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PRELIMINARY NUMERICAL STUDY OF METHANE-HYDROGEN CO-COMBUSTION EFFECTS ON HEAVY-DUTY GAS TURBINES

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ABSTRACT

The use of hydrogen as a fuel for gas turbines is certainly an opportunity to limit greenhouse gas emissions, despite the still not-negligible limitations related to materials and the management of combustion processes.

This work aims at proposing a numerical model, developed by the Authors by means of the link between Matlab™ environment and Aspen HYSYS™ environment, able to evaluate the performance of commercial gas turbines with a set geometry in response to the co-combustion of CH_4 and H_2 for different fuel mixture compositions.

The purpose of this study is therefore to propose a modelling approach to assess the use of hydrogen injection in the combustion chamber of commercial gas turbines, already in operation, as a short-term strategy for reducing carbon emissions.

The preliminary results obtained by the simulation process show that the increase in the hydrogen content in the fuel mixture, up to a maximum of 62% by volume, corresponds to a slight decrease in the performance of the machine, in particular net electrical power and efficiency, of about 3%, but also to a reduction in CO_2 emissions of more than 30%.

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Keywords: hydrogen, gas turbines, calculation code, hydrogen turbines, green hydrogen

NOMENCLATURE

k	Specific heat ratio [-]
$\dot{m}$	Mass flow [kg/s]
p	Pressure [bar]
$\dot{Q}$	Thermal power [kW]
R	Gas constant [J/kgK]
RH	Relative Humidity [%]
T	Temperature [K]

<u>Subscripts and Superscripts</u>

air	air
des	Design
in	Inlet
f	Fuel
p	Primary zone
T	Temperature

Greek letters

η	Efficiency [-]
φ	Flow coefficient
Φ	Equivalence ratio [-]
ψ	Pressure coefficient

Acronyms

CC	Combustion Chamber
CCS	Carbon Capture and Storage
EU	European Union
GT	Gas Turbine
HD	Heavy duty
ISO	International Organization for Standardization
LHV	Lower Heating Value [kJ/kg]
MFF	Mass Flow Function [m^2]
RES	Renewable Energy Source
SF	Shape Factor
TIT	Turbine Inlet Temperature
TOT	Turbine Outlet Temperature

1 INTRODUCTION

Following the latest international agreements, the EU's 2050 roadmap has set the target for the next 30 years to reduce greenhouse gas emissions by 80 - 95 % (compared to 1990 levels) through investment plans to replace traditional power plants and introduce less impactful technologies [1-3]. However, the advent of COVID-19 and the recent war in Ukrainian territory have strongly impacted the global economy, creating disruption also to the energy sector and causing an increase in investments in the extraction of fossil fuels, including coal [4, 5]. Compared to oil or coal-fired thermal power plants, current generation systems based on natural gas-fueled turbines have relatively low emissions of pollutants, especially in the combined cycle configuration. However, in order to remain within the limits imposed by the Paris Agreement it is necessary to further reduce the levels of polluting emissions in the energy sector and to encourage the introduction of carbon capture and storage technologies (CCS). According to the European Investment Bank, no more funds will be granted to build up power plants exceeding 250 gCO2/kWhel (the previous limit was set at 550 gCO2/kWhel) and this entails the need of a massive adoption of CCS and decarbonization techniques for traditional energy systems, even the most innovative CCs (typically around 330 gCO2/kWhel) [6]. To accelerate again the general process of transition and decarbonization of the energy sector would require an economic boost capable of increasing the penetration of Renewable Energy Sources (RES) and innovative low-emission technologies but also increase the efficiency of existing production systems [7]. In this context, the use of hydrogen for energy applications therefore remains a promising prospect in response to the uncertainty of the RES and the need to decarbonize energy production through the latest generation of internal combustion engines and turbines [8].

In the last decades, the use of hydrogen as a co-fuel in a blend with natural gas for gas turbines (GTs) has been widely studied in the literature. However, although the combustion of hydrogen in GTs represents an interesting perspective in terms of greenhouse gas emissions, especially if compared to conventional fuels, the GT pure hydrogen cycle is still limited by several factors such as the thermal and mechanical resistance of the materials and the operating conditions of the machine [9].

In this context, Chiesa et al. [10] investigated the possibility of using hydrogen as fuel for gas turbines designed to operate with natural gas. In more detail, the authors identify three main issues related to the use of hydrogen as fuel for gas turbines: i) the enthalpy drop variation in the expansion; ii) variations in the gas flow rate at the turbine inlet and thus variations in the turbine/compressor coupling; iii) variations in the heat transfer coefficient of the exhaust gases and thus impact on the cooling system of the machine. In addition, Meziane et al. [11] studied, through a numerical simulation, the effect of adding hydrogen to natural gas in a rich/quench/lean combustor for a gas turbine. and demonstrated that – with 10 % of hydrogen (volume based) in a hydrogen-natural gas blend – it is possible to reduce emissions of NO and CO respectively by 14 % and 60 %, compared to the use of 100 % natural gas. In his study Pashchenko [12] reported as main result of a numerical study that 20% (vol) and 75% (vol) H2 gives a reduction in CO2 emission of 7.2% and 51.1%, respectively.

The industrial world is also focusing on the study of hydrogen as a fuel for existing or innovative gas turbines. In 2022, Siemens Energy inaugurated a new demonstration plant (ZEHTC) with the objective of testing the use of hydrogen as fuel up to 100% by volume in zero-emission gas turbines by 2030 [13]. At the same time, ENGIE selected and installed in the port of Antwerp (Belgium) the industrial gas turbine Siemens SGT-600 (Alstom legacy GT10B) 24 MW as a demonstrator for cofiring natural gas with hydrogen to perform tests up to 25% hydrogen by volume in the mixture [14], while Burnes et al. [15] from Solar Turbines analyzed the performance impact of the adaptation of industrial gas turbines for carbon capture and the use of low carbon fuels, such as hydrogen and natural gas-hydrogen mixtures.

In this study, a calculation code, developed by the Authors, is presented and discussed. The code is based on the bidirectional interaction between MatlabTM and Aspen HYSYSTM for the macroscopic simulation of Brayton thermodynamic cycles and is able to evaluate the main operational parameters and thermodynamic variables as well as the performance of a gas turbine in response to hydrogen injection in the combustion chamber.

The software has also been applied to study the effect of CH4-H2 co-combustion on the behavior of a mid-size heavy-duty (HD) gas turbine. In particular, starting from the developed calculation routine, it is possible to propose some operational strategies based on the co-combustion of CH4 and H2, depending on the composition of the fuel blend, with the aim to reduce carbon emissions from exhaust gases without changing the geometry of the system. The numerical model allows to determine the performance of each commercial gas turbines in order to evaluate the trend of the main thermodynamic variables for machines with fixed geometry.

The purpose of this study is therefore to propose a modelling approach to assess the use of hydrogen injection in the combustion chamber of commercial gas turbines, already in operation, as a short-term strategy for reducing carbon emissions.

2. METHODOLOGY

In this section, the new calculation model developed in order to assess the performance of commercial GTs when fueled by a blend of natural gas and hydrogen (up to 100% hydrogen) is presented, along with its validation. Furthermore, a preliminary application of the code to a commercial heavy-duty machine is shown, to highlight the calculation code potentialities. To this respect, it must be pointed out that the developed calculation model is able to evaluate the performance of any gas turbine in response to hydrogen injection in the combustion chamber. Indeed, due to the used approach, the model is completely general, and it can be applied to a large variety of industrial gas turbines.

2.1 Numerical approach

The GT numerical code has been developed in an innovative environment integrating Matlab^{TM} [16] environment and Aspen HYSYS^{TM} [17] environment. In particular, the main model has been developed in Matlab^{TM} environment, where specific sub-models have been implemented for the key components of the GT. On the other hand, Aspen HYSYS^{TM} – being a commercial tool for numerical lumped-parameter modelling of complex energy systems – has been applied to the main model in order to evaluate the full profile of thermodynamic quantities in each point of the gas turbine. With this purpose, a link between the Aspen HYSYS^{TM} environment and the Matlab^{TM} environment has to be established; the link, in this study, is based on the creation of a virtual server in Matlab^{TM} environment, that returns a handle to the default interface of the server.

The calculation procedure starts with the thermodynamic design analysis (fuel constituted by pure methane), performed within Matlab^{TM} environment, based on the set point design power input and on the basis of the other system boundary conditions. Then, the off-design calculation of the same system performance under different conditions of fuel blend can be performed, by interrogating specific sub-models. More in detail, concerning the thermodynamic analysis under design and off-design conditions, with pure methane as fuel to feed the machine, reference is made to a previous study by the authors, in which a calculation code was developed and validated to simulate the wet compression in gas turbines [18].

The simplified flow chart of the calculation model is presented in Fig. 1, in which sub-routines of the compressor, the combustion chamber (CC) and of the turbine are highlighted; they contain the physical-mathematical models used to evaluate the respective performance. The input section includes general inputs and gas turbine specific inputs (i.e., GT set-up data), that are provided to the fuel blend evaluation block by means of the Matlab^{TM} main model. It should be observed that the data in the inputs array must be defined by the program user and that the gas turbine set-up array contains all the parameters that characterize the machine.

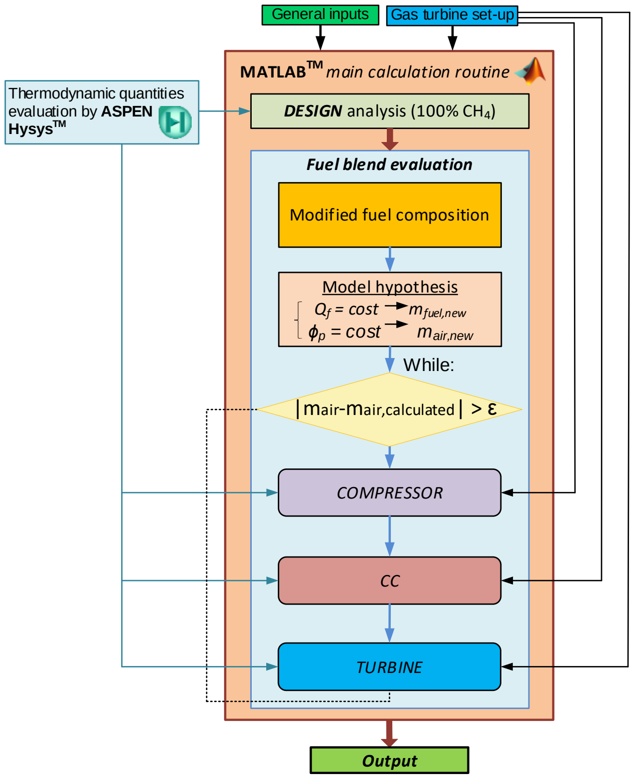

FIGURE 1: SCHEMATIC BLOCK DIAGRAM OF THE NUMERICAL PROCEDURE FOR GT PERFORMANCE EVALUATION

In more detail, the general inputs are:

- ambient conditions (i.e., site temperature, pressure and relative humidity);
- gas turbine control logic (constant turbine inlet temperature, constant turbine outlet temperature, etc.);
- fuel composition, pressure (p_f) and temperature (T_f).

The gas turbine set-up data, instead are:

- gas turbine performance and thermodynamic parameters at ISO conditions;
- number of compressor stages;
- geometrical data of the compressor stages;
- compressor stage performance maps;
- turbine performance maps;
- pressure losses at compressor inlet, through the combustion chamber and at the turbine outlet;
- mathematical model tuning parameters;
- compressor and turbine mechanical efficiency;
- generator efficiency.

Once the fuel blend evaluation has been performed, the parameters calculated by the different modules of the code are collected by the Matlab™ main model and written on the output array. In particular, the output of the code is:

- gas turbine power output and heat rate;
- full profile of thermodynamic quantities in each point of the gas turbine;
- compressor pressure ratio;

- turbine expansion ratio.

The specific modelling of each GT component off-design operation is detailed in the following sub-sections.

2.1.1 Compressor

Regarding the study of the flows inside the compressor, it was chosen to simulate the performance of the machine stage, in off-design conditions, by using some generalized relationships that involve the load coefficient $\psi^* = \psi/\psi_{des}$, the flow coefficient $\varphi^* = \varphi/\varphi_{des}$ and the isoentropic stage efficiency of the compressor $\eta^* = \eta/\eta_{des}$, suitably normalised to design values [19]. Through these relationships it is then possible to evaluate the behavior of each stage of the compressor from the known design point values (ψ_{des}, φ_{des} and η_{des}).

For the first generalized relationship, $\psi^* = F(\varphi^*)$, the Muir et al. [20] model was adopted, in which, by means of an experimental characterization carried out for several stages of the compressor it was possible to reproduce a generalized stage curve. The curve has been approximated by the following relationships:

$$\psi^* = \psi^*_{max} - \frac{\psi^*_{max} - 1}{\left(\varphi^*_{\psi_{max}} - 1\right)^2}\left(\varphi^*_{\psi_{max}} - \varphi^*\right)^2 \qquad (1)$$

Subsequently, in order to take into account the different fluid dynamic characteristics for each type of stage considered, it was decided to introduce the shape factor (SF) in such a way as to cover all the experimental data points reported by Muir et al. [20] through a set of generalized curves $\psi^* = F(\varphi^*, SF)$. The equation of the generalized stage curves is then adjusted as follows [19]:

$$\psi^* = \psi^*_{max} - \frac{\psi^*_{max} - 1}{\left(\varphi^*_{\psi_{max}} + SF\left(\varphi^*_{\psi_{max}} - 1\right) - 1\right)^2}\left(\varphi^*_{\psi_{max}} + SF\left(\varphi^*_{\psi_{max}} - 1\right) - \varphi^*\right)^2 \qquad (2)$$

For the second generalized relationship, $\eta^* = F(\psi^*/\varphi^*)$, it was chosen to adopt the model proposed by Howell and Bonham [21] which obtained the generalized stage efficiency curve:

$$\eta^* = 1 - \frac{1 - \eta^*_{(\psi^*/\varphi^*)_{min}}}{[1 - (\psi^*/\varphi^*)_{min}]^{3.5}}(1 - \psi^*/\varphi^*)^{3.5}, \qquad (3)$$

$$\psi^*/\varphi^* \in [(\psi^*/\varphi^*)_{min}, 1]$$

$$\eta^* = 1 - \frac{1 - \eta^*_{(\psi^*/\varphi^*)_{max}}}{[(\psi^*/\varphi^*)_{max} - 1]^2}(\psi^*/\varphi^* - 1)^2, \qquad (4)$$

$$\psi^*/\varphi^* \in [1, (\psi^*/\varphi^*)_{max}]$$

For both relationships presented so far, $\psi^* = F(\varphi^*, SF)$ and $\eta^* = F(\psi^*/\varphi^*)$, it has been experimentally verified [22] that for *k* values between 1.1 and 1.4 they do not change significantly, and it can therefore be assumed that they do not depend on the gas composition.

2.1.2 Combustion chamber

To evaluate the thermodynamic variables affected by combustion processes within the combustion chamber, the fundamental chemical reactions of fossil fuels combustion have been simulated in Aspen HYSYS™ environment:

$$C + O_2 \rightarrow CO_2$$

$$H_2 + \frac{1}{2} O_2 \rightarrow H_2O$$

$$S + O_2 \rightarrow SO_2$$

$$N_2 + O_2 \rightarrow 2NO$$

Pure methane combustion reaction:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

More in detail, starting from the design mass flows and the chemical composition of the fuel chosen by the user, Aspen HYSYS™ is able to simulate the combustion processes that occur during the machine operation and to return within the Matlab™ code the main thermodynamic variables associated with the gas mixture that will expand in turbine.

2.1.3 Turbine

With regard to the behavior of the turbine, instead, the performance has been evaluated by considering the machine in chocking conditions. Under this hypothesis, the mass flow function (*MFF*) then assumes a constant value at the turbine inlet, which can be defined by the following relationship [23]:

$$MFF = \frac{\dot{m}_{T,in}\sqrt{T_{T,in}}}{p_{T,in}} \sqrt{\frac{R}{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}} \qquad (5)$$

2.1.3 Turbine cooling system

In order to evaluate the air flow purged by the compressor for the turbine cooling system, it was assumed a constant value mass flow function (MFF_{des}) defined by the design point of the machine. In particular, the air mass flow extracted along the compressor stages ($\dot{m}_{air,ex}$) can be defined as:

$$\dot{m}_{air,ex} = \frac{MFF_{des}}{\frac{T_{air,ex}}{p_{air,ex}} \sqrt{\frac{R}{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}}} \qquad (6)$$

To take into account losses in the cooling system, the value of the air flow actually used for the blade cooling ($\dot{m}_{air,cool}$) was finally obtained by introducing a corrective coefficient $Z_{air} \in [0, 1]$. It is therefore possible to isolate the mass flow of extracted air ($\dot{m}_{air,lost}$) that is lost throughout the machine before reaching the turbine blades.

$$\dot{m}_{air,cool} = Z_{air} \cdot \dot{m}_{air,ex} \qquad (7)$$

$$\dot{m}_{air,lost} = (1 - Z_{air}) \cdot \dot{m}_{air,ex} \qquad (8)$$

2.2 Validation procedure

The calculation code has been designed to simulate the behavior of a medium-sized commercial gas turbine (see Fig. 2) with the same properties of the one analyzed by the authors in a previous study aimed at the simulation of wet compression in natural gas-fired gas turbines [18]. In particular, the chosen machine is developed for 60 Hz applications and consists of a seventeen-stage axial compressor and a three-stage axial turbine, with a power output of about 86 MW and an electrical efficiency of about 32.2%. The validation of the Matlab^{TM}/Aspen HYSYS^{TM} code on this machine was carried out throughout a comparison with a THERMOFLEX^{TM} model of the plant, considering the same environmental conditions adopted for the validation of the previous IN.FO. G.T.E. software [18].

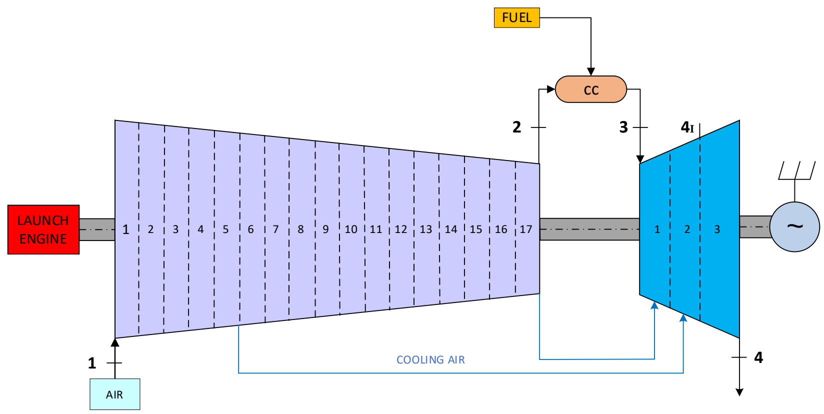

FIGURE 2: SCHEMATIC OF THE INDUSTRIAL GAS TURBINE USED IN THE STUDY FOR THE VALIDATION PROCEDURE

2.3 Main hypotheses for the preliminary analysis

In order to perform the calculations with a modified fuel composition and then evaluate the response of the considered commercial gas turbine to the co-combustion of CH_4-H_2 in variable percentage without any changes to the machine geometry, the preliminary analyses presented in this study were carried out starting from the introduction of the following two hypotheses:

1. the consistency of the fuel thermal power ($\dot{Q}_f = cost$), since it has been hypothesized that the machine operates the same with the variation of the fuel composition;
2. the consistency of the equivalence ratio ($\Phi_p = cost$) in the primary zone of the combustion chamber, defined as the ratio of the stoichiometric air/fuel ratio to the actual air/fuel ratio.

For both the introduced hypotheses, the reference case study corresponds to the use of a 100% CH_4 fuel and of a gas turbine operating under ISO conditions (T_{amb} = 15 °C, P_{amb} = 1.013 bar, RH_{amb} = 60%) with constant cooling mass flow rates equal to the design values.

With this hypothesis, it is possible to calculate the mass flows of air and fuel in the new conditions ($\dot{m}_{air,new}$ and $\dot{m}_{f,new}$) which are then transferred to specific sub-models for the new performance evaluation of the machine.

These hypotheses allow therefore to fix the design architecture of the combustor chamber but at the same time to analyze the general behavior of the machine in response to hydrogen injection. In this way it is therefore possible to evaluate the opportunity to operate with moderate percentages of hydrogen by maintaining the current architectural characteristics of the machine.

A summary table of the main input parameters provided to the calculation code and related to the environmental conditions and the selected commercial machine is shown below (Table 1).

TABLE 1: INDUSTRIAL HEAVY DUTY GAS TURBINE MAIN DATA, USED AS INPUT PARAMETERS FOR THE SIMULATION PROCEDURE.

Ambient conditions		
Ambient temperature	[°C]	15
Ambient pressure	[bar]	1.013
Relative humidity	[%]	60
Gas turbine parameters		
Number of compressor stages	[#]	17
Number of total stages (Compressor + Turbine)	[#]	20
Rotational speed	[RPM]	3600
Design power output	[MW]	86
Design electrical efficiency	[-]	0.322
Design pressure ratio	[-]	12.67
TIT (Temperature Inlet Turbine)	[°C]	1113
Design air flow rate (ISO)	[kg/s]	297.2
Design Mass Flow Function (MFF)	[m2]	0.175
Ref temperature	[°C]	25
Fuel temperature	[°C]	25
Combustion chamber (CC) efficiency	[-]	0.995
Pressure drop in CC	[-]	0.04
Mechanical efficiency	[-]	0.99
Alternator efficiency	[-]	0.97
Auxiliary efficiency	[-]	0.97
Pressure drops along the suction pipe (air filter + exhaust)	[-]	0.00
Pressure drops along the exhaust system	[-]	0.00

3. RESULTS AND DISCUSSION

In this section, the results of the carried-out analyses are presented. Firstly, the validation process applied to the developed numerical model is shown. Then, the preliminary results regarding the injection of hydrogen in the combustion chamber of the selected commercial machine are presented.

3.1 Validation results

In this section, the validation results of the calculation routine applied to an industrial HD gas turbine are presented by means of a comparison with the Thermoflex software results. In Fig. 3 to Fig. 6 the performances of the analyzed machine as a function of the ambient temperature are shown for both the calculation models. A summary table of the deviation values between the predictions of the numerical code and the values obtained by means of the Thermoflex software is shown below (Table 2).

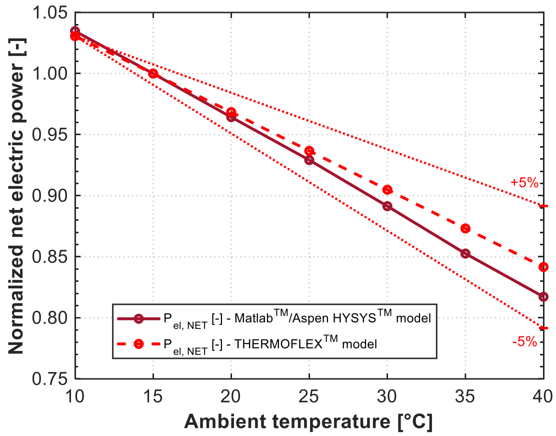

FIGURE 3: NORMALIZED NET ELECTRIC POWER AS A FUCNTION OF THE AMBIENT TEMPERATURE

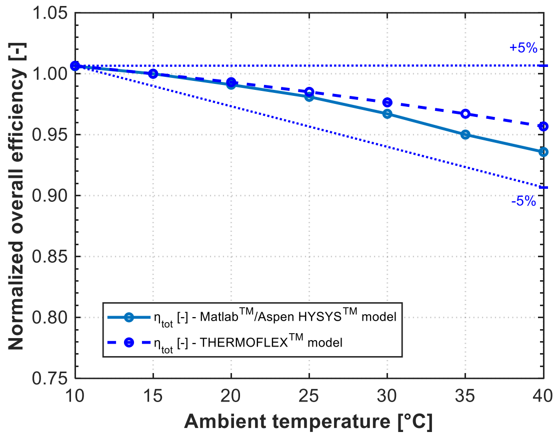

FIGURE 4: NORMALIZED OVERALL EFFICIENCY AS A FUCNTION OF THE AMBIENT TEMPERATURE.

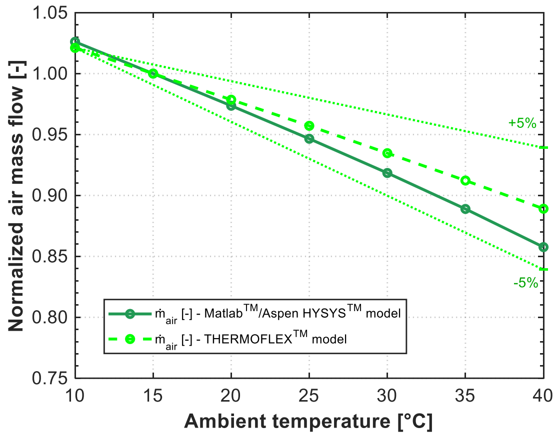

FIGURE 5: NORMALIZED AIR MASS FLOW AS A FUCNTION OF THE AMBIENT TEMPERATURE.

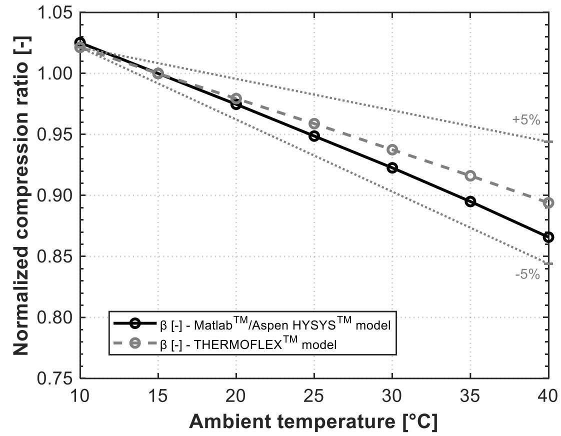

FIGURE 6: NORMALIZED AIR COMPRESSION RATIO AS A FUCNTION OF THE AMBIENT TEMPERATURE.

TABLE 2: DEVIATION (A) AND AVERAGE DEVIATION (B) BETWEEN THE PREDICTIONS OF THE NUMERICAL MODEL AND THE THERMOFLEX RESULTS.

(A)

$\varepsilon P_{el,NET}$ [%]	$\varepsilon \eta_{tot}$ [%]	εm_{air} [%]	$\varepsilon \beta$ [%]
0.410	0.037	0.499	0.390
0.000	0.000	0.000	0.000
0.411	0.208	0.495	0.469
0.751	0.405	1.061	1.016
1.346	0.936	1.626	1.485
2.058	1.708	2.331	2.111
2.443	2.098	3.151	2.816

(B)

$\tilde{\varepsilon} P_{el,NET}$ [%]	$\tilde{\varepsilon} \eta_{tot}$ [%]	$\tilde{\varepsilon} m_{air}$ [%]	$\tilde{\varepsilon} \beta$ [%]
1.060	0.770	1.309	1.184

3.2 Numerical model application results

In this section, the preliminary results of the co-combustion of CH_4-H_2, in variable percentage, for the considered gas turbine are presented. Fig. 7 shows the main combustion parameters as the percentage of hydrogen in the fuel blend varies, while in Fig. 8 to Fig. 13 the trends of the main operating parameters and the performances of the considered machine as a function of the hydrogen content are shown. In more detail, due to the hypothesis of the model, the air and fuel mass flows decrease with the increase in the hydrogen content. Indeed, due to the consistency of the thermal power, the fuel mass flow tends to decrease with respect to the case of pure methane since hydrogen shows a higher LHV than the methane and then the mass flow has to decrease to keep the relationship constant ($\dot{Q}_f = \dot{m}_{CH4} LHV_{CH4} = \dot{m}_f LHV_f$). The same explanation can be applied to the decrease in the air mass flow with the increase in the hydrogen content but considering the second hypothesis of the model (consistency of the equivalence ratio).

It can be noted that, under ISO operating conditions, the increase in the hydrogen content corresponds to a decrease in the net electrical power produced by the machine and its total efficiency, about 3% (Fig. 8), up to the point of the compressor stall, which occurs at about 62% of hydrogen by volume in the fuel mixture. The air mass flow processed by the machine is also reduced by more than 5% (Fig. 8) while the compression ratio rises from 12.67 to

14.2 (Fig. 9). In Fig. 10 and Fig. 11 it can be seen the effect of the co-combustion of CH_4-H_2 on the operating temperatures of the cycle, that, for the maximum content of hydrogen shown, can reach, respectively at the turbine inlet and outlet, values of about 7% and 10% more than the design case for the considered machine. Finally, the reduction effect on CO_2 formation as a consequence of the H_2 injection into the combustion chamber is shown in Fig. 12. It can be reduced by more than 30% compared to the case of pure methane

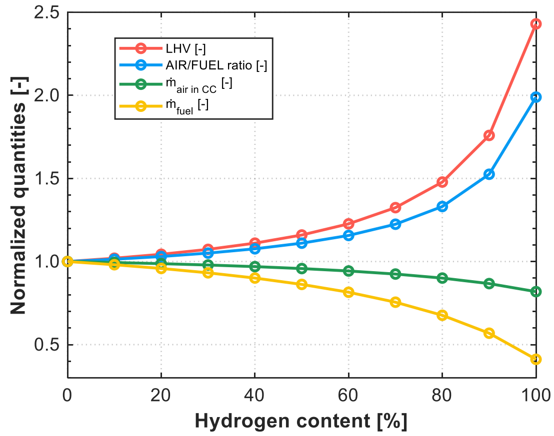

FIGURE 7: LOWER HEATING VALUE, AIR/FUEL RATIO, AIR AND FUEL MASS FLOW AS A FUCNTION OF THE HYDROGEN CONTENT BY VOLUME.

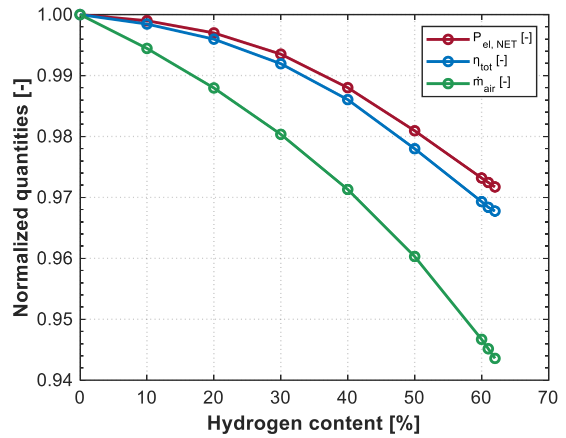

FIGURE 8: NET ELECTRIC POWER, OVERALL EFFICIENCY AND AIR MASS FLOW AS A FUCNTION OF THE HYDROGEN CONTENT BY VOLUME.

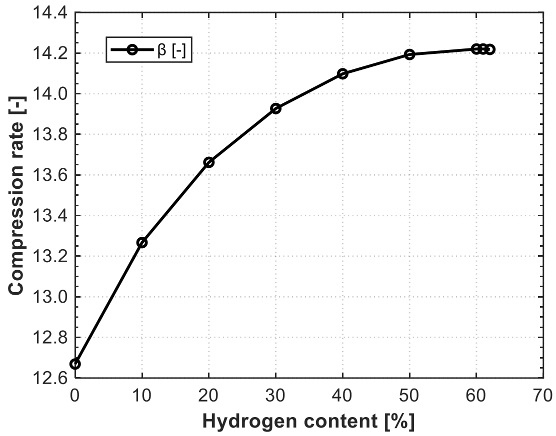

FIGURE 9: COMPRESSION RATIO AS A FUCNTION OF THE HYDROGEN CONTENT BY VOLUME.

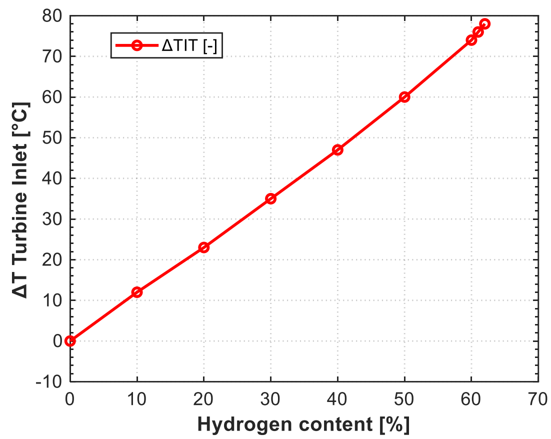

FIGURE 10: $\Delta T = TIT - TIT_{des}$ AS A FUNCTION OF THE HYDROGEN CONTENT BY VOLUME.

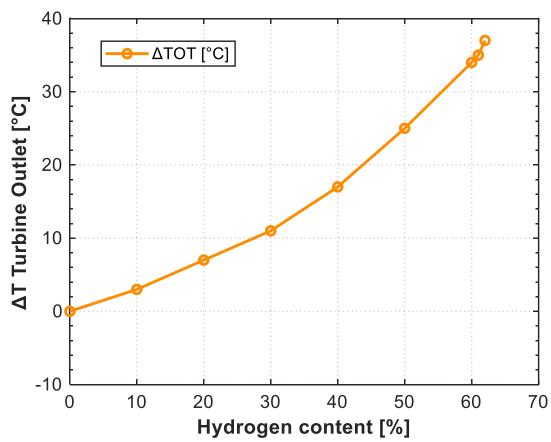

FIGURE 11: $\Delta T = TOT - TOT_{des}$ AS A FUCNTION OF THE HYDROGEN CONTENT BY VOLUME.

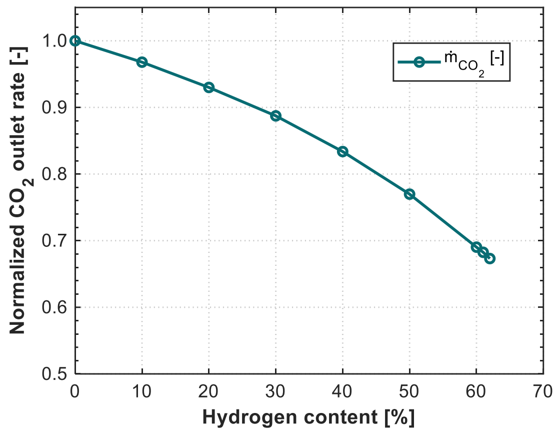

FIGURE 12: NORMALIZED CO_2 MASS FLOW AT THE EXHAUST AS A FUNCTION OF THE HYDROGEN CONTENT BY VOLUME.

In support of the above considerations, Table 3 summarizes the main output parameters of the simulations performed by varying the hydrogen content in the fuel mixture with reference to the considered heavy duty gas turbine.

To underline the preliminary results obtained from the carried-out simulations, the main consequences of hydrogen injection in commercial GTs, for percentages between 10% and 30% in the CH4-H2 blend (volume based), are summarized below:

- by increasing the hydrogen rate by 10%:
  - the net electric power and the overall efficiency decrease by 0.1%;
  - the air mass flow at the compressor inlet decreases by 0.5 %;
  - the compression ratio increases by 0.6%;
  - TIT and TOT rise respectively by 11 °C and 3 °C;
  - the CO_2 emissions decreases by less than 4%;

- by increasing the hydrogen rate by 20%:
  - the net electric power and the overall efficiency decrease by less than 0.5%;
  - the air mass flow at the compressor inlet decreases by 1.2 %;
  - the compression ratio increases by about 1%;
  - TIT and TOT rise respectively by 23 °C and 7 °C;
  - the CO_2 rate at the exhaust decreases by 7%;

- by increasing the hydrogen rate by 30%:
  - the net electric power and the overall efficiency decrease by respectively 0.7% and 0.8%;
  - the air mass flow at the compressor inlet decreases by 2%;
  - the compression ratio increases by about 10%;
  - TIT and TOT rise respectively by 34 °C and 12 °C;
  - the CO_2 rate at the exhaust decreases by more than 11%.

TABLE 3: MAIN OUTPUT PARAMETERS OF THE SIMULATION PROCEDURE FOR THE CONSIDERED INDUSTRIAL HEAVY DUTY GAS TURBINE.

H_2 [%VOL]	0	10	20	30	40	50	60	61	62
LHV_{fuel} [MJ/kg]	50.000	50.985	52.178	53.654	55.527	57.982	61.339	61.744	62.165
Air/Fuel ratio [-]	52.3	52.9	53.8	54.8	56.2	57.9	60.3	60.6	60.9
$\dot{m}_{airCC}$ [kg/s]	278.28	276.63	274.70	272.44	269.75	266.48	262.44	261.98	261.51

$\dot{m}_{fuel}$ [kg/s]	5.32	5.22	5.10	4.97	4.80	4.60	4.35	4.33	4.30
$P_{el,NET}$ [MW]	85.798	85.711	85.539	85.240	84.768	84.162	83.497	83.434	83.371
η_{tot} [-]	0.3225	0.3220	0.3212	0.3199	0.3180	0.3154	0.3126	0.3123	0.3121
$\dot{m}_{air}$ [kg/s]	297.20	295.55	293.62	291.36	288.67	285.40	281.36	280.90	280.43
β [-]	12.67	13.27	13.66	13.93	14.10	14.19	14.22	14.22	14.22
TIT [°C]	1113	1124	1136	1147	1159	1172	1187	1189	1190
T_{4I} [°C]	1075	1086	1098	1109	1121	1134	1148	1150	1151
TOT [°C]	534	537	541	546	552	560	569	570	571
$\dot{m}_{CO2}$ [kg/s]	14.72	14.25	13.69	13.07	12.28	11.34	10.16	10.05	9.91

CONCLUSIONS

This work aims at proposing a numerical model, developed by the Authors by means of the link between Matlab™ environment and Aspen HYSYS™ environment, able to evaluate the performance of commercial gas turbines for different operating strategies based on the co-combustion of CH_4 and H_2, varying the composition of the fuel blend for machines with a set geometry.

From the preliminary analysis, the increase in the hydrogen content of more than 60% by volume in the fuel mixture corresponds to a decrease in the net electrical power produced by the machine and its total efficiency of about 3%, as well as the air mass flow processed by the machine which is reduced by more than 5%, while the compression ratio rises from 12.67 to about 14.2. In addition, the effect of co-combustion CH_4-H_2 on the operating temperatures of the cycle leads to a maximum increase in values, at the inlet and outlet of the turbine, respectively 7% and 10% more than in the case of design for the machine considered with a maximum reduction in CO_2 emissions of more than 30%.

By increasing the percentages of hydrogen in the mixture between 10% and 30%, instead, the performance of the machine decreases almost negligibly (net electrical power and efficiency are reduced by less than 0.8%), while temperatures at the turbine inlet and outlet rise by 34 ºC and 12 ºC, respectively (corresponding to 3% and 2.2% respectively). Finally, there is a reduction in CO_2 emissions of about 11% in the best case.

This study can therefore be intended as the preliminary phase of a more in-depth analysis aimed at identifying optimized management strategies for hydrogen injection in commercial GTs as a short-term strategy to reduce carbon emissions. The future steps could be represented by the implementation of a more detailed combustion model in Aspen HYSYS™ environment, with a focus on pollutant emissions, in particular NO_x and CO_2, and related strategies to reduce them.

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