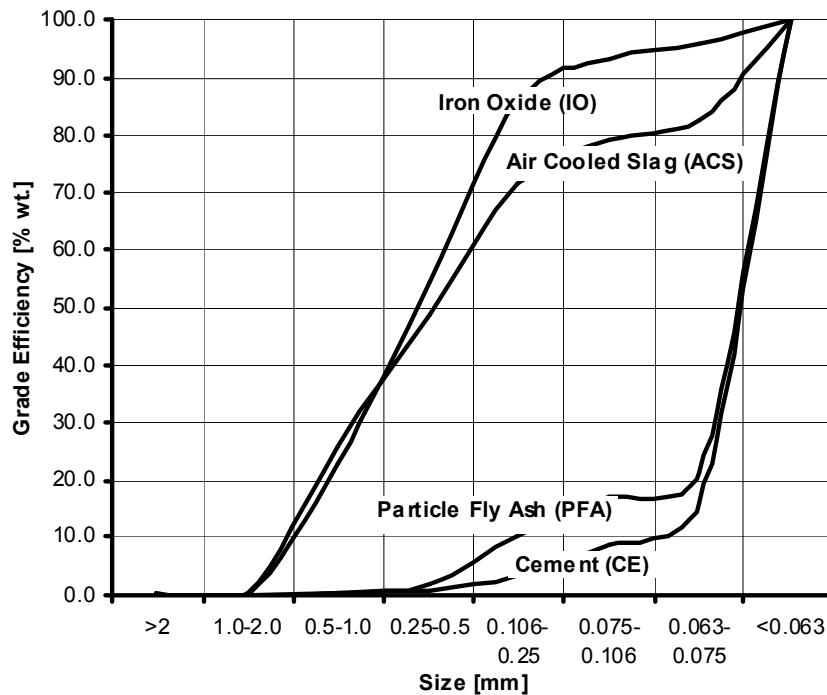
Figure 9: Grade Efficiency Curves for Inert Particles

Isothermal velocity measurements (Fick et al. 1999a) found that the minimum losses of tangential velocity at the cyclone inlet and the best conservation of tangential velocity in the main cyclone body were achieved with a geometrical swirl number of 5.1 regardless of the inlet flow rate. Thus only this configuration was in this work, resulting in tangential velocity of up to 20% higher than the inlet velocity.

3.2.2 Gasification Trials

3.2.2.1 Fuel size distribution

Experimental studies were carried out on three different fuels:

a) Fuel A: Commercial Austrian sawdust

b) Fuel B: Commercial British sawdust

c) Fuel S: Commercial Swedish wood powder

Mean size distributions of the fuels were determined using sieves of different sizes. Figure 10 shows the size distributions of the three fuels.
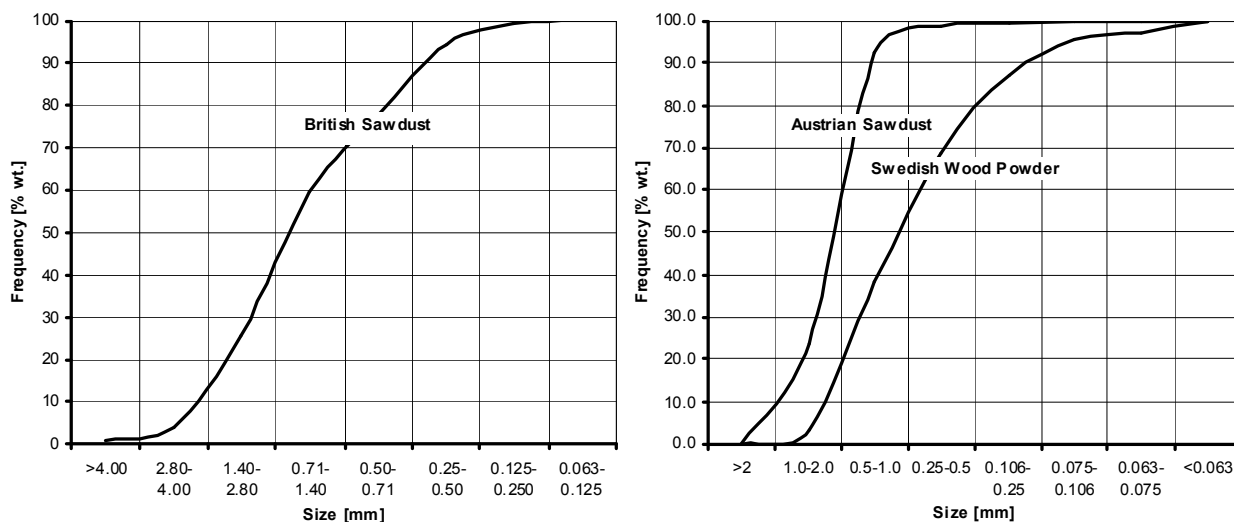

Figure 10: Size distribution of saw dust (A) British (B) Austrian and Swedish (S)

Fuel S contains the smallest particles with a wide distribution ranging from 2 down to 0.063 mm. The average size was found to be 0.5mm. Fuel B has an average size of 1 mm with a reasonably equal proportion of fine and coarse particles. Fuel A has a similar average size as Fuel B however the size distribution is very different with 70% of all particles in the 0.5-1 mm range. This results in a slightly unstable start-up condition for Fuel A when the fuel is first introduced to the rig and the gas is turned off. This is most likely due to the finer particles are initially needed to quickly provide the combustion heat for stable start-up operation. With the absence of finer particles more time is needed to change the wall temperature patterns of the cyclone from preheat to gasification processes.

3.2.2.2. Experimental conditions

A large number of tests were taken to characterise the cyclone gasifier. These tests used a wide range of wood powder feed rates and equivalence ratios and resulted in a range of exhaust gas temperatures. All tests resulted in stable steady conditions. A detailed investigation of four conditions for trial VIII are shown in the following section. Table 4 shows the gasification conditions investigated.

Table 4: Gasification Conditions

Trial No	VIII-a	VIII-b	VIII-c	VIII-d
Fuel Type	Fuel A	Fuel A	Fuel S	Fuel S
Total Air Flow Rate [l/min]	550	660	550	660
Wood Fuel Feed Rate [kg/h]	26	37	26	40
Equivalence Ratio [-]	0.243	0.205	0.243	0.190

3.2.2.3 Exhaust Gas Temperatures and Wall Temperature Distributions

A large number of trials were carried out with fuel A and S at different mixture ratios to characterise the cyclone gasifier. The exhaust gas temperatures are shown in Figures 11 a and b as a function of the primary air and equivalence ratio respectively.
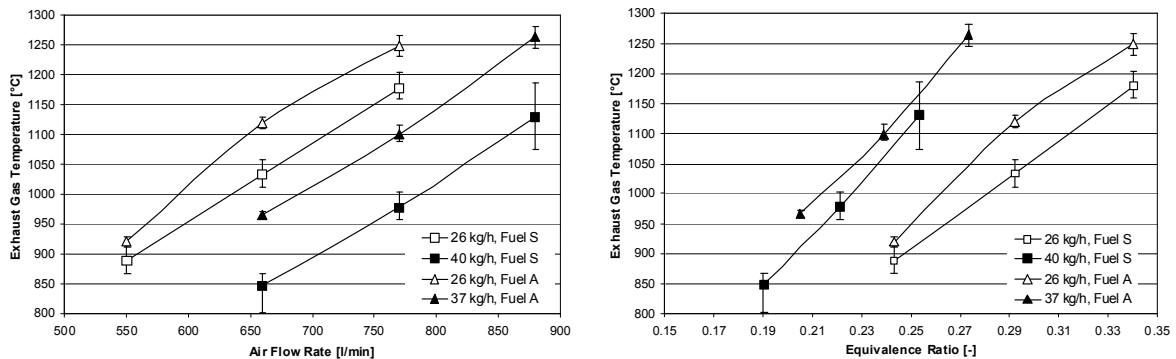

**Figure 11: Exhaust gas temperatures as a function of (a) air flow rate (b) Equivalence ratio**

The exhaust gas temperature increases from 850°C to 1250°C as the primary air flow rate is increased from 550 to 880l/min (the lowest exhaust gas temperature achieved at the smallest equivalence ratio). Exhaust gas temperatures above 1000°C do not represent realistic gasification conditions, regardless of the mixture ratio. Thus the lowest equivalence ratio at any feed rate is selected for further investigation. Exhaust gas temperatures were generally slightly higher for fuel A but fluctuations were higher with fuel S. This was due to fuel S being composed of smaller particles that are more reactive and hence release heat for the gasification reaction more quickly. The lowest equivalence ratios were obtained for higher fuel feed rates (Figure 11 b). Wall temperatures increased as the equivalence ratio was increased. This effect was more pronounced at lower fuel feed rates and not present at all for fuel S at 40 kg/h. The inlet and exhaust sidewall temperature distribution are different with cyclic variations of about 50°C. However, local fluctuations of wall temperatures were very low.

3.2.2.4: Velocity Distribution under Gasification Conditions

Detailed experimental characterisation of the flow pattern within the cyclone gasification chamber were undertaken by Fick *et al.* (1999a, b). Two dimensional LDA measuring techniques were used. The isothermal and combusting flow patterns were shown to be generally independent of Reynolds and swirl number, however, negative or positive peak values were affected. It was found that regardless of the inlet flow rate, minimum losses of tangential velocity at the cyclone inlet and the best conservation of tangential velocity in the main cyclone gasification chamber were obtained for a geometrical swirl number of 5.1. The LDA data for axial and tangential velocity distributions measured by Fick *et al*., for isothermal and combusting (natural gas) conditions, was used as guides for the velocity distribution to be expected for the relevant gasification cases, Figures 12 a and b.

Figure 12 c shows the measured temperature distribution under combustion conditions (the system was probed with fine wire thermocouples). An air flow rate of 1900 l/min and a gas flow rate of 100 l/min (mixture ratio ~ 2) were chosen to simulate the conditions for the gasification trials at realistic wall temperatures of around 850°C. With a Reynolds number of 30055 in the combusting case (Figure 12 c) and an inlet velocity of 6.17 m/s (for a total inlet flow rate of 2000 l/min), the Reynolds number for the relevant gasification conditions at about 650 l/min decreases to 12755 based on an inlet velocity of 2 m/s for Figure 12 a and b. Figure 12 a and b highlight the axial and tangential velocity distributions for these conditions.

Figure 12: Mean (a) Axial and (b) Tangential velocity contour plots for Re=12755 under gasification conditions (calculated) and (c) average temperature distribution for Re=30055 across the cyclone under combustion conditions (measured)
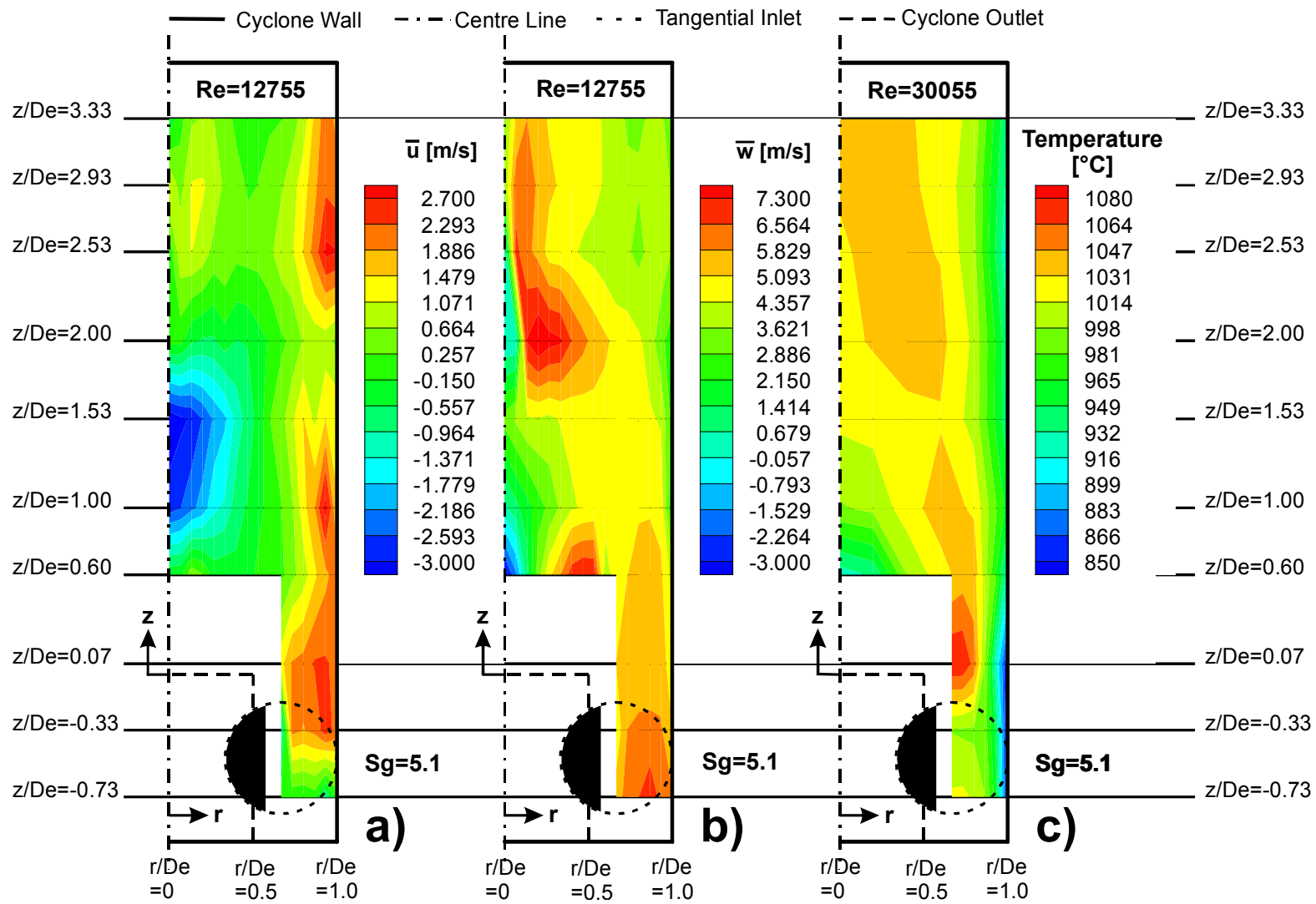
3.2.2.5 Energy Balance

An energy balance was performed for the four test cases to evaluate thermal efficiency and energy losses involved in the process. The summary of the energy balance, static and dynamic inlet and outlet pressures, input and output flow rates and thermal energy is found in Table 5.

Table 5: Summary of Energy Balance

	Fuel A	Fuel A	Fuel S	Fuel S
Experimental Condition	**VIII-a**	**VIII-b**	**VIII-c**	**VIII-d**
Fuel feed rate [kg/h]	26	37	26	40
Energy from fuel [MJ/h]	422	601	457	703
Thermal input [kW]	117	167	127	195
Air flow rate rotameter reading [l/min]	500	600	500	600
Air flow rate correction factor [-]	1.2	1.2	1.1	1.1
Air flow rate corrected [l/min]	600	720	550	660
Equivalence Ratio theoretical [-]	0.2653	0.2237	0.2431	0.1724
Inlet Velocity [m/s]	1.848	2.218	1.694	2.033
Static pressure inlet cold [Pa]	100	160	100	160
Pressure difference exhaust hot [Pa]	38.0	53.5	42.0	55.0
Exhaust gas temperature [°C]	819	915	818	842
Exhaust gas velocity hot [m/s]	7.80	9.44	7.88	9.60
Exhaust gas velocity cold [m/s]	16.6	19.3	16.8	19.5
Calorific value exhaust gas [MJ/m^3]	0.78	2.99	4.26	5.91
Losses [%]	**64**	**41**	**30**	**24**
Efficiency [%]	**36**	**59**	**70**	**76**

The highest calorific value was achieved with trial VIII-d. The exhaust gas composition included 52% by volume of nitrogen and 13% by volume of carbon dioxide. The products of gasification included methane, carbon monoxide and hydrogen add up to the remaining 33%. The losses for trial VIII-d are the lowest at 24%, followed by trial VIII-c at 30%, then trial VIII–b at 41% and trial VIII-a at 64%. The higher calorific values and lower equivalence ratios for the fuel S result in lower losses. At a lower equivalence ratio condition VIII-b could certainly reach higher efficiencies. However conditions VIII-a and VIII-c are approaching the lower operating limit of the gasifier where the percentage of input fuel needed to heat the gasifier gets too high and subsequently the losses increase. Trial VIII-c represents the highest efficiency possible for such a low load condition. Table 6 shows the exhaust gas composition for the four cases.

Table 6: Exhaust gas composition

Sample	Units	Fuel A	Fuel A	Fuel S	Fuel S
Rate of Feed	Kg/h	28	40	40	26
Temperature	^{o}C	819	915	842	818
Hydrogen	%v/v	3.71	7.14	8.56	9.06
Oxygen	%v/v	1.97	1.69	1.48	1.17
Nitrogen	%v/v	68.34	60.33	52.09	56.87
Methane	%v/v	1.06	1.95	3.31	1.95
Carbon Monoxide	%v/v	7.05	12.08	18.18	15.37
Carbon Dioxide	%v/v	16.47	14.55	12.63	13.39
Acetylene	%v/v	0.2	0.4	0.8	0.4
Ethylene	%v/v	0.2	0.6	1.2	0.4
Nitrogen Dioxide	Vpm	<0.5	<0.5	<0.5	<0.5
Ethane	Vpm	1.3	2.5	2.6	5.9
Propane	Vpm	124	210	900	101
Hexanes	Vpm	4	11	40	6
Propylene	Vpm	88	188	690	65
Butenes	Vpm	<2	6	15	<2
Sulphur Dioxide	mg/m^3	1.6	3.8	3.1	4.4
Water content	g/m^3	178	199	207	180

3.2.2.6 Elemental/Proximate Analysis and Separation Performance

The separation of ash from the flow is very important to minimise damage to the turbine and additional hot gas clean up for use of the gas in the gas turbine. The VCP and CCP were designed to remove these particles from the flow as described in section 3.2. To determine separation/retention performance of the VCP and CCP a reference source to the ash was produced by forming 100% burnout ash from each fuel. This was done by burning a set amount of fuel in a stainless steel pot. The size of the 100% burnout ash for fuel A was typically 3 times smaller than the fuel and 5 times smaller for fuel S, Figure 13. The ash collected in the VCP and CCP was generally larger than the 100% burnout ash. For case VIII-d the CCP residual is of very similar size to the fuel with some 10% of the particles bigger than 2mm. As fuel S contains no particles larger than 2 mm (Figure 10 b) there is clear indication that agglomeration of the particles is occurring. The axial and tangential velocity distribution (Figure 12 a & b) can explain this occurrence. Fuel particles move upstream and gasify close to the cyclone wall from r/De=0.7 to r/De=1.0. The particles not being separated by the VCP situated at r/De=1.0 and z/De=3.33 join the downstream flow which leads into a reverse floe region from r/De=0 to r/De=0.5 and z/De=1.53 to z/De=1.00. With low tangential velocities also in this region, low shear stresses are present assisting agglomeration in this region. This effect is however beneficial to the separation performance since the larger particles are more likely to be separated by the CCP collector pocket situated further downstream. Although particles collected in the VCP are

generally smaller than the CCP ones, they are typically 30% to 70% heavier. This is because the centrifugal force field in the VCP region of the flow had tangential velocities identical to the inlet air velocity; the heavier particles are more likely to be separated. The remaining lighter particles in the flow are then separated in the CCP although heavier particles not captured by the VCP are again more likely to be separated.
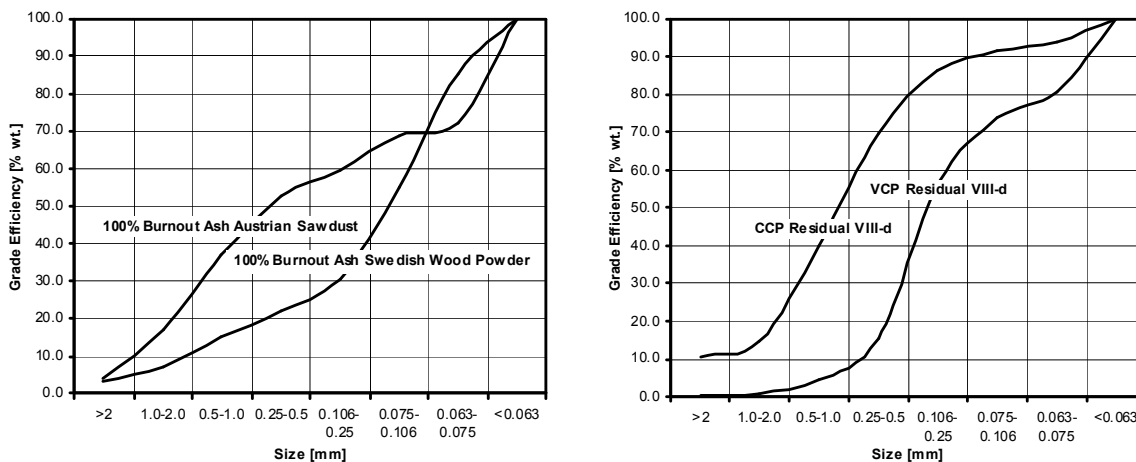

Figure 13: Grade efficiency curves for (a) 100% burnout ash fuel A and fuel S (b) VCP and CCP residual trial VIII-d

The VCP and CCP collected different percentage of the retained ash for each separate trial, highlighting the importance of both VCP and CCP collector pockets to provide good separation performance for a wide range of fuels and operating conditions.

The overall gasification performance of the rig was characterised. The carbon burnout was calculated for the fuels and 98 to 99% burnout was achieved (Table 7.). The variations in efficiency that occur are not due to incomplete fuel carbon extraction during the gasification process but due to burn out effects of the gas produced as a result of the excess oxygen present (high equivalence ratios). This was demonstrated by comparing the volatile and fixed carbon content of the fuel with the one of the VCP and CCP ash. A volatile content of 75/77% wt. for fuel A/S reduces down to typically 0.2% wt. in the VCP and CCP ash related to the total fuel input. A slightly higher percentage of fixed carbon was detected in the VCP and CCP residual which when compared to the total input was also very small. As to be expected, the non-combustible content of the fuel was mainly found in the VCP and CCP ash varying from 46% to 99% as shown in Table 7.

Table 7: Elemental/Proximate Analysis and Separation Performance

Experimental Condition	VIII-a	VIII-b	VIII-c	VIII-d
Fuel Feed Rate [kg/h]	26	37	26	40
Fuel Type	Fuel A		Fuel S	
Absolute Density [g/ml]	1.59		1.65	
Calorific value - Gross (upper)	17.480		18.840	
Calorific value - Net (lower)	16.249		17.573	
100% burnout ash	From Fuel A		From Fuel S	
Absolute Density [g/ml]	4.91		4.73	
Volatiles [% wt.]	75.300	75.300	77.100	77.100
Fixed Carbon [% wt.]	12.400	12.400	14.700	14.700
Ash [% wt.]	0.400	0.400	0.700	0.700
Non combustibles [% wt.]	0.350	0.350	0.600	0.600
Moisture [% wt.]	11.900	11.900	7.500	7.500
Ash [% wt.]	0.350	0.350	0.600	0.600
VCP percentage of total ash collected [% wt.]	37.650	62.795	52.764	29.350
CCP percentage of total ash collected [% wt.]	62.350	37.205	47.236	70.650
Absolute Density of ash collected in VCP [g/ml]	2.680	2.660	2.610	3.490
Absolute Density of ash collected in CCP [g/ml]	1.940	2.080	2.630	2.030
Particulate emission exhaust gas [g/s]	0.146	0.058	0.129	0.044
Total volatiles CCP & VCP [% wt. of fuel feed]	0.330	0.195	0.151	0.153
Total fixed carbon CCP & VCP [% wt. of fuel feed]	1.604	0.712	1.171	1.713
Total moisture CCP & VCP [% wt. of fuel feed]	0.371	0.171	0.359	0.253
Total non combustibles CCP & VCP [% wt. of fuel feed]	81.566	99.780	77.416	46.223
Carbon Burnout [% wt.]	98.396	99.288	98.829	98.287
Mass balance Na (out/in)	0.578	0.681	0.839	0.645
Mass balance K (out/in)	0.676	0.696	0.617	0.553
Na CCP & VCP retention rate [% wt.]	43.706	58.936	65.742	58.244
K CCP & VCP retention rate [% wt.]	49.997	54.752	46.578	48.516
Al CCP & VCP retention rate [% wt.]	6.724	10.372	50.869	41.925
Ca CCP & VCP retention rate [% wt.]	72.543	67.509	45.354	38.505
Mg CCP & VCP retention rate [% wt.]	45.574	44.358	47.347	45.698
Mn CCP & VCP retention rate [% wt.]	92.988	100.494	65.544	44.802
P CCP & VCP retention rate [% wt.]	63.505	90.380	47.531	44.932
Si CCP & VCP retention rate [% wt.]	24.686	40.031	63.410	66.609

The collection efficiency for Al, Ca, K, Mg, Mn, Na, P and Si with respect to the input of those elements evaluated from the elemental analysis of 100% burnout ash for fuel A and fuel S is shown in Figure 14. The results are presented independent of the fuel feed content, i.e. 100% burnout ash content.
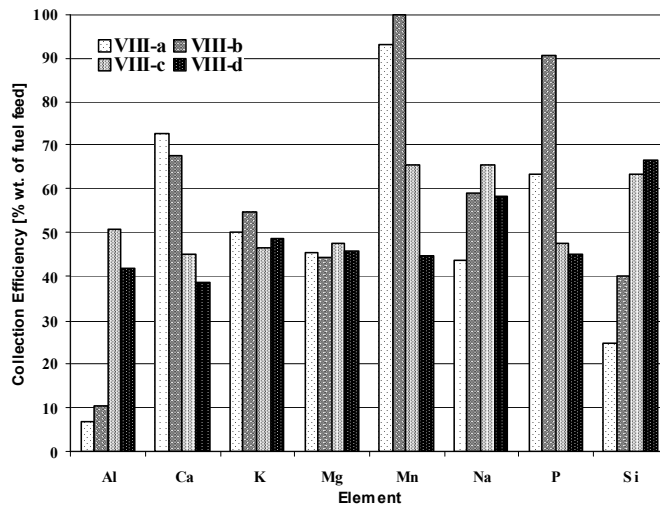
Figure 14: Collection efficiency for Al, Ca, K, Mg, Mn, Na, P and Si in VCP & CCP (%wt. of fuel feed)

Figure 14 shows that, regardless of the air flow rate and the fuel load, there is a certain trend only dependent on the fuel type. For fuel S, trials VIII-c & d, typically 50% of all elements are separated with the VCP and CCP ash. For fuel A, trial VIII-a & b, the trend is also very consistent but with significant differences in separation performance between different elements was observed. Only 10% of Al was removed compared to well above 90% of Mn. Large separation performances were seen between the two fuels for some of the elements and it is clear that the separation rate is much more dependent of fuel type than any other operating condition. The retention rates of Alkali's, Na and K, however were independent of the fuel type and identical for the more relevant cases, trial VIII-b & d, with around 50% retention of Na and K.

As carbon burnout and the ash separation performance are generally very high, low particulate emissions in the exhaust stream can be expected. This was only the case however for the more relevant operating conditions (at lower equivalence ratios) trial VIII b & d. High particulate emissions were obtained for trials VIII a & c, presumably due to the lower flow rates, hence velocities and cyclonic separation. The tar content of the particulate emissions in the exhaust gas was very low, less than 5% for trial b & d, indicating very good carbon burnout throughout the system.

Very good separation and gasification performance was obtained with the inverted cyclone gasifier under atmospheric conditions. The lower fuel load 26 kg/h was found to be far too low for this test rig. The inlet velocities of 2 m/s or less for the proposed conditions were on the lower operating limit of the test rig. Higher fuel feed rates and subsequently air flow rates/ inlet velocities are certainly possible and higher turn down ratios of 5 or even higher can certainly be achieved. This highlights the robustness of the design with clear benefits.

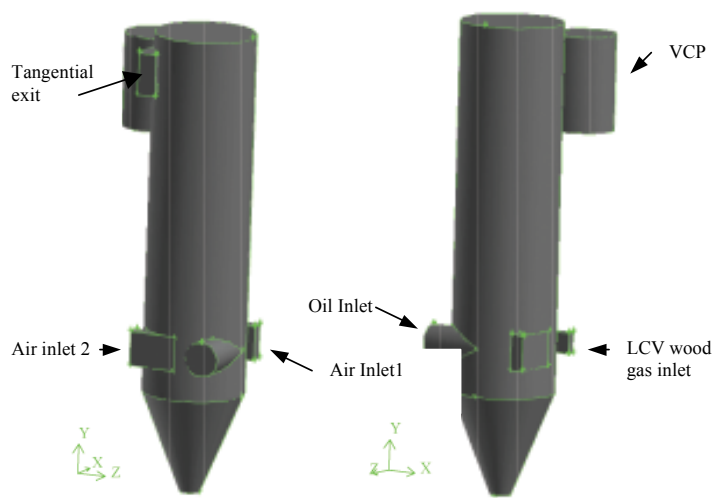## 3.3 Cyclone Combustor
3.3.1 Combustor development

A secondary combustor was developed for integration into the small-scale biomass co-generation process describe above. This combustor should be capable of running on the gasifiers' exhaust gas whilst still be able to utilise natural gas or oil for start up and shut down procedures or as a pilot fuel during certain operational conditions. To minimise pressure loss across the system and maximise

efficiency the use of complex hot gas clean up system should be unnecessary, due to the separation capability of the gasifier. The combustor will be indirectly fired to the gas turbine. Present designs of combustor that can run on multiple fuels are either direct fired or involve complex gas clean up systems. All these designs are essentially derived from conventional gas turbine combustor systems fired on conventional liquid fuels or natural gas. They are all designed to be fired on cleaned bio-gas. This arises because the turbines have sophisticated blades incorporating numerous fine cooling passages susceptible to blockage. Conversely the present combustor address a different problem involved with small scale power systems. Here gas turbine systems are generally of simpler construction with un-cooled turbine blades and can sustain modest levels of fine particulates less than 5 microns in size. Turbine inlet temperatures are up to 900°C. The combustor for this process can be more robust and is indirectly fired. The combustor should be capable of removing any particles remaining above 5µm.

3.3.2 Combustor design

The cyclone type combustor is designed with several tangential inlets as shown in Figure 15, an air inlet, a high CV fuel inlet for oil or natural gas and a low CV gas inlet. The tangential inlets create a strongly swirling flow that gives good mixing and burn out rates. The combustor is to be operated at a maximum thermal input of 550kW. The combustor is mounted vertically and has a cone section at the base to collect larger particles in the flow. The combustor is designed with a long chamber to allow the flame to move up and down with varying thermal input and quality whilst giving sufficient residence time for fuel burnout and thus low emissions of CO and volatile hydro carbons. The central section of the combustor is refractory lined allowing substantial heat storage capacity helping to create stable flames. There is a tangential off take on the combustor that recovers energy from the flow and also forces the exhausting flow tangentially across the VCP aperture, hence increasing separation capability. The combustor is to be used to fire a small gas turbine operating at an inlet temperature of 800°C. The exhaust gas of the combustor has higher temperatures than this, and is diluted by a co-flowing air stream that is passed through a jacket surrounding the combustor. This co-flow air lowers the temperature of the exhaust gas and acts as a diffuser to the flow, lowering the pressure drop across the combustor. The VCP is placed just before the tangential off-take, which removes any fine particle above 5 microns that could damage the turbine. The VCP also removes the need for a cyclone separator to remove the particles, which would increase the pressure drop across the system further.

Figure 15: Design of the cyclone type combustor with VCP

3.3.3 Combustor flow characteristics

A cyclone type combustor was built with several tangential inlets, for air, oil and gasifier LCV gas. The tangential inlets produced a swirling flow with good mixing, creating a stable flame. Figure 16 shows the swirling flame inside the combustion chamber, with the top plate removed.
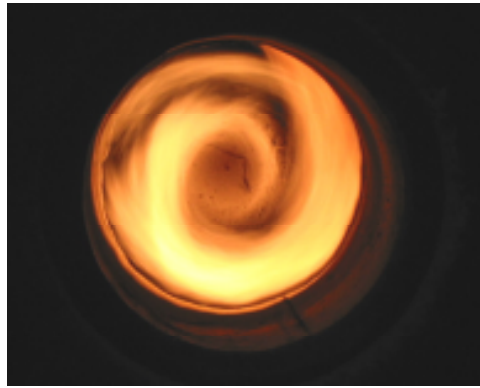

Figure 16: Photo of swirling flame inside combustion chamber

3.4 CFD modelling

The combustor was initially modelled using the CFD package Fluent 6, under atmospheric conditions but with velocities representative of those occurring under pressurised gas turbine operating conditions.` An unstructured three-dimensional grid was applied for this analysis. The Reynolds Stress turbulence model was used as the combustor has strong swirling flow. Experience has shown that this model predicts swirling flow with significantly more accuracy than other models, Fraser (2003). The LCV gas species and gas oil were modelled using the mixture fraction/pdf approach thus assuming chemical equilibrium. The non-adiabatic pdf model was used as radiative heat transfer to wall boundaries would have to be considered. The local thermo-chemical state is also related to enthalpy as well as the mixture fraction. The DTRM radiation model was used throughout the calculation. Particles (assumed to be spherical) were introduced into the LCV gas stream using the discrete phase function. This enables the particles to interact with the flow and predict particle trajectories, hence separation.

3.5 Combustor CFD Results

The combustor was modelled using the gas composition from the cyclone gasifier, trial VIII b**,** Table 6. Two tangential inlets to the combustor were used, the LCV gas inlet and an air inlet.
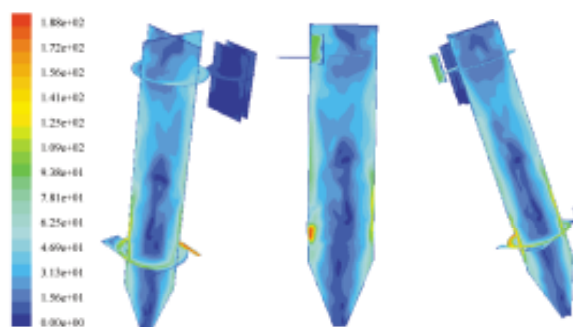

Figure 17: Velocity contours across combustor m/s

The combustor was designed with multiple inlets to allow high to low CV gas to be tested with two inlets used for the modelling described below, a LCV gas inlet and an air inlets. The combustor was modelled at 30% excess air to allow good mixing, produce a clean exhaust gas with low emissions and to allow wall temperatures to stay within acceptable limits. CFD results show using a typical LCV wood gas the combustor produced a strongly swirling flame. The vortex structure of the flow field caused an extended residence time producing complete combustion of the fuel. Experimental tests demonstrated the swirling flame in Figure 16. Velocity contours across the combustor are shown in Figure 17. The LCV gas and air enters the combustor tangentially through their respective inlets. Velocity contours taken at a plane through the inlets, Figure 18a, show high tangential velocity at the inlets and close to the wall. The tangential velocity decreases towards the centre and thus the flow is of forced vortex form. The areas of high tangential velocity near the wall correspond to regions of high temperatures, Figure 18b. The flow has good mixing and produces a stable swirling flame. The swirling flow continues along the combustor, with the majority of the flow staying near the chamber walls. As the flow moves downstream in the chamber the velocity of the flow increases in the centre, indicating that there are complex recirculation patterns and secondary flows in this region. The LCV inlet is designed with a relatively small area to increase mixing with the air. The tangential exit produces a near uniform velocity profile thus creating more stable inlet conditions for the gas turbine.
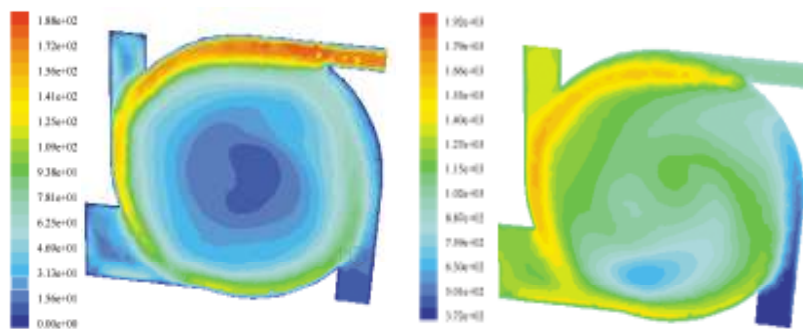

Figure 18: a) Velocities and b) temperature contours through plane of combustor inlet

Temperature contours, Figure 19, show that combustion is initiated in the wall region near the inlets and then spreads across the whole top 50% of the unit, presumably due to the secondary flow effects. The exit temperature profile is quite uniform and when mixed with dilution air will form a good flow to fire into the turbine. Average exit temperatures are around 1350 K.
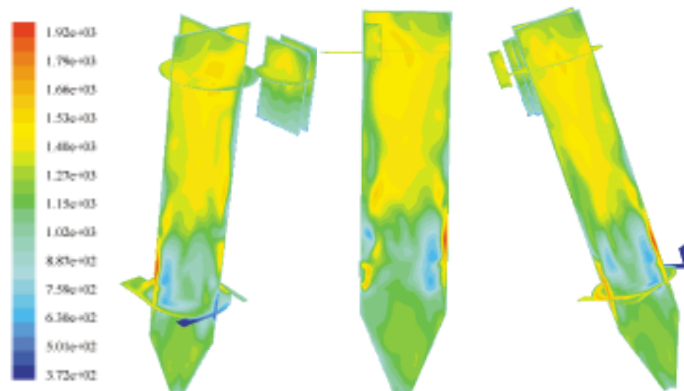

Figure 19: Temperature profiles across combustor inlet and outlet

Minimising pressure drop across a system is important, especially in small-scale plants. The maximum pressure drop across the system should not exceed 0.2 bar. Figure 20 shows the pressure drop across the combustor is typically less than 0.025bar. This pressure drop could further be reduced by placing a 7° diffuser on the tangential off-take (O'Doherty et al. 1992). This could also recover up to 40% of lost kinetic energy. Diluting the exhaust gas with co-flowing air, to lower temperatures suitable for the turbine inlet, would also lower the pressure drop.
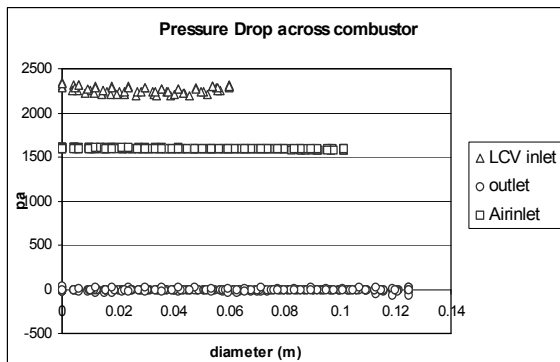

Figure 20: Pressure drop across the combustor

Particles were introduced into the flow through the LCV gas inlet. Figure 21 shows the tracks of 5 microns particles through the combustor. The VCP removes the particles from the flow before they exit in the exhaust. Thus the performance of the VCP is adequate to remove particles from the flow without the addition use of cyclone separators downstream of the combustor.
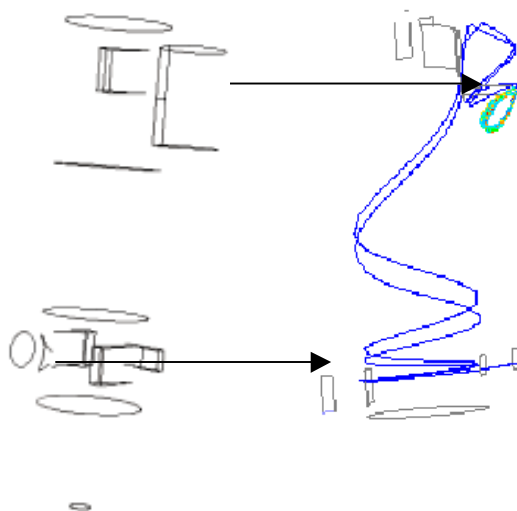

Figure 21: Particle tracks of 5micron diameter through combustor

4 Conclusions

The inverted cyclone gasifier developed by Cardiff University is suitable for gasifying biomass under a range of conditions with varying equivalence ratio between 0.19 and 0.24. The design is robust with good mixing and 98-99% burnout rate. The VCP and CCP removed particles from the flow, with high separation/retention rates. Up to 50% of the Alkali's Na and K were removed from the flow. A good quality LCV gas was produced with maximum calorific value of 5.91 MJ/m^3.

The LCV gas produced is suitable for firing directly into the secondary combustor. The combustor produced a strongly swirling flame with good mixing and burnout. An area of secondary flow and recirculation occurred where combustion was enhanced. The combustion patterns showed good mixing with near full burnout and hence low CO and hydrocarbon emissions. Velocity and temperature profiles at the combustor exhaust were nearly uniform, forming stable conditions for the turbine inlet. The VCP positioned just before the combustor exit effectively removed particles above 5μm from the flow. The system produced gas suitable for indirect firing of the gas turbine. The pressure drop across the system was relatively low at a predicted maximum level of 0.2 bar.

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