Experimental Investigation of Lean Premixed Combustion with Exhaust Gas Recirculation in a Swirl-Stabilized Turbine Burner

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Introduction

Responding to the necessity of a low carbon economy, Gas Turbines (GTs) and micro Gas Turbines (mGTs) coupled with a Carbon Capture (CC) plant have been recently identified as an attractive solution for the energy transition period [1]. As a matter of fact, the future electricity system will be forced to become increasingly flexible and able to adapt to fluctuating conditions, mostly because of the significant share of renewables [2]. Accordingly, GTs and mGTs would play a decisive role due to their high load flexibility. However, although their CO_2 emissions when burning natural gas are lower than those of other fossil fuels (e.g. coal-fired plants), the constraint of a zero/negative carbon emissions will anyhow drive the deployment of CC on large- and even small-scale turbine cycles. Considering their traditional configurations, mGTs and GTs are not suitable for CC applications: the low concentration of CO_2 in the exhaust gases is disadvantageous for the capture process, demanding a high energy input leading to a major overall efficiency penalty [3]. As a matter of fact, the flue gas produced by a GT contains 3-5% CO_2 concentration which is very low compared to 13-15% CO_2 concentration in the exhaust gas produced by coal fired power plants, where CC is traditionally envisioned. At a smaller scale, with mGTs, this problem is even more accentuated due to the lower concentration of CO_2: the absence of turbine blade cooling compels a high excess air in the cycle, reducing the CO_2 concentration in the exhaust gas to around 1.5% [4]. To overcome this obstacle, various innovations have been proposed focusing on the modifications of the traditional configuration of the turbine cycles, such as Exhaust Gas Recirculation (EGR). EGR consists in reinjecting a fraction of the exhaust gas in the compressor inlet of the cycle to increase the CO_2 content of the oxidizer and thus the flue gas. Applying this measure, three main advantages can be achieved: lower NO_x emissions, a reduced exhaust gas mass flow rate and a higher CO_2 concentration in the flue gas [5]. In contrast, it can also reduce combustor performance and affect pollutant emissions of the turbine plant [6]. Several studies addressing the impact of EGR on GTs and mGTs are available in literature. From a large-scale point of view, several numerical studies have been carried out which prove the advantages of EGR [7-9]. On the other hand, only a few experimental investigations are available which analyze the effects of EGR, focused to the turbine combustor (GE's Dry Low NO_x Combustor) [5]. From a small-scale point of view, several studies analyzed the effect of EGR on mGT focusing mainly on the CC unit side [10-11]. Focusing only on the mGT operation, Best et al. performed CO_2 injection in a Turbec T100, considering the effects on turbine and capture performance [6]. However, the amount of CO_2 injected largely overcome the maximum amount of CO_2 that is recirculated with a real EGR. Moreover, they neglected the high fraction of recirculated nitrogen and oxygen in the system. Nevertheless, real experimental studies on EGR application applied to mGTs has not yet been presented, which indicates the importance of the experiments presented in this study. In this abstract, we will present the results of combustion experiments performed with EGR which was reproduced by means of CO_2 and N_2 injection. The experiments were performed in the atmospheric, variable-swirl, premixed turbine combustor of the Thermal Power Engineering division of the Department of Energy Sciences of Lund University, Sweden. In this study, we fo-

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cused on the Lean Blowout (LBO) limit and CO emissions of methane combustion. By measuring the CO emissions and low-frequency acoustic sensors, we tried to predict how the fuel control of the mGT needs to be changed to get a stable combustion under EGR conditions and avoid LBO.

Experimental setup

The experiments were carried out in an atmospheric, variable-swirl, premixed burner with circular cross section. The burner has a total length of 120 mm and an inner diameter of 60mm. A quartz tube with the same inner diameter as the burner provides optical access to the flame, enabling the visual detection of LBO. A swirling flow of air and methane was supplied to the combustor through a centrally located premixing tube. Both axial and tangential air flows are combined in the swirler, located at the entrance of the premixing tube. The swirler allowed for the introduction of air into the premixing tube in axial and tangential directions in varying proportions. In this study, the axial flow has always been set to zero and only the tangential flow has been used to reproduce the high swirl numbers of a mGT combustor (typically around 0.66 [12]). The tangential flow was measured and preheated using a laminar-flow, differential-pressure mass-flow controllers (Alicat MCR250) and feedback-controlled air heaters (Sylvania Sureheat Jet) of 8kW power. The inlet air temperature was controlled by adjusting the air heaters and measuring the temperature in the center of the inlet before and after the experiments using a K-type thermocouple. The desired methane fuel flow was generated by a laminar-flow differential-pressure mass-flow controller (Alicat MCR50). CH_4 (purity 99.98%) was supplied from gas bottles and subsequently mixed with the air flow before the combustion chamber. The exhaust gases from the combustor were discharged into a force-ventilated extractor hood. To reproduce the effect of EGR, CO_2 and N_2 injections have been implemented using external gas bottles. The vitiated air composition as function of the EGR ratio has been calculated with an Aspen Plus simulation and reproduced with a precise combination of air, CO_2 and N_2. As well as in the methane regulation, the desired CO_2 and N_2 flows were generated by a laminar-flow mass-flow controller (Alicat MCR50). The vitiation gases flows have been mixed with the air flow before the heater, in order to maintain the desired CIT.

By expressing the combustion efficiency as a function of the air mass flow rate, the rate of reaction, evaporation and mixing, Walsh and Fletcher [13] have shown that with a limited number of experimental results the combustion efficiency of a GT combustor can be estimated over a wide operating range. For both the mGT combustor and atmospheric combustion chamber used in the experiments of this paper, the fuel is gaseous and premixed and therefore the combustion efficiency can be assumed to be reaction controlled for both burners. Therefore, the expression of efficiency, provided by Walsh and Fletcher [13] can be simplified to the so-called ‘Combustion Loading’ or *CL* parameter to correlate efficiency:

$$CL = \frac{\dot{m}}{\dot{V} \cdot P^{1.8} \cdot 10^{0.00145 \cdot (T_{in} - 400)}}$$

By using this parameter, it is possible to use experimental obtained efficiencies from tests performed on lab scale test rigs (commonly at atmospheric pressure) to estimate the combustor efficiency under real operating conditions (at elevated pressure) when the combustor is installed in an actual test rig. Depending on the load of the mGT (the mGT is operated at constant power output conditions by changing the rotational speed), inlet conditions vary when changing the power output. For different power outputs, ranging from 60 to 100kW$_e$ the air mass flow rate changes from 0.587 to 0.737 kg/s, the inlet pressure of the combustor p_{in} from 3.35 to 4.34 bar, while the Combustor Inlet Temperature (CIT) remains approximately constant (a small reduction from 595 to 590 °C). Making the assumption that only 30% of the total mass flow of air is used to in the combustion (primary air) and the remaining 70% used as subsequent dilution (dilution air), the equivalence ratio along the power output span varies from 0.38 to 0.48. These operating parameters of the Turbec T100 mGT have been obtained from a validated Aspen Plus numerical model [4]. With these parameters the CL at 100kW is 3.29 and at 60kW is 4.10. Hence, the experimental campaigns were there set to be as close as possible to the highest CL as precautionary limit. The following parameters have been used: mass flow of air of 0.0039 kg/s (200 SLPM), pressure of 1 atm, CIT of 454 ^{o}C, with a resulting *CL* parameter of 0.412. This indicates that similar combustion efficiencies can be expected between the atmospheric combustion chamber and the Turbec T100 combustion chamber.

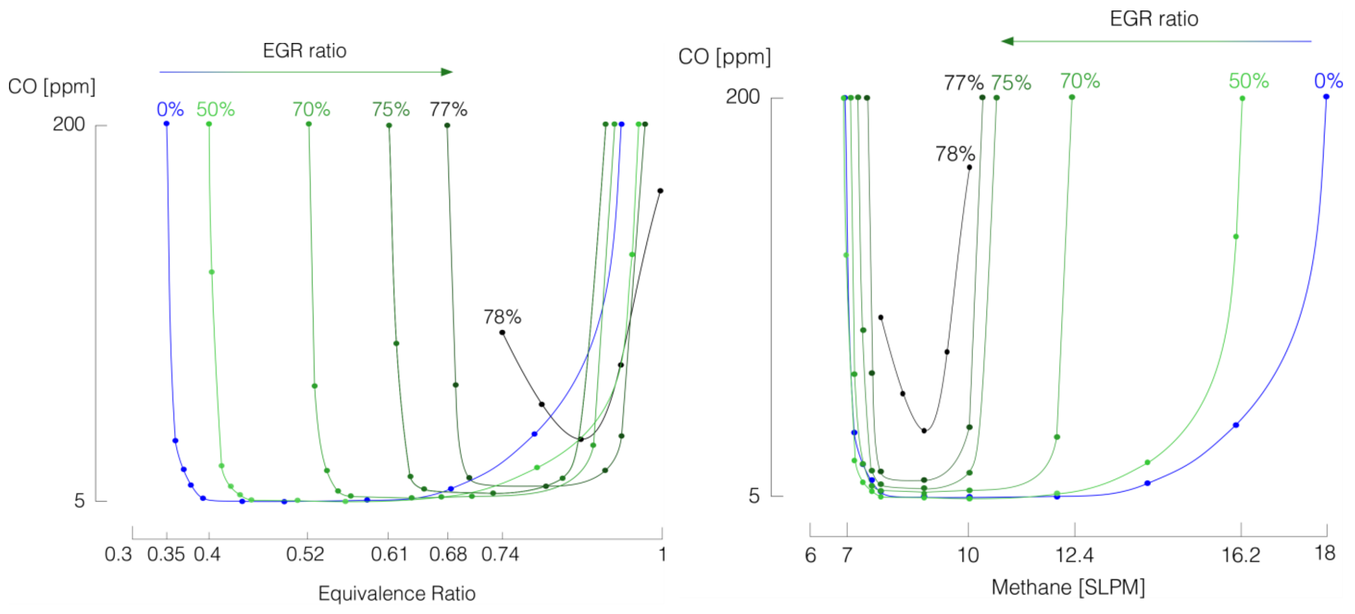
LBO and unstable combustion detection

The focus of the presented work was the investigation of the effect of EGR on the flame stability and LBO limit for methane combustion at different equivalence and EGR ratios (EGR ratio is defined as the ratio between the recirculated volumetric flow and the total volumetric flow of flue gas). The swirl number, the total volumetric flow and CIT were kept constant. Tests were only performed at high swirl numbers, which are representative for mGT combustion chambers.

To the author's knowledge, no standard procedure for LBO-limit determination exists. Hence the following procedure has been adopted (the same as in [12]): a rich mixture (equivalence ratio close to 1) of pure air and methane was ignited. After ignition, the air flow was gradually replaced with vitiated gases in the right proportion calculated by the Aspen numerical model. Once a steady condition is achieved, the equivalence ratio was slowly reduced by gradually reducing the methane fuel flow rate until LBO occurred. After each change of the fuel flow rate, the combustion was maintained unvaried for at least 2 minutes to stabilize the flame and to allow the combustion chamber to reach thermal equilibrium. CO emissions were captured for each equivalence ratio and EGR ratio. The sampling probe consisted out of a metal tube with different openings along the burner radius, to get a representative sample of the total exhaust gas emissions. The water in the flue gases was condensed before entering the CO measuring device (Rousemount BINOS). The device was calibrated each experimental campaign and compensated for the changing atmospheric pressure. The CO emissions were measured and averaged over a period of 2 minutes in which the air dilution and equivalence ratio were kept constant. The maximal CO concentration that could be measured was 950 ppm with an accuracy below 1% of the full scale at constant temperature and pressure.

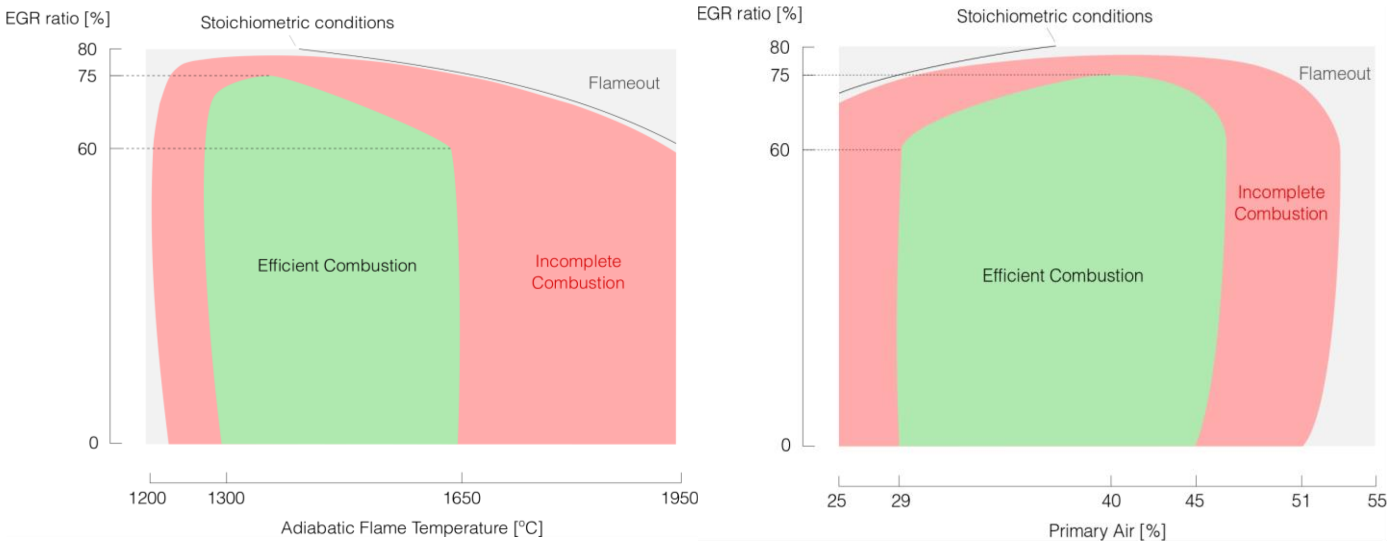
Results

Correcting CO to 15% O_2 may not be appropriate, since the O_2 in the exhaust when applying EGR will be lower than in the case of traditional combustion; the lower O_2 results mean NO_x appear to be higher [5]. Therefore, the values have been reported as shown in the CO analyzer. While in traditional combustion, it is customary to represent CO emissions as a function of the equivalence ratio, in EGR condition this may be misleading since changing EGR ratio influences the concentration of oxygen and thus the equivalence ratio. For this reason, the resulting CO emissions have been represented both as function of the equivalence ratio (Figure 1a) and of the volume flow of methane in SLPM (Figure 1b). It is important to note how the increasing EGR ratio gradually decreases the operational range of our combustor. This is mainly due to the lack of oxygen in the rich region which does not permit an efficient combustion. Nevertheless, the effect of EGR on the lean region is relatively limited and the curves overlap up to an EGR ratio of around 60%. Considering the operating conditions of the burner where the CO emissions were limited to a value below 10 ppm and no combustion stabilities were detected, it is possible to draw area charts which represent the *efficient combustion* zone, *incomplete or unstable combustion* zone and *flameout* zone. These areas can be drawn as function of the EGR ratio and either the adiabatic flame temperature (Figure 2a) or, from a real mGT point of view, the fraction of total flow of air which participates to the combustion process, the aforementioned primary air (Figure 2b). Considering the primary air ratio of a real mGT approximately 30% of the total mass flow rate, the maximum EGR ratio that can be applied maintaining an efficient combustion is around 60%. This result is in agreement with that of Elkady et al. [5] obtained for a DLN F-class combustor (General Electric).



a) b)

Figure 1: The operational range of the burner gradually decreases while increasing EGR ratio, up to a maximum of 78% where it was not possible to find an efficient combustion regime.



a) b)

Figure 2: The efficient operation (CO emissions below 10 ppm and no flame fluctuations) of the burner under EGR conditions can be represented as function of the adiabatic flame temperature or as function of primary air which is involved in the combustion in real conditions. With an EGR close to 60%, the richer boundary of the efficient combustion area shows a “knee”, where leaner conditions are required to maintain a high combustion efficiency.

References

[1] M. A. Gonzalez-Salazar, "Recent developments in carbon dioxide capture technologies for Gas Turbine Power Generation", International Journal of Greenhouse Gas Control 34 (2015) 106-116.
[2] W. F. Pickard, "Addressing the intermittency challenge: Massive energy storage in a sustainable future", Proceedings of the IEEE 34 (2012) 106-116.
[3] M. M. Majoumerd, H. Nikpey Somehsaraei, M. Assadi, P. Breuhaus, "Micro gas turbine configurations with Carbon Capture – Performance assessment using a validated thermodynamic model"Applied Thermal Engineering 73 (1) (2014) 172-184.
[4] S. Giorgetti, L. Bricteux, A. Parente, J. Blondeau, F. Contino, W. De Paepe, "Carbon capture on micro gas turbine cycles: assessment of the performance on =dry and wet operations".
[5] A. M. Elkady, A. Evulet, A. Brand, T. P. Ursin, A. Lynghjem, "Application of exhaust gas recirculation in a DLN F-class combustion system for post-combustion carbon capture", Journal of Engineering for Gas Turbines and Power 131 (3) (2009) 034505.
[6] T. Best, K. N. Finney, D. B. Ingham, M. Pourkashanian, "CO_2-enhanced and humidified operation of a micro-gas turbine for carbon capture", Jorunal of Cleaner Production, 176 (2018) 370-381.
[7] A. M. Alcaraz-Calderon, M. O. Gonzalez-Diaz, A. Mendez, J. M. Gonzalez Santalo, A. Gonzalez-Diaz, "Natural gas combined cycle with exhaust gas recirculation and CO2 capture at part load operation", Journal of the Energy Institute.
[8] A.D. Rao, W.H. Day, "Mitigation of greenhouse gases from gas turbine power plants", Energy Conversion and Management 37 (6) (1996) 909-914.
[9] H. Li, M. Ditaranto, D. Berstad, Technologies for increasing CO2 concentration in exhaust gas from natural gas-fired power production with post-combustion amine-based CO2 capture, Energy 36 (2) (2011) 1124-1133.
[10] U. Ali, K. J. Hughes, D.B.Ingham, L. Ma, M. Pourkashanian, "Effect of the CO2 enhancement on the performance of a micro gas turbine with a pilot-scale CO2 capture plant", Chemical Engineering Research and Design 117 (2017) 11-23.
[11] M. Akram, U. Ali, T. Best, S. Blakey, K. Finney, M. Pourkashanian, "Performance evaluation of PACT pilot-plant for CO2 capture from gas turbines with exhaust gas recycle", International Journal of Greenhouse Gas Control 47 (2016) 137-150.
[12] W. De Paepe, P. Sayad, S. Bram, J. Klingmann & F. Contino, "Experimental Investigation of the Effect of Steam Dilution on the Combustion of Methane for Humidified Micro Gas Turbine Applications", Combustion Science and Technology, 188 (8), April 2016.
[13] P. P.Walsh, P.Fletcher, "Gas Turbine Performance – Second Edition" Blackwell Science (2004).