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IMPACT OF CO_2-ENRICHED COMBUSTION AIR ON MICRO-GAS TURBINE PERFORMANCE FOR CARBON CAPTURE

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ABSTRACT

Power generation is one of the largest anthropogenic greenhouse gas emission sources; although it is now reducing in carbon intensity due to switching from coal to gas, this is only part of a bridging solution that will require the utilisation of carbon capture technologies. Gas turbines, such as those at the UK Carbon Capture Storage Research Centre's Pilot-scale Advanced CO_2 Capture Technology (UKCCSRC PACT) National Core Facility, have high exhaust gas mass flow rates with relatively low CO_2 concentrations; therefore solvent-based post-combustion capture is energy intensive. Exhaust gas recirculation (EGR) can increase CO_2 levels, reducing the capture energy penalty. The aim of this paper is to simulate EGR through enrichment of the combustion air with CO_2 to assess changes to turbine performance and potential impacts on complete generation and capture systems. The oxidising air was enhanced with CO_2, up to 6.29%vol dry, impacting mechanical performance, reducing both engine speed by over 400 revolutions per minute and compression temperatures. Furthermore, it affected complete combustion, seen in changes to CO and unburned hydrocarbon emissions. This impacted on turbine efficiency, which increased specific fuel consumption (by 2.9%). CO_2 enhancement could therefore result in significant efficiency gains for the capture plant.

Keywords: gas turbine; CO_2 emissions; post-combustion carbon capture; exhaust gas recirculation; energy penalty.

Highlights:

- Experimental investigation of the impact of exhaust gas recirculation (EGR) on GT performance
- Combustion air was enhanced with CO_2 to simulate EGR
- EGR impact was ascertained by CO and unburned hydrocarbon changes
- Primary factor influencing performance was found to be oxidiser temperature
- Impact of CO_2 enhancement on post-combustion capture efficiency

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1. INTRODUCTION

Due to the mounting evidence for anthropogenic climate change and its potentially serious impacts, there is an increasingly strong impetus to tackle the key contributor of carbon dioxide [1]. In the UK policy discussions often avoid picking technology 'winners' and 'losers' to deal with the problem, with electricity market reform allowing provision of contracts for difference for a spectrum of low carbon generation options, wind, solar, biomass nuclear and potentially carbon capture plants [2]. In truth though we are already locked into a 'loser' with the ever growing consumption of carbonaceous fuels, and the thermal power infrastructure. Global energy consumption is projected to increase with population and economic growth. A 34% increase in primary energy demand is expected by 2035, of which 80% will be met by fossil fuels. During this time it is thought that although coal will make up a smaller percentage of the energy mix, natural gas consumption will increase significantly, with over 45% of it being used for power generation [3-5]. To not exceed projected global carbon budgets and to keep predicted temperature increases in the atmosphere to within 2°C, it is critical that carbon capture is chosen as a 'winner'. This is because of the global economic fossil fuel dependence and the potential for retrofitting capture systems to mitigate the "lock in" of existing thermal power plants [6], and have negative emissions that may now be necessary [7]. A solution no other low carbon generation technology can offer.

1.1 Carbon Capture in the UK

The UK has shown a stronger interest than most countries in the various capture technologies, with research funding, policy and pledges of financial support for commercial demonstration. Geographically, the UK also has excellent potential large offshore storage sites in relatively close proximity, e.g. in the North Sea.

Front End Engineering Design – FEED – studies for two UK full-scale demonstration plant were carried out from late 2013 due to be complete in late 2015 to aid the decision in the award of a large substantial grant from the government for commercialisation of one. The Peterhead proposal was a gas turbine power station with post-combustion capture, whilst the White Rose proposal was an oxy-fuel coal and biomass project [8]. However funding was scrapped from the UK government budget in November 2015, and both commercial enterprises announced they would not progress the projects without government support, continuing uncertainty.

The UK Carbon Capture Storage Research Centre (UKCCSRC) has funded national specialist R&D facilities for combustion and carbon capture technology research – the Pilot-scale Advanced CO_2 Capture Technology (PACT) Facilities – which have the capacity to demonstrate both the concept processes (post-combustion and oxyfuel capture) at a pilot scale much larger than previously established in academia. The UKCCSRC PACT Core Facility near Sheffield houses two 100kW natural gas-fuelled microturbines; the CO_2 from the exhausts of which can be captured by the onsite post-combustion solvent-based carbon capture plant. The capture plant comprises an 8m column absorber which can capture up to 1 tonne of CO_2 a day using a monoethanolamine solution. The PACT Core Facility also houses a 250 kW down-fired combustion test facility, which can burn solid fuels, including coal and biomass feedstocks under air and oxyfuel operating regimes. The conventional combustion rigs, gas turbine and capture plant have access to gas mixing facilities which can be used to create synthetic process gas, and flue gases for capture, enhancing with CO_2, O_2 and N_2 possible, and trace gases through additional ports.

1.2 Background

There have been a significant number of high quality modelling studies such as those by Li, et al [9,10] and Mansouri, et al [11] into the potential benefit of exhaust gas recirculation (EGR) implementation on gas turbines to increase the CO_2 partial pressure, and create net efficiency gains through reboiler duty reduction. The optimal EGR ratio is the highest degree to which combustion is kept stable and does not require additional vitiation with oxygen to maintain the flame. Mansouri, et al [11] used a validated thermodynamic model to show the impact EGR would have on a T100 microturbine. However only baseline data was from experimental values, it showed the relationship between the turbines mechanical operation, in terms of turbine frequency, turbine inlet temperature (TIT), compressor pressure and the impact of ambient temperature. The model predicted the impact EGR would have on these.

Experimentally, there have been limited investigations into gas turbine EGR combustor performance, with papers published by ElKady, et al [12] and Evulet, et al [13]. These researchers investigated the performance and operability of a combustor from a GE F-Class turbine. Another by Rokke and Hulstad [14] investigated the impact of the addition of oxides of nitrogen (NOx), O_2 and CO_2 on a smaller 65kW burner, and similar investigations were undertaken by Jansohn, et al [15]. However these focused on the combustor performance isolated from the turbine, and not the whole system. These studies have shown that EGR can take the exhaust emissions of a Combined Cycle Gas Turbine (CCGT) from 3.8% up to 10% with 55% EGR [10]. These percentages are possibly even pessimistic as Evulet, et al [13] experimentally achieved over 8% CO_2 in the exhaust at the equivalent 25% EGR, and ElKady, et al [12] achieved over 10% CO_2 at 35% EGR. However gains in reboiler duty efficiency have been shown to become less significant over 6% mol CO_2 at the absorber inlet, and the hence possible reduced combustion efficiency and potential necessity of energy intensive oxygen vitiation may not be worthwhile at higher EGR ratios [9].

The critical limiting factor to high EGR ratios is generally considered to be the oxygen percentage at the turbine inlet for combustion. Li, et al [9, 10] found that 60% EGR meant only sufficient oxygen for stoichiometric combustion, and at 55% EGR, only 11% oxygen for combustion. O_2 at 11% is insufficient for a stable flame to be maintained , and for complete combustion resulting in lean blow out, excess CO and unburnt hydrocarbons [9].

Ditaranto, et al [16] showed flame blow out at O_2 levels below 14%, and like ElKady, et al [12], this produced higher NOx, CO and some unburned hydrocarbons (UHC) below O_2 levels of 16% vol. Due to the oxygen limitations, 35%-40% EGR is generally considered the maximum achievable. Evulet, et al [13] showed recirculations in the GE F-class burner of 50% and this resulted in an inlet oxygen of 13.2%, and EGR at 40% gave an inlet O_2 15.9%. ElKady, et al [12] showed that EGR at 35% gave an O_2 concentration of 17% for the oxidizer, much above the lean blow out limits, but at 14-16% oxygen at higher recycle ratios, unacceptably high CO and NOx emissions have been observed. However ElKady, et al [12] suggest that with modifications for pilot flames, higher EGR ratios may be achievable.

Combustor studies also examined the impact of EGR on combustion quality and emissions. Evulet, et al [13] observed higher CO at lower flame temperatures corresponding to lower powers [17]. This CO represents incomplete combustion that can be attributed to incomplete combustion, resulting from flame temperature below 1250°C where oxidation reactions are more significant, but also lack of oxygen, pressure or residence time. Though Evulet, et al [13] found relatively low CO emissions, there was a significant trend of higher CO with EGR, above the baseline combustor performance.

EGR has also been shown to impact other combustion species. Though the recirculation of CO_2 reducing peak flame temperatures, and hence turbine inlet temperature and efficiency. The reduction in combustion temperature reduces thermal NOx creation. Lee, et al [18] discussed the thermal and radiative effect of CO_2 addition having the largest influence on reducing combustion temperature, with the kinetic effect being negligible. The higher heat capacity of CO_2 than air reduces the temperature, and the increased rate at which heat was radiated, results in overall lower combustion temperatures. This dilution effect of CO_2 addition causing lower temperature causes a reduction in the production of thermal NOx.

The only published research that looks at the impact of EGR on turbine performance is by De Paepe, et al [17], who discuss the impact of the observed accidental recirculation of exhaust gases on a T100 microturbine. In this case, poor turbine performance was noted to be caused by the exhaust being adjacent to the air inlet for the gas turbine. De Paepe graphed the direct correlation between rises in turbine inlet temperature and reductions in power generated due to a reduction in turbine frequency. Power is a function of combustor temperature, which is correlated to fuel flow. Increases in fuel flow means increased combustion and CO_2 formation. The temperature CO_2 relationship is shown by Evulet, et al [13], and the corresponding power and turbine inlet temperature for the T100 micro gas turbine is reported by Mansouri, et al [11]. This is equates to the expectation increased turbine power will results in increases in CO_2 production.

The recirculation observed by De Paepe, et al [17] was not controlled and therefore the paper focused on the mechanical impact of the increased temperature of inlet air on the turbine output, with no emissions data collected. The key studies by Evulet, et al [13] and ElKady, et al [12] focus only on combustor performance and emissions, not turbine and whole system. For this reason these experimental investigations herein and this paper are novel in field and detail of investigation.

1.3 Aims and Objectives

The aim of this work was to simulate the process of EGR through the enrichment of the combustion air with CO_2, and to analyse the impacts this may have on combustion performance and the gas turbine efficiency. This was achieved by piping in various controlled flow rates of CO_2 into the combustion air from external storage. From the literature explored above, it is expected that 8% CO_2 by volume may be successfully injected into the combustion air without lean blow out or turbine stall – this represents up to 35% EGR [12]. The tests were designed to test the limit to which CO_2 can be added to the Turbec T100 PH gas turbine at PACT and what degree of EGR this represents. Since the combustion is inherently lean for the microturbine, it was possible to 'recirculate' more flue gas – or enhance it with more CO_2 than the literature suggests, with the equivalent CO_2 of up to 356% EGR added (essentially, selective exhaust gas recycle). The implications of the changes were evaluated by assessing the variations in the system temperatures and flue gas composition, as well as to the gas turbine performance data. Further the impact of the oxidizer density change and temperature variation has been analysed.

The paper mainly focuses on the evidence of the impact on combustion in terms of incomplete combustion characterised by CO, and unburnt hydrocarbons, and varying combustion temperatures in terms of oxides of nitrogen. Also it investigates the mechanical impact on the turbine, variations in turbine frequency, compression pressure and temperature altered by changes in the oxidiser composition and heat capacity. The third element for analysis is the efficiency and specific fuel consumption changes caused by the addition of CO_2 to the oxidiser.

2. METHODOLOGY AND DATA ACQUISITION

2.1 Turbec T100 PH Gas Turbine

The two natural gas-fuelled microturbines at the PACT Core Facility are both Turbec T100 PH designs. Each turbine produces up to 100 kW of electrical power, and since they contain a combined heat element, they also generate up to 165 kW thermal power, in the form of hot water at 70-90°C. The manufacturers quoted the electrical efficiency as 30%, but the use of heat recovery components, the recuperator and heat exchanger shown in Figure 1, increases the overall efficiency to ~77% [19].

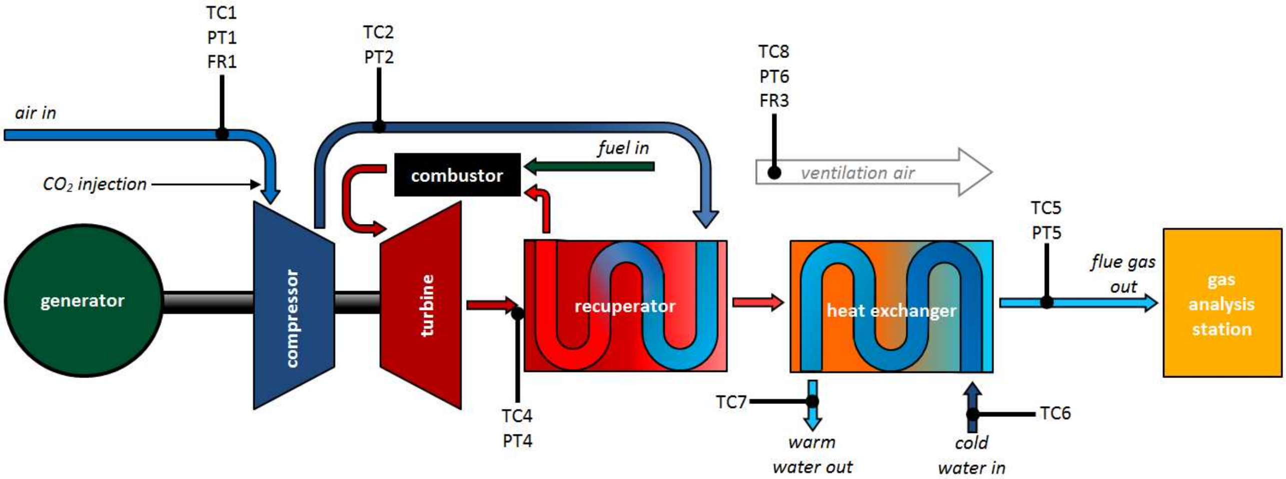

Figure 1: Schematic of the components of the Turbec T100 PH combined heat and power gas turbine system, including the modifications made of the CO_2 injection and the instrumentation (TC – thermocouples; PT – pressure transducers; FR – flowrate meters).

The key components of the turbine are outlined in Figure 1. The engine is a single shaft design, where the compressor is driven by the turbine on the same shaft, with the high-speed electrical generator. A single centrifugal compressor is used to compress the ambient air until the optimal pressure ratio is achieved (ideally ~4.5:1 at maximum power output). The pressurized air is then pre-heated with the hot flue gases via a recuperator before entering the combustion chamber; this significantly increases the electrical efficiency. The natural gas is fed into the combustion chamber and an electrical ignitor is used to light the gas. A fuel lean, oxygen rich, mix ensures emissions of carbon monoxide, unburned hydrocarbons and NOx are low, without the need for flue gas treatments.

The radial turbine drives both the compressor and the generator. The combustion gases at the combustor exit are at a high temperature (~950°C) and an elevated pressure (~4.5 bar). These expand through the turbine, where the pressure decreases close to atmospheric and the temperature drops by about 300°C. The hot gases from the turbine pass through the recuperator, pre-heating the inlet air and further reducing the gas temperature. These then pass through the counter-current water-gas heat exchanger to generate the hot water.

A number of auxiliary systems are included in the power module – a lubrication system, a cooling system, an air intake and a ventilation system (incorporating an external ambient coarse filter and an internal fine filter), fuel gas system with a fuel booster and a buffer air system. Table 1 outlines additional manufacturer technical data for the gas turbine engine.

Table 1: Turbec T100 PH micro gas turbine technical data [19].

INLET AND COMPRESSOR	
Type	centrifugal
Pressure ratio	4.5:1
Maximum inlet air flow at 15°C (kg/s and m^3/s)	1.69 (1.38)
COMBUSTOR AND FUEL REQUIREMENTS	
Type of chamber	lean, pre-mix
Pressure in combustion chamber (bar, a)	4.5
Fuel flowrate (m^3/hr)*	31
Lower heating value (LHV) of fuel (MJ/kg)	38-50
Wobble Index (MJ/m^3)**	43-55
TURBINE	
Type	radial
Inlet temperature (°C)	950
Normal turbine outlet temperature (°C)	650
Maximum turbine outlet temperature (°C)	710
Outlet pressure	~atmospheric
Nominal shaft speed (RPM)***	70,000
FLUE GAS HEAT EXCHANGER	
Type	gas-water
Flow	counter-current
Maximum flue gas flowrate at outlet (kg/s)	0.87
Maximum flue gas temperature at outlet (°C)	325
Minimum and maximum water flowrates (l/s)	1.5-4.0
Minimum water temperature at inlet (°C)	50
Maximum water temperature at outlet (°C)	150

** The fuel flowrate depends on the gas composition – this is specified for a fuel with a LHV of 39 MJ/m^3*
*** m^3 at 288.15K and 101.325kPa.*
**** Revolutions Per Minute*

2.2 Experimental Methodology and Test Conditions

The two key variables in these tests were the power output of the turbine and the level of CO_2 enhancement (the simulated EGR ratio). In total, 66 different conditions were tested, ranging from 0kg to 175kg of CO_2 injection, with power outputs from 50kW to 80kW, thus covering EGR ratios from 0% to 356%; the EGR ratio was calculated based on volume. Each test was carried out for a minimum of 15 minutes of stable operation in accordance with ISO 2314 to determine emissions performance.

2.3 Data Acquisition

Testing was performed at the PACT test facilities, with the above-described T100 PH Series 1 microturbine. The turbine performance was measured using the turbines own instrumentation and recorded by the vendors WinNAP program for the following parameters:

- air inlet temperature (T1 °C)

- calculated turbine inlet temperature (TIT °C)
- turbine outlet temperature (TOT °C)
- power generated by the turbine (kW)
- power set point (kW)
- engine speed (Revolutions Per Minute, RPM, and % of maximum)
- gas pressure (mbar)
- opening of the pilot and main fuel valves (both %)

Additional instrumentation was added to the turbine, taking a range of flowrates, temperatures and pressures throughout the system, and importantly after compression, as shown in Figure 1. The channels were recorded with a LabView program using a National Instruments Compact RIO-9022 Real-Time controller. Table 2 outlines the quantities monitored.

Table 2: Quantities monitored by LabView for the Series 1 gas turbine.

LabView DESIGNATION	PARAMETER	UNIT
THERMOCOUPLES		
TC1	system air inlet temperature	°C
TC2	compressed air temperature (compressor outlet)	°C
TC4	flue gas diffusion zone temperature	°C
TC5	flue gas outlet temperature	°C
TC6	cold water temperature (heat exchanger inlet)	°C
TC7	hot water temperature (heat exchanger outlet)	°C
TC8	ventilation air outlet temperature	°C
PRESSURE TRANSDUCERS		
PT1	system air inlet pressure	bar g
PT2	compressed air pressure (compressor outlet)	bar g
PT4	flue gas diffusion zone pressure	bar g
PT5	flue gas outlet pressure	bar g
PT6	ventilation air outlet pressure	bar g
FLOWRATE MEASUREMENTS		
FR1	system air inlet flowrate (total air in) – measured	kg/min
FR3	ventilation air outlet flowrate – measured	kg/min
FR4	flue gas outlet flowrate – calculated	kg/min

Gas species were measured using a Servomex analyser, and GasMet FTIR. Oxygen and CO_2 concentrations in the flue gas were monitored using a Servomex Servoflex MiniMP 5200 analyser, which uses a non-dispersive infrared sensor for CO_2 analysis and paramagnetic transducers for O_2 detection. The GasMet DX4000 FTIR, Fourier transform infrared spectroscopy, measures the absorbance of infra-red to determine the gas species within a sample. Here, a GasMet FTIR DX4000 was used with its associated conditioning system to quantify the levels of CO_2, CO and various unburned hydrocarbons. A number of other gaseous species were also determined in this manner, including water vapour, and total NOx. It does this by measuring the absorbance at each wavelength so species in the sample can be calculated.

The fuel gas flowrate into the turbine was measured with a Quantometer turbine flow meter. Type K thermocouples were used throughout for additional temperature measurement. Two different Rosemount pressure transmitters were used for instrumentation, PT1 and PT5 are model 2051CDC2A. PT2, PT4, and PT6 are model 2051TG2A.

The maximum uncertainty in instrumental error is listed in Table 3. For type K thermocouples the error is the greater of 0.4% of the measurement or 1.5°C. The maximum standard deviation of results for both baseline experiments and CO_2 enhanced is also reported in Table 4. It should be noted the majority of results fell within a much smaller deviation. Due to the small instrumental error and standard deviation of results, error bars are not plotted on graphs.

Table 3: Instrument errors.

	INSTRUMENTAL ERROR	
INSTRUMENT	**%**	**Unit**
Servomex Servoflex MiniMP	0.10%	
GasMet DX4000 FTIR	n/a	n/a
Quantometer	0.63%	
Type K thermocouples	0.40%	1.5°C
Rosemount pressure transmitters 2051CDC2A	0.07%	0.8 mbar
Rosemount pressure transmitters 2051TG2A	0.07%	7.5 mbar

Table 4: Standard deviation errors.

	MAXIMUM STANDARD DEVIATION	
GASMET DX4000 FTIR READINGS	**BASELINE**	**ENHANCEMENT**
CO ppm	9.2	17.32
CH_4 ppm	1.45	3.91
CO_2 vol%	0.51	0.03
NOx ppm	1.88	1.27
TURBEC T100 READINGS		
RPM	141.65	143.66
kW	1.57	2.19
T1 °C	0.22	0.6
ADDITIONAL INSTRUMENTATION		
PT2 mbar	0.02	0.02
TC2 °C	0.66	0.69

2.4 Emission Corrections and Reporting

Species have been corrected to a dry basis for reporting and comparison with the literature and industrial standards of gas turbine emissions reporting using the following equation:

$$\text{Dry Basis Volume Concentration} = \frac{Wet\ Basis\ Concentration}{1 - H_2O\ Volume} \tag{1}$$

The industrial standard for gas turbine emissions reporting for NOx is corrected to 15% O_2. This is also mandated by the EU Large Combustion Plants Regulation 2012 that requires values to be reported on a dry basis at 273.15K, 101.3kPa and 15% O_2 for gas turbines [20]. This was calculated using the equation:

$$NO_{x_{O_2\,Corrected}} = NO_x\left(\frac{20.9 - O_{2_{Ref}}}{20.9 - O_{2_{Measure\,Exhaust}}}\right) \tag{2}$$

ElKady, et al [12] correctly suggested that NOx reporting corrected to the standardised oxygen percentage is not an appropriate metric with EGR burners. The determination of the recirculation has impacted NOx emissions compared to non-EGR burners is therefore of greater importance in this investigation. This is because the O_2 in the exhaust of the gas turbine will be reduced in turbines with EGR, and hence the small denominator will result in a greater multiplication of the uncorrected NOx volume. This is caused by not having the inlet oxygen conditions to replace the 20.9% value, with O_2 being displaced due to the recirculation of CO_2 and N_2.

Since the inlet oxygen conditions were not monitored for this gas turbine, a more appropriate metric is required. It is suggested that reporting the NOx emissions in terms of the mass of fuel combusted and the net power output would be beneficial in this case. This was not possible for ElKady, et al [12] with the study only utilising a combustor. The calculation for the NOx Emissions Index (g/kg Fuel) is given by:

$$NO_{x_{EI}} = \frac{NO_{x_{mg/Nm^3}} . Exhaust_{Total\,m^3/h}}{Fuel_{\dot{m}}} \tag{3}$$

The NOx/kWh, corrected to 15% O_2, can thus be calculated using:

$$NO_{x_{g/kWh}} = \left(\frac{\left(NO_{x_{mg/Nm^3}} . Exhaust_{Total\,m^3/h}\right)/1000}{Net\ Power_{kWh}}\right) \tag{4}$$

As the EU Large Combustion Plant Directive (LCP) regulation covers larger single cycle gas turbine, and turbines with an efficiency greater than 35%, where the NOx would be limited to 50η/35 (η is turbine efficiency, giving 34mg/Nm3 for the T100), the Turbec T100 microturbines do not fall under the stipulated emissions limits within it [20]. However there are emissions limits for single cycle CHP units within regional air quality guidelines. The Greater London Authority proposes 5g/kWh for compression ignition gas engines [21]. Under normal operating conditions, the turbine used in this investigation is compliant and achieves significantly lower emissions, as detailed in Section 3.

3. RESULTS AND DISCUSSION

All experiments were performed at the PACT Core Facilities, with subsequent analysis, post processing and normalisation of the results completed. This section first outlines the baseline performance of the Series 1 Turbec T100 PH gas turbine, and then compares the results with CO_2 enhancement for comparison.

3.1 Base Performance

The key factor that impacts on the turbine performance is the ambient temperature (monitored as TC1), as higher temperatures reduce the air density and thus increase the energy required for compression and constant mass flow through the turbine. This can be seen with the base performance of the Series 1 gas turbine in Figure 2 where increasing temperatures required increased turbine speed, for a fixed power output, replicating results of De Paepe, et al [17] and Evulet, et al [13]. For an ambient temperature increase of ~10°C, the turbine speed increased by over 2000 rpm. This extra power required for compression reduced the overall system efficiency by more than 12%.

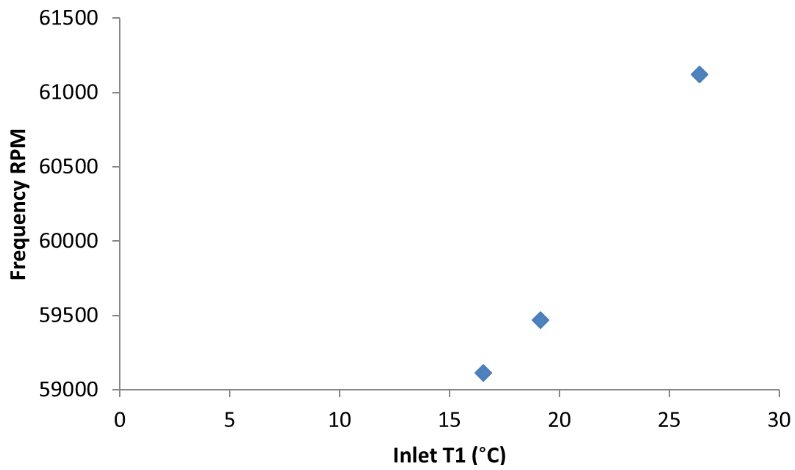

Figure 2: Impact of the oxidiser temperature on the turbine frequency at 50kW.

The baseline performance and combustion of the Series 1 turbine was good across all turn down ratios. This is visible in the low emissions of CO and unburnt hydrocarbons in the form of methane, which was the dominant gas species in the fuel. Only at low power outputs of 50-55kW was there some indication of poor combustion, similar to Evulet, et al [13], seen by the elevated levels of CO and methane in Figure 3 across turn down ratios.

The CO_2 vol% emissions increased linearly with power output, as shown in Figure 4. This is due to increased fuel consumption and hence CO_2 as a combustion product replicating the results of others [11, 13, 17]. However the total emissions intensity was higher here, and this is hidden by the higher mass flow through the system at higher power outputs. The total calculated CO_2 mass flow can be seen in the CO_2 intensity. This is significantly higher for this turbine than a CCGT, which has a carbon intensity of almost 400 gCO_2/kWh [22]. The carbon intensity of generation here was 900-1000 gCO_2/kWh, and this is due to the lower efficiency that was also reflected in the lower combustion efficiency over the same turn down ratios at lower powers, as demonstrated in the higher concentrations of methane and CO.

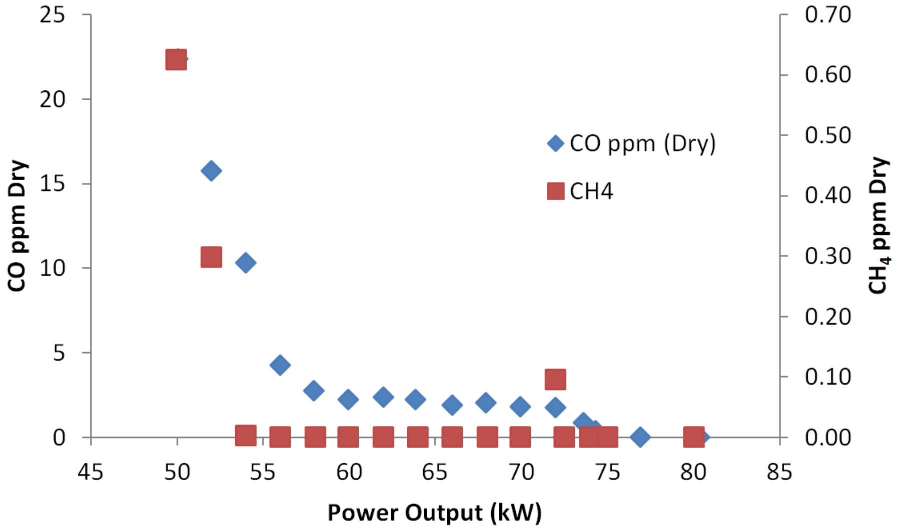

Figure 3: Variations in CO & CH_4 across the turn down ratios.

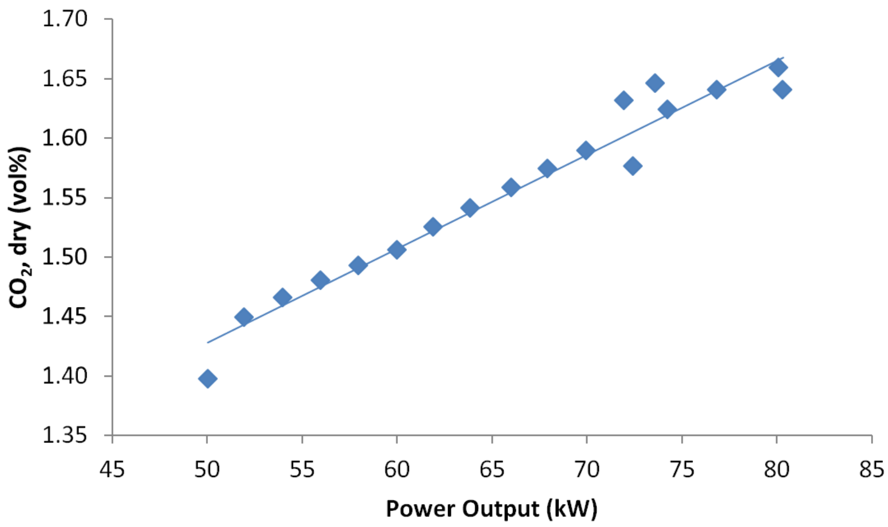

Figure 4: CO_2 emissions across turn down ratios.

3.2 CO_2 Enhancement Performance

Having established base turbine performance, facility and turbine modifications were made to improve capacity to add up to 175kg/h oxidiser enhancement with CO_2 from the gas mixing facilities.

The addition of CO_2 to the oxidiser, as predicted, had a linear effect on its exhaust gas concentration, as seen in Figure 5. However a slight decrease in CO_2 volume % for the same injection mass at higher power output is attributed to the higher total mass flow at greater turbine speeds for highest power output. The mass flow through the turbine was calculated by measuring the change in volume percentage of CO_2 in the exhaust, and the known CO_2 mass addition for that change above baseline. This revealed the higher mass flow at higher outputs, and a much leaner fuel:air ratio than CCGTs. A CCGT will have a ratio of 1: 42-58 [23], whereas for the T100 microturbine, it is calculated as 1:105-130.

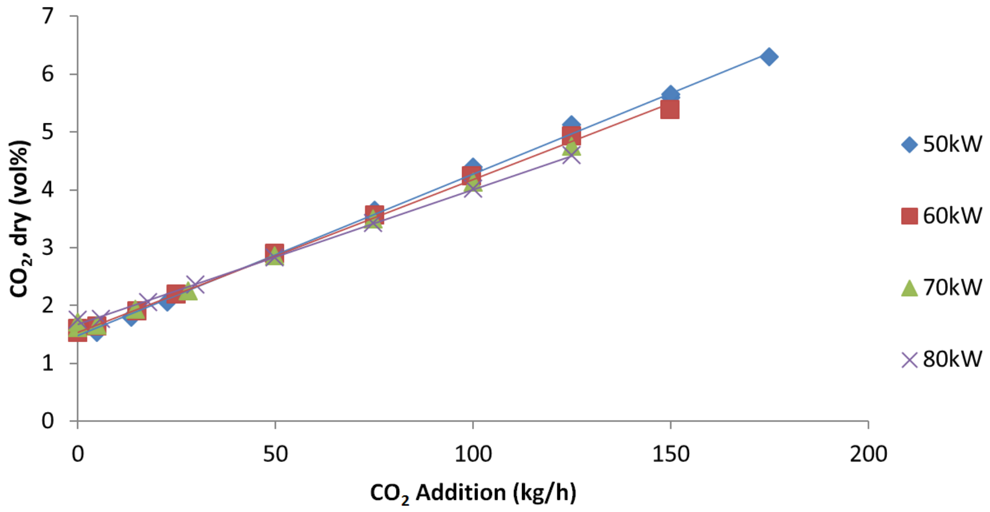

Figure 5: CO_2 vol% with addition of CO_2 to the oxidiser.

3.2.1 Mechanical Impact

Mechanically the impact on the turbine performance due to the CO_2 addition is dominated by the ambient temperature, and impact CO_2 has on the average oxidiser temperature. On the cold tests CO_2 addition raised the oxidiser temperature, and on warmer tests it reduced it, due to a relatively constant delivery from the gas mixing facilities. Figure 6 shows the reduction in frequency with CO_2 addition at 50kW, and the main driver of which is the ambient temperature.

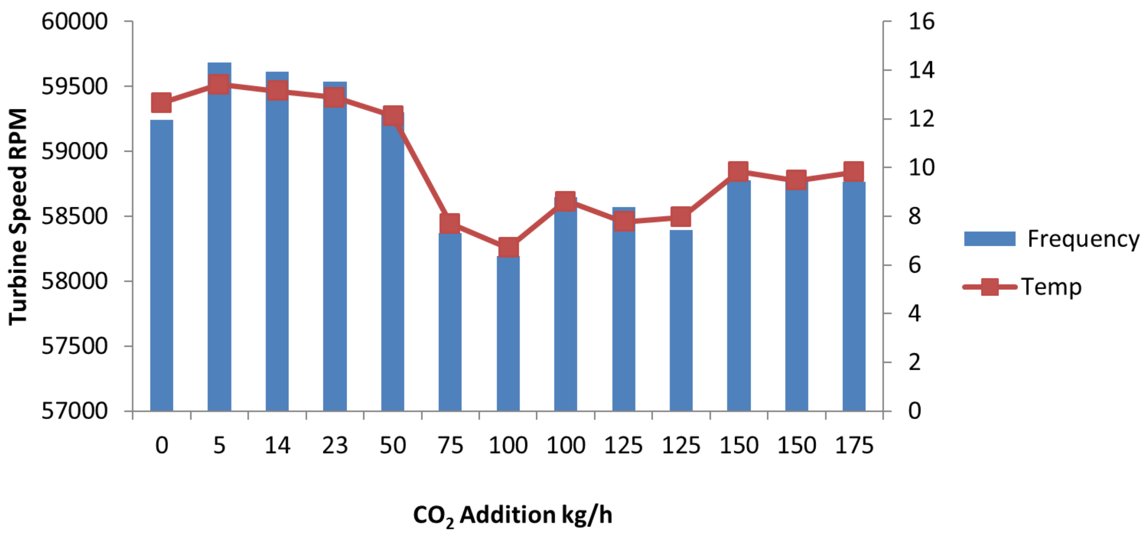

Figure 6: CO_2 mass addition to the oxidiser with turbine frequency and the oxidiser temperature relationship at 50kW.

There is a slight reduction in the turbine revolutions, due to the higher density CO_2 displacing the air, the lower temperature denser air had the lowest revolutions per minute (RPM), not the highest CO_2 mass flow, due to the ambient and overall oxidiser temperatures. This provides further evidence for the temperature frequency relationship depicted in Figure 2. If a higher volume percentage of CO_2 enhancement was possible it could be expected to have a more significant impact on turbine speed.

The relationship of the consistent pressure at lower RPM with higher CO_2 mass enhancement is well depicted in Figure 7 across turn down ratios from 50 to 80kW. All these results replicate those modelled by Mansouri, et al [11], and experienced in the turbine speed variations of De Paepe, et al's EGR microturbine investigations [17].

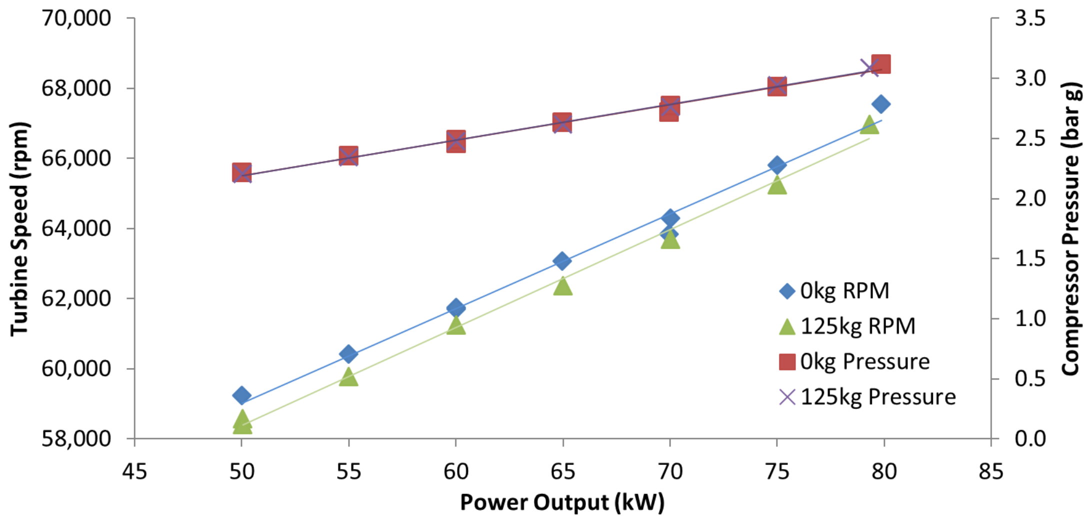

Figure 7: Impact of CO_2 on the turbine frequency.

Post compression the CO_2 enhancement has a significant impact on the oxidiser altering the heat capacity and the density. Figure 8 outlines the influence CO_2 injection has on the compressed temperature TC2. This impact on the temperature is due to the higher heat capacity of CO_2 than that of the air it displaces. Therefore it takes more energy to increase the temperature by the same amount, and so with the same energy input (compression), the same temperature is not achieved [18].

On inspecting Figure 8, it shows the CO_2 impact on compression temperature; the temperature post compression is consistently lower, and the only anomalous dips apparent are from low ambient temperatures on the testing days. Though there is insufficient instrumentation to monitor the impact after heat recovery through the recuperator or in the combustion chamber, the TC2 is a strong indication that CO_2 injection has a significant impact throughout the system. This is due to the increased heat capacity, radiation, and the changes in the combustion characteristics.

3.2.2 Emissions Impact

Lower oxygen levels in the oxidiser of 14-16% have been shown to cause unstable flames [12]. The increased CO_2 at high injection levels displaces the oxygen, which leads to poorer combustion. As the fuel:air ratio was so lean in the Turbec system, the oxygen percentage was calculated at the lowest to be 19.4vol% in the combustor. At this level blow out is not a risk and it is expected that there will not be as significant impact on the combustion performance as observed by ElKady, et al [12]. This is why this had a reduced impact on the combustion performance.

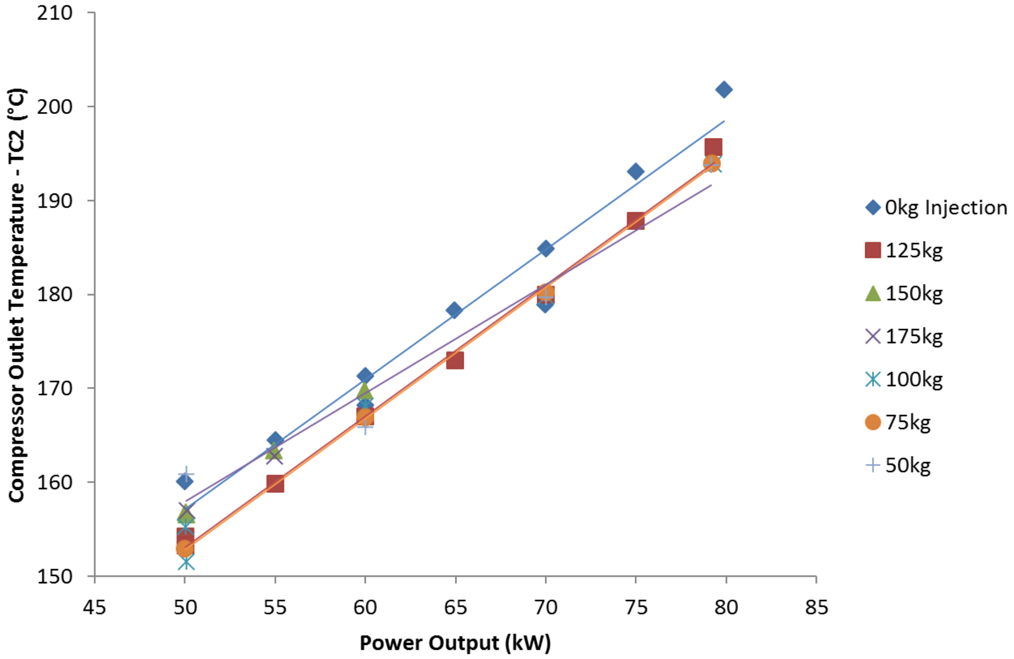

Figure 8: The CO_2 addition impact on post compression temperature (T2).

However the lower O_2 level and higher heat capacity of CO_2 still has an impact on complete combustion as evidenced at low powers of 50-55kW, where the fuel:air ratio is lean. This is the oxygen reduction having an impact. As shown in Figure 9, there is higher CO, from the reduced level of oxygen. This is caused by the incomplete combustion of the natural gas resulting in higher methane and TOC emissions. The addition of CO_2 to the oxidizer as discussed increasing the heat capacity, reducing reaction rates and hence flame speed. This reduction in combination with the depletion of oxygen contributes to the evidence of incomplete combustion observed.

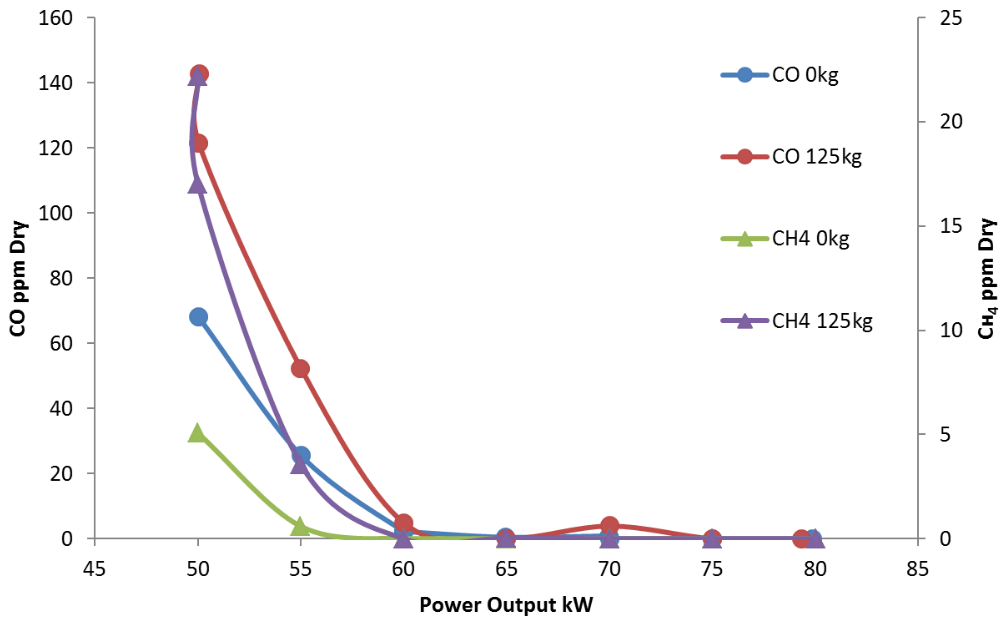

Figure 9: CO and CH4 emissions, comparing the baseline with 125kg/h CO_2 enhancement across turn down ratios.

The reduced combustion efficiency both with and without CO_2 enhancement at the low turn down ratios, can probably be attributed to the turbine design and temperature, with standard continuous operation expected above 60kW. These results correspond with similar findings of Evulet, et al [13] with elevated CO at lower temperatures, and higher with EGR. The lower total oxidiser mass flow, and reduced pressure due to the lower turbine speed at the bottom turn down ratios may be inadequate for full premixing of the fuel in the burner [18]. There is a slightly anomalous result with 125 kg/h CO_2 enhancement at 70kW for which there was also a low standard deviation of 0.14. On the test date all results of CO were 2-3ppm higher than other dates, this may be due to an unusual concentration in the fuel, or more likely there were slightly raised ambient levels of CO, such a value would not be exceptional [24, 25].

NOx is a tightly regulated and monitored emission. It was measured via the FTIR in terms of mg/Nm^3, using Equations 3 and 4; this has been converted to g/kWh (kg/MWh) at 15% O_2 [12]. The results, outlined in Figure 10, show an interesting consistent reduction in NOx with CO_2 enhancement. This trend can probably be attributed to the increased heat capacity of the oxidizer reducing peak temperatures and hence the production of thermal NOx [18]. All Turbine Inlet Temperatures recorded were higher without CO_2 enhancement, than with. The calculated values were 6-13°C lower, which will only be indicative of the true temperature reduction if measured. The lowest NOx emissions pattern also mirrored the indication that most efficient combustion and best performance of the combustor is 60-70kW.

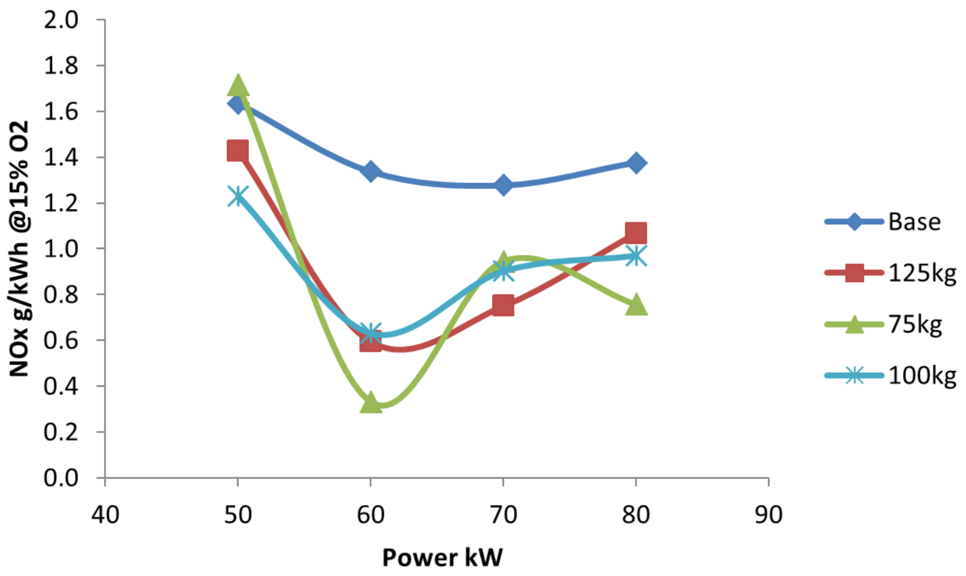

Figure 10: NOx emissions (g/kWh) at 15% O_2, with CO_2 enhancement.

Though 11 of 12 results show the same general behaviour of reduced NOx with CO_2 enhancement, the 75 kg/h case is more erratic. On further investigation of the raw data no evident reason such as variations in ambient temperature, TIT or fuel consumption could be found to explain this. For this reason the results are taken to be accurate.

3.2.3 Efficiency

Significantly poorer combustion would be expected to result in an increase in fuel consumption for maintaining the same power output with enhanced CO_2. Although fuel consumption data is in line with the combustion species observed, in that there is not a strong trend of increased fuel consumption with increased CO_2 enhancement. The apparent nominal increase in terms of specific fuel consumption (SFC) is a clear trend, detailed in Figure 11, with enhancement of 125kg/h increasing SFC 1.7 – 2.9%. This means CO_2 enhancement replicating EGR did reduce turbine power generation efficiency. It can be expected that this influence would be much more significant in a CCGT with significantly higher fuel flow rate and a richer fuel:air mix.

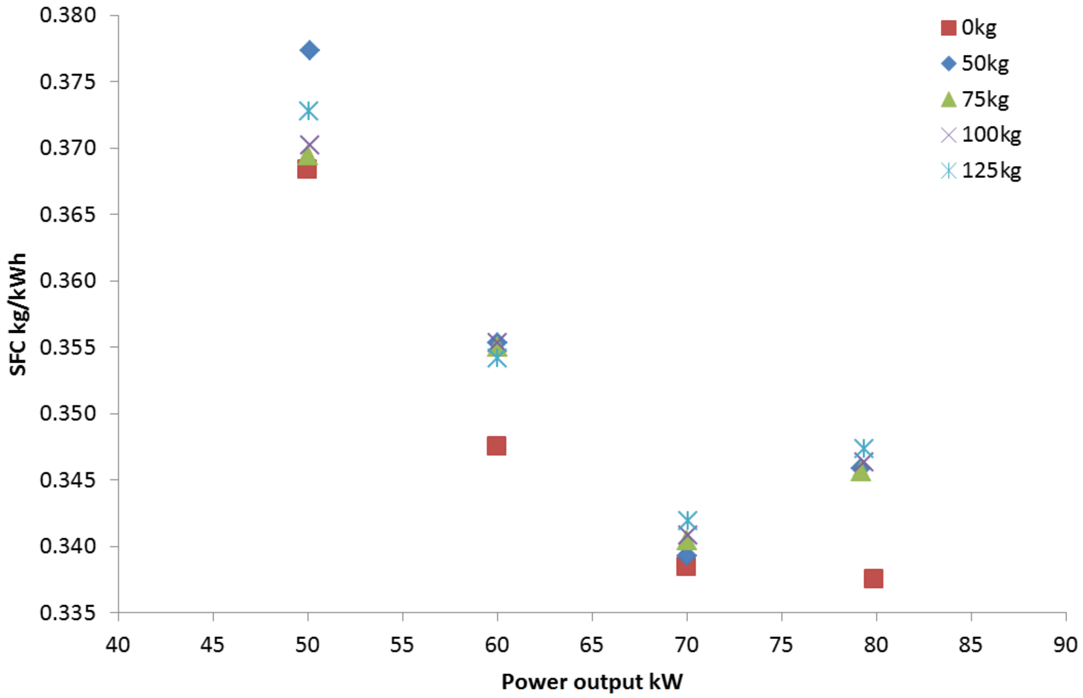

Figure 11: Specific fuel consumption with varying CO_2 enhancements.

The impact on combustion from enhancing the CO_2 concentration is not as disruptive as expected due to the low fuel:air ratio. This meant the CO_2 vol% was low, and a large enhancement of CO_2, representing over 50% EGR, up to 356%, caused a relatively small increase in CO_2, displacing less oxygen and nitrogen, thus leaving sufficient oxygen for combustion, which had less impact on the heat capacity of the air and the flame speed. The mass of fuel is calculated from the volume, and some of the points fall within the maximum potential instrumental error, however there is a significant visible trend that is unlikely to be anomalous with so many data points observed. Again the highest specific fuel consumption at 50kW and 50kg enhancement was from a test conducted on the warmest day, indicating the dominant influence ambient temperature has on the efficiency.

3.3 Impacts of CO_2 Enhancement on Post-Combustion Carbon Capture from Gas Turbines

As the regeneration of solvent is currently the most energy intensive step of a post combustion capture system, so reducing the relative mass flow for CO_2 captured is beneficial [10]. Models have shown increasing the concentration in the flue for absorption from less than 2 vol% to over 5 vol% can reduce the specific reboiler duty from 7.5 MJ/kg of CO_2 to below 4.0 MJ/kg of CO_2, a modelled reduction of over 45%.

The capture performance at the PACT facility is significantly less efficient than that. Tests were not conducted at 1.5% vol CO_2, but from 4.5% upwards in the exhaust gases, as to better represent CCGT emissions. At an exhaust gas concentration of 4.5% CO_2, regeneration was achieved at 8.3 MJ/Kg, whilst at 6.5% CO_2, it was 7 MJ/kg [26]. The closest comparable numbers in these tests are at 50kW, with 100kg of CO_2 enhancement giving 4.34%, and 125 kg giving 5.11 vol% CO_2. Akram, et al [26] extrapolates energy consumption dropping 7.5% per unit increase of CO_2, giving an estimate for regeneration of 8.4 MJ/kg at 4.34% and 7.92 MJ/kg at 5.11%, representing a 5.7% reboiler efficiency gain. When compared to the 1.7% to 2.9% increase in specific fuel consumption, this is a relative efficiency gain for the system with CO_2 enhancement simulating EGR at pilot scale. On a larger demonstration scale, much more significant gains can be expected, due to more efficient and less CO_2 intensive generation, and larger more efficient regeneration plant, and design.

4. CONCLUSIONS AND FUTURE WORK

The main results and their implications show a quantification of the expected results – reduced combustor performance with increased CO_2 addition that represented high levels of EGR. The CO_2 enhancement resulted in a clear impact on the 3 key areas investigated: on the turbine mechanically, on the emissions, and ultimately on the efficiency indicated by the specific fuel consumption.

The turbine performance, both pre and post enhancement, was significantly affected by the temperature of the oxidiser drawn into the turbine. Higher temperatures resulted in higher turbine speeds for consistent mass flow. With the addition of CO_2 to the oxidiser, lowering the temperature post compression relative to other 20°C ambient results. This can be clearly seen in Figure 8 where enhancement with 125 kg/h of CO_2 results in compression temperatures consistently 5°C cooler. The higher heat capacity of the CO_2 may have improved the efficiency of the heat exchanger being at a lower temperature. However it is to be expected the higher heat capacity and radiativity [18] also resulted in lower exhaust gas temperatures than without the addition, and hence lower temperatures at the hot side of the heat exchanger. This contributes to the overall lower efficiency of the turbine with enhancement, thus simulating the impact of exhaust gas recirculation.

Although there were small changes in the emission trends, the enhanced CO_2 results still showed an impact on the emissions. With high CO_2 addition, the higher CO and methane emissions evidenced an impact upon the combustion. At 50kW with 125 kg/h of CO_2 enhancement, CO emissions were up 109%, and unburnt CH_4 the primary constituent of the fuel up 338%. This is predominantly due to a mix of two factors, firstly through the displacement of oxygen used for combustion, and secondly due to increasing the heat capacity of the oxidiser thus causing lower temperatures, slower flame speed and reaction rates for complete combustion.

Also it is observed that there is an impact on NOx, where a decreased flame temperature can be expected to result in a reduction in the thermal NOx creation. This is apparent for 11 out of 12 results in Figure 10. The reduction in thermal NOx was small, varying from 0.20 g/kWh to 1.01 g/kWh, but there was a definite trend. Both with and without CO_2 addition, the turbine fell within the Greater London and German air quality emission limits of 5 g/kWh and 75 mg/m^3 at 15% O_2.

There were small changes in fuel efficiency and carbon intensity. Although a trend was observed that shows a decrease in efficiency, it was within the potential instrumentation error. However when examined in terms of specific fuel consumption, there was a definite trend seen, though these changes were relatively small, they were consistent over all data points with 125 kg/h increasing SFC 1.7 – 2.9%. This is the effect that was expected before the experimentation.

With the addition of CO_2 it was possible to calculate more precisely the total oxidizer mass flow through the system; this revealed that it is significantly higher than first postulated with the fuel:air ratio as lean as 1:105-130, compared to a CCGT of 1: 42-58 [23]. As a result the lean ratio means there was a less significant impact on the oxidizer composition than expected and hence the combustion was less dramatically affected.

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Nomenclature

CHP	Combined Heat & Power
CCS	Carbon Capture & Storage
EGR	Exhaust Gas Recirculation
LHV	lower Heating Value
PACT	Pilot-scale Advanced Capture Technology
PT	Pressure Transducer
RPM	Revolutions Per Minute
TC	Thermocouple
TIT	Turbine Inlet Temperature
TOT	Turbine Outlet Temperature
UHC	Unburnt Hydrocarbon
UKCCSRC	UK Carbon Capture Storage Research Council