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Enabling efficient and low emissions deep part load operation of combined cycles and combined heat and power plants with external Flue Gas Recirculation

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ABSTRACT

Gas Turbine (GT) Combined Cycles (CCGT) and Combined Heat and Power (CHP) plants show in general a significant drop in cycle efficiency and increase in emissions with part load control techniques such as turbine inlet temperature regulation, inlet guide vanes or compressor air recirculation. Through the use of Flue Gas Recirculation (FGR), hot stack flue gases are recirculated to the GT inlet, leading to lower GT mass flow and lower power production at a better efficiency and lower NOx emissions. Information on FGR as a part load technique is limited. Hence, we investigate the effects of traditional controls versus FGR on CCGT and CHP cycle part load efficiency from a thermodynamic and combustion point of view. Results show that FGR could increase CHP cycle efficiency at part load up to 11% and CCGT efficiency up to 2% compared to other control techniques. Combustion stability limits, NOx emissions and other technical limitations are discussed. The CHP case is used to assess the potential economic and environmental benefits from the use of FGR.

INTRODUCTION

Gas turbine based combined heat and power (CHP) and Combined Cycle (CCGT) plants have originally been designed for full load operation, especially those older than 10 years. However, the increasing share of renewable energy and its intermittent nature pushes the electrical power generation towards more flexibility and forces CHP and CCGT plants to operate at part load conditions. Traditional load control strategies first reduce the load by changing inlet guide vane (IGV) positions until around 80% load, at which compressor surge limit is reached. The turbine inlet temperature (TIT) is then reduced for lower load operations. Under these conditions, the burner switches from premixed to non-premixed mode to prevent unstable combustion due to lower equivalence ratios. This results in increased NOx emissions. The standard operational part load range for gas turbines with lean premixed combustion is therefore limited to typically 80%-100%. Non-premixed operation limits are given by NOx emissions and the constant power required by the connected steam or heat cycle.

A solution to this dilemma is the use of Flue Gas Recirculation (FGR) for which flue gas from the exhaust of the Heat Recovery Steam Generator (HRSG) are recirculated to the compressor inlet, as shown in Fig. 1 for a CHP plant. A CCGT configuration is similar. FGR has the potential to enable efficient deep part load operation and to reduce emissions compared to traditional control techniques as IGV, TIT control and compressor air recirculation (CAR, bleed heat). Indeed, it changes the content of the gas turbine working medium, which results in lower oxygen and thereby lower NOx emissions. If the recirculated flue gases stay uncooled, the inlet temperature of the gas turbine will rise resulting in a lower mass flow through the installation and lower power production at a higher efficiency due to stack losses reduction. Furthermore, FGR ramp rate is high and modulated or switched off easily. All these aspects make FGR a very prospective technology to be applied to gas turbine based CHP and CCGT installations.

Yet, FGR application for part load operation is not straightforward. The recirculation of flue gases is a common technology for achieving very low NOx and CO emissions in furnaces. The combustion process is then characterized by strong dilution of the (pre-heated) oxidizer and/or fuel flows

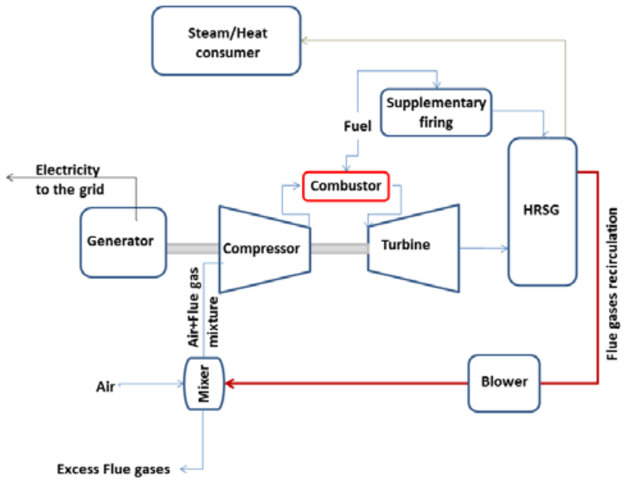

Figure 1 Flue Gas Recirculation for a gas turbine CHP plant

through internal recirculation, which results in an increase of chemical time scales and a more gradual heat release (Wunning and Wunning(1997)). However, the application of (external) FGR to gas turbines is less common and most studies are mainly motivated by increasing CO_2 levels in flue gases for capture (Bolland and Mathieu(1998); Rokke and Hustad(2005); Cameretti *et al.*(2009); Evulet *et al.*(2009); Li *et al.*(2011); Merkel *et al.*(2013); Lewis *et al.*(2014); Akram, Blakey and Pourkashanian(2015)). A small number of experiments on real scale gas turbine combustors have been performed. Investigations of a GE-F class combustor (ElKady *et al.*(2009)) showed that it can operate in a range of FGR rates up to 30% to 35% with standard components. Tanaka *et al.*(2013) further showed that a combustor with FGR could be operated at up to 1700°C with NOx emissions below 50 ppm and CO emissions below 10 ppm. However, Guethe, de la Cruz Garcia and Burdet(2009) showed that FGR increases the combustion time scales significantly leading to flame stability problems and increased CO emissions. Other investigations (Evulet *et al.*(2009); Best *et al.*(2016)) have reported on combustion stability issues and increased CO emissions when FGR is applied. FGR with bio-fuels showed the same promising results in terms of CO_2 capture (Cameretti, Tuccillo and Piazzesi(2013)) or with different control strategies to increase the revenues in small CHP (100kWe) (Camporeale *et al.*(2014)).

However, there is a lack of detailed study of the effect of the thermodynamic cycle variation on part load operation efficiency and emissions. Hence, in this paper we investigate the effects of traditional controls versus FGR on CCGT and CHP cycle part load efficiency with thermodynamic considerations and thermodynamic cycle simulations. Combustion stability limits, NOx emissions and technical limitations are discussed. The CHP case is used to assess the potential economic and environmental benefits from the use of FGR

METHODOLOGY

A CHP and a CCGT cycle are studied. Thermodynamic considerations and Thermoflex simulations help to assess the impact of FGR for part load operation for both cycles. The results from these simulations provide data for the estimation of GT NOx emissions. The cycles and the methodology are presented in this section.

Cycles and design conditions

The cycles are based on generic plants. The CHP is based on the General Electric GT GE 6531B. At the design point the TIT is 1104 °C, for a pressure ratio of 11.7 and a mechanical power of 38.27 MW. When the exhaust gases exit the GT, they are heated up in the supplementary firing unit (SFU) before entering the HRSG. The latter is made of three heat exchangers: the economizer, evaporator and super-heater. The water pressure is increased by a water pump up to the evaporation pressure of 23.5 bars and heated up to 285ºC in the HRSG. The HRSG recovers 65 MW of heat from the flue gases temperature by decreasing the temperature from 551.6 to 141ºC. The steam is produced with a flowrate of 22 kg/s and is useful for any process application (district heating, chemical applications, food industry, etc.). The pinch point conditions are verified for each off-design working condition.

For the CCGT plant, the main difference is the conversion of HSRG heat into electricity. In the studied Rankine cycle, the steam produced is expanded in a steam turbine (ST) with three pressure level (4.2, 25 and 100 bars at base load) in order to generate electrical energy. The CCGT load is defined as the sum of the GT load and the Steam cycle load (ST). Given that no supplementary firing unit is placed at the turbine exhaust, the GT exhaust temperature is the same as the boiler inlet temperature and will change with the control technique. The boiler inlet temperature is limited by the wall insulation material properties of the boiler and of the turbine, a maximum value of 650 ºC is assumed. For this case study, a General Electric 6101FA GT is selected. At the design point, the TIT is 1287.8 ºC, for a pressure ratio of 14.8 and a mechanical power of 69.29 MW. The boiler inlet temperature at base load is 598 ºC. Turbine efficiencies are computed with Thermoflex and their off design behavior has been validated in Van den Oudenalder(2016).

Part load control technique and their limits

Different control techniques allow to operate the gas turbine at part load. They present different limits that define their operational range. The following techniques will be discussed here:

- Flue gas recirculation (FGR)
- IGV control, the standard part load technique
- TIT control
- Compressor air circulation, bleed heat (CAR)
- Air Preheating

With FGR a controlled splitter divides the flue gases stream into two main flows as in Fig. 1. The FGR ratio controls directly the percentage of flow recirculated. At full load, no flow is recirculated and all the exhaust gases head to the stack. Mixing a fraction of exhaust gases with the inlet airflow (at ambient temperature) causes the compressor inlet temperature to rise and reduces the mass flow rate through the compressor. The pressure ratio decreases due to the lower turbine flow. This links the part load operation with the FGR ratio, as shown in Fig. 2 for the CHP cycle. However, when mixing the exhaust gases with ambient air, the molar composition of the inlet flow changes since the flue gases are rich in CO_2, H_2O, etc. Therefore, O_2 concentration at the GT inlet diminishes with the FGR ratio as shown in Fig. 2. This has an important impact on combustion stability, as discussed in ElKady *et al.*(2009). The minimum oxygen concentration for stable combustion was found to be 14.5% at the GT inlet and this value is taken as a limit for FGR application in this study. Turbine cooling due to a new medium and aerodynamics effect another potential limit on the amount of FGR. They might also affect gas turbine components and their performance and temperature.

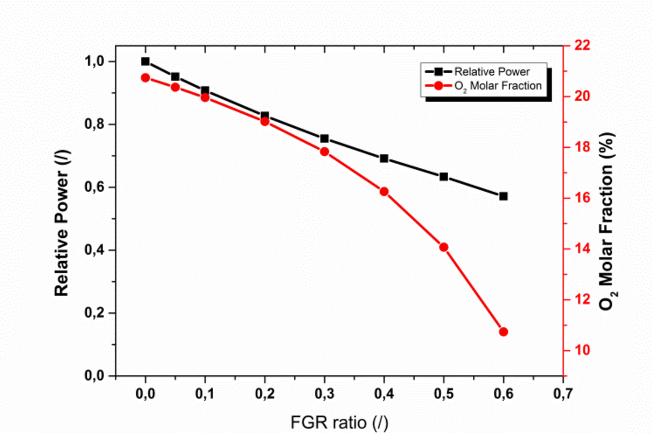

FIGURE 2 Effect of Flue Gas Recirculation ratio on the power generation and oxygen content in the inlet gases. FGR ratio is the recirculated flow over the total flow in the CHP plant

The different part load control techniques are described hereafter. IGV controls make use of variable stator blades in the first section(s) of the compressor. Changing the angle of these blades changes the axial velocity and thus the mass flow rate. This leads to a lower power production of the GT. However, IGV can only be used down to an operation of around 80% load. When the angle is increased furthermore, compressor surge limit is normally reached.

Turbine Inlet Temperature control (TIT) aims at reducing the inlet temperature of the turbine by reducing fuel injection in the combustor (while the air flow rate stays constant). For a pre-mixed combustor, TIT control can only be used in a very limited range of flame temperatures, as the flame becomes unstable due to the very lean operation. The combustor has then to be switched from pre-mixed to non-premixed operation. This has a negative impact on NOx emissions.

Compressor Air Recirculation (CAR, also known as bleed heat), aims at increasing the compressor inlet temperature and to reduce power production. The recirculated compressor exit hot flow is expanded in a valve and mixed with ambient air. Power production is thus reduced by the combination of the reduction of the inlet air mass flow rate and the lower flow in the turbine due to CAR. However, losses are important due to the expansion of the gases through the valve.

In the case of air preheating, gas turbine inlet air is preheated by an external source, in most cases low pressure steam via a separate heat exchanger network. Air preheating is normally designed for anti-icing and has a small operational area. As for FGR and CAR, the increase in inlet air temperature leads to a lower mass flow through the compressor. If the heat source is freely available (e.g. waste heat), the thermodynamics of the preheating are comparable to FGR. If the preheating induces a reduction in steam cycle power, it will affect cycle efficiency. Due to the similarities with FGR, air preheating is not discussed separately.

When these control techniques are used in a CHP plant, the GT exhaust gas flow rate and temperature change and the supplementary firing unit needs to be adapted to ensure a constant steam flow to the application. These cycle and control limits are implemented in a Thermoflex simulation to assess realistic operation and controls of the gas turbine.

Emissions

Nitrogen oxides emissions (NO_x) are a major concern in modern gas turbines. They are generated in the combustor according to mainly 4 pathways (Correa(1993)): Thermal NO_x; prompt NO_x; Nitrous Oxide Mechanism and NNH. These mechanisms show that NOx formation is dependent on temperature, pressure, species concentrations and residence time in the combustor. They will thus change according to the part load technique used. Advanced chemical reactor network modelling or CFD simulations allow for deeper insights on NOx generation in the combustor, but are out of scope of this study.

In gas turbines both premixed and non-premixed combustion are used. For premixed traditional combustion, emissions at base load are much lower than for non-premixed combustion due to the lower flame temperature of the fuel-air mixture versus the stoichiometric flame temperature for non-premixed combustion. With premixed FGR, experimental studies (Evulet *et al.*(2009); Tanaka *et al.*(2013)) indicate that flue gas recirculation has a very positive effect on the reduction of NOx emissions compared to fresh air combustion. This goes up to a factor of 2 at the same flame temperature and is mainly due to the lower reaction rates due to the lower oxygen concentration in combination with the increase of H_2O and CO_2 concentration. ElKady *et al.*(2009) show that at same flame temperatures and pressures, there is a reduction of 40% of NOx emissions compared to standard combustion, for a FGR ratio of 25%. Tanaka *et al.*(2013) and Evulet *et al.*(2009) furthermore show that for the same recirculation ratio, higher flame temperatures lead to higher reduction in NOx emissions. These results can be used to have a first indication of how the emissions will be affected by FGR compared to base load operation.

No literature is available for the impact of FGR on the NOx emissions of non-premixed combustion in gas turbines, although FGR application in non-premixed combustion in boilers and furnaces has been applied very successfully the last decades for the reduction of NOx (Cavaliere and de Joannon(2004); Cho *et al.*(2011)). For a non-premixed gas turbine, Fig. 3, adapted from Becker and Perkavec(1994), shows that the emissions are exponentially dependent on the variation of the stoichiometric flame temperature. With FGR in non-premixed gas turbines, the main impact will be due to the flame temperature (as residential times stay roughly constant). The trend of Fig. 3 can thus be used to characterize the effect of FGR on the stoichiometric adiabatic flame temperature and provides an indication on NOx emissions due to oxidiser composition changes, compared to base load operation.

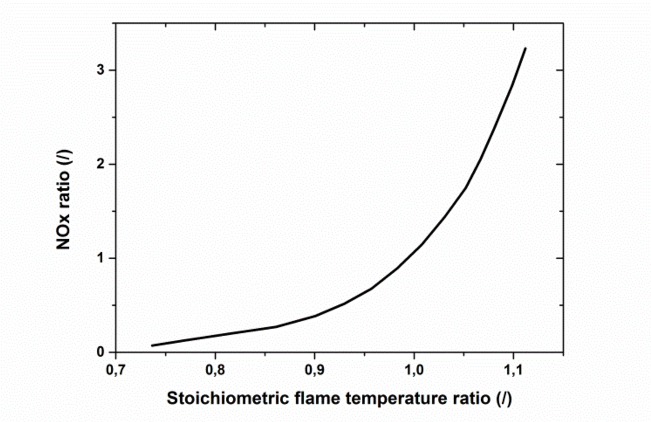

FIGURE 3 Evolution of the NOx ratio in function of the ratio of stoichiometric flame to fresh air combustion with natural at base load, as taken from Fig. 2 of Becker and Perkavec(1994).

To estimate base load and part load emissions with fresh air and traditional controls in industrial gas turbines, Becker and Perkavec(1994) developed a semi-empirical correlation able to simulate part load emissions. This formulation has been validated for three gas turbine types operating on a non-premixed mode, including the Frame 6 used in this study. In part load operation, emissions are dependent on changes in adiabatic mixture temperature in the primary zone, pressure and changes in residential time in the combustor. For this study, the combination of the thermodynamic cycle simulations for fresh air combustion on the Frame 6 gas turbine and the semi-empirical correlation have been validated against experimental data of Becker and Perkavec(1994).

CO emissions will be increased due to depletion in oxygen and increase of CO2 content at the inlet. UHC are related to combustion stability and will be affected by FGR. These emissions are studied in a future paper to assess the exact impact of FGR.

THERMODYNAMIC ANALYSIS

System energy balance

By considering the CHP system as a black box and assuming no other losses than stack losses, the system energy balance can be derived as Eq. 1.

$$\dot{m}_f LHV + \dot{m}_a h_{amb} + \dot{m}_w h_{in} = P_{el} + (\dot{m}_a + \dot{m}_f) h_{stack} + \dot{m}_w h_{out} \quad (1)$$

The system net fuel input ($\dot{m}_f LHV$) is the sum of the fuel required in the combustion chamber and in the SFU. It is converted into electrical power (P_{el}) in the GT and in process heat ($Q_{HRSG} = \dot{m}_w h_{out} - \dot{m}_w h_{in}$) in the HRSG. Equation 1 is transformed in Eq. 2 by considering that the stack enthalpy can be neglected compared to fuel LHV ($h_{stack} / LHV \simeq 0.31\%$).

$$Q_f = P_{el} + Q_{HRSG} + \dot{m}_a (h_{stack} - h_{amb}) \quad (2)$$

The efficiency of the CHP plant is thus defined as follows.

$$\eta_{CHP} = \frac{P_{el} + Q_{HRSG}}{Q_f} \quad (3)$$

At part load, the electrical power of the CHP plant is diminished by

$$\Delta Q_f = \Delta P_{el} + \Delta Q_{HRSG} + \Delta Q_{stack} \quad (4)$$

For a CHP operation, process heat must be kept constant ($\Delta Q_{HRSG} = 0$) and at part load the efficiency becomes

$$\eta_{CHP,PL} = \frac{P_{el,FL} + \Delta P_{el} + Q_{HRSG}}{\Delta P_{el} + \Delta Q_{stack} + Q_{f,FL}} \quad (5)$$

For different control techniques of the same CHP plant at part load, the terms $Q_{HRSG}, \Delta P_{el}\ P_{el}$ are constant. Therefore, the efficiency will be directly related to Q_{stack} of each technique.

For the combined cycle, based on the same reasoning and the fact that there is no heat export from the HRSG, it can be found that part load efficiency can be expressed as

$$\eta_{CC,PL} = \frac{P_{el} + \Delta P_{el}}{\Delta P_{el} + \Delta Q_{losses} + Q_f} \quad (6)$$

In this case, losses are depicted as the sum of stack losses and condensation loss. The latter depend on the off design steam/water flows, thus on the steam turbines load.

Part load operation

Computing the cycles using the Thermoflex software allows to study the GT and cycle off-design conditions. It provides a direct insight in thermodynamic parameters of the complete cycle. Figure 4 shows the evolution of the temperature at different locations in the CHP cycle, relative to design values at base load. It is given for the maximum part load limits (=minimum loads) achieved by each control technique. This figure helps to understand how the different control techniques affect the cycle. TIT control reduces the GT exhaust temperature down to 70% of the design value and the required corresponding additional heat in the SFU is thus high. As the stack mass flow rate does not change at part load, the HRSG temperature stays constant compared to base load. Therefore, the stack losses reduction due to part load operation is almost null. For IGV, CAR and FGR controls, the exhaust gases of the GT present a higher temperature than at design, but as the mass flow is reduced, additional heat is provided in the SFU to enable the HRSG to deliver 65 MW process heat. As the final HRSG temperature and stack flow are lower than at design, stack losses are reduced. Pinch points conditions are ensured by Thermoflex, and a warning signal is send if the condition is not respected. However, as the variation according to the design point are different for IGV, CAR and FGR, the stack loss reduction will not be the same. Figure 5 shows this for the complete range of part load of each control technique. It shows that the highest reduction in stack losses is provided by FGR control technique, while TIT almost does not change the stack losses.

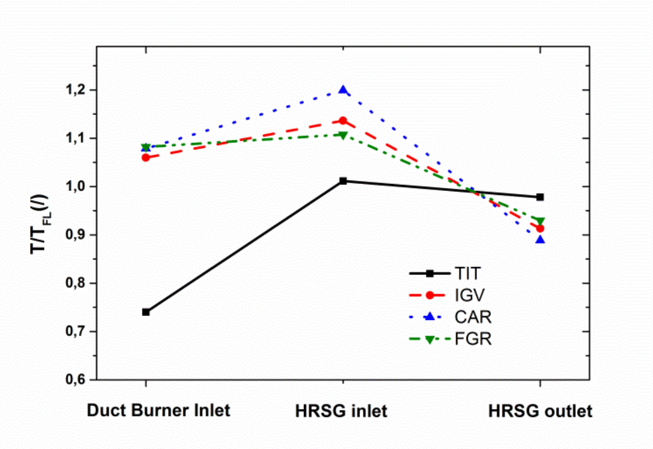

FIGURE 4 Evolution of temperature over the full load temperature through the CHP cycle for different control techniques at minimum part load operation (TIT: 0.5; IGV: 0.82; CAR: 0.5; FGR: 0.68)

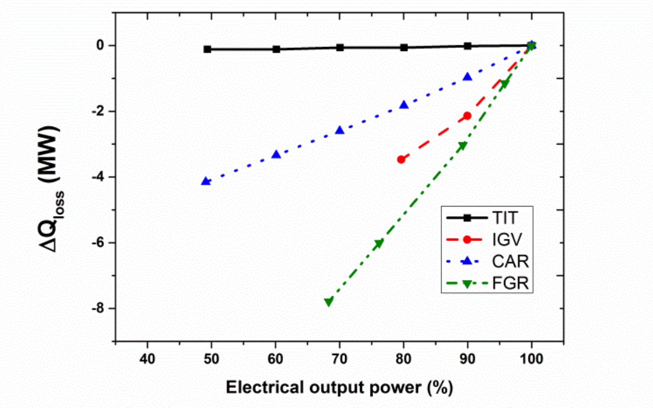

FIGURE 5 Evolution of stack losses for different control techniques with part load for the CHP plant

These considerations also allow to compute the cycle efficiency which, as explained in Eq. 6, will depend on the stack losses. Figure 6 shows that the efficiency gains are the highest with FGR. The efficiency increases for FGR, CAR and IGV controls at part load, but decreases with TIT control.

The same analysis can be performed for the Combined Cycle. Figure 7 shows the evolution of the CC efficiency at part load conditions. All the off-design operation with Thermoflex of the CC have been validated in Van den Oudenalder(2016). Although condensation losses are higher for the FGR control as ST load stays almost constant, Recirculating exhaust gases to the GT inlet is found to be the most efficient control technique. However, since the steam turbines load stay almost constant, the combined cycle load can be decreased only down to 86% to keep the O_2 levels at the inlet of the gas turbine higher than 14.5%. The second most efficient control is found to be IGV followed by CAR and TIT controls.

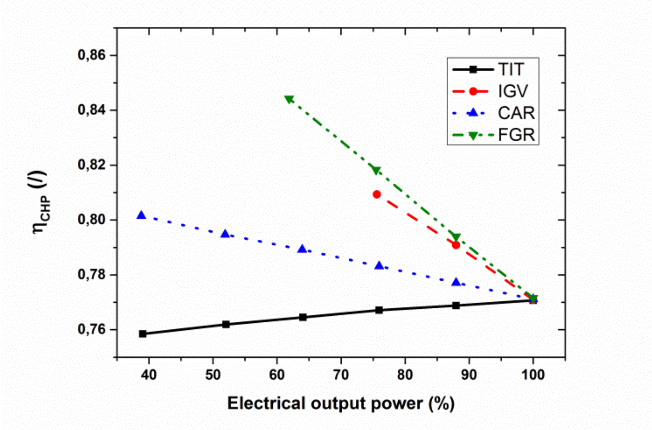

FIGURE 6 Evolution of the CHP efficiency for different control techniques with part load

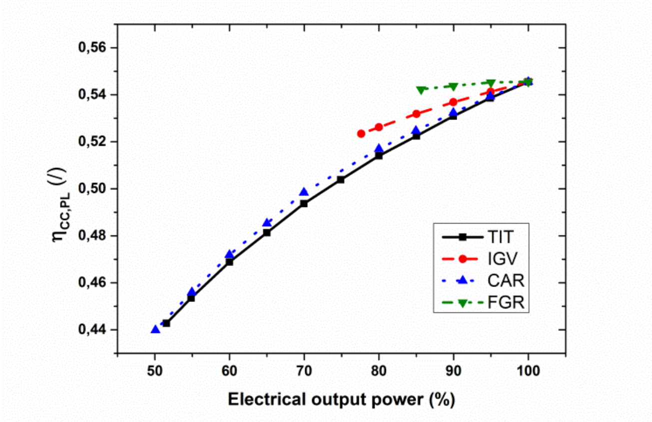

FIGURE 7 **Evolution of the CC efficiency for different control techniques with part load**

Optimal control cycle

Although FGR and IGV controls are the most efficient controls, they cannot be used on a deep part load basis as the GT and CC load can only be reduced to limited values due to surge limits and oxygen depletion as discussed before. The controls can be combined to go to deeper part load performance. Traditional operation of a GT works first on an IGV basis, and when surge limit is reached, switches to TIT control to the detriment of efficiency, as shown in Fig. 6.

This also comes with a penalty in emissions as the combustion mode switches from pre-mixed to non pre-mixed due to the reduction in combustion temperature. Using FGR enables another control combination for deep part load operation, as shown in Fig. 7. The load is first reduced with FGR, down to the O_2 concentration limit in the inlet gases. IGV cannot be used at this point as reducing the air flow rate would also reduce the O_2 concentration. The second step thus uses TIT control. This reduces the fuel flow rate and increases O_2 levels and is used down to a part load operation of 58%. Finally, IGV control is applied and the load decreases down to 50%. This enable an efficient deep part load operation of the GT in a CHP cycle. Figure 8 shows that an efficiency gain of 11% can be expected compared to traditional control techniques.

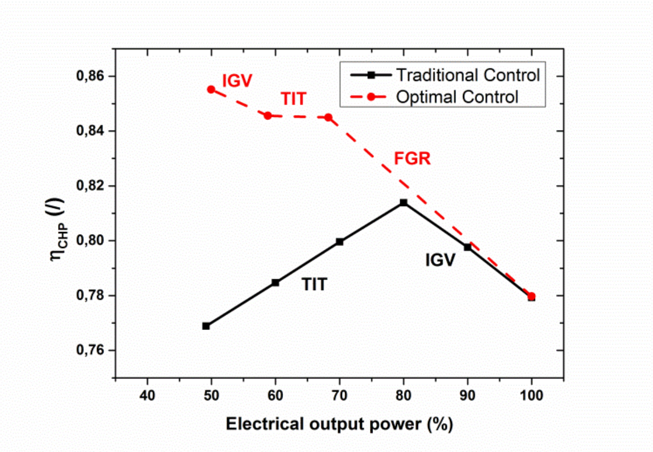

FIGURE 8 Traditional control for CHP GT part load operation compared to optimal control with FGR

For the combined cycle, the minimum O_2 molar concentration, the maximum boiler inlet temperature and the maximum guide vane closure limit the part load operation. In accordance with these limitations, the most efficient (part load) control combination was found to be a combination of FGR, IGV and TIT controls as shown in Fig. 9. Firstly, flue gases are recirculated up to the minimum achievable load of 86%. Secondly, IGVs are closed up to 83% of the site rating, reaching maximum allowed boiler inlet temperature (650 ºC). Thirdly, this temperature is maintained constant by decreasing the TIT and closing the IGV simultaneously until surge limit is reached. Finally, the load is decreased to 50% reducing the TIT. Figure 9 shows that compared to standard controls, an increase of around 2% of efficiency is achieved.

NOx EMISSIONS AT PART LOAD

In this paper, only estimates of the impact of FGR on NOx emissions are presented. Better understanding and prediction of NOx emissions from premixed and non-premixed combustion through modelling and experiments is part of new research at TU Delft.

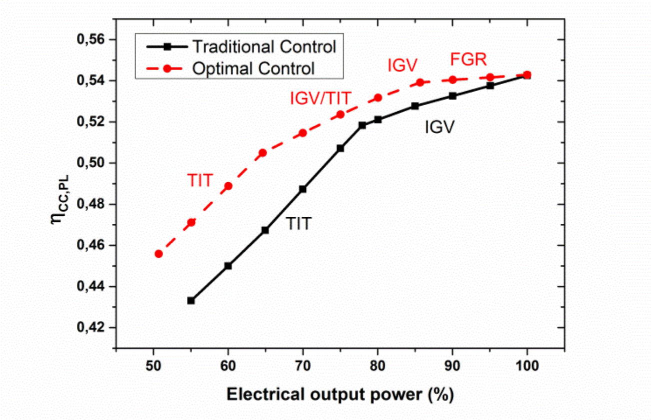

Figure 9 Traditional control for GT part load operation compared to optimal control with FGR

In FGR non-premixed combustion, the main influence on NOx emissions is due the reduction of the stoichiometric adiabatic flame temperature caused by oxidiser composition changes with the load. The stoichiometric adiabatic flame temperature corresponding to the oxidiser composition entering the combustor for the studied CHP cycle is shown in Fig. 10. On the basis of the ratios of Fig. 3 showing the reduction of emission with flame temperature, the NOx emissions with FGR are estimated compared to fresh air combustion at base load. This provides the trend shown on Fig. 11. At deep part load operations, NOx emissions are almost reduced by a factor 3. The reduction in NOx at deeper part load with FGR are also interesting to reduce water injection during non-premixed operation. Furthermore, NOx mitigation is likely improved by reaction pathway modifications that are not captured in this first estimation. The base load emissions is of 300mg/Nm^3.

For pre-mixed combustion, FGR emissions is estimated at a FGR ratio of 25% based on the experimental results of ElKady *et al.*(2009). They show that emissions are reduced, compared to standard combustion, of 40% at 25% FGR ratio (around 80% load). Figure 12 shows that FGR is being very advantageous at lower loads when traditional combustion switches to non-premixed operation (same absolute value as Fig. 11). Further improvements are expected by taking into account more detailed kinetics of the combustion.

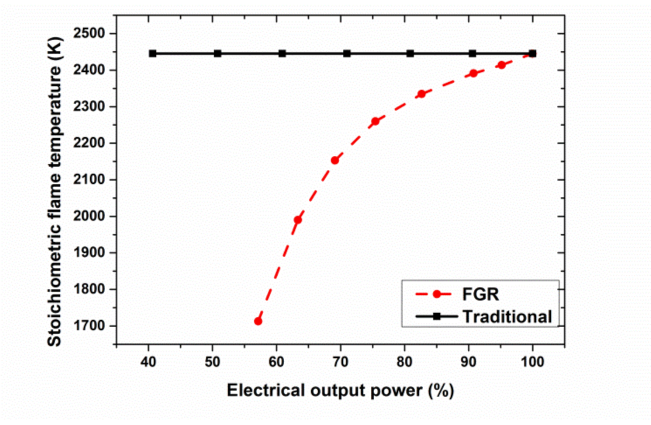

FIGURE 10 Influence of inlet air composition on the Stoichiometric flame temperature

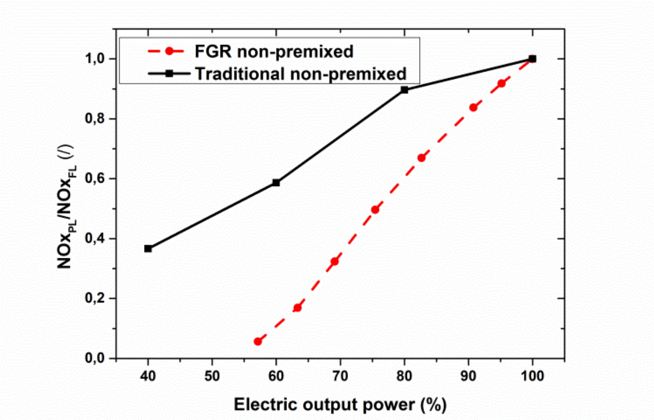

FIGURE 11 Relative NOx emissions for non-premixed combustion for traditional (Becker and Perkavec(1994)) and FGR controls estimated with flame temperature reduction

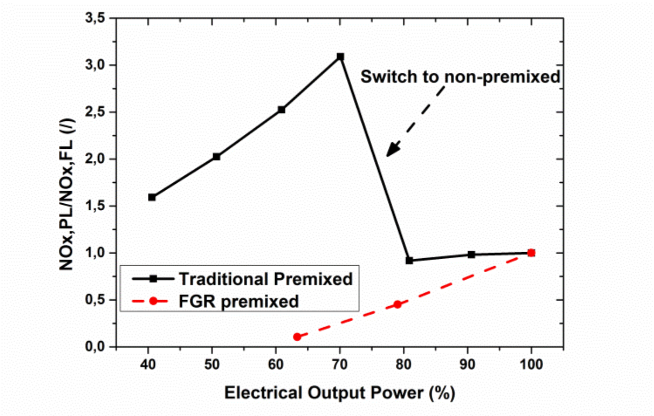

FIGURE 12 Relative NOx emissions for premixed combustion function of the GT load for traditional and FGR controls (estimated with ElKady *et al.*(2009), base load emissions of 50mg/Nm^3); the switch of traditional controls from non-premixed to non-premixed is shown at 80% load.

CASE STUDY

An economical study linking the CHP plant operation with the hourly spot market prices for natural gas, CO_2 and power in 2016 in the Netherlands shows another advantage of FGR. The CHP plant has been chosen as the load of the GT can be decoupled better from the steam production and with a higher efficiency. The model takes account of the GT operation at 100 and 50% load for FGR operation and between 100 and 80% for premixed standard operation (with the respective efficiency and NOx emissions regulation limits). Figure 8 provides the efficiency for the corresponding operation. By operating the GT at the optimum economic point of view in regards of max and min load, Fig. 13 shows that with FGR, the plant is more often operated at part load as it is more economically preferable. Compared to a DLN combustor, 4% yearly savings in steam generation costs can be expect with this usage.

Despite these advantages, the practical use of FGR needs to be studied further. Flame stabilization at low O2 levels could enhance the part load range of FGR. Turbine blade cooling will be affected as the composition and the temperature of cooling air will change. A trade-off between improvements in efficiency and emissions and the operation and maintenance of the gas turbine might have to be found. In addition to this, a blower needs to be installed to recirculate the gases to the GT inlet. Pressure drop computations show that it would reduce the efficiency at the lowest part load by maximum 2%.

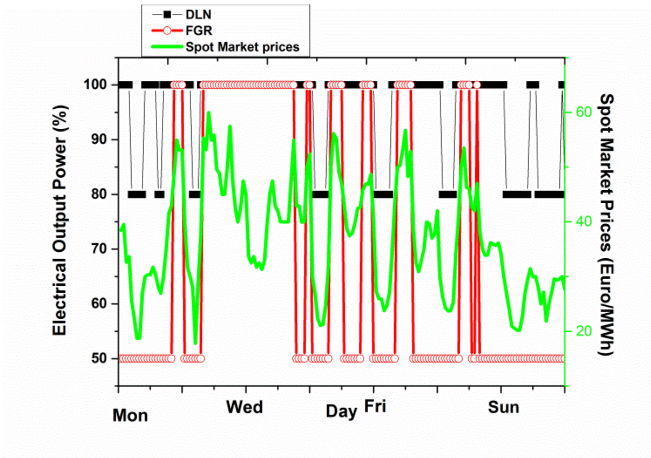

FIGURE 13 GT load during the course of a week for the CHP plant with FGR and traditional controls

CONCLUSIONS

The use of external Flue Gas Recirculation has been studied to enable deep part-load operation of Gas Turbine in Combined Cycles and Combined Heat and Power plants. It has been compared to other control techniques as Turbine Inlet Temperature control, Inlet Guide Vanes and Compressor Air Recirculation. The results of thermodynamic simulations show FGR increases the cycle efficiency for both CHP and CC, compared to traditional control techniques. The range of application of FGR in this study is limited to oxygen content of 14.5% defined as the minimum oxygen level for stable combustion. By combining FGR with traditional control techniques, it has been shown that the cycle efficiency at deep part load conditions can be increased of 11% for the CHP and 2% for the CC plant. This is primarily due to the reduction of stack losses. FGR also presents a potential to reduce NOx emissions of pre-mixed and non-premixed GT combustion. It allows a lower part load operation in pre-mixed mode due to an almost constant equivalence ratio. In non-premixed mode, FGR also shows interesting prospect to enable operation without steam injection at part-load, as NOx emissions can be reduced up to 3 times compared to traditional control techniques with non-premixed combustion. These first estimates show the potential for NOx mitigation with FGR. This will be further studied through chemical reactor network modelling and experimental efforts at TU Delft.

From an economical point of view, computations with 2016 natural gas and spot market data showed that a 4% financial saving can be expected for a CHP operated with FGR compared to a pre-mixed combustor.

However, limitations as CO and UHC emission, or aerodynamics and cooling effects on the gas turbine components might limit some of the benefits highlighted.

Flue Gas Recirculation is thus a very promising technique for part load operation of CHP cycles. Future studies will focus on enhancing FGR part load range by addressing potential combustion stability issues.

NOMENCLATURE

Symbols

P	Power
H	Enthalpy
Q	Heat
m	Mass flow rate
NOx	Nitrogen Oxides emissions

Subscripts

Fl,ad	Adiabatic Flame Temperature
Fir	Firing Temperature
Fl	At full load
Pl	At part load
stack	Related to the stack
amb	Related to ambient conditions
el	Electric
f	fuel
HRSG	Heat Recovery Steam generator

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