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Towards Zero-Carbon Emission Cogeneration Through Hydrogen Fueling: Assessment of the Impact of CH_4/H_2 Blends on the Thermodynamic Performance of a Gas Turbine CHP Unit[#]

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ABSTRACT

To attain a zero-carbon emissions energy system, we need dispatchable and flexible power production means that can respond to the variability of wind and solar. Hydrogen-fired gas turbines could have a place to fit this demand, but research and development are needed to assess the impact of the fuel change on the systems. This study focuses on a cogeneration unit composed of an aero-derivative industrial gas turbine within a range of 25-35MWe and an HRSG to generate steam for heat and industrial purposes. Setting aside existing challenges for hydrogen-fired gas turbines, we have found that hydrogen has a slightly positive impact on the thermodynamic performance of the considered system. The gas turbine cycle's efficiency and net output power increase by about 2.69% and 4.84% respectively with the H2 ratio, while the heat utilization factor for the whole system does not show significant improvements.

Keywords: hydrogen, gas turbines, thermodynamic performances, combined heat and power, energy systems.

NONMENCLATURE

Abbreviations	
CCGT	Combined Cycle Gas Turbine
CH_4	Methane
CHP	Combined Heat and Power
ER	Expansion Ratio
H_2	Hydrogen
HRSG	Heat Recovery Steam Generator
HUF	Heat Utilization Factor
IEA	International Energy Agency
IGV	Inlet Guide Vanes
LHV	Lower Heating Value
MWI	Modified Wobbe Index
NG	Natural Gas
OEM	Original Equipment Manufacturer
PR	Pressure ratio
TIT	Turbine Inlet Temperature
TOT	Turbine Outlet Temperature
VRE	Variable Renewable Energies
Symbols	
ϕ	Equivalence ratio
$\dot{m}_{fuel}$	Fuel mass flow rate
C_p	Heat capacity
γ	Heat capacity ratio
$\dot{Q}$	Heat duty
P_{net}	Net power output

1. INTRODUCTION

One of the key milestones of the International Energy Agency (IEA) to reach net-zero by 2050 is that almost 70% of electricity generation will need to come from Variable Renewable Energies (VRE) [1]. However, these technologies come with a set of challenges for the operators of the energy systems, who need to ensure the security of supply and the reliability of the system. To be able to integrate them while avoiding power shortcuts and instabilities, the system must be flexible with a firm capacity and have a high level of response for frequency and voltage control [2].

Today's electricity mix varies deeply depending on each country and its strategies. Thanks to their characteristics, gas turbines can operate as base load or peak load. This makes them particularly interesting in a system with a high share of renewables. When sun or wind are not available and the demand needs to be fit rapidly, gas turbine power plants are one of the most efficient and fast responding production means [3].

Despite these facts, the use of natural gas will probably be diminished in the next few decades, either for environmental or geo-political issues. From the end of 2021, the European Union no longer finance traditional fossil fuel projects and it has set a threshold of 250gCO_2 per kW of electricity generated [4].

Nowadays, an interest in hydrogen as a fuel has grown. The IEA forecasts a significant increase in the penetration of hydrogen in the power sector, notably as a mechanism to balance the electricity grid and provide seasonal energy storage. The agency foresees that, to achieve the net-zero energy scenario, approximately 17% of the total demand for H_2 will need to be addressed in the power sector [5].

The combustion of hydrogen, however, brings several challenges in terms of NO_x, combustion instabilities, and flashback. Although the discussion and handling of these issues is out of the scope of this specific study, it is important to comment on it considering the technology deployment. At present, no dual-fuel gas turbine can handle pure hydrogen while respecting NO_x emissions limitations [6].

Hydrogen-fueled gas turbines could play a part in the future energy system because of their dispatchable and flexible operation characteristics and zero carbon emissions. Öberg et al. [7] have explored the competitiveness of hydrogen-fueled gas turbines in future energy systems finding that hydrogen gas turbines may have an important role in the energy system, particularly for very low carbon emissions scenarios.

Besides the challenges related to the hydrogen combustion, altering the fuel will also have an impact on the performance of the gas turbines. In this field, some studies focusing on the thermodynamic cycle of hydrogen-fueled gas turbines have already been carried out. Pyo et al. [8] compared different configurations of Power-to-Gas-to-Power integration for a 427MW Combined Cycle Gas Turbine (CCGT) power plant and analyzed the thermodynamic cycle. However, they focused on CO_2 emissions and economic feasibility, showing that CO_2 emissions could be reduced up to 95.5% and a shorter payback period for the Power-to-H_2 plant than for the Power-to-Methane system. Arsalis [9] and Lopez-Ruiz et al. [10] modeled a Joule-Brayton cycle and a regenerative Brayton cycle respectively varying the ratio of H_2 in a blend with Natural gas (NG), to understand the impact of these blends on the thermodynamic cycle. Both showed a slight improvement in the efficiency of the cycle with the increase of H_2, 1% for a 0.9 ratio H_2/NG [9]and 4% for pure H_2 [10]. Despite this first attempt to assess the impact of fuel conversion on gas turbine performances, more in-depth research remains necessary, not only to characterize the impact of hydrogen on the gas turbine cycle itself, but also on its potential towards heat production.

In this paper, we will assess thus the impact of the use of H_2 as fuel on the thermodynamic performance of a 25-35MW_e [11,12] gas turbine coupled with a heat recovery steam generator (HRSG) for a combined heat

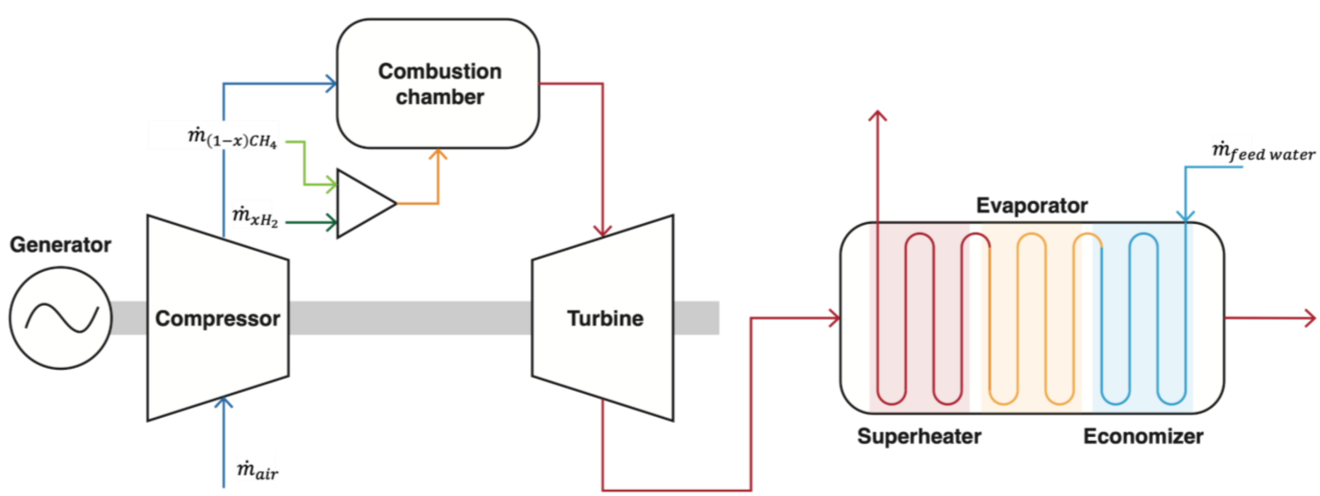

Fig. 1 The simulated gas turbine is represented by a simple Brayton cycle. The compressed air and the fuel blend enter in the combustion chamber, generating flue gases at high temperature. They continue to the turbine to produce electrical power and finally go into the HRSG to produce steam for industrial or heating purposes.

and power (CHP) utilization. We have focused on the thermodynamic performance only and hence, an in-depth discussion on the combustion performance is outside of the scope of this work.

2. METHODOLOGY

In this paper, we simulate the effect of the increasing H_2 ratio in the fuel blend on the thermodynamic performances of a CHP system composed of a gas turbine and a heat recovery steam generator (HRSG) (Fig. 1).

For this simulation we considered the GE LM2500 RD as a reference. This turbine has a nominal power output of 32.69MW$_e$. Under iso-conditions and nominal operation burning CH_4, the electric efficiency accounts for 38.3% and the pressure ratio (PR) for 23:1, the temperature at the outlet of the turbine (TOT) reaches 525°C, and 91kg/s of exhaust mass flow goes through the system [12]. These values are summarized in Table 1.

Table 1. Summary of OEM datasheet for the LM2500 RD.

Output power	32.69MW$_e$
Pressure ratio	23
Electric efficiency	38.3%
TOT	525°C
Exhaust mass flow rate	91 kg/s

2.1 *Simple gas turbine cycle model*

Firstly, we modeled the gas turbine using the parameters present in the datasheet of the OEM [11,12]. To do it, we used the process simulation software Aspen plus®. For the compressor, we fixed the PR at 23 and an isentropic efficiency of 83%. The PR was obtained from [12]. Considering that the isentropic efficiency was unknow, the value was obtained through an iterative procedure (Fig. 2), where we aimed to match the OEM data. The combustion chamber was simulated with a Gibbs reactor, assuming complete combustion and no pressure and heat losses. As there are no by-passes for blade cooling, meaning all compressed air is going through the combustor, we set a first design specification by limiting the temperature at the inlet of the turbine to 1500K, to prevent material damage. The turbine was modeled by setting the discharge pressure at 1.05 bar and an isentropic efficiency of 0.925, found by following a similar procedure as for the compressor. Both mechanical efficiencies, for the compressor and the expander, were set at 0.99. A second design specification included setting the given power output of 32.69MW$_e$. To respect these design specifications, Aspen plus® adapts the fuel mass flow rate to limit the TIT and the air mass flow rate to attain the requested power output. This whole procedure allows us to obtain an estimate of the exhaust mass flow and the TOT. The set of input parameters is shown in Table 2.

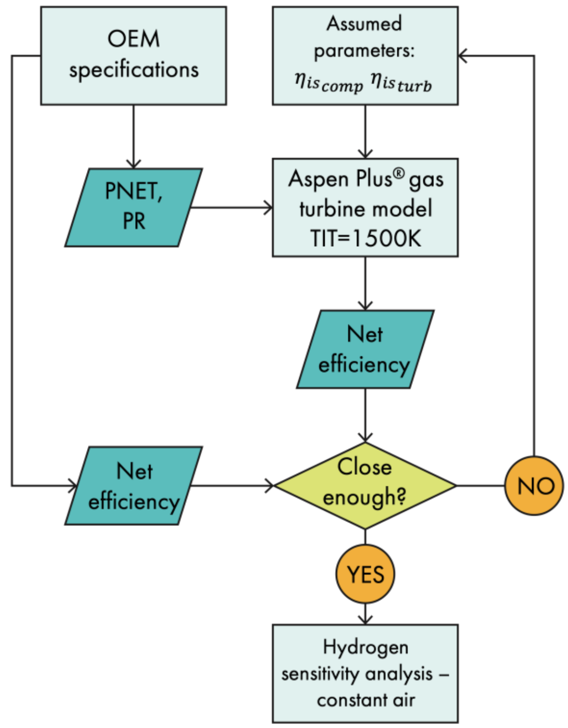

Fig. 2 Scheme of the iterative process to approach real cycle parameters with a simple cycle model.

It is worth noting the uncertainty inherent to the simple cycle calculation. A real cycle must include pressure, heat and mechanical losses, cooling flows, performance maps, true operation TIT or TOT, and PR. For the matters of this study, we started with a simple gas turbine model as described in the preceding paragraph.

Table 2. Summary of main parameters values used in the model.

Compressor	*Isentropic efficiency*	0.83
	Pressure ratio	23
	Mechanical efficiency	0.99
Turbine	*Isentropic efficiency*	0.925
	Discharge pressure	1.05 bar
	Mechanical efficiency	0.99
TIT		1500K
Output power		32.69MW$_e$

To simulate the impact of fuel blending, the molar fraction of H_2 in the fuel was altered from 0 to 100%. Assuming ISO conditions and inlet guide vanes (IGV) fully open for the same compressor, the air entering the system remains constant for the analysis. No modifications have been done to the turbine. Although for the validation, we considered operation at constant power output, for this analysis of the impact of H_2 on the cycle, we only consider constant TIT (constant power design spec was deactivated). Hence, Aspen plus® exclusively adapts the fuel mass flow rate to limit TIT. This choice was made to have a better understanding on the behavior of the cycle after the combustion chamber.

2.2 *HRSG model*

For the HRSG part, we have included three generic heat exchangers at one pressure level. Firstly, an economizer, where cold feedwater enters the system to be heated until reaching the saturation temperature for the input pressure. At this point, flue gases leave the system at a low temperature. The evaporator generates steam, while finally the superheater increases the temperature till 450°C using the flue gases coming out of the gas turbine.

In this HRSG, a pinch point of 10°C was set between the temperature of the flue gases leaving the evaporator and the saturated water coming from the economizer and entering the evaporator. For this design specification, Aspen plus® adapts the feedwater mass flow rate by respecting the pinch point for a given pressure. It enables to obtain an optimized heat exchange by maximizing the mass flow rate.

2.3 *Comparison of different cases*

To illustrate the impact of the chosen parameters used to model the systems, we decided to compare different settings shown in Table 3. Case 1 is the base case, explained in section 2.1. Pressure ratio and component efficiencies remain the same. We simulate the cycle by varying all these parameters thanks to the design specifications in Aspen plus®. As previously stated, this are simple cycle calculations purely used to perform a preliminary assessment of the impact of H_2 on the cycle.

Table 3. Summary of compared cases for preliminary assessment

	TIT (K)	*TOT (°C)*	*Output power (MW_e)*	*Energy input (kJ/s)*
Case 1	1500			
Case 2	1500		32.69	
Case 3		525		
Case 4		525	32.69	
Case 5			32.69	85509,17

3. RESULTS

3.1 *Impact of the H_2 blend on the cycle performance*

Increasing the H_2 fraction on a fuel blend with CH_4 has no immense effect on the thermodynamic cycle performance for nominal operation and ISO conditions. We notice an almost 5% increase in power output and a 2.69% relative increase in electric efficiency of the cycle, as shown in Fig. 3 and Fig. 4 The power needed to drive the compressor is constant given that this component is not impacted by the fuel alteration, and the air flow rate was kept constant.

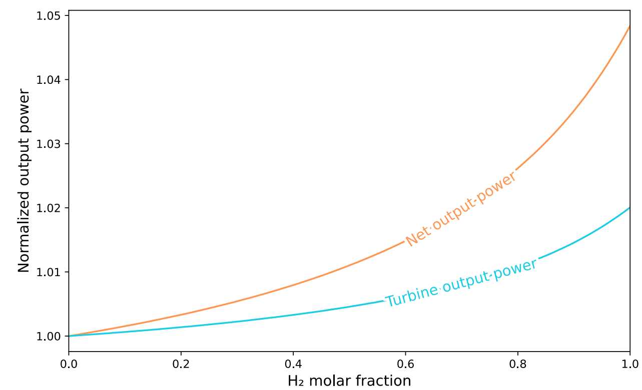

Fig. 3 The H_2 fraction has a positive effect on the net power output of the gas turbine.

As described in the methodology all parameters are kept constant, except for the fuel mass flow rate, which alters due to the TIT control. It means that the increase in the power output is only related to the fuel change. Complete combustion of H_2 generates only water, so the higher the H_2 fraction, the higher the quantity of water in the flue gases is.

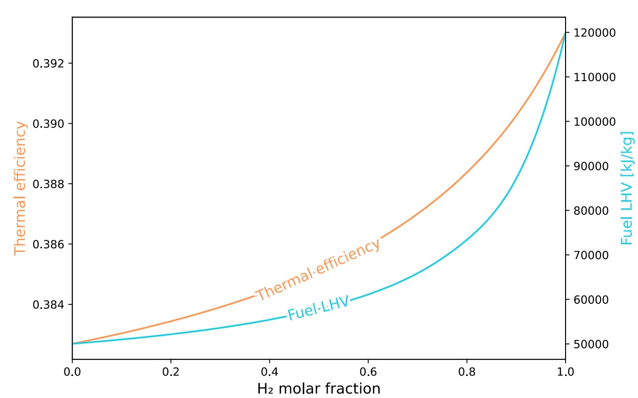

Fig. 4 The electric efficiency of the gas turbine increases 2.69% with respect to the base case (relative increase).

The heat capacity ratio of flue gases increases with the rise of the water fraction from the combustion products as shown in Fig. 5. If we analyzed the isentropic relation for an ideal gas in the Eq. 1, we could notice that TOT decreases for a bigger heat capacity ratio (γ). In this order of ideas, the enthalpy of the hydrogen combustion products is higher than the methane one. Hence, the observed increase in the power output is due to the increase of the water content on the flue gases, and the reduction of CO_2 with the introduction of H_2.

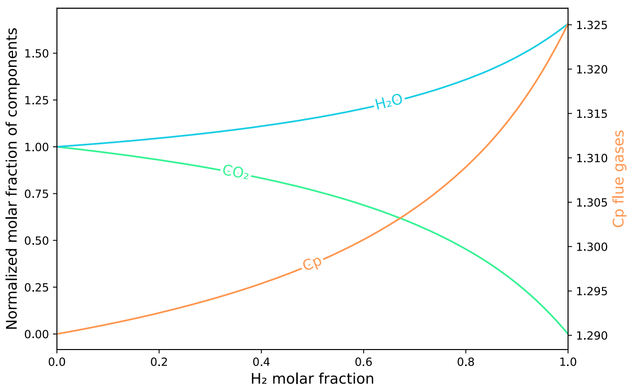

Fig. 5 The composition of the flue gases changes with the increase of the H_2 ratio for a TIT=1500K.

$$\left(\frac{TOT}{TIT}\right)^{\frac{1}{\gamma-1}} = (ER)^{\frac{1}{\gamma}} \qquad [13] \qquad (1)$$

Considering that gas turbines require a very lean operation to limit emissions and too high temperatures, they run with about 100% excess air [14]. In this case, as we operate at a lower TIT [15], the excess of air exceeds 200% for each fuel. This means that γ will only display rather limited changes. Still, as we have seen small variations can impact the performance of the gas turbines.

We can observe in Fig. 6, that indeed the mass flow rate of the fuel decreases with the increase of the H_2 fraction. This is due to the higher low heating value (LHV) in terms of the mass of H_2, in counterpart, the volume flow rate (at ISO conditions) growths, and it is the parameter that determines the design of the fuel system. The modified Wobbe Index (MWI) indicating a measure of the energy injected into the combustor, allows to compare different fuels. For heavy-duty gas turbines variation up to ±5% can be allow, still maintaining stability [14]. Considering that H_2 has a 40.9 WI, 15% lower than the WI of CH_4. For this kind of situation, a retrofit of the existing fuel system will be required e.g., the difference of WI between fuels can be compensated with a lower fuel temperature or a larger fuel nozzle [14].

Flue gases leave the gas turbine at a temperature of 521°C for 100% H_2 and 526° for 100% CH_4. At first, we estimate the potential of heat production from these flue gases for cogeneration first by using a generic heat exchanger in Aspen Plus®. In this heat exchanger, flue gases are cooled down till 15°C, allowing to assess the full thermal potential for cogeneration. Results indicated that hydrogen combustion flue gases have a potential increase of 12% relative to the methane base case.

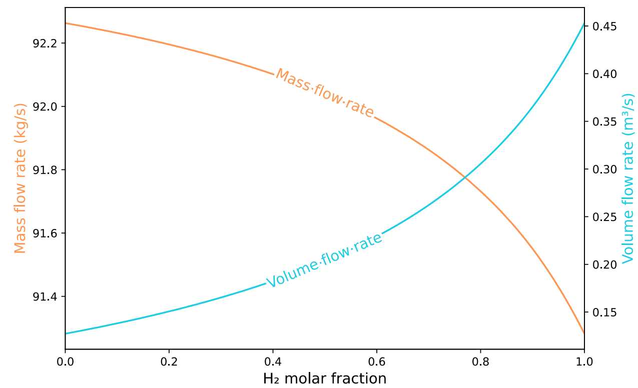

Fig. 6 Due to the characteristic of gases, the volume flow rate is more significant for the fuel system design.

Although the potential has increased for cogeneration, in practical applications, cooling till 15°C is not possible and hence the temperature of the cold source will determine the potential. To assess this potential, the generic heat exchanger was replaced with a HRSG, simulated as explained in section 2.2. As indicated before, one important parameter for the thermal power production is the feedwater temperature. In real application, this temperature depends on the returning state (pressure) of the steam loop.

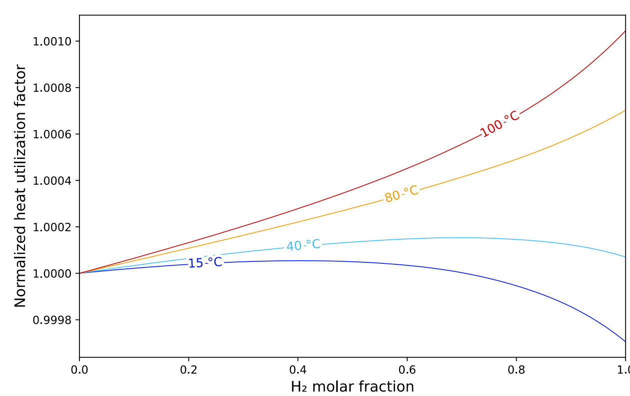

Fig. 7 Normalized heat utilization factor for a range of feedwater inlet temperatures at constant inlet pressure of 8 bar.

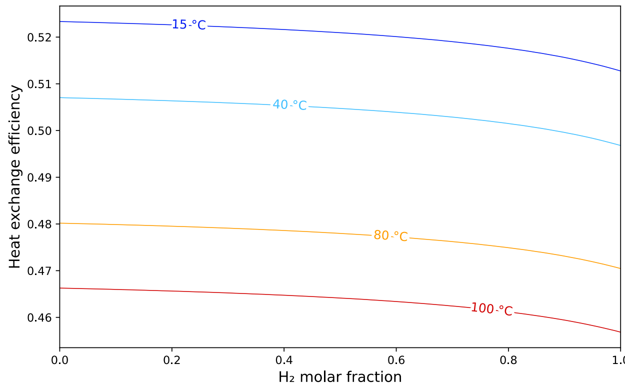

Fig. 8 Heat exchange efficiency term for a range of feedwater inlet temperatures at constant inlet pressure of 8 bar.

We have analyzed several settings of temperature and pressure of the feed water to see the impact of H_2 on the heat exchange. The heat utilization factor defined by Eq. 2, depends somewhat on the H_2 fraction. For a constant pressure of 8 bar (Figs. 7-8), we remark a difference between the high and low feedwater temperatures. The thermal power term in the efficiency decreases with the increase of H_2 for every case. It is evident that for higher temperatures, the thermal power term in the efficiency decreases, but this translates into an unfavorable impact on the HUF that is smaller for higher than for lower temperatures. It means that the efficiency of the gas turbine cycle preponderates over the HRSG one for higher temperatures.

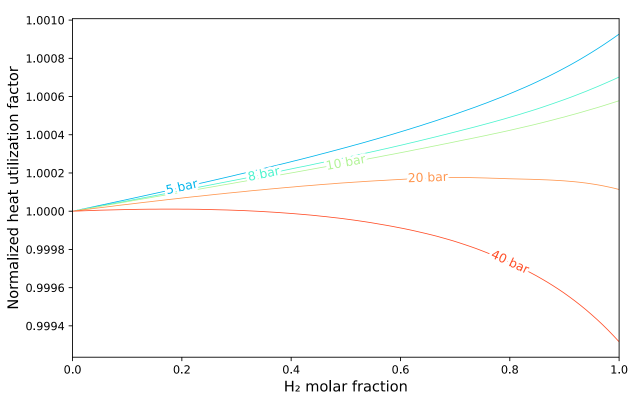

Fig. 9 Heat utilization factor for range of feedwater inlet pressure at constant inlet temperature of 80°C.

For a constant temperature of inlet feedwater (Fig. 9), we also observe a distinct different impact on the thermal power production between low and high pressures. Contrary to the previous example, less thermal power translates into a negative impact that is greater than for lower pressures. We can see in Fig. 10, that for higher pressure, the flue gases of the CH_4 case exchange slightly more heat with the cold fluid. But for the lower pressure, though barely noticeable, the inverse happens. It is due to the heat exchange curves related to the mass flow rate and the Cp of the flue gases.

$$HUF = \frac{P_{net_{GT}} + Q_{HRSG}}{\dot{m}_{fuel} LHV} \quad [16] \qquad (2)$$

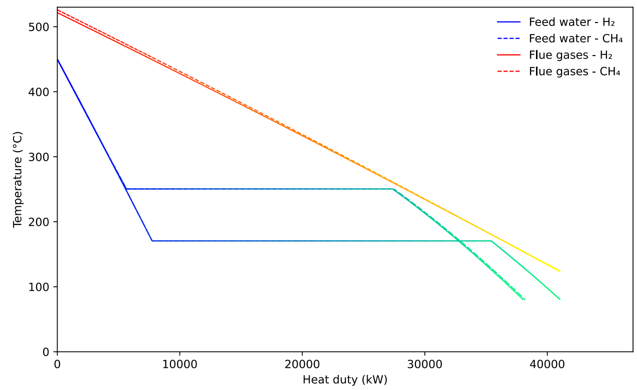

Fig. 10 Heat exchange curves for inlet pressure of 40 and 8 bar for the cold side.

For the outlet temperature of flue gases, we can note that it remains nearly constant with the variation of the H_2 fraction. But the dew point temperature increases with the change in the composition of the flue gases, so it is essential to verify it for HRSG design.

3.2 *Impact of the control parameters*

A modification of the control parameters of the model can lead to significant changes in the system behavior. In Fig. 11, TIT and TOT don't vary significantly for cases 2 and 4. Fixing one of the temperatures and the isentropic efficiency of the turbine will limit their variation. But for case 5, we see a significant drop in the temperature. It is directly related to an augmentation of the air entering the system (Fig. 11). Evidently, as we modified the air flow going into the system for case 2, 4 and 5 (Fig. 12), the compressor work varies. The change of flue gases composition and the diminution of the fuel flow rate impact the turbine output power (Fig. 13). Finally, the sum of the compressor work and the turbine leads to either constant output power for cases 2, 4 and 5 or an increase for the two others (Fig. 13). For every case, where the efficiency is not imposed, the efficiency increases with the increase of the H_2 fraction (Fig. 14).

Lopez Ruiz et al. [10] commented on the fact that pure hydrogen needs less air than methane to do complete combustion and the need to evaluate the size of the compressor. The statement is not completely accurate, since the H_2 air/fuel ratio stoichiometric (AFR) is twice the CH_4 [17]. In case 2, to generate the same amount of power with a hydrogen-fired gas turbine, we would need less air mass flow rate entering the system, but less amount of fuel is needed. In contrast, case 5 needs more than a 10% increase of air to obtain the same efficiency as the CH_4-fueled gas turbine. But, with the high dilution, the drop of TIT and TOT, the system is losing the potential efficiency against these constrains.

4. CONCLUSIONS

Hydrogen offers large potential for decarbonization of our current electricity production. However, its impact on the thermodynamic performance of gas turbines still needs to be analyzed. Throughout this preliminary assessment of a CHP configuration, we concluded that H_2 has a slight positive impact on the thermodynamic performance of the system. It may provide a minor relative increase of around 2.69% of the electric efficiency and 4.8% of the nominal power output of the gas turbine, but the validity of the made assumptions still needs to be confirmed. Although, hydrogen has a higher heat potential it is not reflected in the HRSG system performance and depending on the conditions we can notice a decline of the HUF in respect to the base case with the CH_4 fired gas turbine.

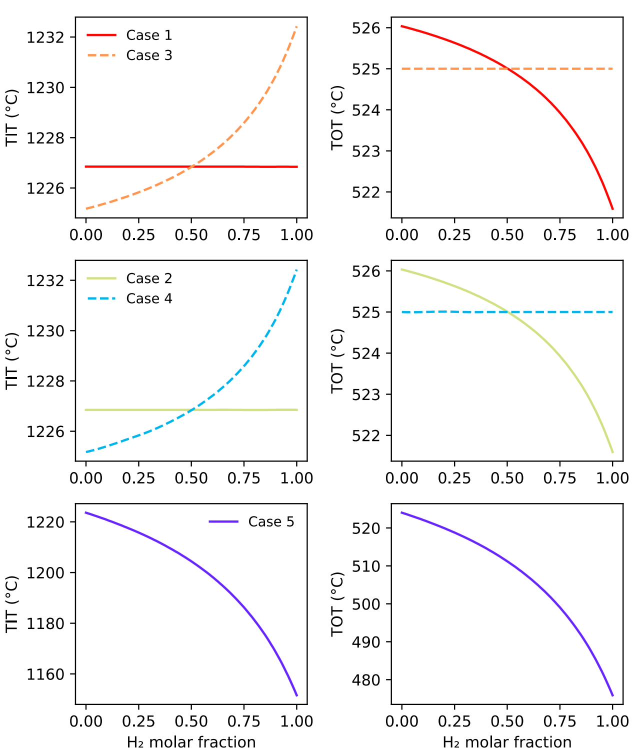

Fig. 11 Variation of temperature of the flue gases for the different cases.

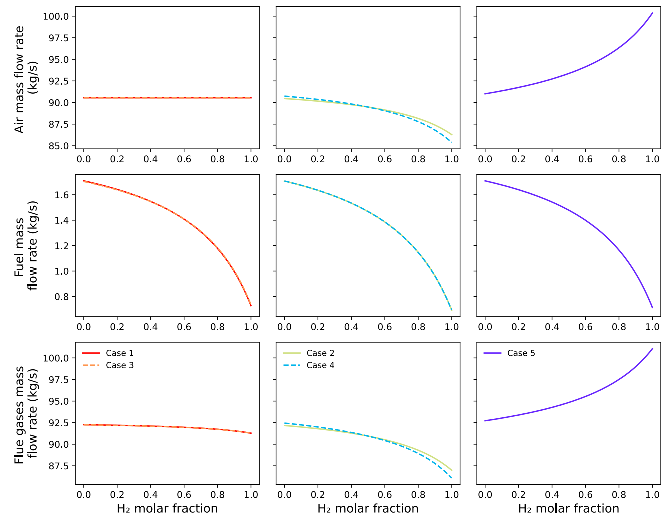

Fig. 12 Variation of mass flows for the different cases. For the first four cases TOT or TIT are controlled by the fuel flow rate. The last case the energy input = $\dot{m}_{fuel}LHV$ is constant, as the LHV of H_2 is higher than the CH_4 one, the mass flow rate needs to decrease.

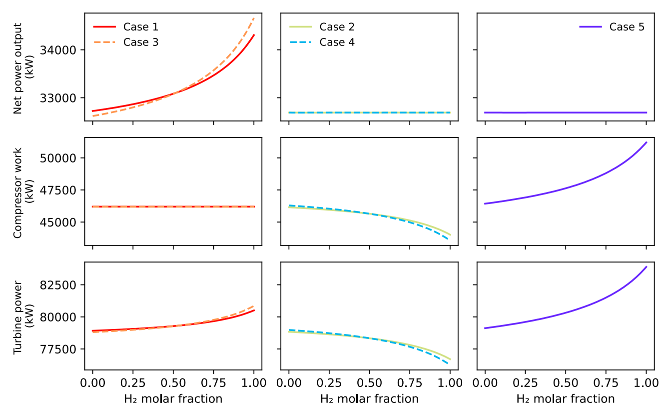

Fig. 13 Impact of the hydrogen fraction on work and power outputs.

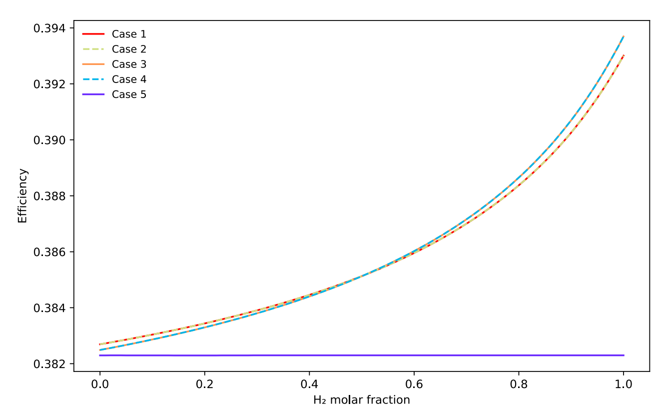

Fig. 14 The first four cases show that it is possible to obtain higher efficiencies for H_2-fired gas turbines. For case 5, we impose constant efficiency by fixing output power and energy input. For this case, we identify a significant increase in the air entering the system, this means that the turbine will be working under off-design conditions, or it will not be able to handle the increase of air.

In this study, we focus in the analysis on the impact of flue gas composition associated with the combustion products of each fuel. Particularly, for the expander, the flue gas composition may influence the isentropic efficiency. This model is performed under very simplified conditions. It is imperative to use the component's performance map to have a more realistic image of the system, as well the cooling flows and pressure losses.

5. DISCUSSION AND FUTURE WORK

Despite all the challenges that today represent H_2 combustion, it seems to be here to stay as a zero-carbon emission alternative fuel for the future energy system. In this instance, it is important to continue to evaluate its impact on different configurations of gas turbines, CHP and CCGT. In the next work, more detailed models that reflect the real operation of the systems will be done, especially at off-design and part-load operations.

ACKNOWLEDGEMENT

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REFERENCE

[1] IEA. Net Zero by 2050 - A Roadmap for the Global Energy Sector. Paris: 2021.
[2] IRENA. Planning for the Renewable Future: Long-term modelling and tools to expand variable renewable power in emerging economies. International Renewable Energy Agency (IRENA); 2017.
[3] IRENA. Innovation landscape brief: Flexibility in conventional power plants. Abu Dhabi: 2019.
[4] EIB. EIB energy lending policy - Supporting the energy transformation. Luxembourg: 2019.
[5] IEA. Global Hydrogen Review 2021. Paris: 2021.
[6] ETN Global. Hydrogen Gas Turbines. Brussels: 2020.
[7] Öberg S, Odenberger M, Johnsson F. Exploring the competitiveness of hydrogen-fueled gas turbines in future energy systems. International Journal of Hydrogen Energy 2022;47:624–44.
[8] Pyo MJ, Moon SW, Kim TS. A comparative feasibility study of the use of hydrogen produced from surplus wind power for a gas turbine combined cycle power plant. Energies (Basel)
[9] Arsalis A. Thermodynamic modeling and parametric study of a small-scale natural gas/hydrogen-fueled gas turbine system for decentralized applications. Sustainable Energy Technologies and Assessments 2019;36.
[10] Lopez-Ruiz G, Castresana-Larrauri J, Blanco-Ilzarbe JM. Thermodynamic Analysis of a Regenerative Brayton Cycle Using H2, CH4 and H2/CH4 Blends as Fuel. Energies (Basel) 2022;15:1508.
[11] LM2500 & LM2500XPRESS Gas Turbines | GE Gas Power n.d. https://www.ge.com/gas-power/products/gas-turbines/lm2500 (accessed May 23, 2022).
[12] The Market for Gas Turbine Electrical Power Generation n.d. https://www.forecastinternational.com/samples/F646_CompleteSample.pdf (accessed May 23, 2022).
[13] Cengel YA, Boles MA. Thermodynamics (Asia Adaptation): An Engineering Approach 2010.
[14] Gülen SC. Gas turbines for electric power generation. Gas Turbines for Electric Power Generation 2019:1–720.
[15] Lugo-Leyte R, Zamora-Mata JM, Toledo-Velázquez M, Salazar-Pereyra M, Torres-Aldaco A. Methodology to determine the appropriate amount of excess air for the operation of a gas turbine in a wet environment. Energy 2010;35:550–5.
[16] Matjanov E. Gas turbine efficiency enhancement using absorption chiller. Case study for Tashkent CHP. Energy 2020;192:116625.
[17] Catapano F, di Iorio S, Magno A, Sementa P, Vaglieco BM. A comprehensive analysis of the effect of ethanol, methane and methane-hydrogen blend on the combustion process in a PFI (port fuel injection) engine. Energy 2015;88:101–10.