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Preliminary zero-dimensional thermodynamic analysis of a gas turbine using EBSILON Professional software

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Abstract

The application of zero-dimensional mathematical modelling offers significant advantages in the domain of energy systems. Firstly, it serves as a crucial support tool for the design process. Secondly, it facilitates the optimisation of operating parameters, thereby enhancing the efficiency of energy systems. Furthermore, it offers, depending on the accuracy of a model, a comprehensive analysis of the impact of various factors on the performance of a specific system. The present article focuses on a model of a gas turbine unit that has been developed using the EBSILON Professional software. The model was employed to assess the operating parameters that are characteristic of gas turbines, including the turbine inlet temperature and compression ratio, that are crucial for the efficiency and unit power of the abovementioned power system. Moreover, this approach enables the determination of the optimal operating points for the considered assumptions. Next, for the selected case, an analysis was conducted to determine the impact of changes in atmospheric air parameters on the operation of the gas turbine, applying two different approaches. Research has demonstrated that daily, and particularly seasonal, fluctuations in temperature can exert a substantial influence on the efficiency and capacity of the system, while fluctuations in pressure and relative humidity have a much smaller impact on turbine operation. Finally, the simplified model was utilised to ascertain the effect of incorporating hydrogen into fuel on the operational parameters of a gas turbine, indicating the effect of both quantitative and qualitative changes of stream in the combustion chamber and turbine.

Keywords: Gas turbine; Thermodynamic analysis; EBSILON Professional; Zero-dimensional analysis; Numerical modelling

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1. Introduction

The ongoing development of natural gas-based energy is driven by several factors [1]. These include the need to diversify generation sources, reduce direct industrial emissions, and reorganise the geopolitical landscape of existing technology parks [2]. Consequently, gas installations are implemented to generate usable forms of energy, such as electricity and heat. These systems are designed for the high-efficiency conversion of chemical energy from fuels and energy carriers in a gaseous state into mechanical or electrical energy, on a given scale and with a given degree of system complexity (including combined cycles [3−6]). These primarily include gas turbines, which are energy systems that enable the implementation of a real cycle similar to the Joule-Brayton cycle [7,8].

The simplified gas turbine unit consists of an inlet, compressor, combustion chamber, gas turbine and an exhaust system [4,9]. Its thermal efficiency is not far from that of other energy systems, such as piston engines and the Rankine steam cycle. However, they also have several advantageous features, including ecological benefits (the combustion of natural gas results in lower emissions of carbon dioxide and sulphur oxides per unit

Nomenclature

h – specific enthalpy, kJ/kg
LCV – low calorific value, MJ/kg
$\dot{m}$ – mass flow, kg/s
p – pressure, Pa
P – power, kW
TIT – turbine inlet temperature, °C
t – temperature, °C

Greek symbols

ε – relative pressure drop, %
η – efficiency, %
κ – heat capacity ratio
π – compression ratio
φ – air relative humidity, %

Subscripts and Superscripts

a – air
C – compressor
el – electrical
f – fuel
g – flue gas
G – electric generator
GT – gas turbine (turbogenerator)
m – mechanical
n – polytropic
s – isentropic
T – turbine
u – per unit

Abbreviations and Acronyms

CCGT – combined cycle gas turbines
LCV – low calorific value
TIT – turbine inlet temperature
UAV – unmanned aerial vehicle
UPS – uninterruptible power supplies

of chemical energy introduced into the combustion chamber), operational benefits (quick start-up), logistical benefits (transportation, fuel availability, and extraction costs), and technological benefits (the ability to create cascade installations, such as combined cycle gas turbines (CCGT), with efficiencies above 60%, as well as high development potential). Consequently, an increasing number of countries worldwide, including Poland (as evidenced by the PEP2040 document [2]), have chosen to prioritise research and implementation projects focusing on technological resources and fuel reserves in favour of gas carriers based on methane and other hydrocarbons.

Additionally, there is a growing interest in micro gas turbines, defined as systems dedicated to small stationary installations with a capacity of several hundred kW. These systems are most often coupled with electric generators, which are used in uninterruptible power supplies (UPS) and so-called SKI-Dach systems. Micro gas turbines also include mobile units, such as military vehicles, including the M1 Abrams tank family, which is equipped with Avco-Lycoming AGT1500 turboshaft multi-fuel engines. Other mobile units include guided missiles and unmanned aerial vehicles (UAVs), which burn aviation kerosene-type fuel [10].

Hydrogen, regarded as the fuel of the future, possesses technological potential related to combustion in gas turbines (both stationary and mobile, including aircraft). Dedicated combustion chambers, turbines and exhaust systems, along with solutions that allow for satisfactory temperatures (strength and thermal issues of expansion devices) and composition (environmental issues) of exhaust gases, have led to its current consideration as a component of the machinery park of many countries in the second half of the 21st century, similar to fuel cells. However, the technical implementation of hydrogen comes with challenges related to production, storage and use, including in gas turbines.

The research domain of every development process is becoming increasingly dependent on the availability and quality of computer-aided tools. The second of the aforementioned features is increasingly identified not only with the distinctive convergence of results generated by a given program or solver, but also with their satisfactory level in terms of the time needed to perform them and the computer resources used. Consequently, a wide array of models, ranging from complex 2D and 3D numerical ones to their one-dimensional or dimensionless counterparts, have become integral to research endeavours. These models provide a foundation for decision-making processes and the utilisation of advanced tools. Additionally, they play a crucial role in the education and training of engineering and technical personnel, as well as in the construction of computer models for parametric diagnostics.

In recent years, as a result of the aforementioned progressive development and popularisation of various classes of computer-aided design tools and ongoing control of energy systems, there have been many numerical analyses of gas systems and power machines and devices that consider gas-fuel-based systems, implementing real cycles identified with gas-steam units [3] or aviation turbine power units [9,11,12]. They have been part of various research and development initiatives, focusing on a selected group of physical quantities and issues related to the ongoing operation of technical systems. This confirms their important role in the broadly understood decision-making system. A review of the extant literature reveals the existence of commercial tools that facilitate the aforementioned identification work in both stationary and aviation contexts. These include programs such as EngineSim (manufacturer or distributor – NASA Glenn Research Center) [13], ASTRA (Samara National Research University) [14], DVIGwT (Ufa State Aviation Technical University) [14], GasTurb (J. Kurzke, MTU) [15], GSP (Royal Netherlands Aerospace Centre) [16], Aspen HYSYS [6,17], and Uni_TTF (A. Lyulka Experimental Design Bureau) [14].

Nevertheless, despite its undeniable and steadily growing popularity, it remains advisable to repeat previously published numerical studies and verify the theses put forward with their help [6−9,11,12,18−22]. This is both because of the multi-threaded but often limited nature of the objects described by them in the course of computer tests and because of the simplifications used in many programs. For instance, existing literature continues to employ estimations that disregard the moisture content of the ambient air introduced into the flow machine during the estimation process. Additionally, these analyses often underestimate the influence of climatic characteristics on the resulting operating parameters, treating them as negligible factors.

One of the software programs currently available that addresses the aforementioned issues is EBSILON Professional. It has been identified by numerous independent researchers as an interesting and practical tool for simulating and evaluating the operation of power systems, including those with a high degree of technological complexity and non-linear operating characteristics of the power machines and equipment included in it [23,24]. The software is characterised by its graphical interface, which has been in development since the early 1990s. It is equipped with a range of function codes dedicated to simulations of various systems, including gas and gas-steam units [25]. The model under consideration is 'based on a structured model of connections and time-operational dependencies. This provides a realistic basis for identifying a number of optimisation and research directions for the modelled gas turbine. In addition, it offers insight into the specifics of the energy conversion chain and its accompanying technical implementations.

In this article, the EBSILON Professional program was implemented to conduct preliminary analyses of the selection of operating parameters (combustion chamber outlet temperature, compression) for a model stationary gas turbine. The impact of environmental conditions (ambient temperature and pressure) on the indicators obtained by the associated energy system was also investigated, including cycle efficiency, gross unit power, and the air flow necessary to feed the turbine compressor. Additionally, the impact of enriching conventional fuel (methane) with hydrogen is presented using a previously created parameterised gas turbine model. The results obtained were subsequently subjected to a comprehensive evaluation, encompassing both comments and an assessment of their applicability in terms of the simplifications adopted in the software.

2. Mathematical model of the gas turbine

As illustrated in Fig. 1, the considered gas turbine unit consists of a compressor, a combustion chamber, a turbine and an electric generator. The flows of individual gases: air, fuel and flue gas, are marked with separate colours, while the numbers indicate characteristic points in the system where thermodynamic parameters are determined. The computational model of the system for its individual components is presented in the following subsections.

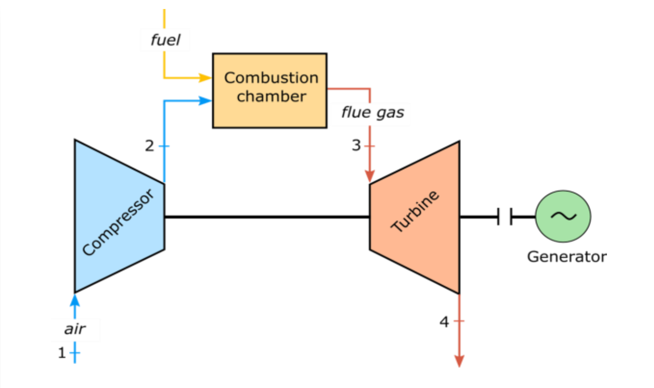

Fig. 1. Scheme of the modelled gas turbine.

The EBSILON Professional software facilitates the execution of numerical simulations of a variety of predefined thermodynamic processes and cycles. In addition, it enables the construction of user-defined models of energy systems, which can be employed for the purpose of thermodynamic analysis. In the latter case, the development of a model principally consists of the appropriate selection and combination of components available in the program and the determination of preliminary parameters and assumptions for the model. The calculations are based on the principle of mass and energy conservation, among other considerations, and are performed iteratively. These calculations are informed by the relationships that result from the thermochemical parameters of fluids. During the computational process, the program notifies the user of its convergence status. The mass and energy balance equations employed, along with the assumptions underlying the individual elements of the system, will be presented below.

The modelled gas turbine comprises four fundamental functional elements available in the program, which were combined into a single system, as illustrated in Fig. 2.

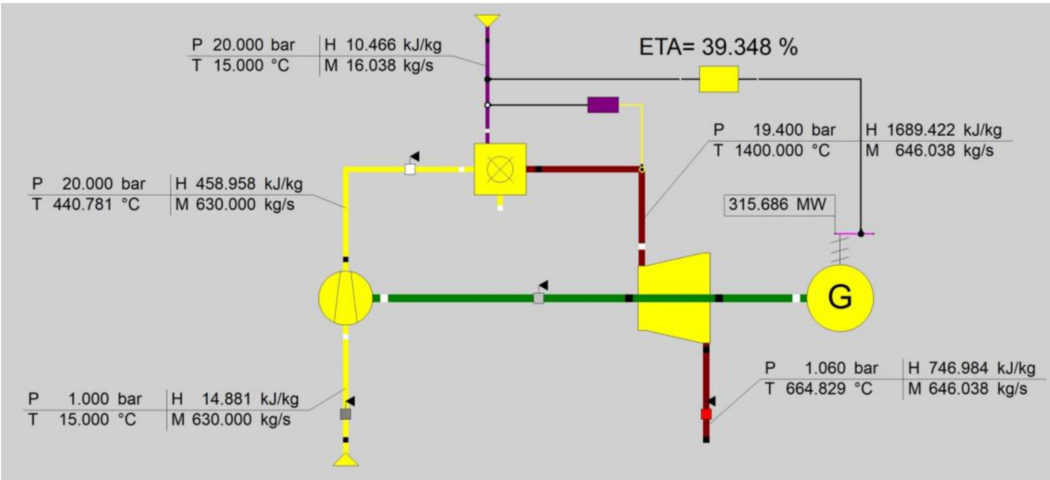

Fig. 2. Scheme of the gas turbine model implemented in the EBSILON Professional program with operational parameters.

Each component was assigned specific input parameters, which constituted the boundary conditions for the entire model. In the case of air parameters at the compressor inlet, standardised parameters for gas turbines according to ISO-2314 were utilised. The remaining parameters were assumed, and their values were mostly based on the actual parameters of similar facilities. Pure methane, the primary component of methane-rich natural gas (type E according to the PN-C-04750:2011 standard), was presumed to be the fuel utilised during the combustion process within the gas turbine. Furthermore, in order to execute a more extensive evaluation of the gas turbine's performance under diverse operating conditions, the predetermined turbine inlet temperature (TIT) was achieved through the automatic adjustment of the air or the fuel mass flow rate supplied to the combustion chamber alternatively. The program employs a dedicated control component for this purpose (forcing the implementation of the set change function). A comparative analysis was conducted on the results obtained from the implementation of both methodologies.

The following subsections contain a detailed description of the equations and the input values implemented in the program to solve the entire analysed system. The mathematical model is based on the balance equations of individual machines and devices forming the gas turbine.

2.1. Air compressor

The initial component of the modelled gas turbine is the compressor. According to the calculation code employed in the program, the variables necessary to determine the specifics of the process taking place in the compressor are considered to be:

1) air inlet (static) temperature t_1,
2) air inlet (static) pressure p_1,
3) air inlet relative humidity φ,
4) air outlet (static) pressure p_2,
5) polytrophic efficiency of the compressor $\eta_{n,C}$,
6) mechanical efficiency of the compressor η_m,
7) air mass flow $\dot{m}_a$.

Assuming no bleed flows and a leak-tight compressor, the calculations presume a constant air mass flow throughout its entirety, as depicted by (control cross-sections are delineated in Fig. 1):

$$\dot{m}_a = \dot{m}_1 = \dot{m}_2. \tag{1}$$

In addition, under the assumption that compression occurs without heat exchange with the environment (adiabatically) and that the change in kinetic energy of the flow is negligible within the scope of the analyses conducted, the energy balance of the compressor is calculated using the following equation:

$$\dot{m}_a\, h_1 + P_c = \dot{m}_a\, h_2, \tag{2}$$

where P_c denotes the power supplied to the working fluid in the compressor, and $h_i = f(t_i, p_i)$ denotes the specific enthalpy of the stream determined in accordance with the fluid properties libraries adopted in the program.

The procedure for determining the parameters behind the compressor (in point 2) necessitates the calculation of the isentropic compression efficiency. In the case of the analysed program, this efficiency is calculated using the following formula:

$$\eta_{s,C} = \frac{\pi^{\frac{\kappa_a - 1}{\kappa_a}} - 1}{\pi^{\frac{\kappa_a - 1}{\kappa_a\, \eta_{n,C}}} - 1}. \tag{3}$$

In order to determine it, it is therefore necessary to assume a directly correlated polytropic efficiency of compression $\eta_{n,C}$, specific heat ratio for air κ_a, and to identify the static compression ratio resulting directly from the assumed values of static pressure before and behind the compressor:

$$\pi = \frac{p_2}{p_1}. \tag{4}$$

Accordingly, the specific enthalpy of the air behind the compressor is determined by the following formula:

$$h_2 = h_1 - \Delta h_{12}, \tag{5}$$

where Δh_{12} is the actual increase in specific enthalpy, taking into account isentropic efficiency, determined from the following relationship:

$$\Delta h_{12} = \frac{h_{2s} - h_1}{\eta_{s,C}}, \tag{6}$$

where $h_{2s} = f(s_1, p_2)$ is the enthalpy after the compressor during isentropic compression. Knowing the specific enthalpy for the non-isentropic case, it is possible to determine the air temperature behind the compressor $t_2 = f(h_2, p_2)$, and the power supplied to the air flow in the compressor (Eq. 2). After taking into account the mechanical efficiency of the machine, the value of the mechanical power required to drive the compressor is obtained with:

$$P_{m,C} = \frac{P_C}{\eta_m}. \tag{7}$$

2.2. Combustion chamber

The second component of a gas turbine system, situated between the compressor and the turbine, forming a single rotor assembly, is the combustion chamber. This device facilitates the controlled oxidation of the supplied fuel in a compressed air atmosphere. The following assumptions were made in order to conduct this analysis:

- Combustion is considered to be complete;
- The entire air flow from the compressor goes to the combustion chamber (there are no air leaks between the two components of the system);
- The fuel is pure methane, and its lower calorific value (LCV), specified in the program's physico-chemical tables, is 50.015 MJ/kg.

In the calculation of the combustion chamber balance, it is essential to ascertain values of the following quantities:

1) fuel mass flow $\dot{m}_f$,
2) fuel pressure – assumed to be: $p_f = p_2$,
3) relative pressure drop in the combustion chamber ε.

In accordance with the principle of energy conservation, the mass flow of exhaust gases leaving the combustion chamber is determined using the following formula:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f. \quad (8)$$

Given an assumed fuel flow rate, the gas parameters behind the combustion chamber can be determined from the energy balance equation:

$$\dot{m}_a\, h_2 + \dot{m}_f\, (h_f + LCV) = \dot{m}_g\, h_3, \quad (9)$$

where $h_f = f(t_f, p_f)$ is the specific physical enthalpy of the fuel.

Knowing the enthalpy of the exhaust gas determined from Eq. (9), its temperature $t_3 = f(p_3, h_3)$ can be calculated. A pressure drop in the combustion chamber is assumed according to the formula:

$$p_3 = p_2\, (1 - \varepsilon). \quad (10)$$

The temperature of the exhaust gases behind the combustion chamber is described in the literature as the turbine inlet temperature (TIT), and will be presented in Section 3 using this acronym. This is a parameter of significant importance, as it directly influences the operational dynamics of a gas turbine and, by extension, the constraints imposed on materials and designs.

2.3. Gas turbine

The next component is the turbine (expander), which is the machine responsible for driving both the compressor and the generator by extracting enthalpy from the stream and converting it into effective work. In addition to the values calculated for the previous components, the model of the turbine requires determining the following quantities:

1) p_4 – outlet pressure,
2) $\dot{m}_g$ – flue gas mass flow,
3) $\eta_{s,T}$ – isentropic efficiency of the turbine,
4) η_m – mechanical efficiency of the turbine.

According to the principle of conservation of mass, the flow rate through the turbine remains constant:

$$\dot{m}_g = const. \quad (11)$$

Assuming adiabatic expansion, the energy balance of the turbine itself is as follows:

$$\dot{m}_g\, h_3 = \dot{m}_g\, h_4 + P_T, \quad (12)$$

where P_T is the theoretical (effective) power of the turbine, and h_4 is the specific enthalpy of the exhaust gas, given by the formula:

$$h_4 = h_3 - \Delta h_{34}, \quad (13)$$

where Δh_{34} is the actual enthalpy drop, taking into account isentropic efficiency, determined from the formula:

$$\Delta h_{34} = \frac{h_3 - h_{4s}}{\eta_{s,T}}, \quad (14)$$

where $h_{4s} = f(s_3, p_4)$ is the enthalpy behind the turbine, which assumes isentropic expansion. Knowing the actual specific enthalpy, similarly to the case of a compressor, it is possible to determine the temperature of the stream (exhaust gases) behind the turbine $t_4 = f(h_4, p_4)$ and the theoretical power of this machine from the energy balance, Eq. (12). After taking into account the mechanical efficiency of the turbine, the value of the mechanical power output to the shaft and directed to the compressor is obtained as:

$$P_{m,T} = P_T\, \eta_m. \quad (15)$$

2.4. Electric generator

In a gas turbine, the mechanical power obtained during expansion is utilised for two primary functions: to drive the compressor and to power the electric generator. Therefore, given the specified generator efficiency η_G, the electrical power is calculated using the following formula:

$$P_{el} = P_{m,GT}\, \eta_G, \quad (16)$$

where P_{mGT} is the net mechanical power of the gas turbine unit given by the formula:

$$P_{m,GT} = P_{m,T} - P_{m,C}. \quad (17)$$

The total efficiency of the gas turbine can ultimately be expressed as:

$$\eta_{GT} = \frac{P_{el}}{LCV \cdot \dot{m}_f}. \quad (18)$$

2.5. Model validation

In order to verify the mathematical model of the gas turbine, it is necessary to refer to real data. In this regard, actual data from a GE 9HA.01 turbine, referring to the ISO-2314 standard [26], were implemented. The input data for the model are presented in Table 1.

Table 1. Operational parameters of the gas turbine 9HA.01 representing input variables for the model validation.

Parameter	Symbol	Unit	Value
Temperature of the air at the inlet to the compressor	t_1	°C	15
Pressure of the air at the inlet to the compressor	p_1	bar	1.014
Relative humidity ratio	φ	%	60
Pressure of the air at the outlet of the compressor	p_2	bar	23.81
Pressure at the outlet of the turbine	p_4	bar	1.027
Compressor polytropic efficiency	$\eta_{n,C}$	%	93.8
Turbine isentropic efficiency	$\eta_{s,T}$	%	90
Electric generator efficiency	η_G	%	98.9
Mechanical efficiency of the compressor and the turbine	η_m	%	99
Pressure drop in combustion chamber	Δp_{23}	bar	1.19
Temperature of the fuel	t_f	°C	25
Pressure of the fuel	p_f	bar	36.3
Fuel mass flow	$\dot{m}_f$	kg/s	20.9
Temperature of the gas at the outlet of turbine	t_4	°C	628.9
Air mass flow	$\dot{m}_a$	kg/s	832.7

In order to compare the data declared by the producer with the theoretical results of the model, the values of the achieved electrical power of the gas turbine and the turbine inlet temperature are compared in Table 2. The calculated results for the implemented actual parameters of the selected turbine are nearly identical to the data declared by the producer, with a relative error of approximately 0.1%. In consideration of the aforementioned factors, it can be concluded that, although certain assumptions and simplifications have been made, the model accurately reflects the actual process occurring within the machine.

Table 2. Comparison between the specifications of the gas turbine 9HA.01 and the model results.

Parameter	Symbol	Unit	Specifications	Model result
Electrical power	P_{el}	MW	446.0	445.6
Turbine inlet temperature	t_3	°C	1412.2	1410.3

3. Results and discussion

The mathematical model of a gas turbine presented in Section 2 was utilised to conduct a series of simulations of the operation of the aforementioned power system. The simulations analysed the impact of variables such as the compression ratio, exhaust gas temperature at the combustion chamber outlet, ambient pressure and relative humidity of the ambient air inlet to the compressor on the obtained operating parameters.

In the course of this study, the so-called base parameters (references) of the system were initially established, thereby serving as a reference point for further analysis and comparison. The values in question were derived from empirical data concerning actual turbines, including the 9HA.01 model from General Electric, and were also assumed on an intuitive basis. Subsequently, to execute fundamental simulations and ascertain the impact of selected variables, one or two parameters were modified while the values of the remaining parameters were maintained constant.

The input parameters at the base point are presented in Table 3, while the calculation results for the adopted parameters and assumptions of the modelled gas turbine are presented in Table 4.

Table 3. Input variables for the model reference point.

Parameter	Symbol	Unit	Value
Temperature of the air at the inlet to the compressor	t_1	°C	15
Pressure of the air at the inlet to the compressor	p_1	bar	1
Relative humidity ratio	φ	%	60
Turbine inlet temperature (TIT)	t_3	°C	1400
Pressure at the outlet of the turbine	p_4	bar	$1.06p_1$
Compressor polytropic efficiency	$\eta_{n,C}$	%	92
Turbine isentropic efficiency	$\eta_{s,T}$	%	90
Electric generator efficiency	η_G	%	98.6
Mechanical efficiency of the compressor and the turbine	η_m	%	99
Relative pressure drop in combustion chamber	ε	%	3
Temperature of the fuel	t_f	°C	15
Pressure of the fuel	p_f	bar	20
Fuel mass flow	$\dot{m}_f$	kg/s	16.038

Table 4. Calculated parameters for the model reference point.

Parameter	Symbol	Unit	Value
Specific enthalpy of the air at the inlet of the compressor	h_1	kJ/kg	14.9
Air mass flow	$\dot{m}_a$	kg/s	630
Specific enthalpy of the air at the outlet of the compressor	h_2	kJ/kg	459.0
Temperature of the air at the outlet of the compressor	t_2	°C	440.7
Flue gas mass flow	$\dot{m}_g$	kg/s	646
Pressure of the flue gas at the inlet of the turbine	p_3	bar	19.4
Specific enthalpy of the flue gas at the inlet of the turbine	h_3	kJ/kg	1689.4
Temperature at the outlet of the turbine	t_4	°C	664.8
Specific enthalpy at the outlet of the turbine	h_4	kJ/kg	747.0
Electrical power of the gas turbine	P_{el}	MW	315.7
Total efficiency of the gas turbine	η_{GT}	%	39.35

In the base model, the gas turbine attained an electrical power of 315.7 MW with an electrical efficiency of 39.35%, for a compression ratio of 20 and a turbine inlet temperature of 1400°C. The subsequent subsections delineate the impact of these final two parameters on the performance of the gas turbine.

3.1. The influence of compression ratio on TIT and efficiency

In order to examine the impact of compression ratio and turbine inlet temperature on the efficiency of a gas turbine, simulations were carried out for TIT values ranging from 1200°C to 1600°C with increments of 100°C, and for various compression ratios ranging from 10 to 80. The results of this study are presented in the graph shown in Fig. 3.

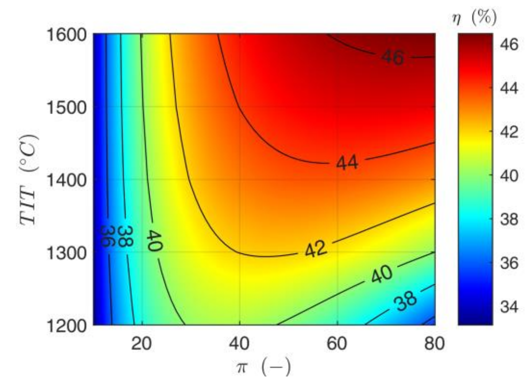

Fig. 3. Dependence of gas turbine efficiency on compression ratio and turbine inlet temperature (TIT).

As illustrated in Fig. 3, the efficiency of the gas turbine cycle is shown to vary depending on the compression ratio and the turbine inlet temperature (TIT). Consequently, it is feasible to ascertain the maximum efficiency of the gas turbine unit for the specified parameters. It is evident that efficiency increases with the temperature at the turbine inlet, an effect that arises from raising the temperature of the heat source. This increase in efficiency can reach up to 46% for a temperature of 1600°C. Furthermore, for each TIT, there is a specified compression ratio that guarantees maximum efficiency at the given operating parameters. It is observed that, at lower temperatures, the turbine attains maximum efficiency at a compression ratio of 40−50, a range that is consistent with axial compressors comprising multiple subsonic stages. Additionally, the TIT has been observed to directly correlate with the optimal compression ratio, with values approaching 80 at temperatures exceeding 1500°C. It should be noted that the current gas turbines are incompatible with such high compression ratios, as depicted in Fig. 3. In practice, values below 50−60 or even lower for heavy gas turbines are assumed. This is a consequence of the relatively flat characteristic curve depicted in Fig. 3, or the complexity involved in operating or constructing multi-stage compressors (particularly in the context of aviation). It is also difficult to maintain such a high temperature without a sophisticated cooling system for the turbine blades.

In order to comprehensively assess the performance of a model gas turbine system, it is advantageous to incorporate an additional parameter: unit power, defined as the ratio of the obtained electrical power to the air mass flow:

$$P_u = \frac{P_{el}}{\dot{m}_a}. \tag{19}$$

As illustrated in Fig. 4, the power characteristics exhibit a noticeable divergence from the efficiency characteristics. The unit power also increases with temperature, reaching values of over 600 kJ/kg in extreme cases. However, for the highest temperature, the maximum unit power is achieved for much lower compression ratios – in the range of 15−25. The optimal compression ratio is consistent across the entire range of assumed temperatures. This phenomenon has been extensively studied in the context of aircraft turbine engines, where the optimal compression ratio, which maximises specific thrust and consequently minimises unit air mass input per thrust output in the engine, differs considerably from the economic compression ratio, which enables the lowest possible specific fuel consumption, defined as the fuel consumed per unit of thrust generated.

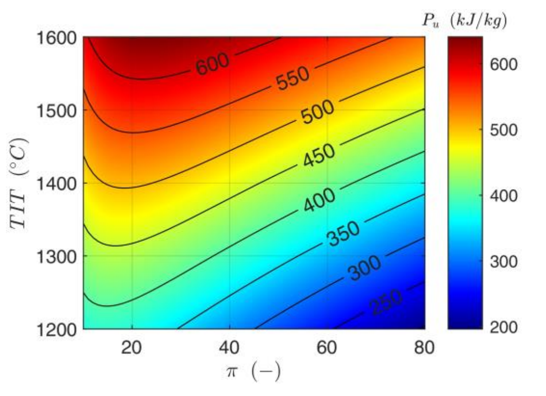

Fig. 4. Dependence of gas turbine unit power on compression ratio and turbine inlet temperature (TIT).

Considering the variations in power and efficiency characteristics, the subsequent step entailed the generation of a set of summary characteristic curves, incorporating the key parameters that characterise the operation of a gas turbine. These curves are presented in the form of a graph in Fig. 5. The method of presenting the collective impact of the aforementioned parameters is frequently employed in the context of gas turbines. It facilitates the selection of operating parameters for the gas turbine based on a systematic, integrated approach.

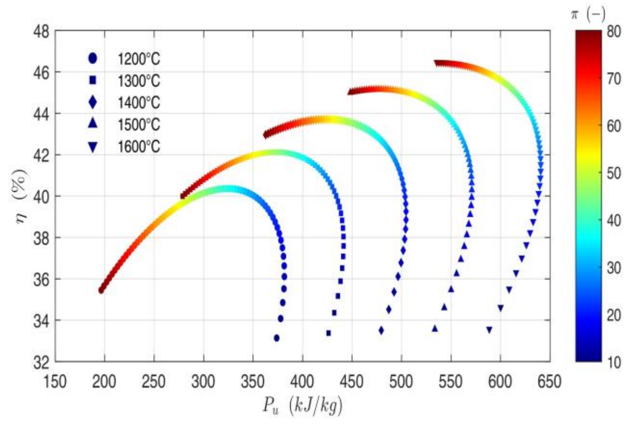

Fig. 5. Dependence of gas turbine efficiency on unit power for different values of compression ratio and turbine inlet temperature (TIT).

The graph presented in Fig. 5 shows the relationship between the efficiency of the gas turbine and the unit power in the form of curves determined for specific temperatures. The temperatures are marked with different shaped points, while the compression ratio scale is marked with a colour map. The graph shows the discrepancy between the maximum thermal efficiency and unit power, which are achieved for significantly different compression ratios.

Usually, the aim is to achieve the highest possible unit power, which, however, is far from the maximum efficiency of the gas turbine. Therefore, when designing such a system, a compromise must be found between maximum efficiency and maximum unit power. Using model characteristics may be helpful in this matter.

3.2. The influence of inlet air parameters on turbine performance

The gas turbine operating parameters presented in Section 3.1 and their dependence on design solutions can be controlled and influenced to a certain extent during the design or operation stage. In contrast, the parameters of the air at the gas turbine inlet (i.e. temperature, pressure and relative humidity) significantly affect operation and often result directly from atmospheric conditions (partially regulated from the air intake level). These parameters also clearly impact the operating parameters of single-shaft aircraft turbines, which are susceptible to changes within the so-called climatic characteristics.

Ambient air parameters may change in an uncontrolled manner depending on the season or the geographical location (climate zone) of the gas turbine unit. Therefore, it is important to understand how it behaves in different atmospheric conditions, which can be simulated using the aforementioned model.

To assess the impact of the environmental parameters, simulations were conducted for TIT = 1400°C and π = 20, as they are considered representative within the scope of the analyses undertaken. The other input parameters, enumerated in Table 3, remained constant during the course of these simulations, with the exception of the fuel mass flux, which was constant in one case and variable in the other.

The results obtained, presented in Fig. 6–11, demonstrate the impact of successive atmospheric air parameters (static temperature at the compressor inlet, static pressure at the compressor inlet, and relative humidity at the compressor inlet) on the most important parameters characterising the operation of the gas turbine. In accordance with the established mathematical model, there are two distinct methodologies for ensuring a constant temperature at the turbine inlet. The primary strategy involves the regulation of the air mass flow supplied to the compressor, while maintaining a constant fuel mass flow rate into the combustion chamber. In the second, the fuel stream is selected automatically, and the air mass flux is maintained at a constant level. The second manner is more aligned with the established practices in real installations. However, a theoretical comparison of these two possibilities is worthwhile in the context of the consequences of different process control.

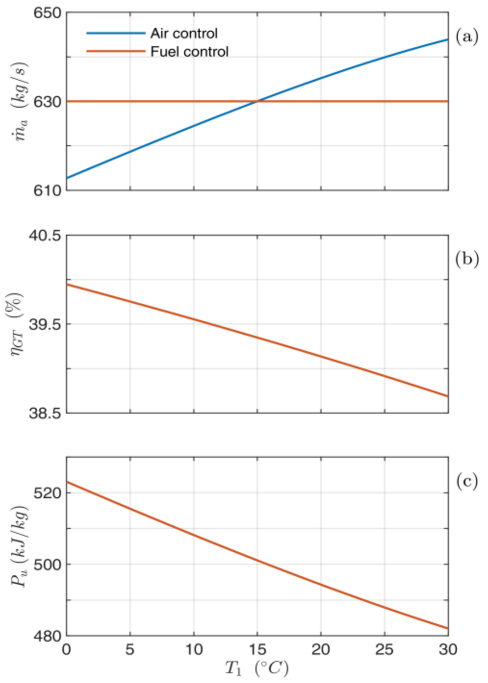

Fig. 6. Influence of the air temperature T_1 at the compressor inlet on: (a) mass flow of the inlet air $\dot{m}_a$, (b) total efficiency η_{GT}, (c) unit power P_u of the gas turbine.

Figure 6 illustrates the influence of the compressor inlet air temperature T_1 on the air mass flow $\dot{m}_a$, gas turbine total efficiency η_{GT}, and unit power P_u. A change in temperature from 0°C to 30°C was found to be accompanied by an almost linear decrease in efficiency, resulting in a reduction of nearly 1.5 percentage points (Fig. 6b), and a linear decrease in unit power of approximately 40 kJ/kg within the considered range of changes (Fig. 6c). Irrespective of the method selected for maintaining a constant TIT, the efficiency and unit power undergo analogous alterations. Consequently, differences are apparent in the individual powers of the combustion chamber, compressor and turbine (Fig. 7).

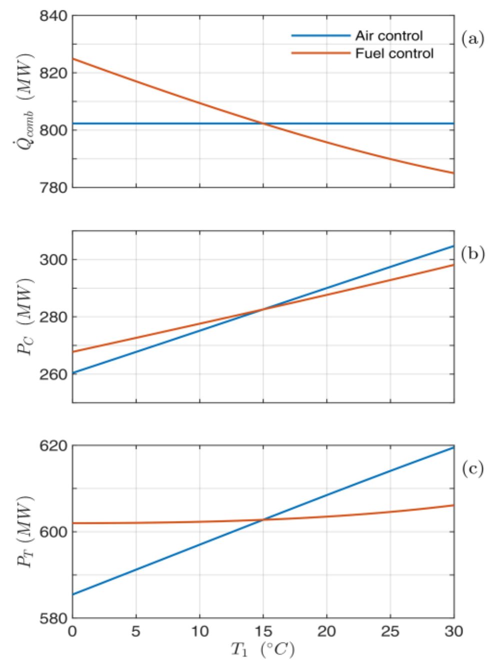

Fig. 7. Influence of the air temperature T_1 at the compressor inlet on: (a) heat flux supplied in the combustion chamber $\dot{Q}_{comb}$, (b) internal power of the compressor P_C, (c) internal power of the turbine P_T.

In the case of air control, and thus the constant fuel flow, the change of efficiency, according to Eq. (11), depends only on the change of the total electrical power of the gas turbine. The decrease in the total power can be explained by the decrease of the air density (due to the temperature change) and the increase of the mass flow of the inlet air to the compressor (Fig. 6a).

Although the decrease in density with temperature is self-evident, an explanation is necessary to understand why an increase in temperature at the compressor inlet leads to an increase in air mass flow. This is due to the assumption of a constant TIT and a constant fuel mass flow $\dot{m}_f$. The inlet air temperature t_1, with the condition of fixed compression ratio ($\pi = 20$), causes an increase in temperature at the compressor outlet t_2. This results in higher temperatures of the air supplied to the combustion chamber. Since the same amount of heat is obtained from fuel combustion, the air flow must be increased so that the flue gas temperature remains at the same assumed level.

The aforementioned factors result in the compressor requiring greater drive power (see Fig. 7b), with an increase of 17% for a 30-degree temperature growth. Concurrently, there is a slight (6%) increase in turbine power (see Fig. 7c), leading to an elevated back-work ratio and a reduction in the net power of the gas turbine.

In the second case, which pertains to fuel control, the air mass flow rate remains constant (see Fig. 6a). Consequently, an increase in air temperature leads to a reduction in the necessary heat input to the combustion chamber to maintain the desired TIT (see Fig. 7a). The increase in compressor power is analogous to the initial case, with the distinction that the air mass flow rate remains constant. Given the air-fuel ratio of approximately 40, fluctuations in fuel mass flux result in a negligible alteration in exhaust gas production. Maintaining constant thermal conditions at the turbine inlet consequently yields a negligible alteration in turbine power. In conclusion, the discrepancy in gas turbine net power in comparison to the initial case is counterbalanced by maintaining a constant air mass flux. This results in the same characteristics of the unit power.

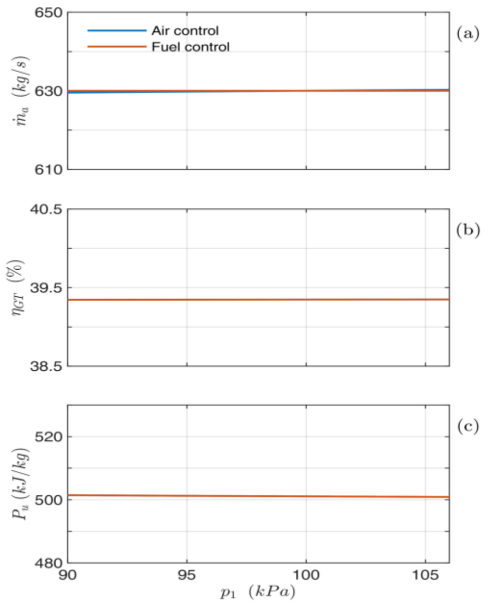

Fig. 8. Influence of the air pressure p_1 at the compressor inlet on: (a) mass flow of the inlet air $\dot{m}_a$, (b) total efficiency η_{GT}, (c) unit power of the gas turbine P_u.

As demonstrated in Fig. 8, the previously specified operational parameters exhibit a minor sensitivity to variations in air pressure at the compressor inlet. As illustrated in Fig. 8a, an increase in pressure p_1 at a constant assumed temperature t_1 results in a decrease in the air mass flow, which remains constant. Maintaining a constant compression ratio leads to a negligible change in temperature behind the compressor. The minor discrepancies can be explained by the variation in absolute pressure values, which consequently impacts the specific enthalpy. According to the implemented model, the specific enthalpy is a function of both temperature and pressure. However, for low pressures, this does not significantly affect enthalpy. Consequently, its value at characteristic points (h_2 behind the compressor and h_3 before the turbine) varies only slightly for different values of the inlet pressure. Therefore, the influence of the inlet pressure (which varies within the range of atmospheric air) on gas turbine efficiency and unit power is hardly noticeable.

As with variations in temperature, controlling the air or fuel mass flow rate so as to maintain a constant turbine inlet temperature (TIT) results in precisely the same characteristics of efficiency and the unit power of a gas turbine. A marginal discrepancy can be discerned in net power and, consequently, in the heat supply, compressor and turbine power (Fig. 9). Nevertheless, these deviations are primarily attributable to minor variations in air mass flux incorporated into the cycle.

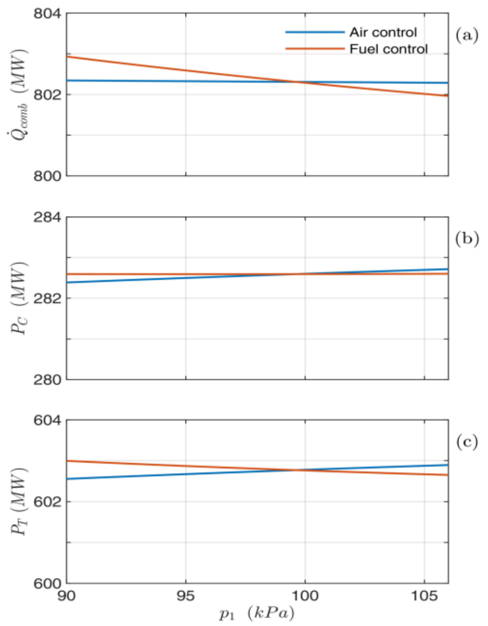

Fig. 9. Influence of the air pressure p_1 at the compressor inlet on: (a) heat flux supplied in the combustion chamber $\dot{Q}_{comb}$, (b) internal power of the compressor P_C, (c) internal power of the turbine P_T.

Another parameter analysed in terms of turbine operation is the relative humidity φ of the air at the compressor inlet. Its impact on the discussed parameters is shown in Figs. 10 and 11. A growth of the relative humidity slightly increases density, resulting in a decrease in air mass flow (Fig. 10a) in the case of air control. Conversely, the effect on enthalpy is more pronounced than the effect on pressure change. This results in a relatively modest yet substantial impact on gas turbine efficiency and unit power. The decline in efficiency is only marginally perceptible (Fig. 10b), and a slight rise in unit power (Fig. 10c) is evident.

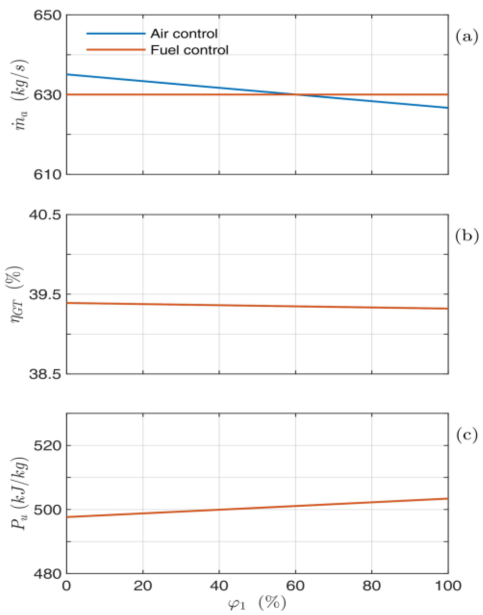

Fig. 10. Influence of the air relative humidity φ_1 at the compressor inlet on: (a) mass flow of the inlet air $\dot{m}_a$, (b) total efficiency η_{GT}, (c) unit power P_u of the gas turbine.

The discrepancies in the air control and fuel control cases are illustrated in Fig. 11. Given the identical thermal parameters at characteristic points, it is the air mass flux that is responsible for the observed discrepancies in compressor and turbine power.

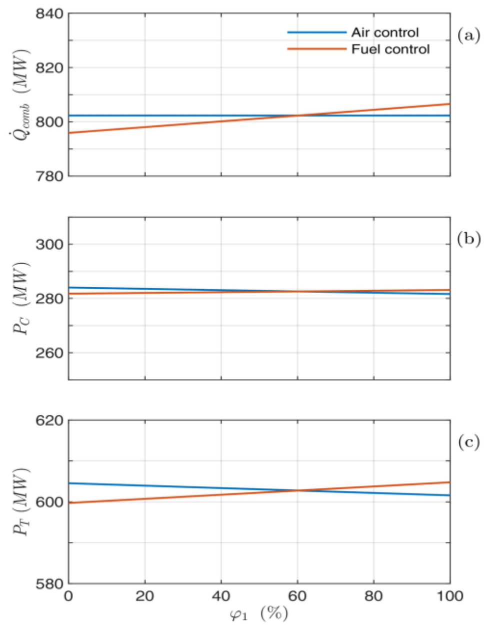

Fig. 11. Influence of the air relative humidity φ_1 at the compressor inlet on: (a) heat flux supplied in the combustion chamber $\dot{Q}_{comb}$, (b) internal power of the compressor P_C, (c) internal power of the turbine P_T.

The findings presented above indicate that, among the examined air parameters, the ambient temperature exerts the most significant influence on the turbine efficiency and unit power. The ambient pressure exerts a comparatively minor influence on the unit power fluctuations and the gas turbine efficiency. It is important to note, however, that the ambient pressure does not undergo as rapid changes during the day or over the course of several hours as the temperature does. Consequently, its impact may be discernible in extreme situations when the pressure deviates from its standard value. It has also been demonstrated that the relative humidity exerts a small but noticeable influence on the efficiency and unit power of a gas turbine.

3.3. The influence of hydrogen addition to the fuel on turbine performance

The created model was utilised to execute a preliminary investigation into the impact of enriching conventional fuel (methane) with hydrogen at varying mass percentages ranging from 0% to 100%. The simulation was performed with a constant air supply to the system, as illustrated in Table 3.

As demonstrated by the analyses, the rising proportion of hydrogen (Fig. 12a) in the fuel utilised in the modelled installation is accompanied by an increase in both the thermal power of the combustion chamber (heat supply) and the power of the turbine. This rise in power, coupled with the constant compressor power, results in an increase in the power output of the entire unit.

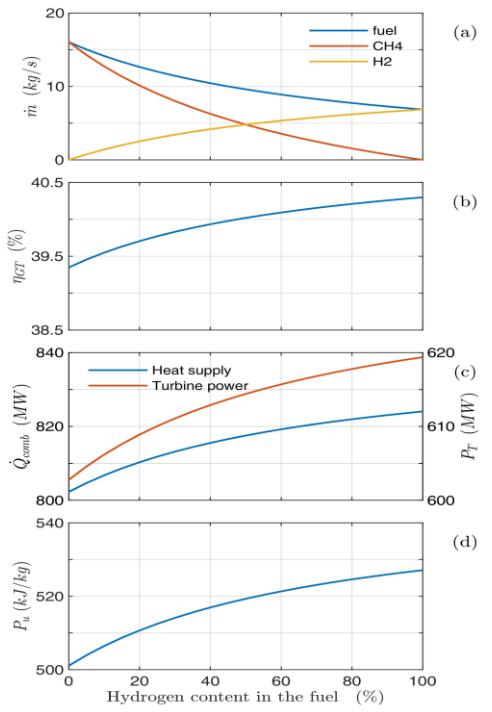

Fig. 12. Influence of the hydrogen content in fuel on: (a) mass flow of the fuel and its components, (b) total efficiency η_{GT}, (c) heat flux supplied to the combustion chamber and turbine power, (d) unit power of the gas turbine P_u.

As illustrated in Fig. 12c, the second increase exceeds the first in percentage terms. Consequently, the total efficiency of the gas turbine increases by less than 1 percentage point (see Fig. 12b). Additionally, the unit power increases by nearly 30 kJ/kg, due to its reference to a constant air flow supplied to the compressor.

It should be noted that the obtained results are strictly dependent on the simulation settings and model parameterisation. For instance, the aforementioned increase in thermal power of the combustion chamber stems from two factors: the higher calorific value of hydrogen compared to methane, and the increased proportion of water vapour in the exhaust gas stream, which raises its specific enthalpy.

An increase in the exhaust gas stream's specific enthalpy is necessary to reach the desired temperature at the turbine inlet. Consequently, more heat must be supplied, introducing more chemical energy into the cycle with the fuel. While the transition from pure methane to pure hydrogen combustion decreases the stream leaving the chamber by 9.182 kg/s (less than 1.5% of the initial mass flux), the increase in the exhaust gas stream's specific enthalpy is 58.54 kJ/kg (nearly 3.5% more). A similar situation occurs in the turbine. The increase in power is solely the result of the greater difference in exhaust gas enthalpy between the two isobars (assuming isentropic turbine efficiency). This difference is a direct result of the thermochemical properties of the gas mixtures formed in the combustion chamber. In the considered model, the pressure ratio, turbine inlet temperature, and the efficiency of the aforementioned machine are constant.

To enhance the representativeness of the results with respect to real objects, it would be necessary to consider factors such as the combustion chamber efficiency (losses due to incomplete or partial combustion, pressure losses) or turbine efficiency as a function of the hydrogen content in the fuel. While these elements were not incorporated in the present simulations, the utilised tool, EBSILON Professional, enables the implementation of more advanced characteristics of the aforementioned machines and devices. Consequently, they remain viable options for zero-dimensional analyses of gas blocks, including those burning or co-burning alternative fuels.

4. Conclusions

The article presents a numerical model of a gas turbine, prepared using the EBSILON Professional software. This program was used to examine the impact of individual parameters (compression ratio, turbine inlet temperature and ambient air parameters) on the efficiency and the unit power of a gas turbine.

The EBSILON Professional program facilitates the modelling of real objects, including gas turbines, ensuring satisfactory accuracy and convergence with actual operating indicators. This is exemplified by the case of GE's 9HA.01 turbine. The software's functionality encompasses the optimisation of selected parameters, facilitating the observation of their impact on turbine performance. An illustration of this optimisation is provided in Fig. 5, a diagram that can be utilised to identify compression ratio values for which the gas turbine achieves a sufficiently high efficiency and the highest possible unit power.

The operation of a gas turbine is dependent on atmospheric conditions, including the temperature, pressure and relative humidity of the surrounding air. The atmospheric parameter that exerts the most influence on its operation is temperature. As demonstrated in Figs. 6−11, the efficiency and the unit power of a gas turbine are significantly affected by the air temperature, while changes in the pressure and the relative humidity are of lesser importance. Changes in ambient temperature are observed throughout the year, depending on seasonal conditions. Consequently, gas turbines demonstrate optimal efficiency during winter months, when ambient temperatures are sufficiently low. Conversely, gas turbines exhibit minimal efficiency during summer months, when temperatures frequently exceed 30°C, dependent upon the specific climate zone. However, it is important to note that the temperature of the inlet air should not become excessively low during operation, as this can lead to operational problems associated with condensation that can crystallise in the air intake. The presence of ice crystals on the compressor blades can damage this component of the gas turbine, leading to a forced shutdown. Therefore, in case of low inlet temperature (i.e. between −10°C and 10°C), the air supplied to the compressor is pre-heated to prevent this from happening. The impact of adding hydrogen to fuel on the operating parameters of a gas turbine was subsequently determined.

In the context of simulating actual gas turbine cycles, it is necessary to consider the material limitations of the components utilised and the atmospheric conditions characteristic of the climate zone in which the turbine is located. These factors impose greater restrictions that must be taken into account during the calculations.

It should be emphasised that the results obtained and presented in this paper are a direct consequence of the parameterisation of the numerical model described in Chapter 2 and referenced in Subsection 3.2. Although these results should be considered approximations of the actual state of the power system, the software allows for the inclusion of higher-order characteristics of individual machines and devices in the simulation, as well as more complex control systems.

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