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Concept of hydrogen fired gas turbine cycle with exhaust gas recirculation: Assessment of process performance

Mario Ditaranto*, Tarjei Heggset, David Berstad

SINTEF Energy Research, Postboks 4761 Torgarden, 7465, Trondheim, Norway

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A B S T R A C T

High hydrogen content fuels can be used in gas turbine for power generation with CO_2 capture, IGCC plants or with hydrogen from renewables. The challenge for the engine is the high reactive combustion properties making dilution necessary to mitigate NOx emissions at the expense of a significant energy cost. In the concept analysed in this study, high Exhaust Gas Recirculation (EGR) rate is applied to the gas turbine to generate oxygen depleted air. As a result combustion temperature is inherently limited, keeping NOx emissions low without the need for dilution or unsafe premixing. The concept is analysed by process simulation based on a reference IGCC plant with CO_2 Capture. Results with dry and wet EGR options are presented as a function EGR rate. Efficiency performance is assessed against the reference power cycle with nitrogen dilution. All EGR options are shown to represent an efficiency improvement. Nitrogen dilution is found to have a 1.3% efficiency cost. Although all EGR options investigated offer an improvement, dry EGR is considered as the preferred option despite the need for higher EGR rate as compared with the wet EGR. The efficiency gain is calculated to be of 1% compared with the reference case.

1. Introduction

The most efficient way to produce power at large scale from gaseous fuel is by using gas turbine engines. High hydrogen content gas fuel can be found in three possible applications: (i) Integrated Gasification Combined Cycle (IGCC) power plants; (ii) power plants using the pre-combustion CO_2 capture in a Carbon Capture and Sequestration (CCS) context; (iii) power plants in a fully developed renewable energy based society, where hydrogen is used as energy storage in case of excess wind, solar, or other intermittent renewable power. Although CO_2 free, the combustion of hydrogen generates high levels of Nitrogen Oxides (NOx) which are strongly regulated because of they play a major role in the atmospheric pollution leading to smog and are responsible for acid rains. For example, NOx emissions limit in California for 10 MW and higher stationary gas fired gas turbines is 9 ppm @ 15% O_2 and 24 ppm in Europe. Many studies report of NOx emission doubling or more by switching a low NOx burner from methane to pure hydrogen [1]. Indeed, a known characteristics of hydrogen combustion is that it has a high flame temperature. One of the main chemical contribution to the formation of NOx is through a kinetic pathway where the nitrogen of the air is oxidized by oxygen at high temperature. This mechanism is strongly sensitive to temperature and called the Thermal NOx mechanism for this reason (but also known as the Zeldovitch' mechanism) [2]. A small increase in the higher range of temperature results in an exponential increase in NOx production. High NOx values in excess of 200 ppm @ 15% O_2 dry have been reported in Todd et al. [3] in an 85–90% hydrogen fuelled GE's 6 F A test combustor and even 800 ppm @ 15% O_2 dry in Brunetti et al. [4]. These values, compared to the typical emission limits applied to gas turbines, highlight how inadequate the current combustor technology is and therefore the need for innovative solutions.

In modern conventional fossil fuel based gas turbines, the high temperature regions in the flame is avoided by premixing the fuel and air prior to combustion to the point that the adiabatic flame temperature of the mixture is much below than that of the stoichiometric mixture. These burners are known as lean premixed burners or Dry Low NOx (DLN) burners. The technology has initially struggled because the required degree of air – fuel premixing leads to issues related to combustion stability: flashback, extinction, and thermo-acoustic instabilities [5]. The technology is however now commercial and the major gas turbine manufacturers offer engines

* Corresponding author.
E-mail address: mario.ditaranto@sintef.no (M. Ditaranto).

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List of abbreviations

AGR	Acid Gas Removal
ASU	Air Separation Unit
CCS	Carbon Capture and Sequestration
DLN	Dry Low NOx
EBTF	European Benchmark Task Force
EGR	Exhaust Gas Recirculation
GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
IGCC	Integrated Gasification Combined Cycle
LHV	Lower Heating Value
NGCC	Natural Gas Combined Cycles
NOx	Nitrogen Oxides
PTE	Polytropic Efficiency
S(N)CR	Selective (Non-)Catalytic Reduction
ST	Steam Turbine
TIT	Turbine Intlet Temperature
TOT	Turbine Outlet Temperature
TRL:	Technology Readiness Level

that achieve NOx emissions levels within the regulated values without the need of end of pipe abatement installations such as SCR or SNCR. Nevertheless, the application of this technology to high hydrogen content fuels still strives because of the specific characteristics of hydrogen combustion: wide flammability limits, much higher reaction rates, preferential diffusion and higher flame temperatures leading to short auto-ignition times and high flame speed [6]. As a consequence of these particular properties, combustion occurs promptly before air and hydrogen have had the time to be fully premixed. This problem is referred to as flashback, i.e. unwanted propagation of flame in the premixer region not designed for the presence of flame with the risk of component damage.

In existing IGCC plants and those with pre-combustion CO_2 capture where hydrogen is the major fuel component, the NOx formation problem is tackled by using simple diffusive type burners, but by adding large amounts of diluent gas. Nitrogen and steam are both potential diluent candidates because they are available at relatively low cost on site of IGCC plants. Wu et al. [7] experimentally showed that steam is more effective than nitrogen to reduce NOx formation because of the higher heat capacity of steam, hence the larger reducing effect it has on adiabatic flame temperature. For example, steam to fuel ratio of unity was shown to half the NOx emissions from 800 ppm @ 15% O_2 dry in Brunetti et al. [4]. Nevertheless, nitrogen is practically preferred firstly because steam affects significantly the heat transfer properties of the hot exhaust gas flow and reduce components life [6]. Secondly, nitrogen is a readily available by-product of the Air Separation Unit (ASU) present on site of an IGCC plant for producing oxygen for the gasifier.

The use of diluents in industrial cases with syngas as fuel on diffusion type combustors have shown good emissions results in the above references. However, although available at low costs, using nitrogen as diluent induces an expense of up to 20%–30% of the total auxiliary power consumption due to the required compression work for injection at the combustor stage. For comparison, this share is even higher than that of the CO_2 compression power in the case of pre-combustion plant [8]. From a cost perspective the, compressor unit is expensive and bulky. Gazzani et al. [9] showed that dilution used in combination with diffusion type combustors imposes an efficiency penalty of 1.5 %-points as compared to the reference combined cycle plant if the amount of nitrogen dilution is that required to reach a flame temperature similar to that of a natural gas flame. The penalty becomes 3.5 %-points in the case of steam dilution. The selected dilution degree and corresponding efficiency decrease is to be compromised with NOx emissions since these are exponentially proportional to combustion temperature.

The implementation of DLN combustors would avoid the inert dilution to reduce NOx emissions. However, to counteract the aforementioned excessive flashback propensity, high injection velocity and therefore high pressure drop would be needed, which in turns has an efficiency cost as shown in Gazzani et al. [9]. Consequently, DLN burners have not been achieved to date for fuels with hydrogen content larger than approximately 60% without some kind of dilution. In addition, even if lean premixed combustion of hydrogen were achieved through DLN burners, Therkelsen et al. [10] evidenced experimentally that at the same flame temperature, measured NOx emissions were still higher in a hydrogen than in a methane flame. They attributed this effect to the higher propensity of the hydrogen – air chemical kinetic to produce NO through the low temperature NNH pathway [11]. There is therefore a real need for innovative concepts in combustion technology to cope with hydrogen fuels.

The recent review from du Toit et al. [12] on the use of hydrogen in gas turbine describes several burner technologies available and still points out the remaining R&D challenges of tackling the high temperature and NOx emission. In search for alternative ways to burn pure hydrogen, Ditaranto et al. [13] suggested to tackle this challenge not by yet another burner technology, but by setting up a power process that inherently avoids high temperature. The gas turbine cycle concept proposed includes exhaust gas recirculation (EGR) that has potential for low NOx emissions without the need for either fuel dilution or burner technology breakthrough. Further, the authors showed [14] through a first order combustion analysis that the oxygen depleted air entering the combustor - due to EGR - naturally limits the combustion temperature and NOx formation. With this concept, the burner and combustor can be of diffusion type, i.e. simple and reliable, and would avoid the high cost and risks associated with the development of complex DLN systems for high hydrogen content fuels.

The concept of EGR is a common and mature technology in internal combustion engines, mostly diesel, with the aim of reducing NOx formation [15]. For gas turbine applications however, it is only known in two cases related to the CO_2 capture context. One as a mean for increasing the CO_2 concentration in natural gas combined cycles (NGCC) exhaust gas, with the aim of making post-combustion CO_2 capture more efficient [16]. The other is in the oxy-fuel CO_2 capture scheme where CO_2 replaces air as the gas turbine working fluid and the cycle is therefore semi-closed [17]. For power cycles based on hydrogen fuels however it has, to the knowledge of the authors, not been evaluated in the scientific literature, apart from their above-mentioned preliminary studies. In Ditaranto et al. [14] the combustion assessment showed that the flame stability could be achieved at high EGR rates, high enough to maintain low NOx emissions even without dilution of hydrogen. It brought the idea through TRL 1 (based on the EU H2020 TRL scale) and the present study aims at validating TRL 2 by an evaluation of the concept from a process and thermodynamic cycle perspective in order to assess whether the concept is worth further development towards TRL 3.

2. Description of the hydrogen fired gas turbine with EGR concept

The power cycle under investigation is an IGCC plant with CO_2 capture with coal as primary fuel. The basic layout of the IGCC power plant is that of the European Benchmark Task Force (EBTF) [8] which has been designed with the objective to serve as a reference. The main components of the power cycle are the pressurized gasifier producing syngas which undergoes first a two stage shift reaction, followed by Acid Gas Removal (AGR) and H_2S Removal units, and then a CO_2 separation step. CO_2 is then compressed and delivered at the battery limit ready for transport and geological storage, while the hydrogen rich syngas is burned in the power island power island composed of a gas turbine and bottoming steam cycle. As described in the Introduction, the gas turbine when fuelled with syngas or hydrogen fuels generally operates with a stream of dilution nitrogen coming from the ASU in order to control temperature in the combustor and avoid excessive NOx emissions. The goal of the present study is to demonstrate that applying EGR to the gas turbine can replace the dilution stream and make the economy of its compression power.

The layout of the proposed concept in its simplest form, i.e. with dry EGR, is depicted in Fig. 1 in two possible application cases: 1) as a natural gas or coal fired IGCC power plant with CO_2 capture, 2) as integrated in a renewable energy system with hydrogen as energy storage. In addition to the dry EGR case, the study considers two other variations of the cycle with wet EGR and wet EGR with cooling. In wet EGR, the recirculated exhaust gas goes directly into the compressor without any form of condensation or cooling after exit from the HRSG unit, while in the wet EGR with cooling, the recirculated exhaust gas is cooled down to its dew point temperature to optimize compression efficiency.

In the EBTF, the overall system was optimized cycle by heat integration of various components such as the air separation unit (ASU) delivering oxygen to the gasification process generating the syngas, the gasifier, and the shift reactor. Applying EGR and suppressing the nitrogen dilution call for some adjustments in this reference system as will be described in the following section. In the work presented, the process components external to the power island (i.e. the ASU, gasifier, shift reactors, gas treatments and CO_2 separation units) are not modelled in this simulation. The streams of mass flows and energy to these components are considered as inputs and outputs unchanged from the reference cycle, only proportionally adjusted to the syngas flow rate. A modification to the ASU needed however be implemented due to the modified gas turbine operating conditions, as will be explained in detail in the following section.

3. Methodology

The power plant setup is shown in Fig. 1 with the high hydrogen content syngas fired gas turbine (GT) and the bottoming cycle with the HRSG and steam turbine (ST) divided into four turbine stages. All water and steam streams scale with the same factor when the lowest of these streams are changed to match the outlet temperature of the exhaust. The setup and all its variations have been modelled in Aspen HYSYS code. The power island of the IGCC plant with pre-combustion CO_2 capture of the EBTF [8] has also been modelled, the results of which is used as the reference case for this study. It is noted that the composition of some streams differs slightly from the ones given in EBTF [8] since more accurate values were obtained in the working document of the European project (DECARBit, FP7-ENERGY.2007.5.1.1 #211971), which generated the EBTF reference.

3.1. Integration of the ASU with the gas turbine

In the reference power plant, integration of the ASU with the gas turbine is done in order to improve the efficiency of the overall cycle, such as 50% of the air used in the ASU comes from the GT compressor while the remaining air is drawn into a separate smaller compressor that is a part of the ASU. Given that the air compressor is the largest energy user in the ASU [18], shifting the compression energy cost of 50% of the air to the gas turbine with a larger and more efficient compressor, means that maximum 3 MW is spared due to the integration, corresponding to an efficiency improvement of approximately 0.3 %-point on the power island efficiency. Because the present concept necessarily implies that the O_2 concentration in the air flowing the GT compressor is decreased, the amount of oxygen in the air going to the ASU would therefore also be decreased and therefore it cannot be expected that the same gain would be obtained by such an integration as the optimal amount of integrated air would probably be at a different amount than in the reference cycle. In order not to complicate unnecessarily this assessment study, no air is extracted from the gas turbine compressor for the ASU and therefore no potential benefit from any integration is considered. Coincidentally, this amount of air kept in the compressor compensates for the nitrogen flow used in the reference case to dilute the syngas, but avoided in our concept. The overall mass flow through the turbine is thus closer to that of the reference case and permits a fair comparison from a hardware point of view.

Omitting the integration results in an increased power consumption in the ASU. Since the integration concerned 50% of the air, the extra energy consumption at the ASU cannot exceed the reference integrated ASU power consumption (12.13 MW) and is probably lower. In fact, assuming a conservative specific energy

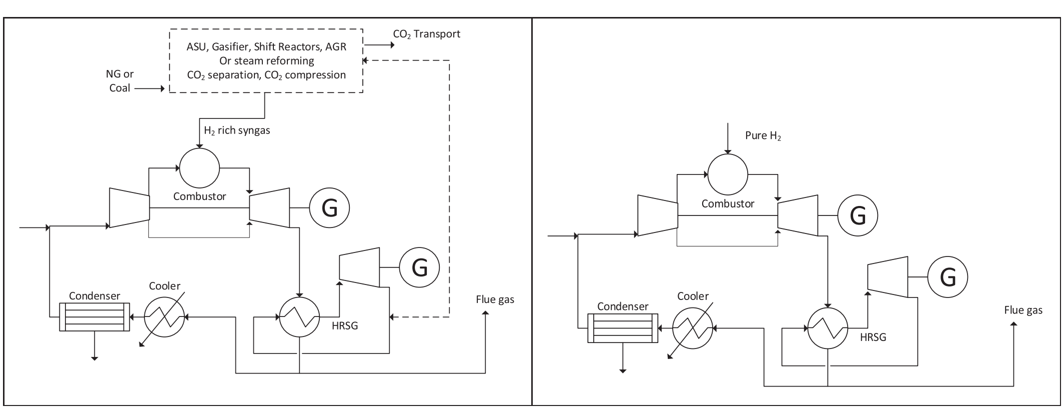

Fig. 1. Power island layout of the power plant with dry Exhaust Gas Recirculation. LHS: IGCC with CO_2 capture case (dashed line symbolizes energy flows between the bottoming cycle and the rest of the process not simulated in this study); RHS: Case with integration in renewable energy system with hydrogen as energy storage.

consumption of 200 kWh/t of oxygen, the production of oxygen needed for the gasifier considered in this study would be of 22.05 MW. Nevertheless, the power consumption for the ASU in the simulation of the EGR cases has been doubled from that of the reference case, thus the results can be considered as conservative or worst-case scenario. Since the syngas production rate varies from case to case as explained latter, the ASU power consumption was further assumed to be proportional to that rate.

3.2. Gas turbine

The typology of gas turbine considered in EBTF [8] is a large-scale "F class" 50 Hz. It is a reference F-class large-scale gas turbine averaged from the largest manufacturers at the time of publication of the EBTF (2008 state-of-the-art). The pressure drop inferred by the air inlet filter was imposed with a valve. The total mass flow of air into to the GT process is 642.1 kg/s. A part of the air (84.6 kg/s) bypasses the combustion chamber and is used as cooling air in the turbine. The pressure ratio across the compressor is 18.11 bar. By setting the temperature outlet to be 409 °C in the reference setup [8] the polytropic efficiency (PTE) could be backwards determined to be 93%. In the simulations this PTE has been applied to the compressor which gives an outlet temperature of 408 °C, considered close enough given the precision of PTE given (no decimal).

The combustion process was modelled as a Gibbs reactor which minimizes the Gibbs energy of the mixture of its inlet streams to produce the outlet stream. A pressure drop of 1.08 bar was set across the reactor, as used in the reference system. In the reference case, a stream of N_2 is injected in the combustor for syngas dilution. The EBTF syngas mass flow is 22.45 kg/s and the N_2 mass flow is 80 kg/s, which when mixed with the air in the Gibbs reactor gives a temperature of 1306 °C after combustion. The syngas and N_2 mass flows were proportionally lowered to 22.34 kg/s and 79.58 kg/s respectively, in order to reach a temperature of 1302 °C after combustion as found in the reference document. After mixing with the cooling air, which is injected as coolant of the first turbine stage, a temperature of 1205 °C is reached. These two temperatures were fixed and used to determine the flow rates of syngas and cooling air in the parametric study. The dilution nitrogen was compressed from 1 bar and −134 °C to 25 bar and 200 °C. In our reference setup this compression power consumption corresponds to 26.28 MW, although in EBTF [8] the pressure after compression is 36 bar with a corresponding power consumption of 27.82 MW. Such a high pressure was deemed unnecessary and the study kept a nitrogen compression power consumption of 26.28 MW as a reference. This choice is further supported by the study of Gazzani et al. [9] who used a dilution nitrogen pressure of 27.1 bar at the mixer inlet. Therefore, the power island efficiency found in our simulated reference is expected to be better than in EBTF.

The gas at the combustor exit was mixed with the cooling air before entering the turbine. By using an expansion of 15.98 bar, as found in the reference system, and setting the turbine outlet temperature (TOT) to 571 °C, a PTE of 84% could be determined which was then specified in our set up to release the condition on TOT. The TOT was calculated to be 569.9 °C, which as for the compressor, was considered to be close enough. The chemical composition and thermodynamic values of the exhaust gas can be found in Table 1.

3.3. Bottoming cycle

The temperatures and pressures of the streams in the steam cycle were specified as indicated in the reference system except for some cases where they had to be changed slightly to have vapour fractions. The water and steam mass flows can be scaled, all with the same factor. This effectively gives the steam cycle one degree of freedom that can be specified to make the exhaust gas reach exactly 100 °C at the cycle outlet by extracting the right amount of heat from it in such a way that the steam cycle operates the same way as described in the reference system. The exhaust gas leaves the bottoming cycle at 1.02 bar as a result of each heat exchanger pressure drop being specified as found in the source documents [8]. To make it reach 100 °C the water inlet flow had to be increased from 129.5 kg/s in the original system to 131.72 kg/s.

The PTE of the steam turbine stages were determined by the specified temperatures and pressures at the inlets and outlets, as found in the reference system. The high pressure stage takes in the high pressure superheated steam and sends a part of its outlet to the gasification process. The rest is combined with the intermediate pressure superheated steam and reheated before entering the intermediate pressure ST stage. A part of this outlet is also sent to the gasification process, while the rest is sent to the next ST stage. This outlet is then combined with the superheated low pressure steam and sent to the last ST stage where it expands below its dew point, leaving with a vapour fraction of 0.899. The steam is then condensed completely before it is pumped into a tank where it is combined with feedwater from other processes (gasification or shift reactors) to return into the steam cycle inlet. The pumps in the steam cycle are implemented as described in the reference material, so this energy consumption is accounted for in the total power calculations.

The exhaust gas from the gas turbine is led through a series of heat exchangers transferring heat to a steam cycle with a four-stage steam turbine. The steam cycle in the reference system is integrated with the gasification and shift reactors processes in that they exchange water and steam from and to the bottoming part of the power island at certain temperatures and pressures. In the reference plant, this carries 49.27 MW of net heat into the steam cycle. It is not desired to simulate the entire power plant including the components external to the power island (i.e. gasifier, ASU, shift reactors, etc.) as that would require to run an entire optimization process on all these components combined together with the power island, with the consequence of losing track of the benefits or losses of the studied concept. Nonetheless, it is necessary to keep the power island boundary conditions equivalent to allow for a common comparison. Decoupling completely the power island from the external processes, would also require a complete redesign of the bottoming cycle and again the basis for comparison would be altered. For example, most of the high pressure water is evaporated in the gasification process and the corresponding latent heat comes so to say "free" for the bottoming cycle. If this heat was to be extracted from the exhaust gas as a result of decoupling of the gasification process, the temperature drop in the exhaust gas would increase significantly and the temperature would become too low to provide sufficient heat to the water in the following heat exchanger stages, and so forth.

In order to keep track of the energy flows between the bottoming cycle and the external components, every stream in and out of the steam cycle were brought to a reference point of 15 °C and 1.01 bar. Water was therefore heated or cooled to the appropriate temperatures and pressures given in the inlets and outlets of the EBTF simulations. These heating and cooling were done with the HYSYS components heater and cooler respectively. Since this was done on all streams entering the steam cycle, the difference in energy flows measured between the heaters and the coolers gives the total net energy flow into the steam cycle from the virtual external processes (symbolized by the dashed arrow in Fig. 1). Because of the variation in total mass flow of water and steam through the steam cycle for the different EGR cases, this net energy

Table 1
Compositions and properties of streams in the reference case.

Property	N_2	O_2	H_2O	CO_2	CO	Ar	H_2	CH_4	T	p	M
Unit	vol. %								°C	bar	kg/s
Ambiant air	77.29	20.74	1.01	0.03	0	0.93	0	0	15	1.01	642.1
Syngas	8.74	0	0.05	03.07	2.55	1.13	84.43	0.03	200	25.45	22.5
GT exhaust	75.17	10.88	12.25	0.79	0	0.91	0	0	569.9	1.05	681.9*

* The residual mass flow (17.3 kg/s) is the net flow of air bled to the ASU and N_2 injection in the syngas.

flow varies. However, this is not necessarily proportional to the syngas mass flow so the net energy extracted from the surroundings in the HYSYS simulation is not necessarily the same as the heat delivered from a calculation using proportionality between syngas mass flow and net heat supply from the gasification and water shift processes. This proportionality constant was calculated in the reference case by measuring net flow of energy divided by the syngas mass flow rate. The difference between the measured net energy flow and the calculated energy flow available from the production of syngas was then multiplied by the weighted average of the efficiencies of the steam turbine stages and the efficiency of the electric generator to find the amount of energy that had been supplied to the total output of the steam turbine, but could not have been provided by the real syngas production. This difference represents an additional virtual energy flow that must be subtracted from the total energy generated by the steam cycle to calculate the corrected efficiency. It is recognized that this correction is a limitation in the comparison work, however this power correction does not represent more than 0.2%, 2% and 1.4% of the gross power output for the cases dry EGR, wet EGR without cooling, and wet EGR with cooling respectively. It will be further seen that the gain observed in terms of efficient are larger that this imprecision.

3.4. Power island efficiency definition

The efficiency of the power island is in this study the main comparison parameter. The battery limit of the net efficiency is as defined in Fig. 1, implying that the syngas production process and CO_2 separation are not included in the power budget, except for the ASU consumption power for the reason described in §3.1. The implementation of EGR in the gas turbine reduces greatly the potential for energy integration of the ASU with the gas turbine compressor, therefore this power loss is included in the efficiency definition in order not overestimate the benefits of the EGR concept. All auxiliary power necessitated by the treatment of the exhaust gas before recirculation must also be included. The net power island cycle efficiency is therefore defined as follows:

$$\eta = \frac{(W_T + W_C)\eta_m\eta_g + W_{ST}\eta_m\eta_g + W_P + W_{N2} + Q_v\eta_m\eta_g + W_{ASU} + W_{EGRcool} + Q_{EGRwe}}{\dot{m}_f\, LHV + Q_{dr}} \quad (1)$$

where:

η : Net efficiency of the Power Island
η_m: Mechanical efficiency
η_g: Generator efficiency
m_f: Fuel flow rate, kg/s
LHV: Lower heating value of coal, kJ/kg
W_T: Turbine work, calculated as fluid enthalpy change, kW (>0)
W_C: Compressor work, calculated as fluid enthalpy change kW (<0)
W_{ST}: Steam turbine work, calculated as fluid enthalpy change, kW (>0)
W_P: Total pump work, feed water pumps, cooling water pumps, etc., kW (<0)
W_{ASU}: ASU power consumption, kW (<0)
W_{N2} Dilution nitrogen compression power - only reference case, kW (<0)
W_{EGRaux} EGR stream cooling power, kW (<0)
Q_{EGRwe} EGR stream water extraction heat, kW (<0)
Qv Virtual heat correction - cf. §3.3 for details, kW (<0)
Q_d Coal drying heat (>0), kW

3.5. Combustion calculations

To calculate the combustion properties of the mixtures in all the EGR and the reference cases, a one-dimensional adiabatic freely-propagating, premixed flat flame reactor case was setup and solved in the kinetic calculation code Cantera [19] under Python. The GRI 3.0 chemical mechanism [20] was used. The fuel and compressor gas composition and temperature calculated by the process simulations at the inlet of the combustor were used as inputs. The adiabatic temperature, laminar flame speed, and NOx concentrations in the burned gases region were calculated under stoichiometric conditions, which is representative of the flame front condition in a simple diffusion flame and considered to be the least favourable conditions from a NOx formation perspective.

3.6. Simulated power plant cases

The reference case was reproduced and simulated, and then adapted to include the different cases of EGR. Two types of EGR are considered: dry EGR and wet EGR, with the EGR rate being defined as the ratio of the volume of the recycled exhaust to that of the exhaust, as in Ref. [13]. Nitrogen dilution of the fuel is only present in the reference case and not in the EGR cases even at low EGR rates. It is therefore obvious that the efficiency of the EGR cycles with 0% EGR has a higher efficiency than the reference case, corresponding to the saving in dilution compression work. The EGR rate was varied between 0 and 65% with increments of 5 %-points.

As described in the previous section, the air stream from the compressor directed to the ASU has been removed as well as the N_2 dilution stream, however the latter (80 kg/s) is not fully compensated by the former (62.5 kg/s) and there is a mass flow reduction of

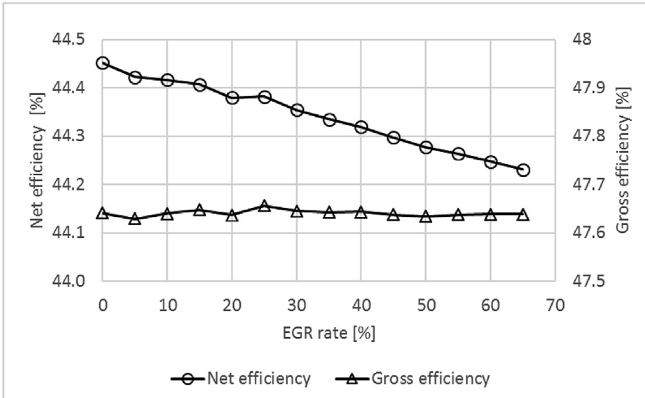

Fig. 2. Net and gross electric efficiency in the dry EGR case.

about 17.5 kg/s through the combustor. Therefore, the mass flow of syngas needs be reduced to achieve the correct TIT compared to the reference case. Due to the changing heat capacity with the varying EGR rate, the syngas mass flow and sometimes the cooling air mass flow, had to be adjusted for each EGR value to achieve a constant TIT value.

The power island efficiency is calculated by dividing the net output power by the coal fuel power input based on lower heating value (LHV). Since the syngas mass flowrate varies with EGR rate, the mass flow of coal has to be accommodated. The gasifier is not simulated in this study which instead focuses mainly on the power island, and it has therefore been assumed and used a mass-based syngas-to-coal ratio as that used in the reference case, equal to 1.688 The heat for coal drying is assumed to be 0.85% of the coal LHV power used, again equal to that used in the reference cycle. In all plant arrangements studied, the ASU power consumption is assumed to be twice as large as that given in Ref. [8] and serves as a “worst case scenario” as argued earlier.

4. Results

4.1. Reference case

The efficiency of the power island calculated with Eq. (1) for the reference case simulated in the present setup gives a value of 43.2% which compares with 41.7% when using data reported in Ref. [8]. The discrepancy between the two values is due to the necessary adjustments made to the cycle setup as explained above, in particular the use of a lower N_2 compression work as explained in §3.2. The reference for comparison in the remainder of the study always refers to the cycle calculated with our plant setup which has the exact same basis.

4.2. Dry EGR case

In the dry EGR case, water was extracted from the exhaust stream before entering the compressor such that the mole fraction of vapour is saturated and the same as in ambient air: 1.01%, even though this may not be the optimal trade-off between acceptable moisture content for the turbomachinery and expense of auxiliary power for cooling, as will be discussed later. The moisture level requires the exhaust to be cooled down to 7.55 °C. The cooling from 100 °C to the ambient temperature of 15 °C is in principle free energy-wise. Further cooling down to 7.55 °C requires some form of heat pump which implicated a certain energy cost. A conservative heat pump efficiency of 4 was chosen to provide a measure of power consumption related to the cooling. The two stages of cooling were therefore separated in two cooler components of different kind. In addition, a water pump was inserted in the model to get rid of the residual water by increasing its pressure by 1 bar.

The net and gross efficiencies for the dry EGR case are shown in Fig. 2. The gross efficiency is based on the power generated by the turbines, while the net efficiency takes into account the auxiliaries power consumption. At 0% EGR the net efficiency is 44.5%, that is 1.3 %-points higher than in the reference case. The efficiency penalty stemming from nitrogen dilution for NOx emission control in an IGCC plant (with or without CO_2 capture) is therefore of around 1.3 %-point. This value compares well with the study of Gazzani et al. [9] who showed that 1.5 %-point of energy efficiency was lost due to nitrogen dilution. For the sake of comparison, the ENEL’s Fusina Hydrogen demonstration project [4] operating a smaller 16 MWe gas turbine fuelled with pure hydrogen achieved 41.6% in combined cycle mode, but did not have to concede the power expense of an ASU for gasification purposes because hydrogen was supplied as a by-product from elsewhere on the industrial site at no energetical cost.

The gross efficiency is rather constant over the range of EGR rate while the net efficiency drops 0.2 %-point from 0 to 65% EGR (cf.

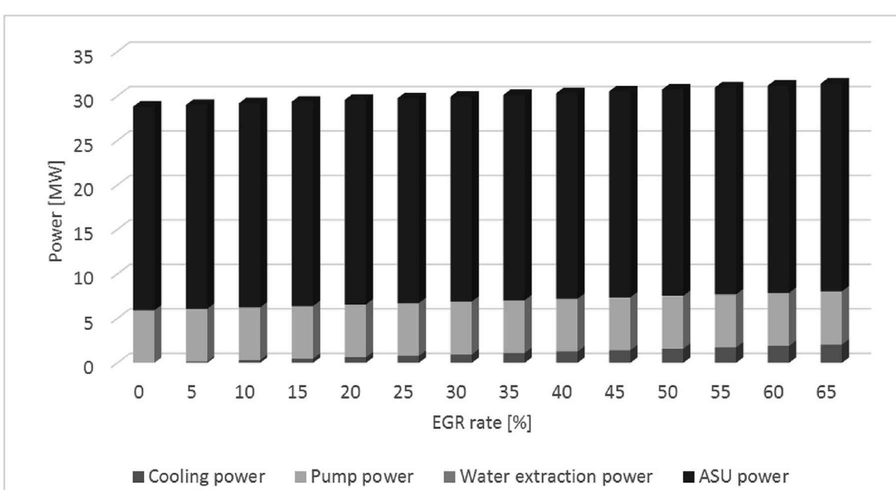

Fig. 3. Auxiliary power consumption posts in the dry EGR case.

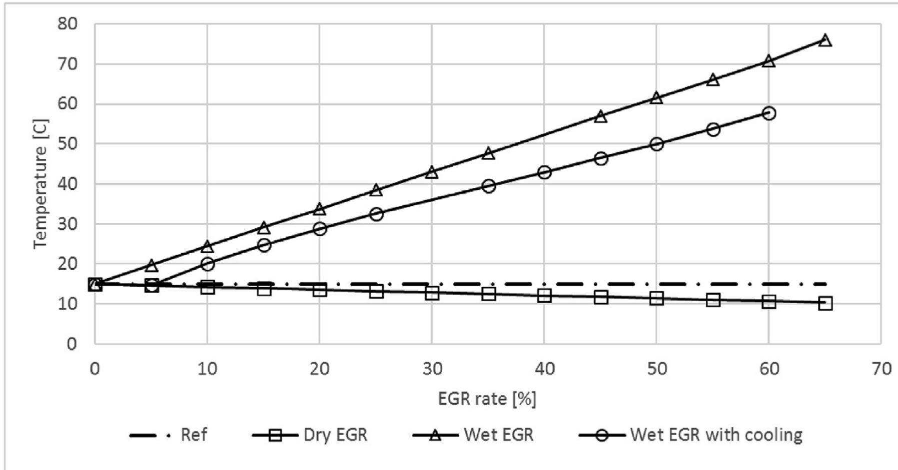

Fig. 4. Gas turbine compressor inlet temperature.

Fig. 2). The decrease in efficiency with increasing EGR is therefore due to the energy penalty related to the various causes of auxiliary power consumption of the power island. Fig. 3 shows clearly that the exhaust cooling is a parasitic power consumer. However, the major contributor to efficiency decrease remains the ASU independently of EGR rate, which accounts for more than 75% of the auxiliary power consumption. The cooling power based on a Coefficient of Performance (COP) ratio of 4 increases also with EGR, but has little impact on the overall efficiency.

4.3. Wet EGR cases

The results of two power plant cases with wet EGR are presented. In one case, the air and exhaust gas mixture is cooled down to the dewpoint to allow for the lowest possible compressor inlet temperature. In the other scenario there is no cooling provided and the compressor inlet temperature is thus allowed to have a higher value as shown in Fig. 4.

Fig. 5 shows the net and gross electric efficiencies for the wet EGR case with cooling. As for the dry EGR case, the net efficiency of the power island declines with EGR rate, but so does the gross efficiency. The presence of moisture in the cycle is therefore affecting the efficiency of the power island, especially that of the gas turbine cycle as shown in Fig. 6. The decrease of power generated in the gas turbine is nonetheless compensated by an increase of power generated in the steam cycle, but only partially.

It is noted that for wet EGR with cooling, it is not possible to achieve 65% EGR since at recirculation rates above 60%, there is not enough oxygen in the working fluid to burn the fuel, which is not a reasonable way of operating a power plant. The reason for this happening only in the case with cooling is that when cooled, the working fluid needs more fuel to reach the specified TIT. In the case of 0 and 5% EGR with cooling the air and exhaust gas mixture had to be cooled below 15 °C, but this energy cost has not been accounted for as it is negligible.

Figs. 7 and 8 show the results for the case of wet EGR without cooling. Results show generally the same behaviour as previously described, but with a greater expense in terms of lost power in the gas turbine when the EGR rate increases. The steam cycle on the other hand remains rather unaffected by the cooling step.

5. Discussion

5.1. Practical implications

The simple gas turbine efficiencies calculated on the basis of the syngas LHV are shown in Fig. 9 showing that all the EGR scenarios fall between the reference cases' efficiencies taking into account (lower bound) or not (upper bound) the nitrogen compression work for syngas dilution. The dry EGR case shows here also the highest efficiency of all the EGR cases, with values very close to that

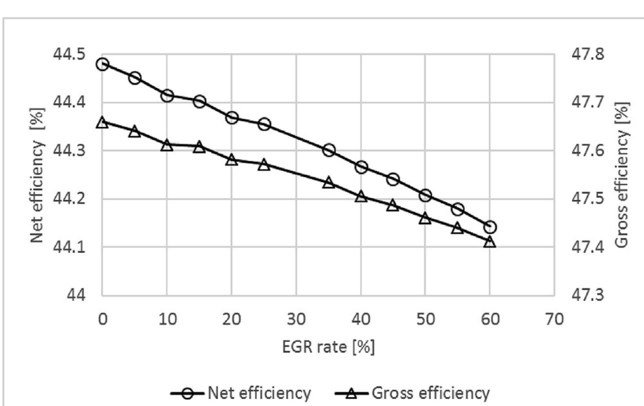

Fig. 5. Net and gross electric efficiency in the wet EGR case with cooling.

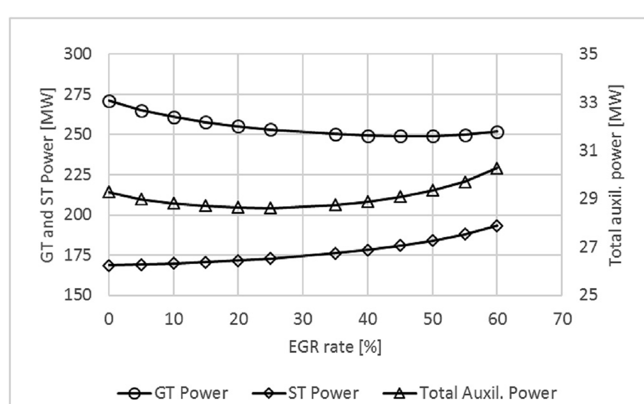

Fig. 6. Power generated in the gas and steam turbines, and consumed in the auxiliaries corresponding to Fig. 5.

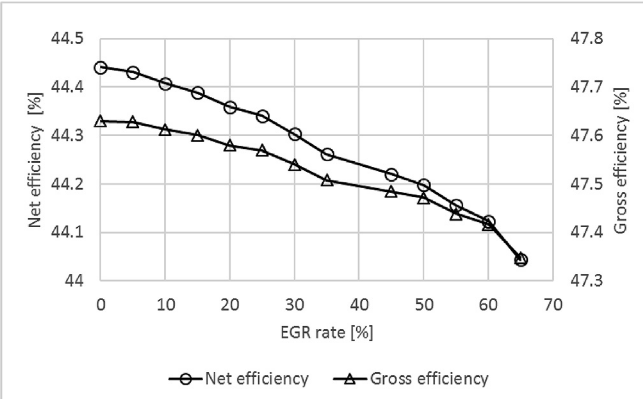

Fig. 7. Net and gross electric efficiency in the wet EGR case without cooling.

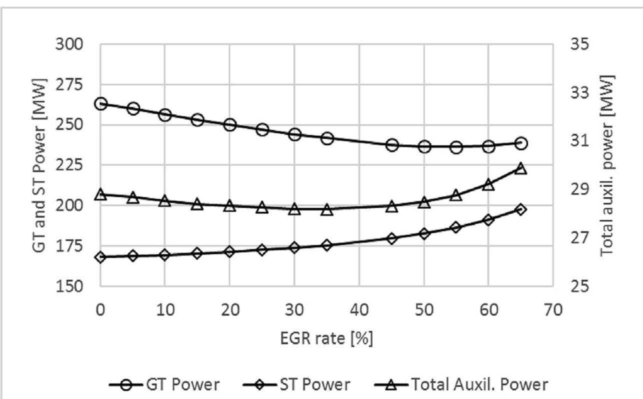

Fig. 8. Power generated in the gas and steam turbines, and consumed in the auxiliaries corresponding to Fig. 7.

of the reference gas turbine without compression work. As a comparison the ENEL's Fusina Hydrogen project had a measured gas turbine net efficiency of 31.1% at 11.4 MWe delivered power [4].

Fig. 10 shows the comparison between the different cases studied. All EGR cases exhibit a better overall efficiency than the reference case showing that avoiding the nitrogen dilution is beneficial. However not all EGR cases can satisfactorily replace the reference case since at low EGR rates, the adiabatic flame temperature in the combustor is high and so are the NOx emissions. The comparison should therefore be made with the EGR rates providing an adiabatic flame temperature outside the potentially high NOx zone. In addition, these conditions must be compatible with flame stabilization properties equivalent to or better than that of the reference case. If such region exists, Fig. 10 shows that the power plant will always have a better overall efficiency, as even at 65% EGR there is nearly 1 %-point difference. A first order combustion assessment of combustor technology applicable to this concept was made in Ditaranto et al. [13] based on two characteristics: 1) calculated flame speed of the different fuel and oxidizer mixtures, which is the conventional property to assess flame stability, hence burner viability; 2) NOx formation. The results showed that premixed technology in dry EGR mode would be possible within a given working fluid distribution with EGR rate of approximately 50% depending on PZ distribution. For the wet case, the domain was found to be larger, but limited to higher EGR rates if low NOx emissions are to be achieved.

Combustion calculations have been updated and presented in Figs. 11 and 12 that would correspond to a diffusion combustor technology (i.e. safer and cheaper). The NOx emissions from a real burner - combustor system is design and hardware dependent and cannot be predicted in general terms from the sole knowledge of fuel and air input compositions and temperature. Kinetic calculations in laminar conditions as those presented herein can only provide the potential for achieving low NOx emissions, which in the case of hydrogen combustion is dominated by the thermal mechanism, hence temperature. Gazzani et al. [9] and Chiesa et al. [6] suggest that if the adiabatic stoichiometric flame temperature (SFT) of a mixture does not exceed 2300 K, state-of-the-art diffusive combustor technology is able to produce low NOx burners, based on ENEL's practical experience in using hydrogen containing fuels in GE gas turbines [21]. In their reference case with dilution they assume that a diffusive combustor operating with a SFT of 2200 K could achieve NOx emissions below 20 ppmvd. The reference case with nitrogen dilution which is used in this study is in agreement with that limit as the corresponding calculated SFT is 2190 K. It is stresses again that this temperature is form a kinetic point of view very and would form very high NO concentration, but it is an indicative flame temperature at which state-of-the-art burner design would manage to "beat" equilibrium and generate low NOx.

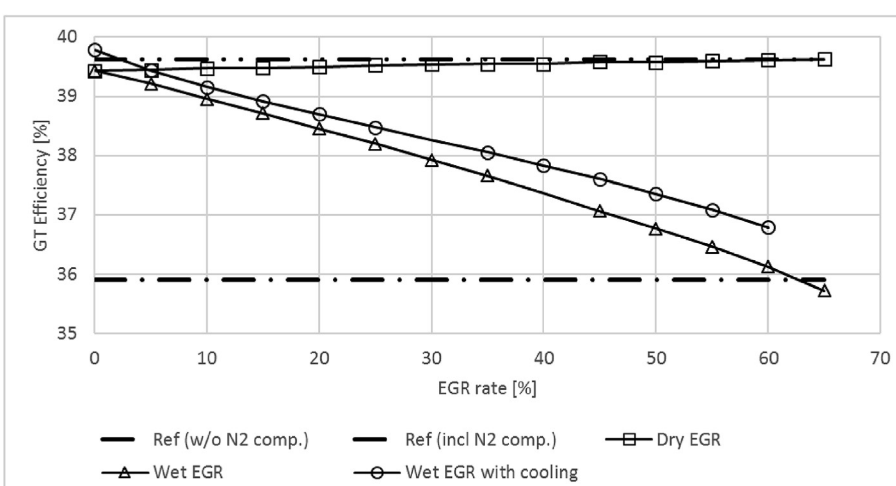

Fig. 9. Simple gas turbine efficiency based on syngas LHV.

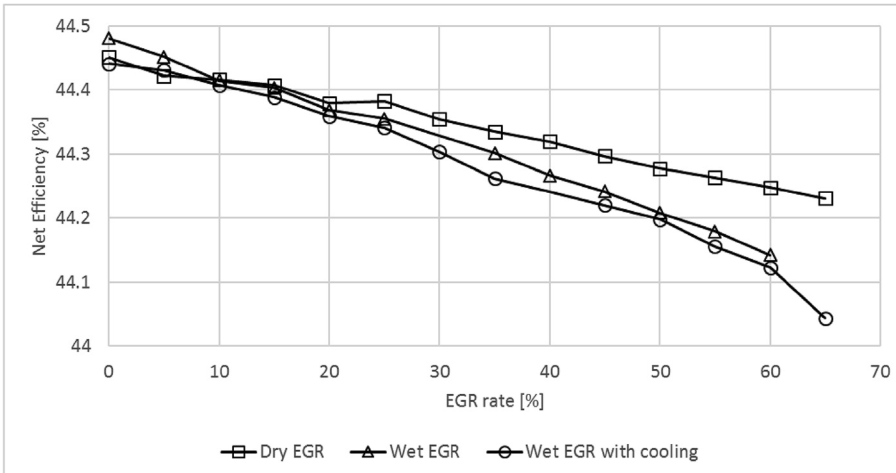

Fig. 10. Net efficiency as a function of EGR rate (reference case with nitrogen dilution has a corresponding efficiency of 43.2%).

For the sake of comparison, the SFT of methane (close to natural gas) in the same combustor conditions is 2464 K. The 2190 K temperature limit, marked in Fig. 11, is achieved above 45% and 55% EGR rates for the wet and dry cases respectively. Reporting these rates in Fig. 10 indicates that the efficiencies are up to 1.0 %-points higher than in the reference case for all EGR cases.

In natural gas fired gas turbine, it is accepted that a maximum of 30%–35% EGR rate could be achieved below which combustion can maintain full combustion efficiency and stability [22]. Nevertheless, Fig. 12 indicates that the reactivity of hydrogen in the high EGR rates required is still sufficient to maintain stable combustion with laminar flame speed being still as high as in the reference case. Noting that the laminar flame speed in these conditions is approximately 110 cm/s also suggest that EGR rates could be further pushed since that value is more than twice that of methane in conventional gas turbine conditions (calculated to be 48 cm/s for the same combustor as in this study).

Although all cases perform approximately equally better than the reference case, but at somewhat lower EGR rate in the wet scenarios, dry EGR is probably a practically better choice from a gas turbine point of view. Indeed, the working fluid has thermodynamics properties that are quasi identical to that of a conventional gas turbine (i.e. air) whereas the moisture content in wet EGR could imply a different choice of materials and a small loss in polytropic efficiency of the turbomachinery. Particularly at the high EGR rates required where water vapour concentration at the compressor inlet reaches 10% at 45% EGR, whereas in dry EGR water vapour concentration is deliberately brought to that of the reference case at all EGR rate.

5.2. Assumptions and accuracy of the study

Several assumptions were made in this study making sure to not over-estimate the potential benefits of the concept. These were:

1) ASU has been attributed double the power consumption as in the reference case with integration, even though we showed that with a conservative specific energy consumption of 200 kWh/t oxygen, the ASU power consumption would be less than that.
2) The need for cooling to lower than ISO ambient temperature for keeping humidity levels in the dry EGR case. The background for this assumption was to make sure that the reference compressor component could be utilized without any modifications and without impeding lifetime and maintenance frequency. In Kakaras et al. [23] simulations showed that the net effect of increasing moisture content is a reduction in the power output, resulting in a gas turbine efficiency reduction of 0.28 %-point for an increase from 0% to 100% humidity, and could thus justify the cooling of the EGR stream. In terms of materials, humidity can represent a corrosion issue for the compressor, however, it must be highlighted that the increased water content is due to hydrogen burning, and in other words clean water without

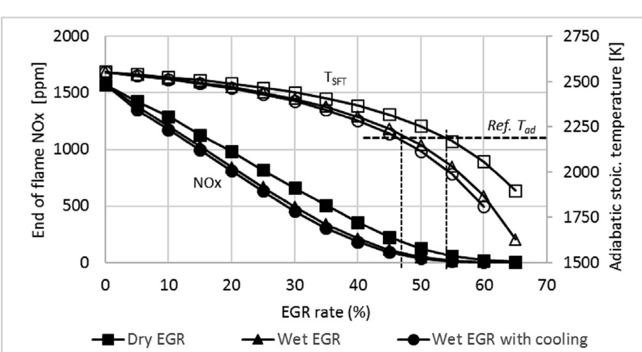

Fig. 11. Calculated stoichiometric adiabatic temperature and end of flame NOx concentration in a laminar free propagating flame.

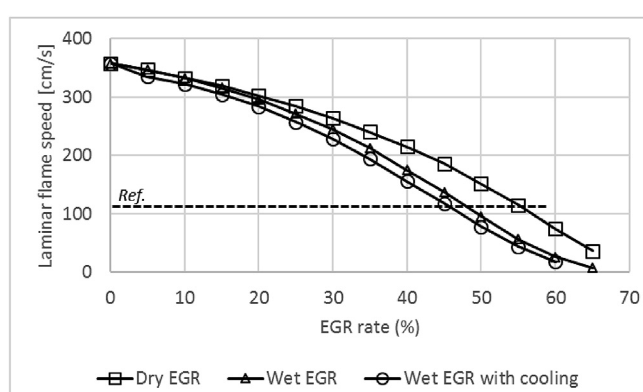

Fig. 12. Calculated stoichiometric adiabatic laminar flame velocity.

traces of impurities or mineral contents. Given that the acceptable relative humidity limit is strongly dependent on local concentration of acidic contaminants [24] which are not presented in the water stemming from the EGR, only efficiency losses should be considered and evaluated in this context. In addition, the deficiency in oxygen in the EGR air stream would probably decrease the aggressivity of potential corrosion issue.
3) In the reference case the working fluid through the turbine has a concentration of 12.25% steam (cf. Table 1) affecting the turbine stage as is well described in Gazzani et al. [9]. At 0% EGR the dry scenario would give approximately 11% steam in the turbine stream, decreasing as EGR rate increases. Therefore, steam content in the EGR concept would not bring more severe degradation than those potentially encountered in the reference scenario with nitrogen dilution.
4) The efficiency of the heat pump to cool the exhaust gas mixed with the air before inlet to the compressor has been given a conservative value.
5) Due to boundary conditions imposed between the power island and the rest of the power plant, a correction for the net heat generated by the syngas production and input to the bottoming cycle had to be applied (cf. discussion in §3.3). This is a recognized, but unavoidable limitation in the comparison work, however this correction represents only between 0.1% and 1.9% of the gross power output depending on the cases and EGR rate, and only 0.2% for the cases which have been considered as optimal (dry EGR with higher than 50% rate). It is therefore not believed that this limitation alters the conclusions of the study.
6) The laminar flame speed calculated and shown in Fig. 12 suggest that the EGR rates assumed to be necessary for achieving equivalent NOx emission performance as in the reference case, could be pushed to higher values and the limitation would come from oxygen availability, more than stability issue

Relaxing even partially some of these assumptions would provide room for improvement on the gain of the concept presented herein and the results can therefore be considered as conservative.

6. Conclusions

To circumvent the difficulties of achieving low NOx emissions when burning high hydrogen content fuels in gas turbines, a concept where exhaust gas is recirculated (EGR) is studied from a process perspective. By forcing exhaust gas back to the compressor inlet, the oxygen concentration in the air decreases as the EGR rate increases to the point where the flame temperature, controlling the NO formation, is limited. The simulations of the power cycle concept were analysed with three options: a dry, a wet, and a wet with cooling of the exhaust gas, all of them built upon and compared against an IGCC power plant with pre-combustion carbon capture with nitrogen dilution as reference cycle. The main findings of the study can be summarized as follows:

- Conventional nitrogen dilution for NOx control in the reference cycle costs 1.3 %-point to the power island efficiency;
- Implementing any of the three EGR options investigated represents a gain in efficiency when nitrogen dilution is eliminated, nevertheless:
- EGR rate needs to be at least 45% and 55% in the wet and dry EGR respectively to meet the same adiabatic flame temperature as in the reference case with dilution and therefore potentially achieve similar NOx levels;
- The gain in efficiency in these EGR conditions is 1 %-point, thus recovering 75% of the loss of efficiency caused by conventional nitrogen dilution of the fuel;
- High laminar flame speed even at high EGR rates indicates that flame stability should not be impaired even at these very oxygen depleted air conditions;

Since the assumptions used are considered as conservative, the results indicate that using EGR instead of nitrogen dilution could very likely increase the total power plant efficiency without compromising on NOx emissions.

Acknowledgements

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