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SCUOLA DI INGEGNERIA INDUSTRIALE
E DELL'INFORMAZIONE

ANALYSIS OF HYDROGEN FIRED GAS TURBINE CYCLES FOR PEAK GENERATION

TESI DI LAUREA MAGISTRALE IN
ENERGY ENGENEERING – POWER GENERATION

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Abstract

Assuming that the growth in the number of renewable plants for the production of electricity continues according to the current trend, there is a need to couple them with flexible and quick start-up power cycles that help balance the grid and meet demand. Furthermore, we need plants capable of contributing to the process of decarbonization and abatement of emissions.

The work proposes first to briefly analyze the current Italian electricity production and the market of large size gas turbines. Subsequently it focuses on the possible key element for the future of electricity production, hydrogen, and its characteristics as a fuel for gas turbines.

The heart of the work is centered on simulations of complex gas cycles based on hydrogen. The simulations are carried out using GS Software, a program developed by Politecnico of Milan. In the different phases the effects on the cycle of intercooling, regeneration and water injection are studied step by step. In each phase, the variations in compression work, fuel consumption, cooling flows and efficiency and power produced are highlighted.

The studied cycles are able to reach performances similar to those of modern combined cycles and turn out to be a possible turning point for the future of power generation.

Key-words: Gas turbine, Hydrogen, Intercooling, Regeneration, Water injection, Peak generation

Abstract in lingua italiana

Supponendo che la crescita del numero impianti rinnovabili per la produzione di elettricità prosegua secondo l'andamento attuale, esiste la necessità di accoppiare ad essi cicli di potenza flessibili e dal rapido avviamento che aiutino a bilanciare la rete e soddisfare la domanda. Inoltre, servono impianti in grado di contribuire al processo di decarbonizzazione e abbattimento delle emissioni.

Il lavoro si propone prima di analizzare brevemente la produzione elettrica italiana attuale e il mercato delle turbine a gas di grande taglia. Successivamente si focalizza sul possibile elemento chiave per il futuro della produzione elettrica, l'idrogeno, e le sue caratteristiche come combustibile per turbine a gas.

Il cuore del lavoro è incentrato su simulazioni di cicli a gas complessi basati sull'idrogeno. Le simulazioni sono effettuate tramite GS Software, un programma sviluppato dal Politecnico di Milano. Nelle varie fasi vengono studiati per step gli effetti sul ciclo di: interrefrigerazione, rigenerazione e iniezione di acqua. In ogni fase sono evidenziate principalmente le variazioni sul lavoro di compressione, sul consumo di combustibile, sui flussi di raffreddamento e su efficienza e potenza prodotta.

I cicli studiati riescono a raggiungere prestazioni simili a quelle dei moderni cicli combinati e risultano essere una possibile svolta per il futuro della produzione di potenza.

Parole chiave: Turbina a gas, Idrogeno, Interrefrigerazione, Rigenerazione, Iniezione di acqua, Produzione di picco

Contents

Introduction

One of the most relevant rights for the mankind, among many others, is that every single person should have easy access to electricity. Since the number of people achieving the access to electricity grows every year it seems that the mankind is moving towards the right direction. Yet, there is an increasing growth of the energy consumption, also linked to the growing wellness of many countries worldwide. This growth has been remarkably fast from the beginning of the XXI century, as we can notice from Figure 0.1. Another relevant aspect easily noticeable is that, until today, most of the energy share has been produced through conventional fossil fuels such as coal, oil or natural gas.

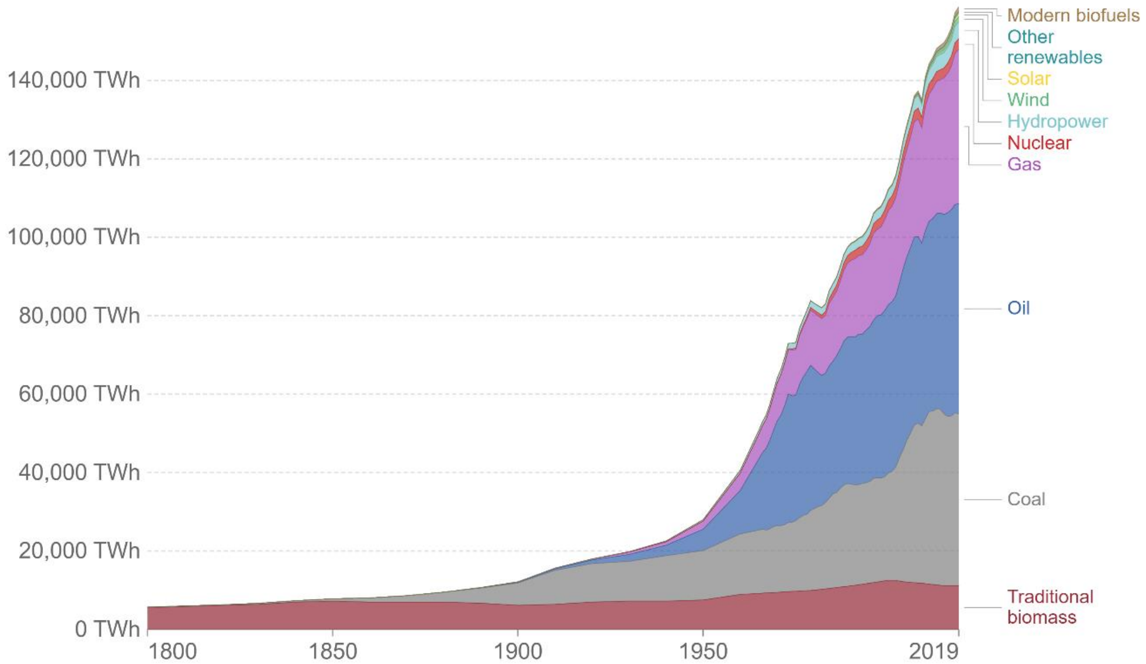

Figure 0.1 - Global energy consumption [1]

One of the main concerns related to fossil fuel combustion is the emission of CO_2, the by-product of the combination in the combustion process of the carbon present in fossil fuels and the oxygen present in the air used as oxidant. As we can infer from Figure 0.2, the direct consequence of the growing energy consumption is the increase of CO_2

emissions, therefore following a very similar trend. Every day, human emissions of carbon dioxide and other greenhouse gases are a primary driver of climate change and determine one of the world's most pressing challenges. This link between global temperatures and greenhouse gas concentrations, especially CO_2, has been consistently true throughout Earth's history.

To set the scene, let's look at how the planet has warmed. In Figure 0.3 we see the global average temperature relatively to an average period between 1961 and 1990.

The red line represents the average annual temperature trend through time, with upper and lower confidence intervals shown in light grey.

We see that over the last few decades, global temperatures have risen sharply, to approximately 0.7°C higher than our 1961-1990 baseline. When extended back our evaluation back to 1850, the temperatures were a further 0.4°C colder than our baseline. Overall, this would amount to an average temperature rise of 1.1°C in slightly less than 150 years.

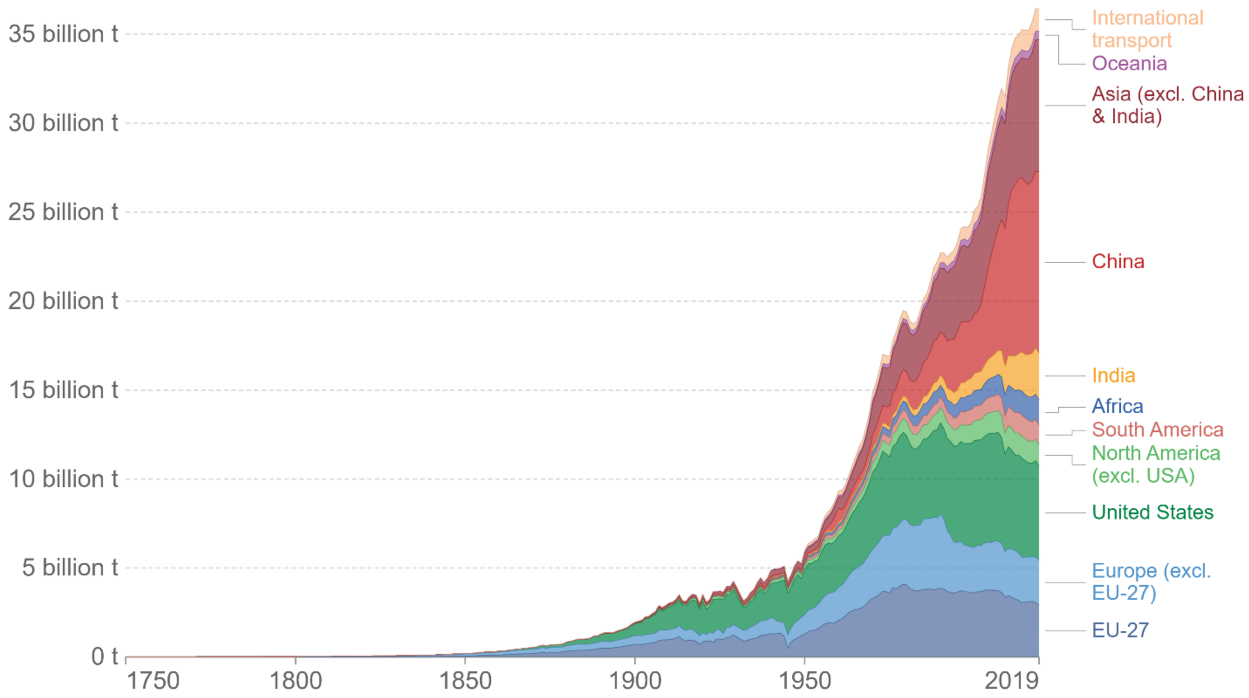

Figure 0.2 – Annual total CO_2 emissions [1]

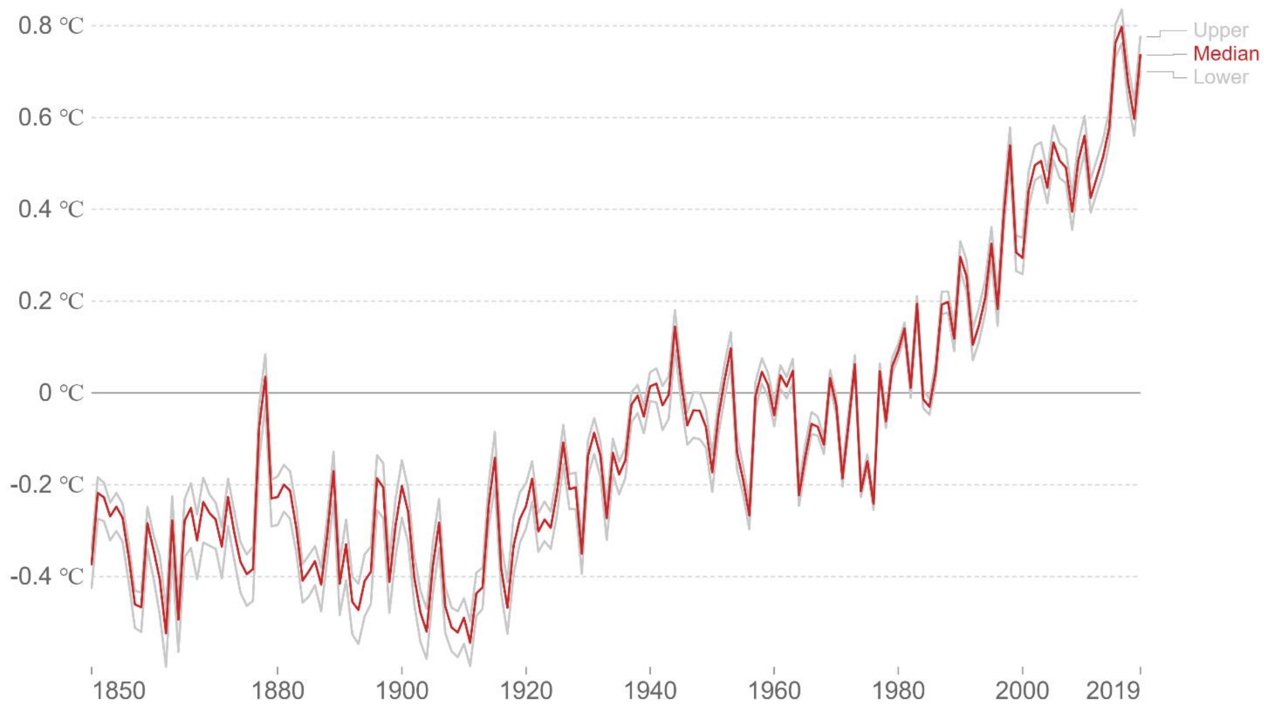

Figure 0.3 – Average temperature anomaly, Global [1]

To slow down the increase of global temperature, with the possible aim of halting it, a stabilization of CO_2 concentrations and other greenhouse gases in Earth's atmosphere is needed. This link between global temperatures and greenhouse gas concentrations, especially CO_2, has been as well true throughout Earth's history.

In Figure 0.4 the global average concentrations of CO_2 in the atmosphere over the past 800,000 years can be appreciated. Over this period, consistent fluctuations in CO_2 concentrations are noticed; these periods of rising and falling CO_2 coincide with the onset of ice ages (low CO_2) and interglacial periods (high CO_2). These periodic fluctuations are caused by changes in the Earth's orbit around the sun, called Milankovitch cycles.

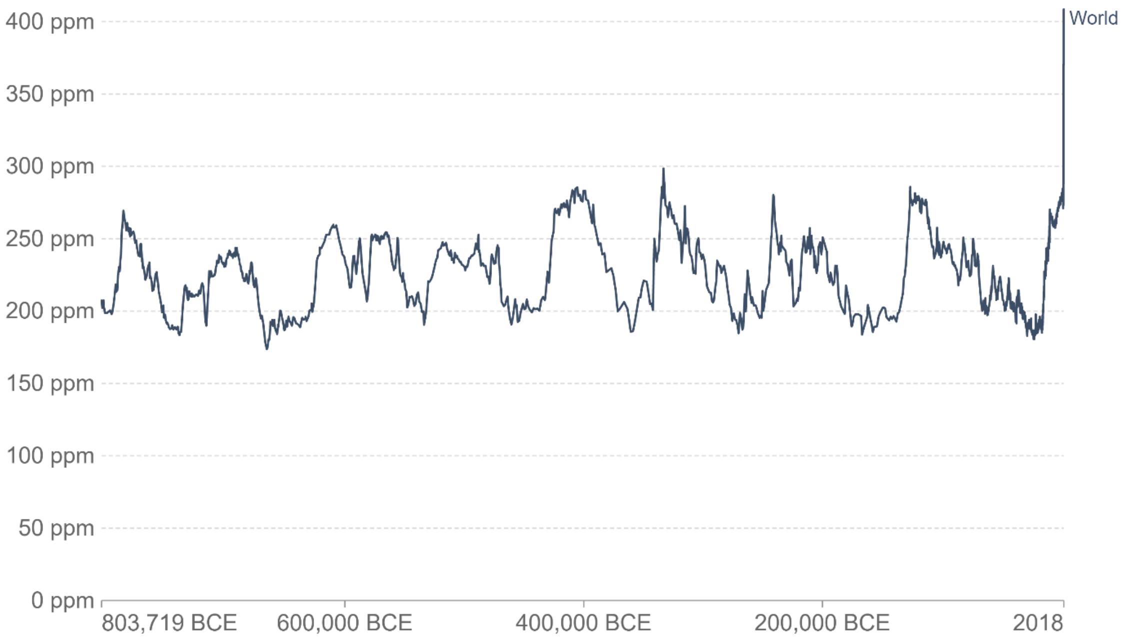

Figure 0.4 – Atmospheric CO_2 concentration [1]

Over this long period, atmospheric concentrations of CO_2 never exceed 300 parts per million (ppm). This pattern changed with the Industrial Revolution and the rise of human emissions of CO_2 from burning fossil fuels. In Figure 0.5 a rapid rise in global CO_2 concentrations over the past few centuries can be seen, and particularly in recent decades. For the first time in over 800,000 years, the CO_2 concentrations not only rose above 300 ppm but are now well over 400 ppm.

It's not only the CO_2 levels change in the atmosphere that matters, but also the rate of the change. Historical changes in CO_2 concentrations tended to occur over centuries or even thousands of years. Conversely and more recently it took a bunch of decades to achieve even larger changes. Obviously, this acceleration gives species, planetary systems and ecosystems much less time to adapt.

For these reasons the process of reducing greenhouse gas emissions has become very important even if it will take some time to become effective. One of the keys to reduce CO_2 emissions in the field of the power generation is represented by the renewables.

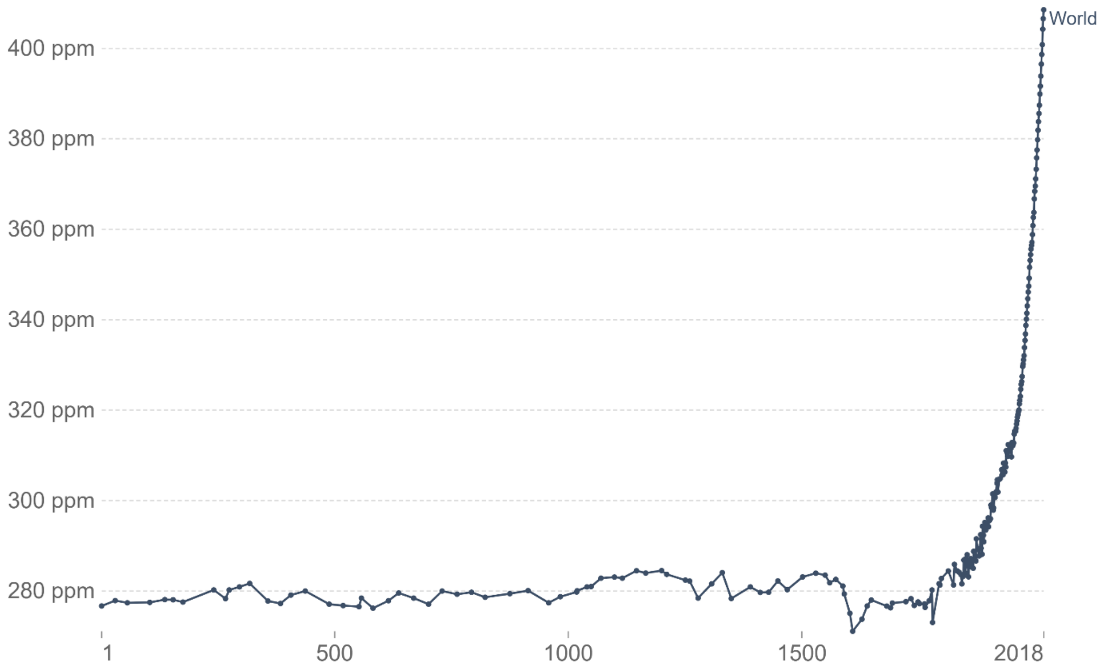

Figure 0.5 – Atmospheric CO_2 concentration [1]

As we can see in Figure 0.1, over the last few decades the share of energy produced through renewables has grown, but the slice of the cake is still very limited, also because the use of fossil fuels keeps increasing, due to the increasing energy demand. Anyway, in the future the quote of electricity produced from renewables is destined to grow more and more.

One of the limiting aspects of renewable plants is linked to their unprogrammable nature. In fact, their functioning is strictly connected to external factors like weather, climate conditions or even moments of the day, month or year. This weakness brings the necessity of additional plants, nowadays based on fossil fuels, that must be coupled with the renewable ones, so that the gap in the energy production of a part of the day, month or year can be covered. Nowadays the most efficient plants used to produce electricity are based on combined cycles, and most of them use natural gas as fuel. An alternative to combined cycle plants should be found, since they need to work for too many hours to be paid back, have long start up times and contribute anyway to global warming due to CO_2 emissions.

One of the most promising technologies in this sense, capable to produce electricity without releasing CO_2 in atmosphere, in the moments in which renewable plants can't generate power or are at partial load, are power cycles based on hydrogen. At the same time the products of its combustion process consist only of water and a little amount of nitrogen oxides.

The aim of the present work is first to highlight the existing problem related to the generation and operational mode of the existing plants in couple with the renewable ones, to have an idea of what the present market of gas turbines offer and to evaluate through simulations the competitiveness from a thermodynamic point of view of different configurations of gas turbine cycles based on hydrogen in respect of conventional combined cycles, highlighting also the possible limitations to the spread of this technology.

1. Peak Generation

1.1 Combined Cycles

Exhaust gasses coming from a gas cycle are characterized by quite high temperatures, easily reaching 550-600°C. Therefore, they retain a high energy content which would be lost whenever they would be released in the atmosphere. For this reason, combined cycle power plants were introduced, creating an assembly of heat engines benefiting in tandem from the heat same source and converting it into mechanical energy first and into electricity thereafter. In fact, CCGT couple a gas turbine to a steam cycle as schematized in the easy'to'understand way in Figure 1.1. The heat entering in the steam cycle is recovered from the exhaust gasses coming from the gas turbine through an HRSG, which stands for Heat Recovery Steam Generator. For this reason, the gas cycle is also called '*topping*' cycle because it works at higher temperatures, while the steam cycle is also called '*bottoming*' cycle, because it works at lower temperatures. In the same Figure 1.1, for simplicity, the steam cycle is represented on a single pressure level. Actually, modern CCGT plants are much more complicated on the water side. In fact, the steam side is divided in two or three pressure levels, depending on the presence of the re-heating, in order to increase the overall efficiency and to reduce the losses due to the difference in temperatures occurring in the heat exchange. This also allows to cool down exhaust gasses to values around 100 °C. Both the gas power and the steam cycles are produced from the same quantity of fuel, and this will increase the overall efficiency of the plant, increasing from values of around 40% to peaks of around 60% in design conditions. So, gas burning combined cycle plants are ideally suitable for the use in heavily populated regions because of their high efficiency and of low emission levels of pollutants. In particular, very low NOx levels of clean combined cycle plants are one of their most attractive features. Furthermore, CCGT plants produce for each kWh only 40% of the CO_2 produced by a coal-fired plant. In the last few years, they have become of crucial importance for power generation and energy sector.

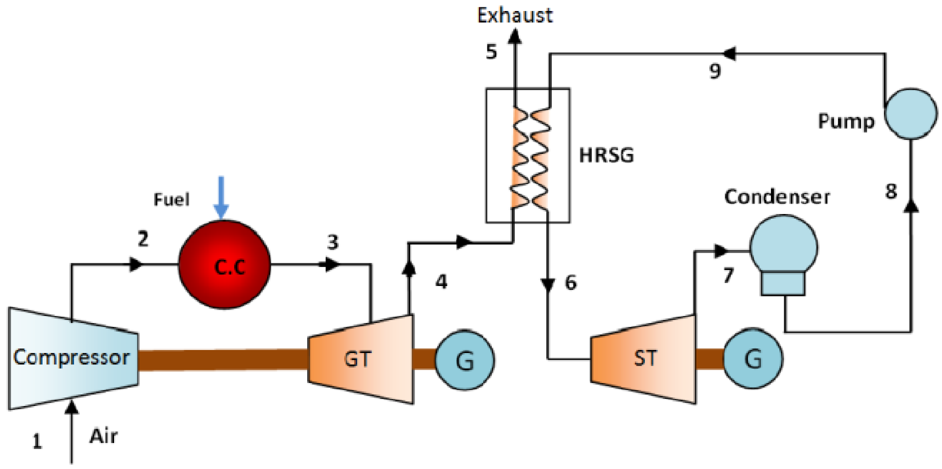

Figure 1.1 – Base CCGT Scheme

1.2 Combined cycles limiting aspects

Anyway, these interesting plants also bear some negative aspects to be considered in the green perspective. In fact, since plants are based on fossil fuels, they contribute to increase pollution with greenhouse gasses, especially with CO_2 emissions. As we can see from Figure 1.2, carbon dioxide emissions produced from electricity production and heat generation represent the most relevant contribution in respect of other sectors like transports, manufacturing, constructions etc. For this reason, the green transition is even more important in the energy sector and adequate alternatives should be found to be coupled with renewable plants to achieve carbon neutrality. Moreover, CCGT are very expensive, the capital cost of these type plants larger than 200 MW ranges from $450 to $650 per kW, so they easily reach costs of hundreds of millions of dollars. Such a relevant cost is mainly due to the presence of two turbines, one for the gas cycle and one for the steam cycle, to the addition of the condenser, to the HRSG and to the construction costs, which fast-growing for such big plants. Given these important investment costs, these plants need to work for some thousands of equivalent hours per year to be paid back within their lifetime and in a renewable context they can't be anymore an optimal solution, as they should work only for hundreds or at least few thousands of equivalent hours each year. Moreover, many of the already present combined cycles have a high nominal power output of hundreds of megawatts and with the spread of renewables they should often work at partial load, which would cause lower performances.

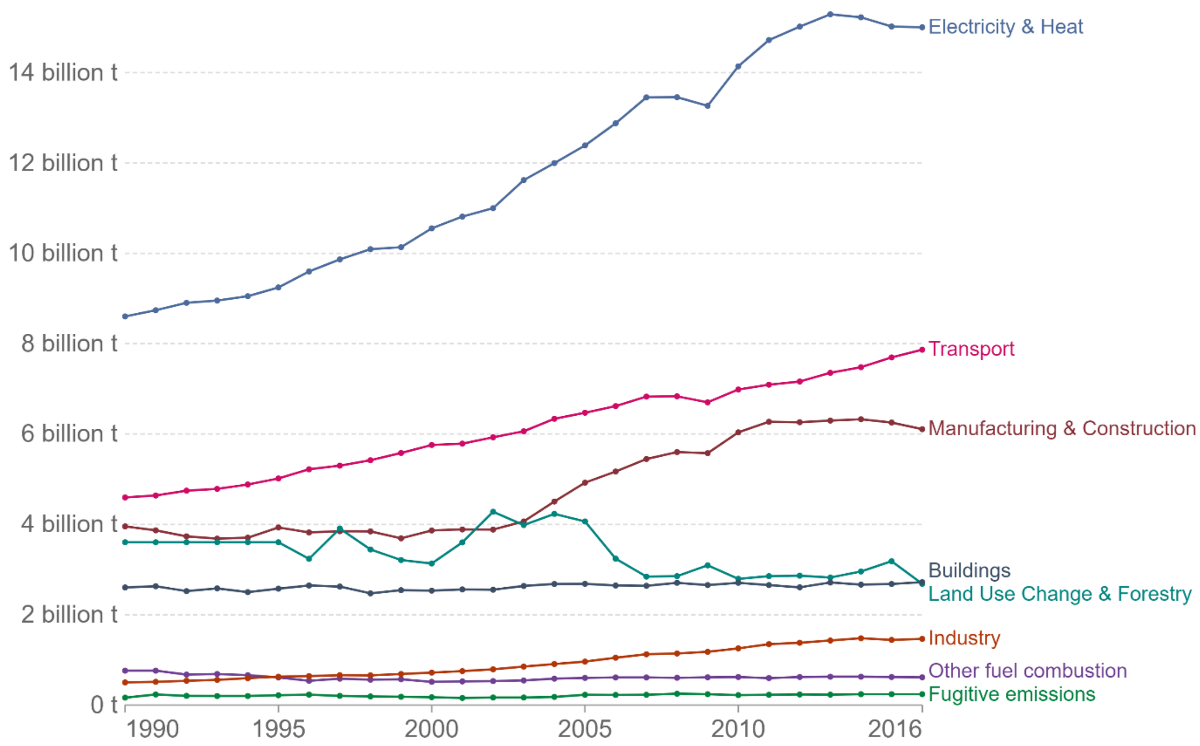

Figure 1.2 – CO_2 emissions by sector, World [3]

Another limiting aspect of combined cycles is the ramp time, i.e. the amount of time needed from the moment a generator is turned on to the moment it can start providing energy to the grid at its lower operating limit, and the minimum run time, i.e. the shortest amount of time a plant can operate after it is turned on. In the case of CCGT, the ramp time is in the order of hours, while for a simple cycle combustion turbine goes from minutes to hours. Instead, the minimum run time can vary from hours to days for CCGT, while it is in the order of minutes for simple cycle combustion turbine. Moreover, turning on and off such big plants is not a great advisable, as these transients are related to important start-up and shut-down costs.

The combustion turbine is the fastest starting component of the combined cycle system. The combustion turbine is started by some auxiliary means and then fuel is introduced. The combustion of fuel generates power which increases the speed. At a certain speed the combustion turbine is coupled or synchronized with the electrical generator to produce electrical power. Generally, combustion turbine takes about ten minutes or less to get to the synchronized speed. Once the combustion turbine and the generator are synchronized, the electrical output is increased to the 100% level. The loading of the generator from 0% load at synchronization to 100% load takes about ten more minutes. Thus, a combustion turbine coupled generator can be brought on line for 100% power output within 20 minutes.

The heat recovery steam generator does not have any moving parts, but it has thermal inertia and rapid heating that may result in high thermal stresses which would affect its the operating life.

In an HRSG the high-pressure drum is most vulnerable to buildup of thermal stresses if heating is done very rapidly. To preclude this possibility the drum is heated in a controlled manner as shown in Figure 1.3. The magnitude of the stress depends on the temperature difference which depends on the material, the operating pressure, the thickness of the component and the fatigue life cycles. The temperature difference can be effectively managed by controlling the pressure inside the drum. If a certain temperature difference is close to the design limit it can be controlled by holding the pressure constant until the temperature difference decreases because of increase in the component temperature due to conduction. This is indicated by the constant pressure or saturation temperature line on the drum heating chart.

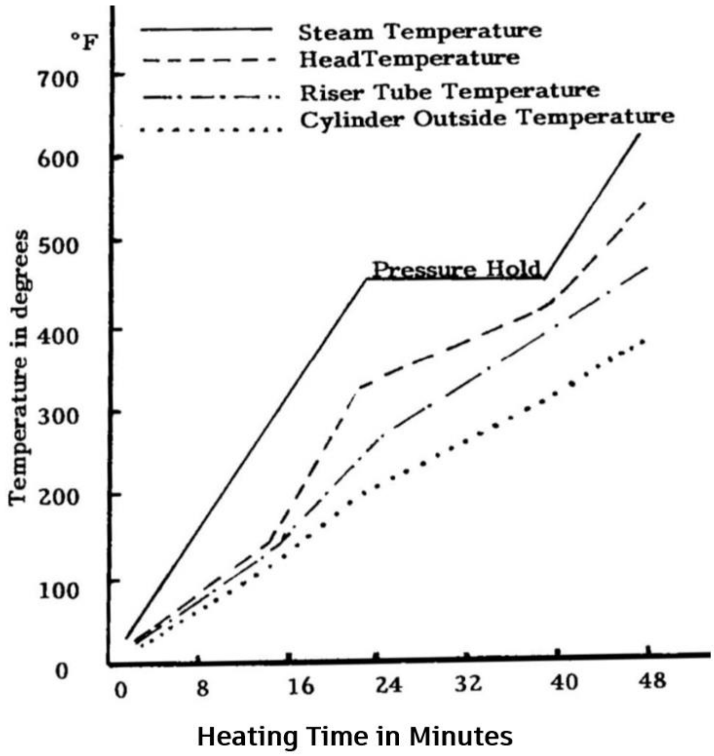

Figure 1.3 – Drum heating criteria [4]

Before an HRSG is put on line, it is first filled in with water and heat is applied thereafter. The cold metal takes some time to get heated. The HRSG starts producing steam after a soaking period of a few minutes. If the steam is not released, then the pressure starts building up. The amount of steam produced and the increase in the pressure depends on the amount of heat supplied. More heat produces more steam and pressure increases at a faster rate. The drum pressure can be controlled either by relieving the generated steam or by controlling the heat input to the boiler. Oftentimes a combination of both the means is used to accomplish the controlled heating of the HRSG. The steam is relieved by venting to the atmosphere or by sending it to a steam sink such as condenser. The heat input is controlled by operating the combustion turbine at a reduced load. A gas side bypass system, which diverts part of the hot combustion turbine gasses to atmosphere, is also used to control the heat input to the boiler. It is not necessary to run the combustion turbine at reduced load if a bypass system is provided.

The steam turbine is also affected by start-up maneuvers because it has by far the most mass and has components with much thicker cross-sections.

The stresses in the rotor during the start-up are caused by:

- temperature gradients in the material (thermal stress)
- centrifugal forces
- pressure gradients

Therefore, it needs the longest warming up time. Initially, steam is applied to warm up and set the seals between the steam turbine and the condenser so that there is no back flow from the condenser when steam is bypassed to the condenser. Various stages of steam turbine are heated in what is called a controlled warm-up by low pressure steam. This generally takes three to five hours. The warm-up steam can be provided by the HRSG as it gets started. Since the steam turbine start-up takes longer, the HRSG needs to be maintained at the low load operation for a much longer time if the steam is supplied for warm-up. The combustion turbine may or may not be run at part load. A combination of preheating by steam bypassing may be used to decrease the steam turbine start-up time. Generally, the steam turbine is a constant volume machine so it can be synchronized at a much lower pressure if the volume of steam is adequate. Under such circumstances, the HRSG is brought on line and full combustion turbine load is applied and the HRSG is operated at a much lower pressure so that the steam turbine can be synchronized with the electrical generator.

In conclusion thick components operating at high temperatures are subject to thermo-mechanic fatigue and for this reason start-up and load change operations have to take place in long periods of time. For sure there are control systems that limit the run-up velocity to avoid exceeding the permissible thermal stress, such as rotor stress evaluators or boiler stress evaluators. Moreover, cyclic stresses, even if under the acceptable limits, may cause damages if we increase too much the number of start-ups. They can reduce the lifetime of some components and increase the maintenance costs.

1.3 Italian power generation

In Italy the installed power related to renewable plants is growing year after year, as we can see from Figure 1.4. We are moving in the right direction to make our world more sustainable.

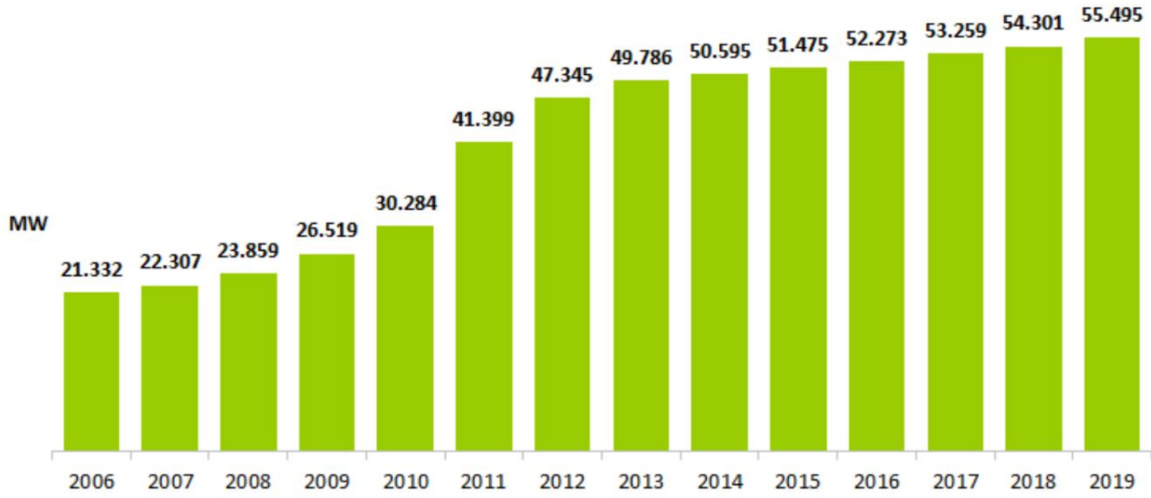

Figure 1.4 – Renewable plants installed power per year, Italy

Renewables, as anticipated before, require the presence of back up plants that must be very flexible to balance in the best way energy supply and demand. This challenge has been called '*filling the duck pond*'. Though electricity storage continues to show promise, the amount of practical, economic storage is limited, so the vast majority of electricity has to be produced as it is consumed.

The data reported in this chapter will be all related to operational year 2019, to realize the mode of operation in normal conditions, without the decrease in the energy demand caused by the pandemic of COVID-19, which has reduced energy consumption in many sectors.

In Italy, in 2019, the 39,5% of the electricity produced has been generated from renewable plants, with a generation of 116 TWh, highlighting a growth of 1,3% respect 2018. The energy from renewables used in thermal and transport sector is also growing year after year.

Most of the electricity is still produced from CCGT, some relevant data are reported in Table 1.1. Today in Italy there are 58 CCGT plants only employed for electricity production, that in 2019 have produced 58 TWh of electricity, 18,6% of electricity production, and 106 CCGT plants used both for electricity and heat generation, that in 2020 have produced 76 TWh of electricity, 24,4% of the internal production.

	% Production	*Power [TWh]*	*Installed power [GW]*
CCGT electricity	*19.6*	*58*	*22.3*
CCGT cogenerative	*26.1*	*76*	*17.5*

Table 1.1 – Generation data of CCGT of 2019, Italy

From this table we can notice that CCGT employed for electricity produce less energy than the cogenerative ones, although their installed power is greater. This means that these plants are not working for long periods like in the past when they were built.

In a future perspective in which electricity from renewables will be more abundant than the one produced from traditional plants, non cogenerative CCGT will probably not be the best idea to contribute for electricity generation, also because they would not work enough to be paid back.

In fact, whereas in the past CCGTs were normally run for lengthy periods of time as baseload power and they were the best option, now they must respond rapidly to variations in energy supply and demand, requiring a more flexible and cyclic pattern of use, with an increasing number of start-ups, and a consequent reduction of the mean electrical efficiency, that in design conditions can be over 60% for modern CCGTs, which is instead closer to 50%. The oscillation of the renewable power availability is evident in Figure 1.5, in which is reported the renewable generation in Italy in the week from 10/04/2019 to 17/04/2019.

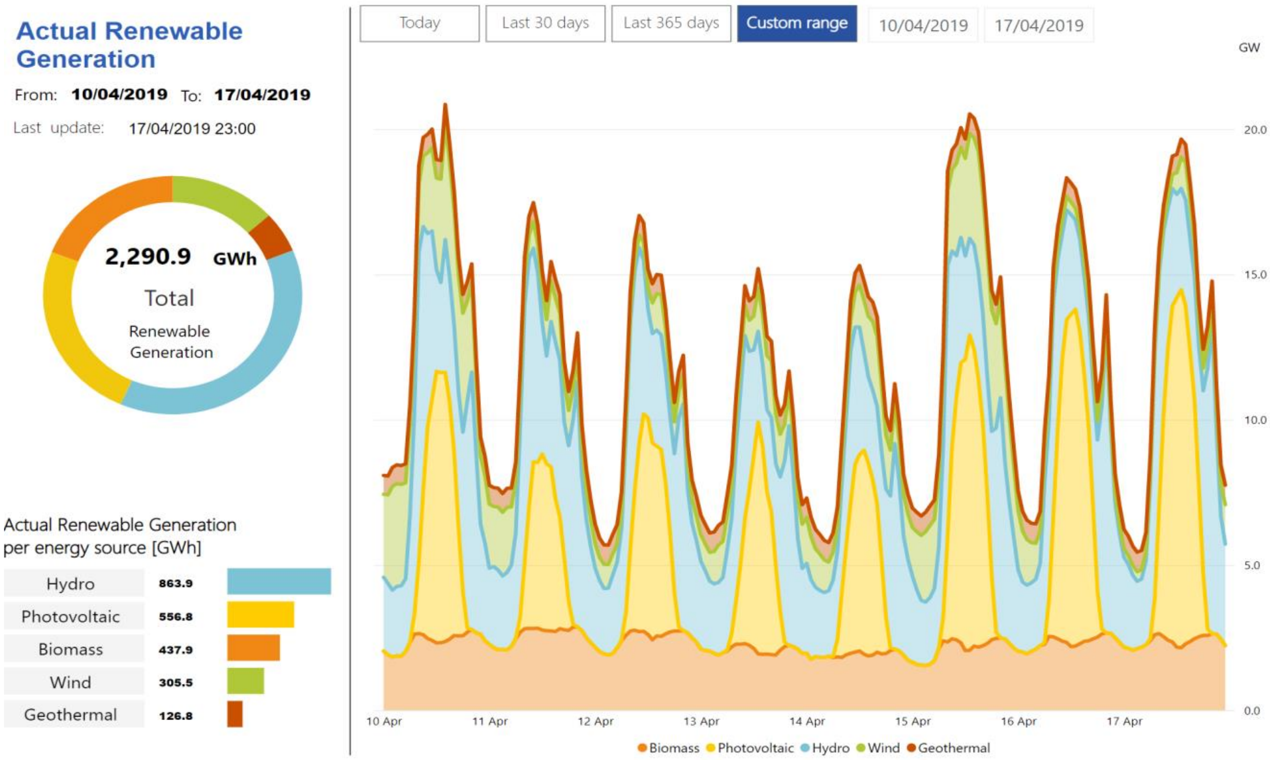

Figure 1.5 – Renewable generation in Italy from 10/04/2019 to 17/04/2019

Nowhere is this more striking than in Europe, where a few CCGT plants have closed. This includes Eon's Irsching CCGT plant in southern Germany, despite the fact that it was the world's most efficient power plant in the past with a 60.8 % rating. The plant was designed for baseload use, and its high combustion temperature and pressures did not allow efficient operation in conjunction with intermittent renewable flows.

To realize of how much a plant is run during the year the best indicator to look at is the number of equivalent hours, that are defined as follow:

$$h_{eq} = \frac{Power\ supplied\ to\ the\ net\ [MWh]}{Nominal\ power\ of\ the\ plant\ [MW]}$$

From Table 1.1 we can calculate the mean equivalent hours of all CCGT plants used only for electricity production, obtaining a value of 2600 equivalent hours, quiet a significant number to understand the direction of the market in relation to these expensive power plants.

In Table 1.2 are reported the installed power, energy injected in the net, the equivalent hours and mean electrical efficiency of the most warning Italian non cogenerative plants related to 2019.

	Inst. Power [MW]	*Gen Power [GWh]*	*Eq. Hours*	*Ren % (LHV)*
A2A Cassano d'Adda	*760*	*1318*	*1734*	*51.5*
A2A Gissi	*840*	*1126*	*1341*	*53.6*
Sorgenia Modugno	*800*	*1364*	*1706*	*53.2*
Torrevaldaliga Sud	*1200*	*1777*	*1480*	*49.8*

Table 1.2 – Warning plants generation, Italy

These plants are working for most of the time at partial load. Obviously not all CCGTs work for such a low number of equivalent hours, but many of them are reducing year after year the time they're turned on and they're increasing the time they work at minimum load.

This brings the necessity of moving the production of electricity towards different plants, like gas turbine cycles, that are much more flexible and less expensive, which means they need to work for less hours to be paid back.

In Italy gas turbines in simple cycle configuration don't contribute that much to power generation. In fact, as we can see from Table 1.3, there are 49 gas turbines used only for electricity production, that in 2019 have produced 1,27 TWh of electricity, only 0,4% of electricity internal production, and there are 77 gas turbines used for cogeneration, that have produced 5,27 TWh of electricity, only 1,8% of the production.

	% Production	*Power [TWh]*	*Installed power [MW]*
GT electricity	*0.4*	*1.27*	*2609*
GT cogenerative	*1.8*	*5.27*	*318*

Table 1.3 – Generation data of CCGT of 2019, Italy

Gas turbines are still not so present in our country because on a simple cycle configuration efficiency are still limited and because there are already many CCGTs that contribute to electricity generation. In the future their number may grow as they will be installed for peak generation, probably in more efficient configurations.

2. State of the art of gas turbines

Gas turbines convert natural gas or other liquid fuels into mechanical energy. In simple cycle configuration air is drawn in the compressor, compressed, mixed with fuel and burnt in the combustor. The hot gasses released expands through the turbine blades connected to the turbine shaft. The shaft turns thus developing mechanical energy which is converted into electrical energy by the generator

2.1 Gas turbines advantages

These plants are small in size, they have an overall lower weight respect CCGTs and lower initial cost per unit of output. This is mainly because expensive components like the HRSG, the condenser and the steam turbine are no more necessary. These mentioned components can have relevant dimensions and both construction and investment costs. So, these plants can be easily installed within short periods. In addition, they have less auxiliaries.

These configurations have lower operational costs. In fact, they are durable and efficient, with lower number of operational failures or downtime respect other plants. Reliability is high, so they require less maintenance. Lubrification costs are less relevant as the lubrification itself is much simpler.

They facilitate distributed power generation, which means they can be used to generate electricity at or near where it will be used, so they may push the development of microgrids.

They're environmentally friendly as they produce low emissions of pollutants through the excess of air, moreover there is no ash content.

Another interesting aspect is that they are quick-starting and smooth running. The loading up is quite fast, and they're perfect to be coupled with renewable power plants.

They allow higher operational speed and have a very high power to weight ratio. They offer flexibility as the load can be changed easily and quite fast, also the control is quite simple. Moreover, by supplying electricity for power generation, if necessary, they can supply compressed air for another process for example. A wide range of liquid and gaseous fuels including synthetic fuels can be used in these plants.

Another point is that water consumption is less compared to steam power plant, as there is not a condenser.

Moreover, modern gas turbine can be used with a percentage of hydrogen as fuel, and in future they will have the possibility to run fully on hydrogen, important aspect to eliminate CO_2 emissions.

All these pros make gas turbine plants suitable in our context of a sustainable world that base power generation on renewable energy plants which need to be coupled with flexible, cheap and fast starting plants.

On the market nowadays there are many kinds of gas turbines, and some of the most performant and modern ones are reported in this chapter.

2.2 GE 9HA

General Electric is one of the most important producers of gas power systems. For more than 125 years, GE has been delivering innovative products and services that create significant value for power generation customers. To have an idea of how much it is important their work in the power sector, in 2014 they have introduced the highly efficient 9HA gas turbine, that in 2016 has been the first to hit 62,22% efficiency at EDF Bouchain with a 9HA.01, and they're looking forward to 65% for the future.

All around the globe their 9HA turbines are delivering flexible and reliable power to millions of people. It is a high efficiency, air-cooled gas turbine, that today is at the heart of the world's most efficient combined-cycle power plants. There are two available models: 9HA.01 with a nominal power output of 448MW and 9HA.02 with a nominal power output of 571MW, both at 50Hz.

Between the most relevant features GE's H-Class combined-cycle power plant has industry-leading flexibility and full combined-cycle plant load in less than 30 minutes for the hot start, making it a great complement to intermittent renewable sources.

The 9HA's DLN 2.6e combustion system offers a step change in performance, emissions and fuel flexibility. The DLN 2.6e maintains many of the elements of GE's DLN 2.6+ combustion system and introduces advanced premixing for reduced NOx emissions while enabling high plant efficiency. Its advanced premixer enables expanded fuel flexibility to operate on both "rich" and "lean" gaseous fuels, a 50% hydrogen capability with a technology pathway to 100%, a gas turbine turndown to 30% load, and an optional park mode at 7-15% load, further reducing customers' operating costs.

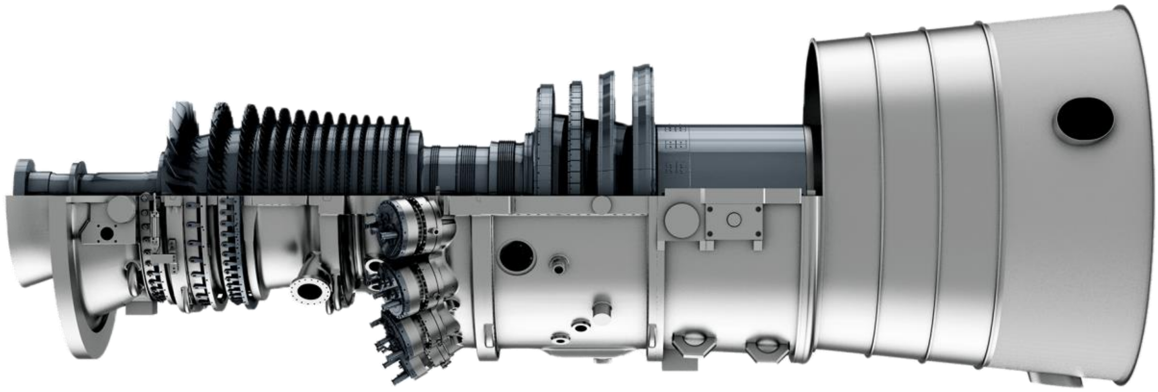

Figure 2.1 – General Electric 9HA

The performances in simple cycle and combined cycle configurations with one or two gas turbines are reported in Table 2.1

Simple Cycle	***9HA.01***	***9HA.02***
SC Net output (MW)	*448*	*571*
SC Net Heat Rate (kJ/kWh, LHV)	*8398*	*8166*
SC Net Efficiency (% LHV)	*42.9%*	*44.0%*
Frequency (Hz)	*50*	*50*
1 x Combined Cycle		
CC Net output (MW)	*680*	*838*
CC Net Heat Rate (kJ/kWh, LHV)	*5661*	*5613*
CC Net Efficiency (% LHV)	*63.7%*	*64.1%*
Plant Turndown – Minimum Load (%)	*33.0%*	*33.0%*
Ramp Rate (MW/min)	*65*	*88*
Startup Time (RR Hot, Minutes)	*<30*	*<30*

2 x Combined Cycle		
CC Net output (MW)	*1363*	*1680*
CC Net Heat Rate (kJ/kWh, LHV)	*5639*	*5598*
CC Net Efficiency (% LHV)	*63.8%*	*64.3%*
Plant Turndown – Minimum Load (%)	*15.0%*	*15.0%*
Ramp Rate (MW/min)	*130*	*176*
Startup Time (RR Hot, Minutes)	*<30*	*<30*

Table 2.1 –9HA series plant performances

2.3 Ansaldo Energia GT36-S5

Ansaldo Energia is a leading international player in the power generation industry, to which it brings an integrated model embracing turnkey power plants construction, power equipment (gas and steam turbines, generators and microturbines), manufacturing and services and nuclear activities.

Ansaldo Energia gas turbines are characterized at the same time by robust and proven design and advanced technology, featuring high performance and low environmental impact, high flexibility and reliability, thus being suitable for several applications and environmental conditions as well as fuel diversification.

Their current production includes E, F and H class technology, ranging from 80MW to 538 MW (ISO Power).

Optimized maintenance plans, low lifecycle costs, easy and reliable operation are some of the key factors for selecting Ansaldo Energia gas turbines, as they affirm in their catalogue.

Built on the evolution of several generations of proven technology and on the GT26 excellence, the GT36 gas turbine offers high efficiency at full and part load, low emissions, a high turn-down capability and high fuel flexibility. Entering in the very large class, the GT36 has been designed to serve evolving customer needs by reducing cost of electricity and CO_2 emissions, increasing operational flexibility and offering outstanding serviceability, therefore ensuring the highest dispatch and consequently the best return on investment.

The GT36 customer's benefits include:

- Fast start and fast ramp

- High part load efficiency
- Optimized part load mode
- High turndown capability with low fuel consumption, providing high reserve power
- High fuel flexibility
- Performance optimized and extended lifetime operation mode to balance power needs with cost saving

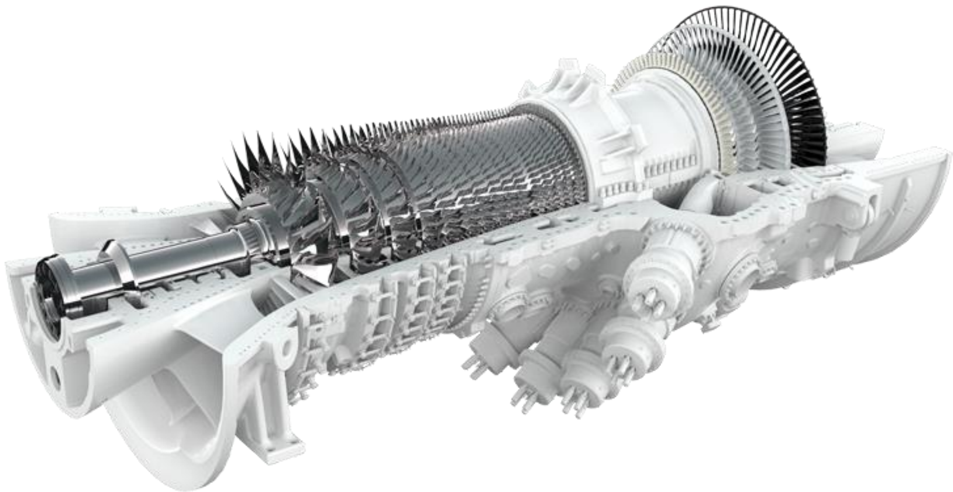

Figure 2.2 – Ansaldo Energia GT36-S5

This machine is characterized by an air cooled 4 stages turbine and between the most interesting features there are a double layer thermal barrier coating on front stages and 3D blade core design to improve cooling air effectiveness. There are 4 variable inlet guide vanes that enhance part load performance, a two stages burner to reduce emissions, modular engine access and turnable vane carriers to access blades and vanes for maintenance.

The performances in simple cycle and combined cycle configurations with one or two gas turbines are reported in Table 2.2.

Simple Cycle ISO Conditions	
Net Power Output (MW)	*538*
Frequency Hz	*50*
Efficiency (%, LHV)	*42.8 %*
Exhaust Mass Flow (kg/s)	*1020*
Exhaust Gas Temperature (°C)	*624*
NOx Emissions (mg/Nm³)	*< 50*
CO Emissions (mg/Nm³)	*< 10*
1 x Combined Cycle	
Net Power Output (MW)	*760*
Efficiency (%, LHV)	*62.6 %*
2 x Combined Cycle	
Net Power Output (MW)	*1525*
Efficiency (%, LHV)	*62.8 %*

Table 2.2 –GT36-S5 plant performances

2.4 Siemens SGT5-9000HL

Siemens Energy is one of the main producers of power products. They work with their clients and partners on the energy systems of the future, supporting the transition towards a more sustainable world. With a wide portfolio of products, solutions and services, Siemens Energy operate across all the energy chain, from generation, to transmission and storage. In fact, for example, they produce steam and gas turbines, hybrid power plants working with hydrogen, generators and transformers.

Depending on the needs they have a wide portfolio of gas turbines in terms of efficiency, reliability, flexibility and environmental compatibility. Siemens heavy-duty gas turbines are robust and flexible engines, designed for large simple or combined cycle power plants. They are suitable for peak, intermediate, or base load duty, as well as for cogeneration applications. The biggest and most efficient one for power generation on a large scale is the SGT5-9000HL, in Figure 2.3.

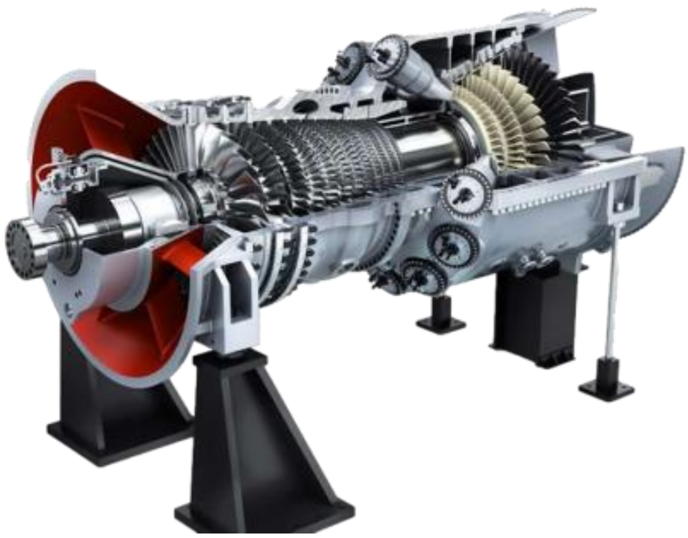

Figure 2.3 – Siemens SGT5-9000HL

Siemens HL-class gas turbines are derived from proven H-class technology in an evolutionary development step, the next generation of advanced air-cooled gas turbines uses a series of new, but already tested technologies like super-efficient internal cooling features for blades or vanes and an advanced combustion system to increase firing temperature.

On their catalogue they report many attributes that make this machine attractive. The gas turbine is capable of fast cold starts and hot re-starts due to the light and stiff rotor with internal cooling air passages and free thermal expansion of rotor and casing parts during transients. The advanced can-annular combustion system with dual fuel capability allows for higher firing temperatures and more operational flexibility. All turbine blades and vanes are equipped with an innovative multi-layer thermal barrier coating. This leads to higher combined cycle efficiency due to less cooling air consumption and reduced operational costs due to higher blade robustness. All rotating compressor and turbine blades can be replaced without rotor lift or rotor de-stacking, to minimize maintenance costs. In conclusion there is the possibility to choose between a 50 Hz or 60 Hz configurations.

The performances in simple cycle and combined cycle configurations with one or two gas turbines are reported in Table 2.3.

Simple Cycle ISO Conditions	
Net Power Output (MW)	*593*
Frequency Hz	*50*
Efficiency (%, LHV)	*> 43 %*
Exhaust Mass Flow (kg/s)	*1050*
Exhaust Gas Temperature (°C)	*670*
NOx Emissions (ppmvd)	*2*
CO Emissions (ppm)	*10*
1 x Combined Cycle	
Net Power Output (MW)	*880*
Efficiency (%, LHV)	*> 64 %*
2 x Combined Cycle	
Net Power Output (MW)	*1760*
Efficiency (%, LHV)	*> 64 %*

Table 2.3 – Siemens SGT5-9000HL plant performances

2.5 Reference plants

The first simulation phase has been related to reproducing the GE9HA.02 in the GS Software, in both combined cycle and simple cycle configuration, to have two reference plants that have been used to compare performances of the proposed gas turbines based on hydrogen and to extrapolate the geometry parameters used in the off-design study, in which the fuel has been changed from natural gas to hydrogen.

2.5.1 GS description

GS Software, which stands for Gas-Steam simulation code, is a program developed by Politecnico of Milan for the thermodynamic simulation of plants used for the power generation, like gas cycles, steam cycles, combined cycles, fuel cells etc. The peculiarity of this program is its modular structure, allows the thermodynamic analysis of a wide variety of energy systems. There is a number of pre-set components that the user can connect as he/she desires to realize the plant or cycle that have to be simulated. The pre-set components are:

0. Pump
1. Compressor
2. Combustor
3. Gas turbine (model 0D)
4. Heat exchanger
5. Mixer
6. Splitter
7. HRSC (Heat Recovery Steam Cycle)
8. Oxygen separation plant (simple model)
9. Shaft
10. Saturator
11. Chemical converter
12. Solid Oxides Fuel Cell (SOFC)
13. Intercooled compressor
14. Steam cycle
15. Gas turbine (model 1D)
16. Molten Carbonate Fuel Cell (MCFC)
17. H2 membrane WGS
18. Multiflow heat exchanger
19. Oxygen separation plant (advanced model)
20. Steam compressor / expander
21. PEM fuel cell

For each iteration, the program solves a system whose equations are the mass and energy balance of each component, following specific correlations for each of them. What we need to know at the end of the iterative process are the thermodynamic conditions of each point of our plant or cycle. Advanced methods of calculation are used because, most of the times, the group of equations is very complex and not linear. More detailed information about assumptions and numerical methods can be found in the GS manual.

2.5.2 Composition and properties

In this kind of simulations, it’s very important to define the flows entering the cycle, in this case being air in ambient conditions, natural gas or hydrogen.

In the GS software the air composition is already defined as in Table 2.4, where also pressure, temperature and absolute humidity are reported. It is also reported the reference mass flow, which is the mass flow entering in the compressor. These assumptions will be the same for all the simulations, so they will not be reported in each step.

In Table 2.5 the molar composition of the natural gas used as fuel in the reference plants. For the other cases 100% hydrogen will be considered.

	p_{amb} [bar]	T_{amb} [°C]	AH [$kg_{water}/kg_{dry\ air}$]		m_{amb} [kg/s]
	1.01325	15.0	0.065		1000.0
	N_2	CO_2	H_20	O_2	Ar
Molar fraction %	77.28	0.03	1.03	20.73	0.92

Table 2.4 – Ambient air properties

	CH_4	C_2H_6	C_3H_8	C_4H_{10}	C_5H_{12}	CO_2	N_2	LHV [MJ/kg]
Molar fraction %	89	7	1	0.1	0.01	2	0.89	46.481

Table 2.5 – Natural gas composition

2.5.3 CCGT reference plant

In Figure 2.4 is reported the scheme used in the GS Software for the CCGT reference plant.

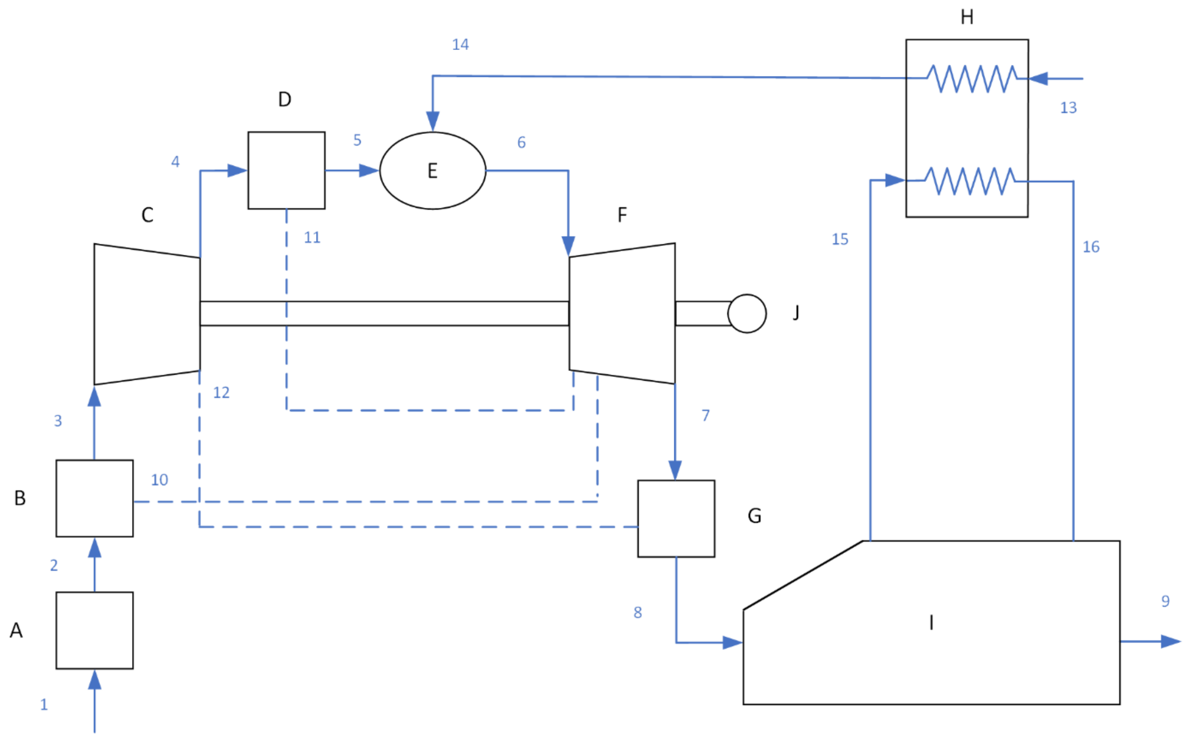

Figure 2.4 – GS scheme of CCGT plant

Component A represents the air filter able to avoid undesired elements from entering inside the compressor. It is associated with a pressure loss $\Delta p/p$.

Components B and D are not real components, but splitters representing the air taken in the compressor to cool down the turbine.

Component C is an 8 stages compressor. The polytropic efficiency is calculated as function of the Size Parameter SP, determined assuming an isoentropic enthalpy drop constant for each stage and equal to the maximum value. The compression ratio is imposed. It is considered a leakage flow, proportional to the entering flow through the coefficient *lk*.

Component E represents the combustor, whose thermal efficiency and pressure drop are imposed.

Component F is a 4 stages turbine, developed in the GS software through a modern 1-D model. The first three stages are cooled down thanks to the flows coming from components B and D coming from open loop circuits. Mass flows used to cool blades are calculated considering the maximal temperature reached by the metal. The geometry of the stages is mainly based on the similitude rules, but some parameters are determined following the convergence variables (will be soon presented).

Component G represents a mixer allowing to mix the exhausted gasses coming from the turbine and the leakage flow coming from the compressor. In this component heat losses at turbine exit are concentrated through a temperature difference.

Component H represents a counter flow heat exchanger used to heat up the fuel before it is burnt. The heat is taken from a hot water flowing from the HRSC, which is calculated in function of the mass flow of fuel.

Component J represents the shaft and generator of the turbogas, working at imposed rotational velocity and electrical and mechanical efficiency.

Component I represents the HRSC, which stands for Heat Recovery Steam Cycle, whose scheme is presented in Figure 2.5. The Rankine cycle is a three level of pressure cycle, the intermediate one with re-heating phase. The heat exchange takes place with a thermal efficiency imposed, it is set at minimum temperature of exhausted gasses at the exit of HRSC and the pressure losses of the gas side. All design conditions are imposed, some of them are reported in Table 2.6.

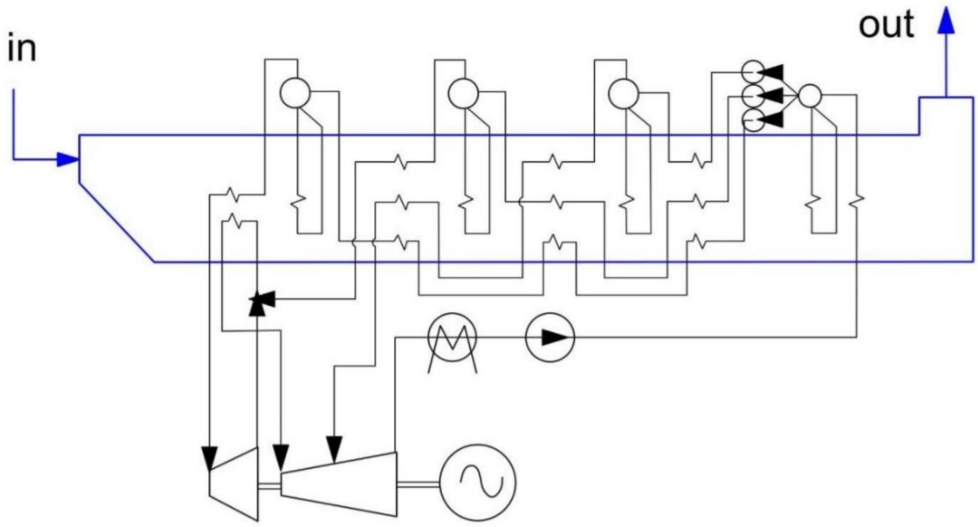

Figure 2.5 – HRSC scheme

Air Filter		***Fuel Heater***	
$\Delta p/p$	*0.007402*	*ε*	*0.9*
Compressor		*ε_{th}*	*0.03*
Compression ratio	*23.6*	*$T_{fuel,\ out}$, °C*	*220*
$\Delta h_{is,\ max\ stage}$ kJ/kg	*25*	*$\Delta p/p$*	*0.000*
lk	*0.004*	***Shaft and Generator***	
η_{org}	*0.99865*	*Velocity of rotation, rpm*	*3000*
Combustor		*Electrical efficiency*	*0.987*
η_{comb}	*0.997*	***HRSC***	
$\Delta p/p$	*0.03*	*η_{th}*	*0.997*
Turbine		*$T_{gas,min}$, °C*	*50*
N. stages	*4*	*Pressure levels, bar*	*185/43/6*
N. cooled stages	*3*	*P_{cond}*	*0.04*
p_{out}, Pa	*104825*	*ΔT_{pp}, °C*	*9*

TIT, K	*1823.15*	*ΔT_{ap}, °C*	*25*
Mach number at turbine exit	*0.55*	*ΔT_{sc}, °C*	*10*
Stanton coefficient	*0.28*	*$T_{max\ vap}$, °C*	*605/603/300*
Polytropic efficiency	*0.93*	*Electric efficiency*	*0.987*
		Velocity of rotation, rpm	*3000*

Table 2.6 – Design conditions relative to CCGT reference plant

The last fundamental part of the GS file is linked to the convergence variables, some laws which link together different single variables. To understand the correct way these laws are assigned, it is important to distinguish dependent and independent variables. A variable is independent if, fixed in input, its value is not changed during the calculation of the program. A variable is dependent if its value is determined during the calculation process.

Heat losses at turbine exit are imposed in component G through a temperature drop, as an independent variable. They're equal to one third of the thermal losses of the combustor.

Peripheral velocity at the first rotor of the turbine is varied to reach the pressure imposed at turbine exit.

Temperature at point 6, which is the TIT, has to converge to the assigned value.

Flow coefficient at turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage is varied to reach the desired h/D_m of 0.3 at turbine exit.

In Table 2.7 are reported the performances of the CCGT reference plant as results of the simulation in the GS Software.

Gas turbine	*Net electric power, MW*	*536.64*
	Mass flow, kg/s	*1000.0*
	Electric specific work, kJ/kg	*536.6*
	GT electric LHV efficiency, %	*42.88*

Power plant	*n. of gas turbines*	*1*
	Steam cycle net power, Mwe	*249.81*
	Overall net power, Mwe	*786.45*
	Electric specific work, kJ/kg	*786.4*
	Net electric LHV efficiency, %	*62.85*

Table 2.7 – CCGT reference plant performances

2.5.4 SCGT reference plant

In Figure 2.6 is reported the scheme used in the GS Software for the simple cycle gas turbine reference plant.

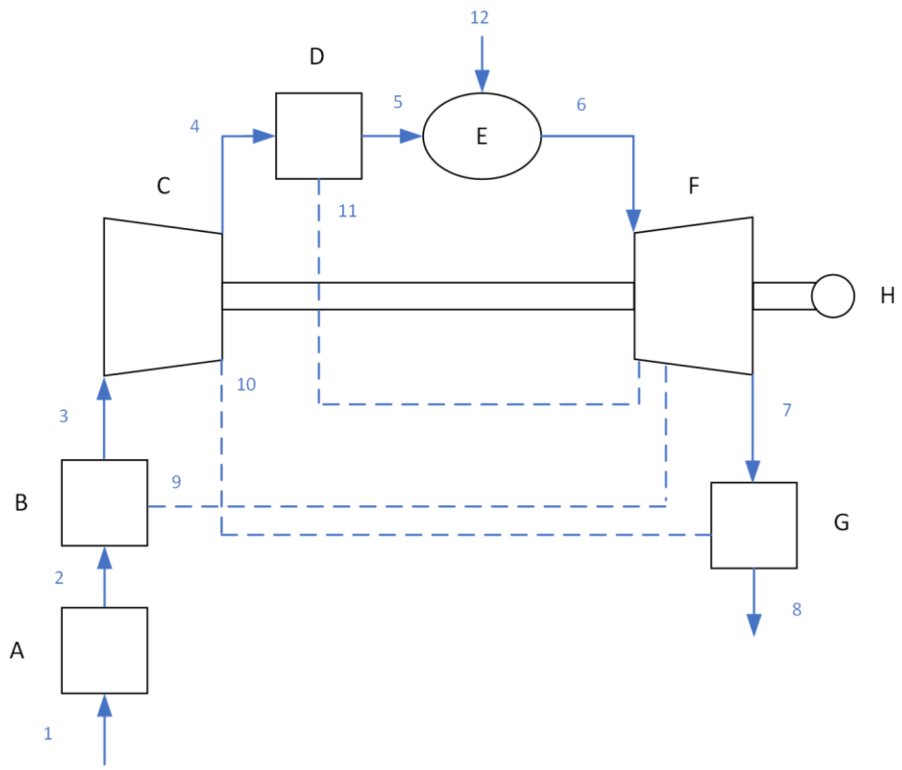

Figure 2.6 – GS scheme of SCGT plant

As we can figure out also looking at the GS scheme, the simple cycle gas turbine based on natural gas is actually the same plant, but without the HRSG and the heat exchanger used to pre-heat the fuel. The consequence is that the little quote of enthalpy of the exhaust gasses used to pre-heat the fuel is not recuperated. So, the fuel that in the reference plant arrived at the combustor at 220°C now is burnt at the ambient temperature of 15°C. Moreover, the elimination of the HRSG will eliminate the pressure losses of the exhaust gasses passing inside this component, reducing the turbine exit pressure, allowing to expand the working fluid a bit more. The pressure drop at the turbine outlet passes from 3500 Pa to 600 Pa, according to existing plants data.

All the component descriptions in chapter 2.5.3 are valid also for this configuration, as they have been maintained the same.

Also the convergence variables are the same ones of the reference plant in the gas side, explained in Chapter 2.5.3. The only changing condition is the one connected to the exit pressure of the turbine, which is now reduced by the elimination of the HRSG, as mentioned before.

In Table 2.8 are reported the performances of the simple cycle reference plant as results of the simulation in the GS Software.

Net electric power, MW	*543.39*
Mass flow, kg/s	*1000.0*
Electric specific work, kJ/kg	*543.4*
GT electric LHV efficiency, %	*43.04*

Table 2.8 – SCGT reference plant performances

3. Hydrogen

Between the elements that compose our planet Earth, hydrogen is the lightest and most abundant of them. The molecule of hydrogen is composed by two atoms and in an atmosphere reach in oxygen it burns almost in the same way as natural gas, but without producing CO_2 emissions. For this reason, hydrogen is one of the clean fuel options that could help to go towards decarbonization.

Hydrogen is also the propellant that feed the nuclear fusion that keep stars burning. So, it is from this extraordinary element that the renewable energy that the Earth receive from the Sun every day is generated. As a fuel hydrogen can also be used in fuel cells to produce electricity without emissions. Like electricity it is considered an energy vector and not an energy source. In fact, they can both be produced from different energy sources and in different ways, and they can be used both in many different applications. It can really be the key factor in the green energy transition. It has a very high energy density, three times petrol's one, but it has a big problem: it is quite difficult to be separated. In fact, it is not available as molecule free in nature, but it is always present with other elements, like in water or hydrocarbons. Hydrogen can be separated but the process requires a big amount of energy, that makes hydrogen expensive and, depending on the separation process, not at all a clean fuel.

Green hydrogen, that is obtained from water through electrolysis using renewable energy, that can really be considered almost a zero emissions fuel with a low environmental impact. Most of the efforts are in this direction, to make it cheaper and more competitive.

Brown hydrogen, which is obtained through carbon gasification, and its production generate CO_2 emissions, in detail more than 20 kg of carbon dioxide per kilogram of hydrogen.

Grey hydrogen, which is obtained through natural gas steam reforming, and its separation produces pollutants, more than 9 kg of carbon dioxide per kilogram of hydrogen.

Blue hydrogen, which is produced with the same methodology of grey hydrogen, but with partial capture, transport and storage of the CO_2 that would be emitted. The production of CO_2 is of 9-10 kg per kilogram of hydrogen, but the emission after the capture process is of 5 kg per kilogram of hydrogen produced.

Rose hydrogen, which is obtained through water electrolysis using nuclear energy. It has a relevant impact on environment due to the radioactive residual waste, although carbon dioxide is not produced.

Unfortunately, nowadays, 95% of hydrogen is produced through methane reforming or carbon gasification, which are the cheapest processes, but which also produce carbon dioxide that is released in atmosphere. For sure, the most promising one is green hydrogen, that would be the most useful of them to reduce CO_2 emissions.

The process of electrolysis to obtain green hydrogen takes place in the electrolyser, and it consists in the decomposition of water into hydrogen and oxygen, with the consequent storage of hydrogen, as schematized in Figure 3.1.

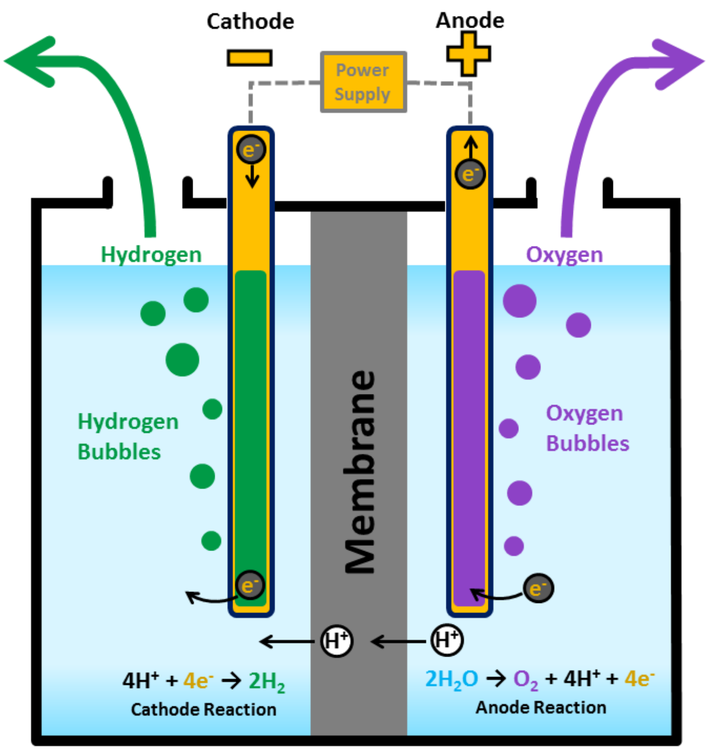

Figure 3.1 – Electrolyser scheme

It requires a relevant quantity of electricity and for this reason, to be a sustainable process, the electricity used in the separation must be produced through renewable plants. Inside the electrolyser water is in contact with two electrodes, a cathode (negative charge) and an anode (positive charge), separated by an electrolyte. There are many electrolysers depending on the electrolyte, for example PEM (Polymer Electrolyte Membrane Electrolysers), Alkaline Electrolysers or Solid Oxide Electrolysers and some others. The electrons current flows in the external circuit. At the cathode the reduction of hydrogen ions takes place and gaseous hydrogen is generated. At the anode occurs the oxidation of water, releasing gaseous oxygen.

3.1 Properties

The properties that contribute to its use as a combustible fuel are its:

- *limits of flammability*: hydrogen has a wide range of flammability in comparison with other fuels. Hydrogen engines, therefore, can be operated more effectively on excessively lean mixtures. As little as 4% hydrogen by volume with air produces a combustible mixture
- *ignition energy*: hydrogen has very low ignition energy, which is the lowest energy required to ignite the flammable material in air or oxygen
- *detonation limits*: hydrogen is detonable over a very wide range of concentrations when confined, however, unlike many other fuels, it is very difficult to detonate if unconfined
- *ignition temperature*: compared to the other fuels, hydrogen has a higher ignition temperature, which is the minimum temperature at which a substance in air must be heated to initiate or cause self-sustaining combustion independent of the heating source
- *flame speed*: hydrogen flame speed is nearly an order of magnitude higher (faster) than that of natural gas
- *high diffusivity*: hydrogen has very high diffusivity. This ability to disperse in air is considerably high and is advantageous for two main reasons: it facilitates the formation of a uniform mixture of fuel and air, and if a hydrogen leak develops, the hydrogen disperses rapidly. Thus, unsafe conditions can either be avoided or minimized
- *very low density*: hydrogen has very low density. This results in a problem: the energy density of a hydrogen–air mixture, and hence the power output, is reduced

3.2 Power to hydrogen

Converting variable renewable energy sources to hydrogen via electrolysis can contribute to power sector transformation in several ways:

- reducing variable renewable energy curtailment, that can be caused by oversupply of renewable electricity in relation to the demand or by local transmission constraints, in relation to insufficient transmission infrastructure
- hydrogen represents a long-term energy storage, for seasonal variations of renewable energy in some specific countries, although nowadays storing power in the form of hydrogen is still not economically convenient

- the electrolyser used to produce hydrogen can be cycled up and down rapidly as a flexible load to balance the grid which is continuously subjected to fluctuations
- clean hydrogen can be used later in other different sectors: as feedstock in chemical industry (for example in ammonia production), as a fuel in transport sector of power generation etc.
- renewable power can be transported over long distances as hydrogen, in region with scarce renewable energy resources

Hydrogen produced during the process of electrolysis can be used as a medium for energy storage and for applications such as producing heat for buildings, refueling fuel cell vehicles and as a source of feedstock for industry, as schematized in Figure 3.2. An important distinction between hydrogen and other forms of energy storage is that hydrogen can be stored and transported through the existing natural gas network. To be more precise, little investment is needed to adapt natural gas infrastructure to transport hydrogen.

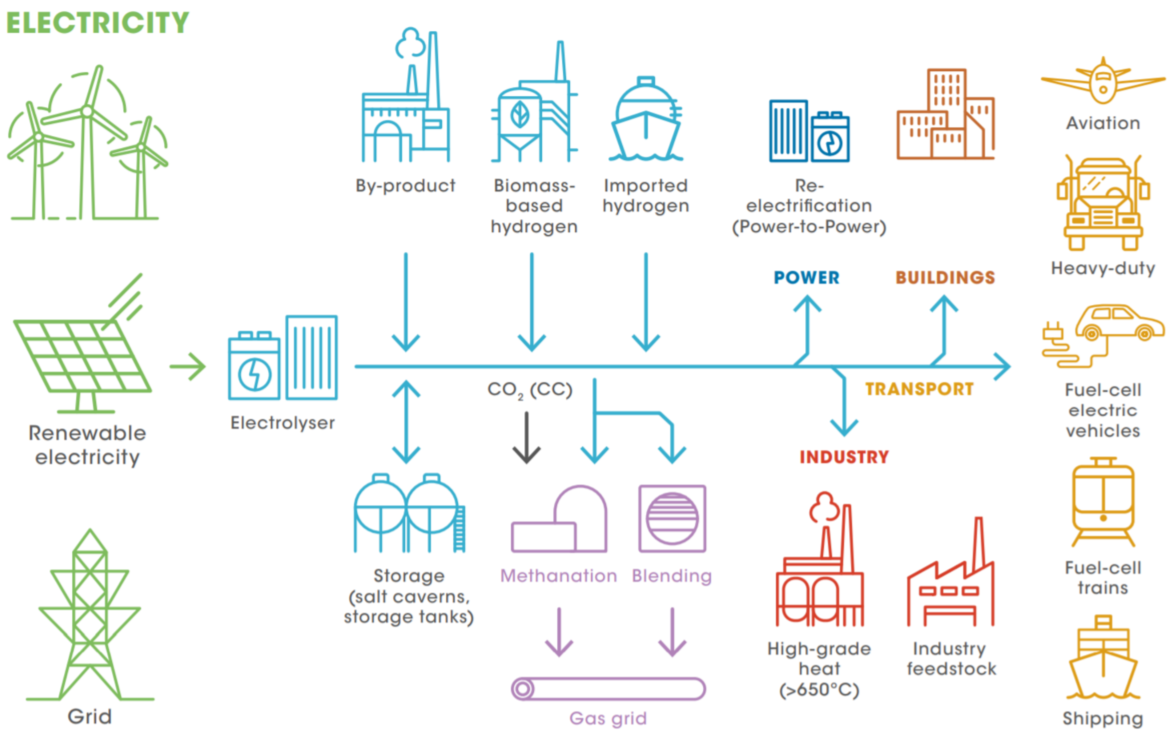

Figure 3.2 – Concept of P2H_2 and the end-use applications of hydrogen

Hydrogen contributes to "sector coupling", on the one hand between the power system and on the other between the industrial, buildings and transport sectors. Sector coupling creates extra loads that represent new markets for hydrogen produced from variable renewable energy sources, furthering the integration of high shares of them in the power system. Well-defined safety standards, appropriate ventilation and leak detection are required to ensure safe operations with hydrogen.

Hydrogen is difficult to store because of its extremely low volumetric density. It is the simplest, lightest and most abundant element in the universe. It is also extremely flammable. All of these qualities combined make its logistics and transportation very complicated.

Hydrogen must become energy dense to be stored. It can be compressed and stored as a gas using high-pressure tanks, or it can be liquefied using cryogenic technology. Hydrogen is typically compressed to between 35 to 150 bar for pipeline transmission whereas the distribution system that provides gas to many end users typically operates at pressures less than ~7 bar. For storage, hydrogen is typically compressed to more than 350 bar. Hydrogen storage and transmission systems may require specialized high-pressure equipment and will require a significant amount of energy for compression. Liquefying hydrogen is even more of a challenge because it condenses from a gas into a liquid at less than -250°C, requiring a significant amount of energy for cooling the gas to this temperature, and special double-walled cryogenic tanks for storage.

3.3 Plant differences

Hydrogen, as a carbon-neutral fuel, is a pre-combustion way to decarbonize a gas turbine. The use of hydrogen as a gas turbine fuel has been demonstrated commercially, but there are differences between natural gas and hydrogen that must be taken into account to properly and safely use hydrogen in a gas turbine. In addition to differences in the combustion properties of hydrogen and natural gas, it's also important to consider the impact to all gas turbine systems, as well as the overall balance of plant. In a power plant with one or more hydrogen-fueled turbines, changes may be needed to the fuel accessories, bottoming cycle components, and plant safety systems. As gas turbines are inherently fuel-flexible, they can be configured to operate on green hydrogen or similar fuels as a new unit, or be upgraded even after extended service on traditional fuels, i.e. natural gas. The scope of the required modifications to configure a gas turbine to operate on hydrogen depends on the initial configuration of the gas turbine and the overall balance of plant, as well as the desired hydrogen concentration in the fuel. For these reasons the simulations of the simple cycle and the combined cycle have been executed in two ways for each of them: first as an off design of the gas turbine of the reference case, secondly as a pure design case optimized for the new fuel which is hydrogen.

The combustor needs to have some characteristics to be run on hydrogen, and there is more than one possible fitting solution:

- Combustion systems with diffusion flames and nitrogen or steam dilution are state-of the art systems which can handle up to 100% vol. hydrogen. Nevertheless, these systems have several disadvantages, including efficiency penalty compared to systems without dilution, higher NOx level compared to lean-premixed technology, higher plant complexity and thereby higher capital and operational costs. In the case of large gas turbines running in a combined cycle or cogenerative configuration, steam dilution performs significantly better than nitrogen dilution with respect to emission reduction and plant efficiency. Overall, premixing of the fuel with the diluent should be preferred, as it allows higher effectiveness in terms of emission reduction.
- The lean-premixed combustor technology has much higher potential. However, this technology is less mature and not sufficiently developed yet with respect to operation on fuels with very high hydrogen contents or even pure hydrogen together with high fuel flexibility

Some of the main problems in hydrogen combustion are:

- *autoignition*: high reactivity of hydrogen inherently increases the auto-ignition risk in the premixing section which needs to be addressed in future combustor developments. This might be of a particular challenge in some systems with very high air inlet temperatures, i.e. in modern high-efficient gas turbines or micro gas turbines due to the recuperator
- *flashback:* burning hydrogen-rich fuels inherently increases flashback risks because of a higher flame speed or a shorter ignition delay time compared to natural gas. This may be a particular challenge in some systems with very high air inlet temperatures
- *thermoacoustics:* compared to natural gas flames, hydrogen flames exhibit significantly different thermoacoustic behavior. This is due to higher flame speed, shorter ignition delay time and distinct flame stabilization mechanisms resulting in different flame shapes, positions and different reactivity. This implies that undesired and dangerous phenomena, such as combustion instabilities, flashback and lean blow out, are likely to occur not only at steady conditions, but also more dangerously during transient operation
- *higher flame temperature, NOx emissions*: the higher adiabatic flame temperature of hydrogen will result in higher NOx emissions if no additional measures are undertaken. Some flexibility might be needed on NOx limits in future, in order to enable the decarbonization. It will be particularly a challenge to achieve even stricter NOx-limits foreseen in the future

3.4 Hydrogen SC and CCGT

As for the reference case based on natural gas, the first simulations hydrogen-based in the GS Software have been related to the simple cycle and to the combined cycle. The simulations have been divided both in two cases: design case and off-design case. The differences of the cases will be highlighted in the following sections.

Being the combined cycle and simple cycle plants the same of the reference plants based on natural gas, the GS scheme of the CCGT cycle and SCGT cycle based on hydrogen, both in design and off-design cases, reported in Figure 3.3 and in Figure 3.4, are the same presented in Chapter 2.5.3 and 2.5.4.

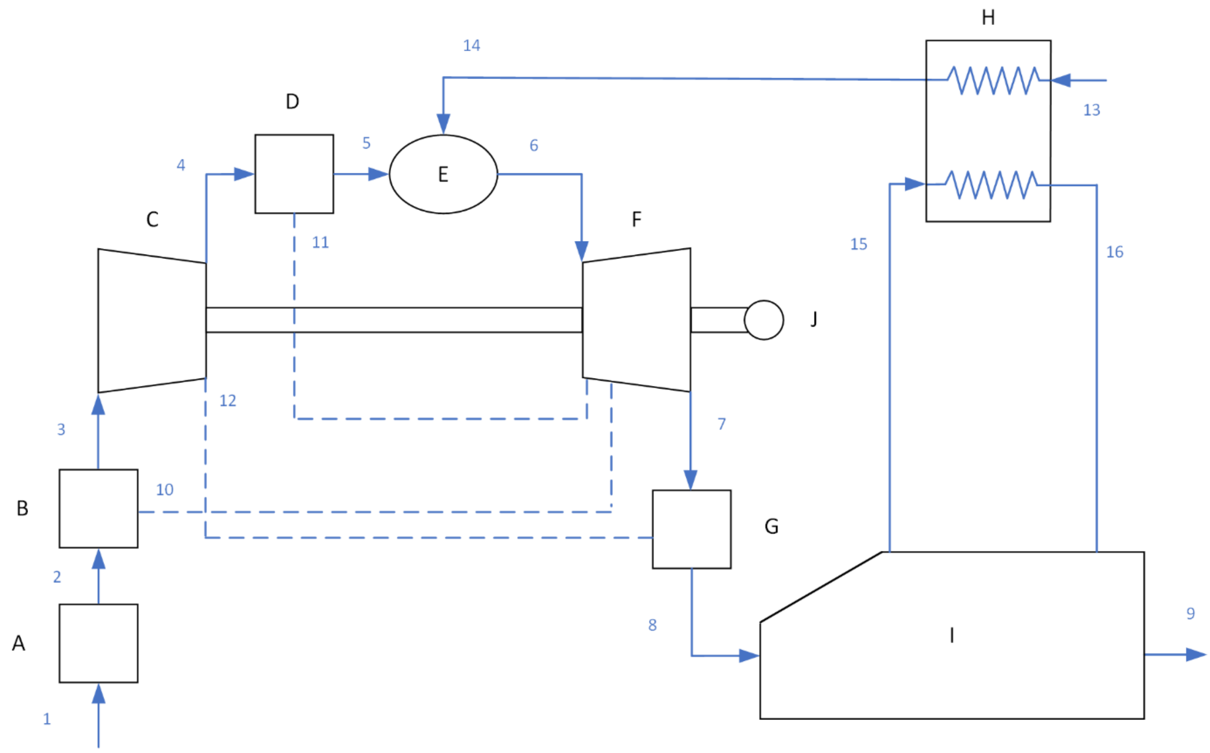

Figure 3.3 – GS scheme of CCGT plant

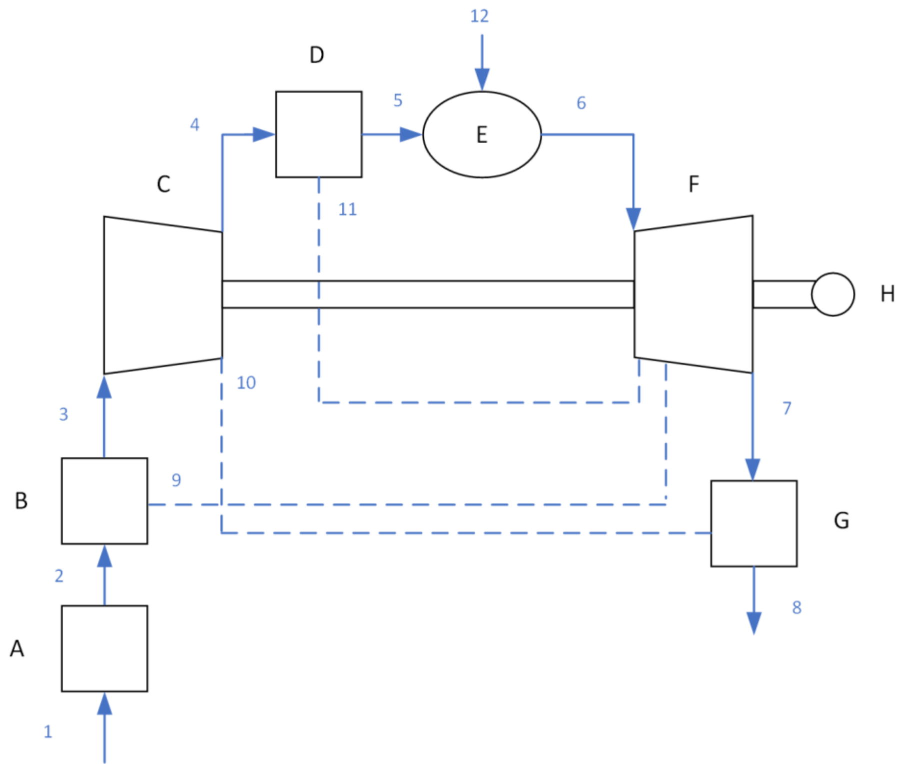

Figure 3.4 – GS scheme of SCGT plant

Also the list and the sequence of the components of the plant is maintained equal and it has already been well described in Chapter 2.5.3. As before the main differences between the two plants are the presence of the HRSG and of the heat exchanger to pre-heat the fuel, that modify the pressure at the exit of the turbine and the temperature at which the fuel is burnt in the combustor.

3.4.1 Hydrogen CCGT and SCGT design case

In the *design case* of CCGT plant the same design conditions of the CCGT reference plant have been imposed, which have been reported again in Table 3.1.

	Air Filter		***Fuel Heater***
$\Delta p/p$	*0.007402*	*ε*	*0.9*
	Compressor	*ε_{th}*	*0.03*

Compression ratio	*23.6*	$T_{fuel, out}$, °C	*220*
$\Delta h_{is, max\ stage}$ *kJ/kg*	*25*	$\Delta p/p$	*0.000*
lk	*0.004*	***Shaft and Generator***	
η_{org}	*0.99865*	*Velocity of rotation, rpm*	*3000*
Combustor		*Electrical efficiency*	*0.987*
η_{comb}	*0.997*	***HRSC***	
$\Delta p/p$	*0.03*	η_{th}	*0.997*
Turbine		$T_{gas,min}$,°C	*50*
N. stages	*4*	*Pressure levels, Bar*	*185/43/6*
N. cooled stages	*3*	P_{cond}	*0.04*
p_{out}, *Pa*	*104825*	ΔT_{pp}, °C	*9*
TIT, K	*1823.15*	ΔT_{ap}, °C	*25*
Mach number at turbine exit	*0.55*	ΔT_{sc}, °C	*10*
Stanton coefficient	*0.28*	$T_{max\ vap}$, °C	*605/603/300*
Polytropic efficiency	*0.93*	*Electric efficiency*	*0.987*
		Velocity of rotation, rpm	*3000*

Table 3.1 – Design conditions relative to hydrogen CCGT plant

In the *design case* of SCGT plant the same design conditions of the SCGT reference plant have been imposed. As for the reference plants, passing to the simple cycle configuration, the little quote of enthalpy of the exhaust gasses used to pre-heat the fuel is not recuperated. So, the fuel that in the reference plant arrived at the combustor at 220°C now is burnt at the ambient temperature of 15°C. Moreover, the elimination of the HRSG will eliminate the pressure losses of the exhaust gasses passing inside this component, reducing the turbine exit pressure, allowing to expand the working fluid a bit more. The pressure drop at the turbine outlet passes from 3500 Pa to 600 Pa, according to existing plants data.

In the *design case* the software has worked following the same convergence variables of the reference plants in both cases.

Heat losses at turbine exit are imposed in component G through a temperature drop, as an independent variable. They're equal to one third of the thermal losses of the combustor.

Peripheral velocity at the first rotor of the turbine is varied to reach the pressure imposed at turbine exit.

Temperature at point 6, which is the TIT, has to converge to the assigned value.

Flow coefficient at turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage is varied to reach the desired h/D_m of 0.3 at turbine exit.

In Table 3.2 and in Table 3.3 are reported the performances of the hydrogen CCGT and of hydrogen SCGT as results of the design simulations in the GS Software.

Gas turbine	*Net electric power, MW*	*562.11*
	Mass flow, kg/s	*1000.0*
	Electric specific work, kJ/kg	*562.1*
	GT electric LHV efficiency, %	*44.26*
Power plant	*n. of gas turbines*	*1*
	Steam cycle net power, MWe	*249.98*
	Overall net power, MWe	*812.09*
	Electric specific work, kJ/kg	*812.1*
	Net electric LHV efficiency, %	*63.94*

Table 3.2 – Hydrogen CCGT design plant performances

Net electric power, MW	*571.03*
Mass flow, kg/s	*1000.0*

Electric specific work, kJ/kg	*571.0*
GT electric LHV efficiency, %	*43.82*

Table 3.3 – Hydrogen SCGT design plant performances

3.4.2 Hydrogen CCGT and SCGT off-design case

In the *off-design case* the main geometrical parameters of the turbine, taken from the natural gas case, have been imposed in the simulation. In the input file the reaction degree of the first stage and its peripheral velocity have been imposed as the same of the natural gas cases. The main job of the GS has been to find the optimal point to have the turbine-compressor matching, that in other words means to find pressure ratio to couple the two machines, and to find the adequate TIT for the imposed cooling channels area.

The variable related to heat losses at turbine exit is the same, they are imposed in component G through a temperature drop, as an independent variable. They're equal to one third of the thermal losses of the combustor.

Flow coefficient at the exit of the turbine is varied to reach the desired h/D_m of 0.3 at turbine exit.

Temperature at point 6 is the dependent variable that must be appropriate in relation to the cooling flow area of the first stator, the same of the reference case working with natural gas.

Head coefficient is varied to reach the imposed pressure at the exit of the turbine.

h/D_m of the first stator of the reference plant working with natural gas is imposed varying the compression ratio.

The only differences in Table 3.1 are the elimination of the design conditions related to the Mach at turbine exit, the imposed TIT and the imposed pressure ratio. In CCGT off-design:

- Mach from 0.55 in design condition, in off-design is equal to 0.545, very close, still an acceptable value
- TIT from 1823.15 K in design condition, in off-design is equal to 1806.4 K a close and lower value, therefore acceptable
- the pressure ratio from 23.60 in design condition, in off-design is equal to 24.01, a very close value

In SCGT off-design:

- Mach from 0.55 in design condition, in off-design is equal to 0.545, very close, still an acceptable value
- TIT from 1823.15 K in design condition, in off-design is equal to 1805.3 K a close and lower value, therefore acceptable
- the pressure ratio from 23.60 in design condition, in off-design is equal to 24.04, a very close value

In Table 3.4 and in Table 3.5 are reported the performances of the hydrogen CCGT and of hydrogen SCGT as results of the off-design simulations in the GS Software.

Gas turbine	*Net electric power, MW*	*555.07*
	Mass flow, kg/s	*1000.0*
	Electric specific work, kJ/kg	*555.1*
	GT electric LHV efficiency, %	*44.72*
Power plant	*n. of gas turbines*	*1*
	Steam cycle net power, Mwe	*237.41*
	Overall net power, Mwe	*792.48*
	Electric specific work, kJ/kg	*792.5*
	Net electric LHV efficiency, %	*63.85*

Table 3.4 – Hydrogen CCGT off-design plant performances

Net electric power, MW	*562.82*
Mass flow, kg/s	*1000.0*
Electric specific work, kJ/kg	*562.8*
GT electric LHV efficiency, %	*44.26*

Table 3.5 – Hydrogen SCGT off-design plant performances

3.5 Considerations

As anticipated, passing to a plant working only with hydrogen as fuel, it is possible to eliminate 100% of the CO_2 that was produced with the reference plants working with natural gas. So, a relevant aspect to highlight are the avoided CO_2 emissions. Under the hypothesis of full combustion of the fuel we can calculate them as follows in nominal condition, passing from the combustion reactions:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$
$$2C_2H_6 + 7O_2 \rightarrow 4CO_2 + 6H_2O$$
$$C_3H_8 + 5O_2 \rightarrow 3CO_2 + 4H_2O$$
$$2C_4H_{10} + 13O_2 \rightarrow 8CO_2 + 10H_2O$$
$$C_5H_{12} + 8O_2 \rightarrow 5CO_2 + 6H_2O$$

$$\dot{m}_{CO_2,\ avoided} = \frac{\dot{m}_{GN}}{MM_{GN}} \cdot (X_{CH4} + 2 \cdot X_{C2H6} + 3 \cdot X_{C3H8} + 4 \cdot X_{C4H10} + 5 \cdot X_{C5H12} + X_{CO2}) \cdot \cdot MM_{CO2}$$

	$\dot{m}_{GN}$(kg/s)	$\dot{m}_{CO_2,\ avoided}$ (kg/s)
CCGT	26.922	71.295
SCGT	27.162	71.930

Table 3.6 – Avoided CO_2 emissions

In Table 3.7 the most relevant data of the analyzed plants are reported all together.

	CC NG	***CC H₂ des***	***CC H₂ off***	***SC NG***	***SC H₂ des***	***SC H₂ off***
$\dot{m}_{fuel}$ (kg/s)	*26.922*	*10.588*	*10.347*	*27.162*	*10.864*	*10.601*
Q_{in} (MW)	*1251.4*	*1270.1*	*1241.2*	*1262.5*	*1303.2*	*1271.6*

η % (LHV)	*62.85*	*63.94*	*63.85*	*43.04*	*43.82*	*44.26*
η % (LHV) GT	*42.88*	*44.26*	*44.72*	=	=	=
Specific work (kJ/kg)	*786.4*	*812.1*	*792.5*	*543.1*	*571.0*	*562.8*
Specific work (kJ/kg) GT	*536.6*	*562.1*	*555.1*	=	=	=
TIT (K)	*1823.15*	*1823.15*	*1806.4*	*1823.15*	*1823.15*	*1805.3*
Press ratio	*23.60*	*23.60*	*24.01*	*23.60*	*23.60*	*24.04*
TOT (°C)	*638.2*	*636.4*	*619.6*	*630.7*	*629.1*	*611.8*
$T_{out, gas}$ (°C)	*80.2*	*73.1*	*75.5*	*629.1*	*627.5*	*610.2*

Table 3.7 – Relevant data related to CCGT and SCGT plants

Natural gas and hydrogen design cases operate at the same imposed TIT, compression ratio and with the same air mass flow at compressor inlet, although the entering thermal power (Q_{in}) is slightly higher in the hydrogen-based plants. Instead, the hydrogen off-design cases, operating at reduced TIT but higher compression ratio and same air mass flow, have a lower entering thermal power (Q_{in}).

The specific work, so the power output, slightly increases passing from the natural gas case to both the hydrogen design case and the hydrogen off-design case, in both combined cycle and simple cycle configurations.

Both hydrogen design and off-design cases have higher efficiency respect natural gas cases. Gas turbines of off-design cases have higher efficiency respect those of the combined cycles, due to lower entering thermal power and lower TOT (Turbine Outlet Temperature). Also, lower TIT means lower cooling flows, therefore lower losses.

Moreover, in combined cycle configuration the design case has higher efficiency respect the off-design case because in the HSRG hot gasses are cooled down to lower temperature, so more heat is recuperated, and more power is produced in the steam cycle in design case.

In both hydrogen design cases and reference plant cases the gas turbines of the combined cycles produce less power respect the ones of the simple cycles because of the higher pressure at turbine exit due to the presence of the HRSG. On the other side, the efficiency of the gas turbines of the hydrogen combined cycles is higher respect the hydrogen simple cycle ones because the most relevant effect on efficiency is given by the presence of the heat exchanger to pre-heat the fuel. In natural gas cases the increase of specific work is more relevant than the presence of the fuel pre-heater.

4. Intercooling

The difference in the power output and efficiency between CCGT and SCGT is quite relevant as demonstrated in the previous chapters. One of the possibilities to increase the power output and the efficiency of a gas cycle, without the necessity to add a bottoming steam cycle, is through intercooling.

4.1 Thermodynamics

The compressor section of the gas turbine consumes a relevant amount of the power produced by the gas turbine. The compression work required may be reduced by dividing the compression process into two stages, separated by an intercooler, that can be accomplished by a conventional heat exchanger, as depicted in Figure 4.1.

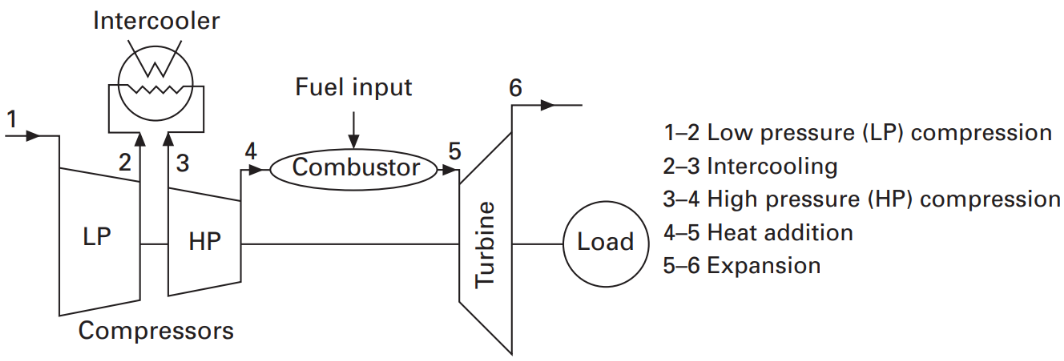

Figure 4.1 – Schematic representation of an ICGT plant

The compressor-specific work requirement to achieve a given pressure ratio is given by:

$$W_{comp} = C_p(T_2 - T_1)$$

which, for an isentropic process, can be expressed as

$$W_{comp} = C_p T_1((\beta)^{\frac{\gamma-1}{\gamma}} - 1)$$

Therefore, reducing T_1 will reduce the compressor work required to achieve a given compressor pressure ratio, β. Thus, intercooling results in a reduction in the compressor work requirement of the HP compressor and hence reduces the overall compression work required to achieve a specific compressor pressure ratio. The consequence is also that the intercooled cycle provides higher specific power, with the same values of β and TIT, and lower temperatures of the air cooling. There are also negative aspects, the combustor inlet temperature is reduced, therefore more fuel is needed to maintain the fixed TIT since energy is lost as heat in the intercooling, as highlighted in Figure 4.2. Moreover, the intercooling process requires a low temperature fluid source, and the cold flow circuit increases auxiliary power requirements. An increase in the efficiency in respect of the simple cycle case is not always true with the same values of β and TIT.

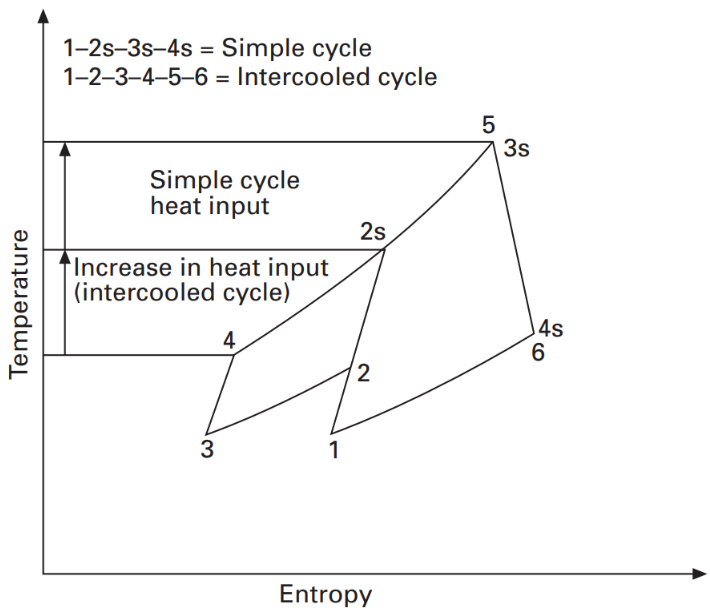

Figure 4.2 – (T,s) diagram of a practical ICGT cycle

4.2 GE LMS100

Many intercooled gas turbines are available in the market. The most famous one is the General Electric LMS100, reported in Figure 4.3. This turbine produces approximately 100 MW at an efficiency of around 43% LHV in open cycle operation. It is currently the largest and most efficient aero-derivative gas turbine. Its intercooled gas turbine system provides rapid startup, with an 8-minute start to full load and emergency ramp speeds of up to 500 MW/minute.

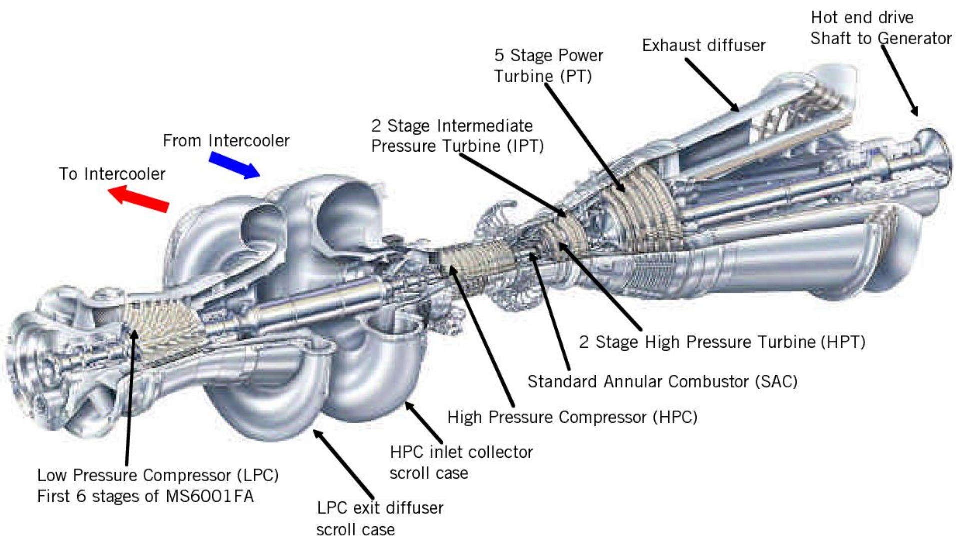

Figure 4.3 – GE LMS100 Gas Turbine

The LMS100 comprises a low-pressure compressor, an intercooler, a supercore and a power turbine. The low-pressure compressor is from the 6FA industrial gas turbine. The supercore (which includes a HP compressor, compressor rear frame, high-pressure turbine and intermediate pressure turbine) is a further development of the LM6000, which in turn is based on the CF6-80C2 and on the CF680E1. Following the intermediate pressure turbine is a 5-stage aerodynamically coupled power turbine (PT) that has been designed specifically for the LMS100.

The compressor discharges through an exit guide vane and a diffuser into an aerodynamically designed scroll case. The scroll case is designed to minimize pressure losses. The air leaving the scroll case is delivered in the intercooler through stainless steel piping. The air exiting the intercooler is directed to the HP compressor inlet scroll case. Like the LP compressor exit scroll case, the HP compressor inlet collector scroll case is aerodynamically designed for low pressure loss.

The combustor can be either a standard annular combustor or an advanced dry low emissions combustor.

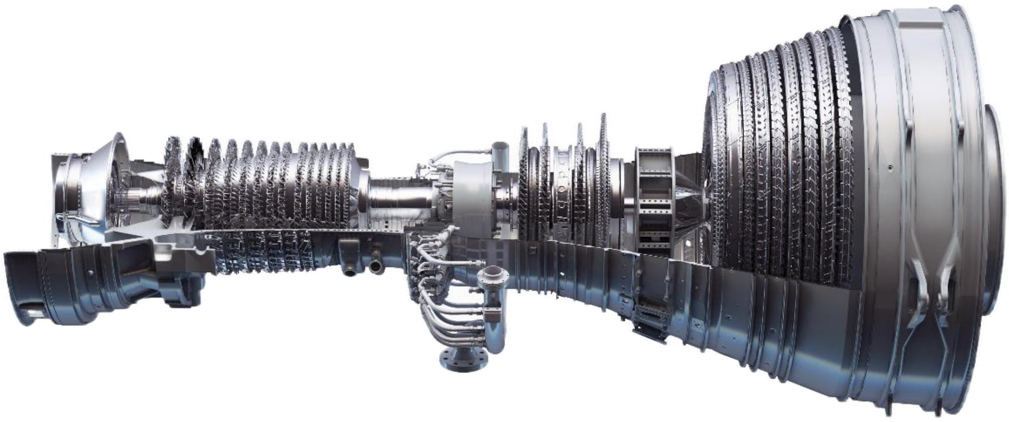

Figure 4.4 – GE LMS100 supercore and power turbine sections

The LMS100 can provide power at part load as efficiently as most gas turbines at full load, and can operate with very little power loss, supporting the grid in times of high demand. Some of the advantages reported on GE catalogue are the extremely high values of reliability, availability and start reliability, each of them higher than 96%.

In Table 4.1 are reported the performances of the GE LMS100.

Simple Cycle	***LMS100 PA+***	***LMS100 PB+***
SC Net output (MW)	*113*	*106.5*
SC Net Heat Rate (kJ/kWh, LHV)	*8371*	*8458*
SC Net Efficiency (% LHV)	*43.0%*	*42.6%*
Frequency (Hz)	*50*	*50*

Table 4.1 – GE LMS100 performances

In this machine the reduced work of compression for the HP compressor allows higher overall pressure ratios, thus increasing the overall efficiency. The cycle pressure ratio is 42:1. The reduced inlet temperature for the HP compressor allows increased mass flow resulting in higher specific power. The resultant lower compressor discharge temperature provides colder cooling air to the turbines, which in turn allows increased firing temperatures at metal equivalent temperatures.

4.3 Intercooling pressure investigation

As for the previous cases the intercooled cycle has been implemented in the GS Software, not only to see the variations in power output and efficiency, but also to find how they change at fixed overall pressure ratio, varying the intercooling pressure. The main design conditions have been maintained the same of the hydrogen design SCGT, while some have been added due to the presence of new components, and they are reported in Table 4.2. The GS scheme is presented in Figure 4.5.

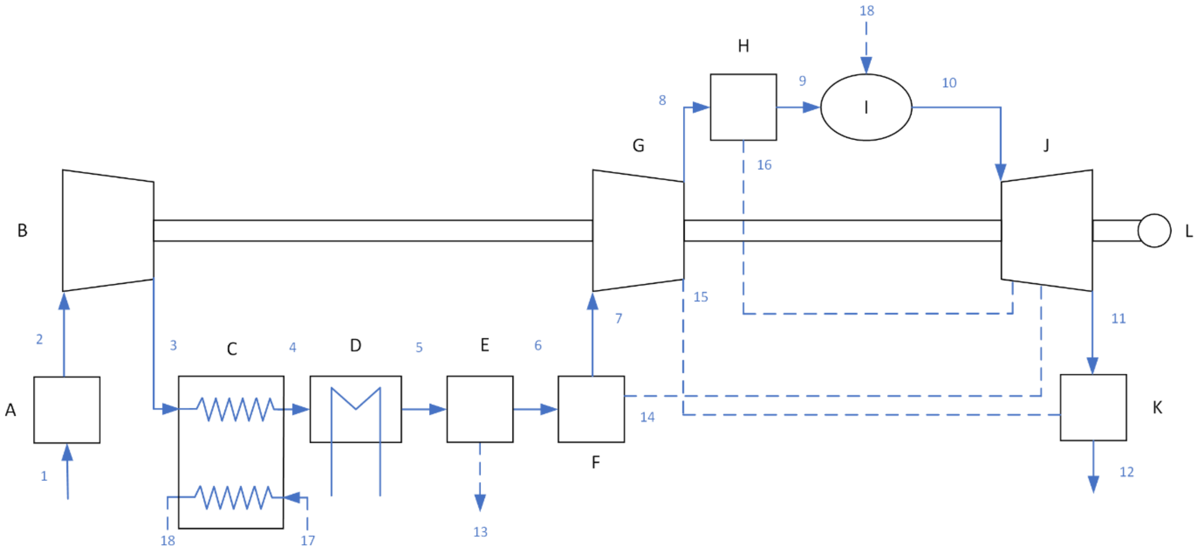

Figure 4.5 – GS scheme of ICGT plant

Respect the simple cycle three new components are added to the plant. The intercooler has been divided into two components, C and D. Component C is a heat exchanger used to pre-heat the fuel (T_{18}) with a temperature difference ($\Delta T_{hot\ end}$) of 30°C respect compressor outlet temperature (T_8), considering a maximum acceptable temperature of 220°C, to avoid the risk of hydrogen cracking. In this manner of the heat taken with intercooling is partly recuperated and not wasted. No pressure losses have been considered in this component. Component D is a mixer in the GS software that represents a heat exchanger used to cool-down air to the design temperature of 30°C. Intercooling pressure losses are concentrated here with a relative $\Delta p/p$ of 2%. It has been implemented as a mixer to avoid the definition of the second flow of the heat exchanger. In the simulation it has been noticed that after the intercooling process some water was present in liquid phase, so it was necessary the addition of component E, a splitter working as a condensate removal.

Air Filter		***Fuel Heater***	
$\Delta p/p$	0.007402	$\Delta T_{hot\ end}$, °C	30
Compressors		ε_{th}	0.03
$\Delta h_{is,\ max\ stage}$ kJ/kg	25	$T_{fuel,\ max}$, °C	220

lk	*0.004*
η_{org}	*0.99865*
Combustor	
η_{comb}	*0.997*
$\Delta p/p$	*0.03*
Turbine	
N. stages	*4*
Cooled stages	*Yes*
p_{out}*, Pa*	*101925*
TIT, K	*1823.15*
Mach number at turbine exit	*0.55*
Stanton coefficient	*0.28*
Polytropic efficiency	*0.93*

$\Delta p/p$	*0.000*
Shaft and Generator	
Velocity of rotation, rpm	*3000*
Electrical efficiency	*0.987*
Intercooler	
$T_{out, IC}$*, °C*	*30*
$\Delta p/p$	*0.02*
No heat losses	

Table 4.2 – Design conditions relative to hydrogen ICGT plant

For the intercooled gas turbine, the convergence variables have been equally maintained as the hydrogen SCGT in the design case.

Heat losses at turbine exit are imposed in component K through a temperature drop, as an independent variable. They are equal to one third of the thermal losses of the combustor.

At the first rotor of the turbine the peripheral velocity is varied to reach the pressure imposed at turbine exit.

The temperature at point 10, which is the TIT, has to converge to the assigned value.

Flow coefficient at turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage is varied to reach the desired h/D_m of 0.3 at turbine exit.

To investigate the effects of the intercooling pressure, the analysis has been performed by considering the same overall pressure ratio of the SCGT of 23.60 and varying the pressure ratio of the LP compressor. The results of the sensibility analysis are reported in Figure 4.6, Figure 4.7 and Figure 4.8.

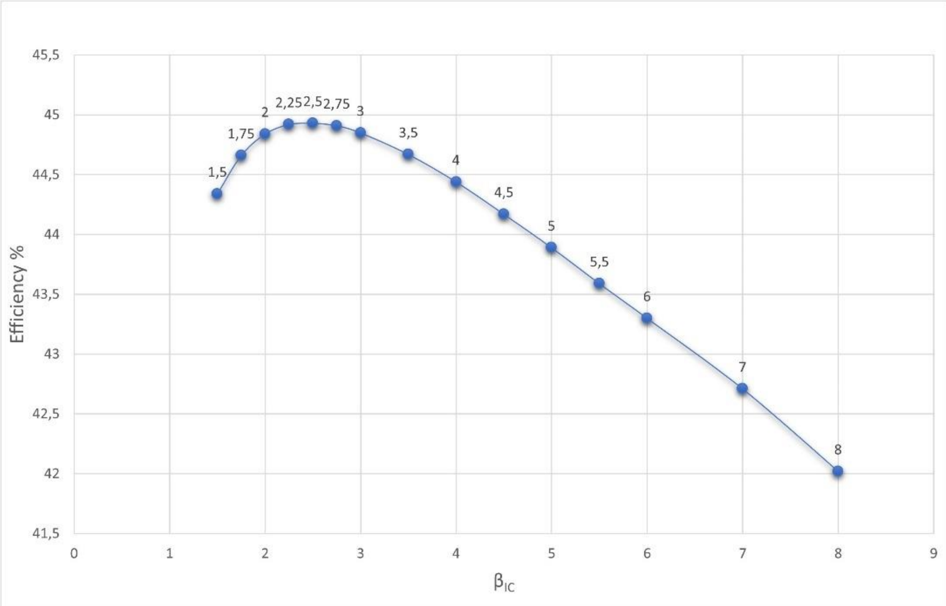

Figure 4.6 – Efficiency, intercooling compression ratio chart of ICGT

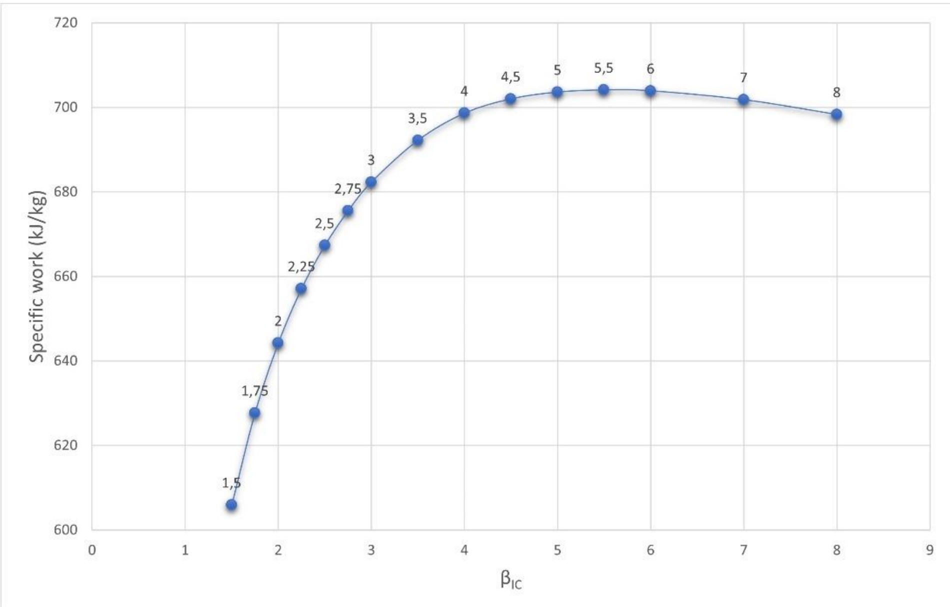

Figure 4.7 – Specific work, intercooling compression ratio chart of ICGT

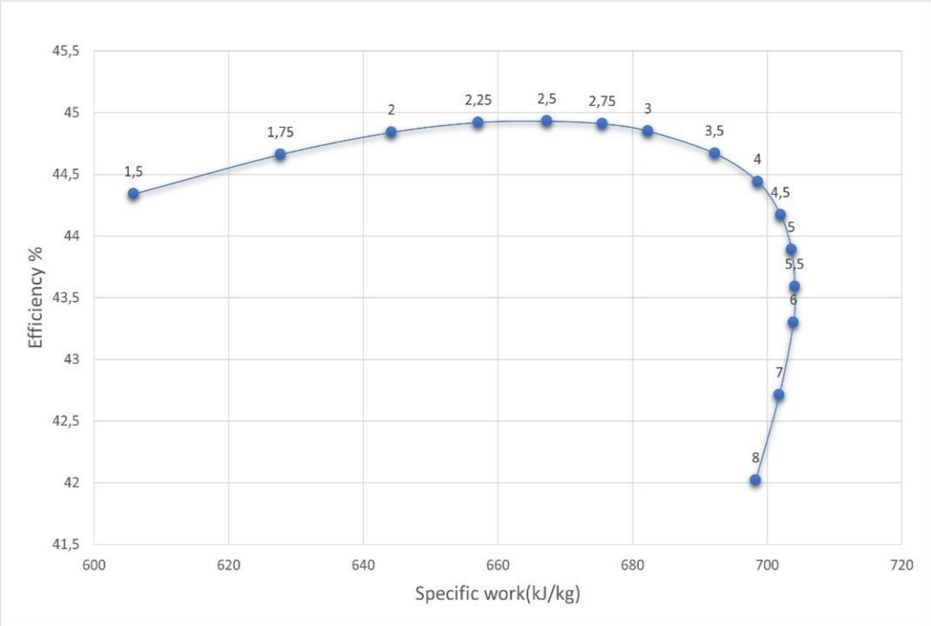

Figure 4.8 – Specific work, efficiency chart of ICGT, varying intercooling compression ratio

The results of the simulations are the same that can be found in literature.

The value of compression ratio that minimizes the compression work and in other words maximizes the specific work of the cycle is found to be around the $\sqrt{overall\ pressure\ ratio}$.

Instead, the value of compression ratio that maximize the efficiency of the cycle is found to be around the $\sqrt[3]{overall\ pressure\ ratio}$.

In the next simulations of this chapter the intercooling compression ratio will be imposed as $\sqrt[3]{overall\ pressure\ ratio}$, because an effect also on efficiency is desired, as the effect on specific work is already given by the intercooler itself.

4.4 Intercooled hydrogen gas turbine

Once the intercooling pressure ratio has been investigated, the next step was to find the optimal overall pressure ratio in order to optimize the performances of the machine and to highlight the main differences in respect of the simple cycle configuration. In this analysis the same GS scheme, design conditions and convergence variables of Chapter 4.3 have been used.

As anticipated, one of the main advantages of intercooling is the reduction of the compression work, with the consequent increase in power output. This variation has been demonstrated also in the GS simulations, as highlighted in Figure 4.9. Moreover, this positive effect becomes more relevant at higher overall pressure ratios.

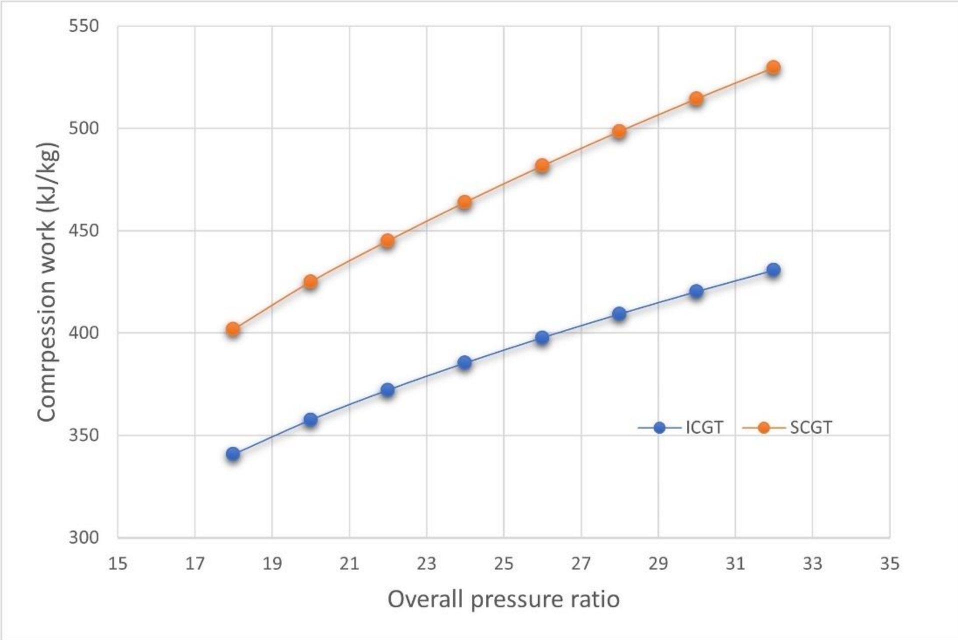

Figure 4.9 – Compression work, overall pressure ratio chart of ICGT and SCGT

Another effect of intercooling is the reduction of the cooling mass flows, thanks to the lower temperature of the air in the HP compressor, with a consequent increase of the working fluid producing power in the turbine. Also this positive effect has been demonstrated in the GS simulations, as reported in Figure 4.10. In the simple cycle configuration higher compression ratios increase the cooling flows because their temperature is higher. The same can be said for the intercooled cycle, although the increase is much smoother. Moreover, the trend is not followed between overall compression ratio of 18 and 20. The difference is caused by the fact that for these values also the stator of the firth stage of the turbine is cooled, while for higher values of compression ratio this is not any more needed, and only three stages of the turbine are cooled down. Also for the cooling mass flows the advantage becomes more relevant at higher overall pressure ratio.

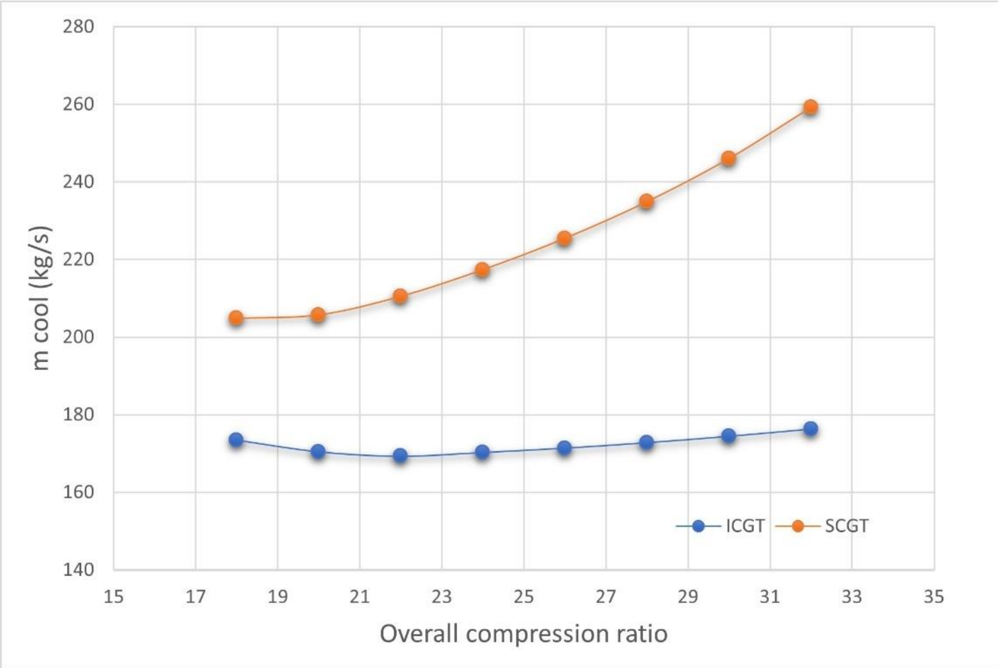

Figure 4.10 – Cooling mass flows, overall pressure ratio chart of ICGT and SCGT

The main negative aspect of intercooling is related to the increase of fuel to obtain the same TIT, being this due to a lower combustor inlet temperature, as demonstrated in Figure 4.11. Increasing the overall compression ratio means increasing the combustor inlet temperature in both intercooled and simple cycle cases, so a reduction of fuel is expected and reported from GS simulations.

Moreover, the higher slope of the SCGT line respect ICGT's one is due to the divergence of the isobar lines in the temperature-entropy diagram, which increases the difference of combustor inlet temperature between the two cases at a given overall pressure ratio.

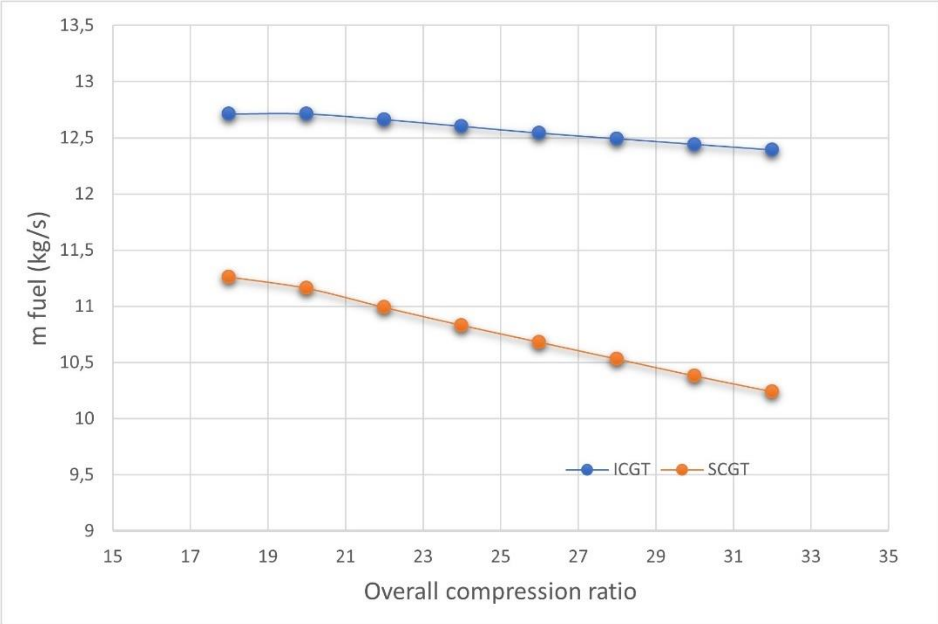

Figure 4.11 – Fuel mass flow, overall pressure ratio chart of ICGT and SCGT

In conclusion, in Figure 4.12 and in Figure 4.13, the trends of specific work and efficiency are reported and highlight the differences between ICGT and SCGT. As anticipated, the great difference stands in the specific work, mainly due to compression work reduction, becoming more important at higher overall pressure ratios. On the other side the addition of the fuel pre-heater, in the intercooled configuration, allows the recovery of a little quote of energy of exhaust gasses. Also because of the choice of the intercooling pressure that maximize efficiency, an increase in respect of the simple cycle configuration is obtained, gaining importance at higher overall pressure ratios. So, the presence of the fuel pre-heater, the reduction of compression work and of cooling flows have a stronger impact on efficiency than the increase of fuel consumption. In addition, the positive effects on efficiency and power output justifies the increase of cost of the plant, represented by the intercooler.

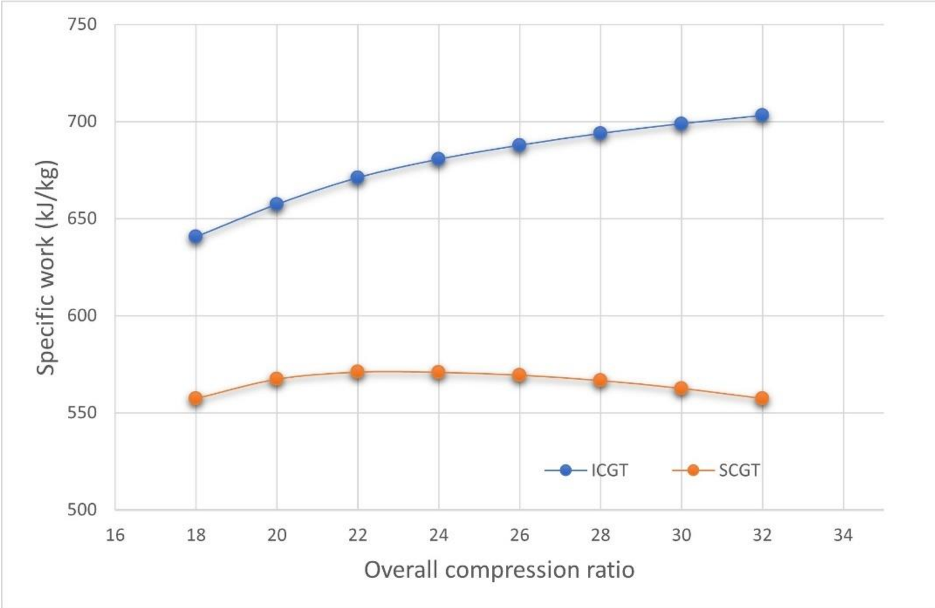

Figure 4.12 – Specific work, overall pressure ratio chart of ICGT and SCGT

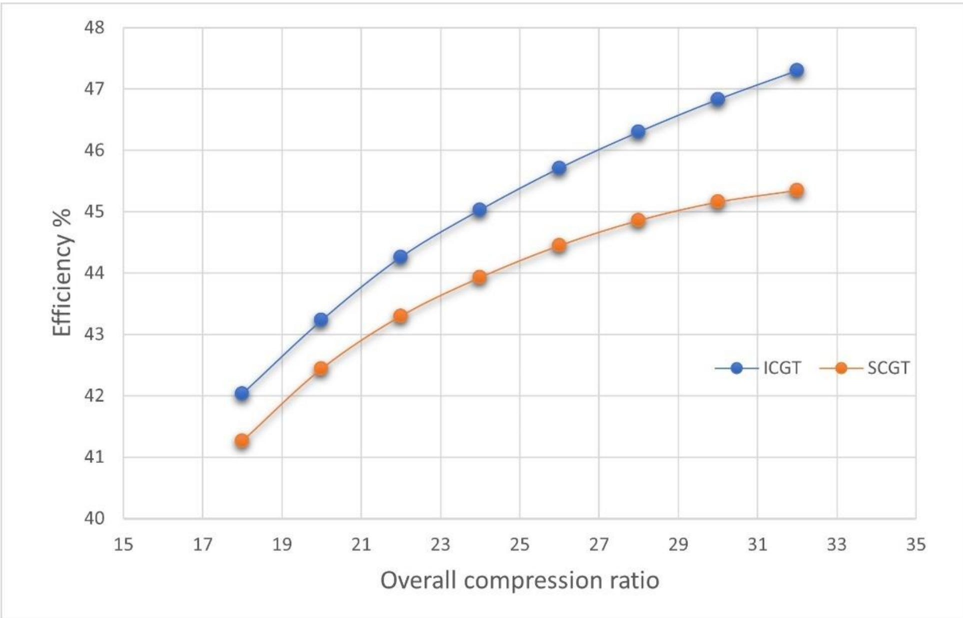

Figure 4.13 – Efficiency, overall pressure ratio chart of ICGT and SCGT

4.5 Twin shaft intercooled configuration

As highlighted in Chapter 4.4 the increase of the overall compression ratio increases the positive effects of intercooling, thus efficiency and power output as well. For this reason, to increase much more the overall compression ratio of the cycle it has been necessary to pass to a twin shaft configuration. This kind of plant would obviously be much more expensive and complex to realize and design.

In Figure 4.14 is reported the GS scheme used in the simulations of the twin shaft configuration.

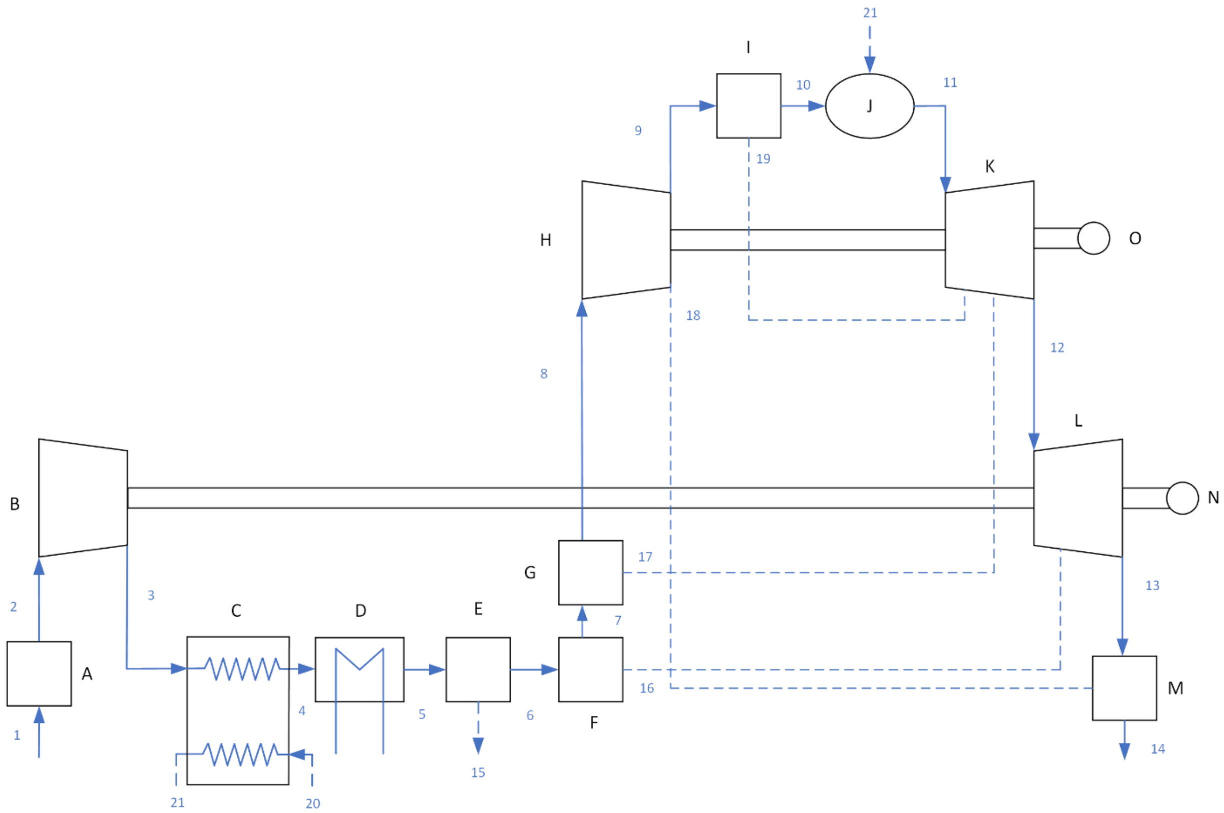

Figure 4.14 – GS scheme of twin shaft ICGT plant

In respect of the single shaft configuration, the expansion now takes place in two different machines. Component K is the HP turbine, connected to the HP compressor through the HP shaft. In the GS scheme the HP generator is present, but a high-pressure speed balance is performed to reduce costs and eliminate its losses, so that in the real plant the HP generator would be eliminated. HP turbine is cooled by flows coming from components I and G, two splitters situated respectively after the HP compressor and before it. Component L is the LP turbine, connected to the LP compressor through the LP shaft, where the generator is also located. It is cooled by the flow coming from splitter F, placed after the intercooler. The other components are the same of the single shaft configuration.

One of the key points of this phase has been the choice of the number of stages of the two turbines and the RPM of the HP shaft. For the choice of the number of stages of the two turbines two possibilities have been tested:

A) 2 stages HP shaft, 3 stages LP shaft
B) 1 stage HP shaft, 4 stages LP shaft

The case A resulted to be more efficient due to a better distribution of the overall compression ratio on the two compressors, resulting in higher efficiencies.

For the design of the HP section, and thus the choice of the RPM, the similitude theory has been exploited. The machine considered is the GE 7E, a 60Hz gas turbine, run at 3600 rpm. It is a smaller machine than the ones considered in Chapter 2, State of the art of gas turbines. In fact, it is characterised by a nominal power output of 89 MW and air mass flows entering the compressor between of 300 kg/s. The RPM has been calculated with the similitude theory maintaining fixed the ratio, scaling the machine:

$$\frac{volumetric\ flow\ (\frac{m^3}{s})}{rotational\ speed}$$

obtaining a value of 4100 rpm for the HP shaft. This procedure is more important to ensure the possibility of the design of the compressor, respect the realization of the turbine which is a less limiting component in the design procedure.

The main design conditions imposed in the twin shaft configuration are reported in Table 4.3.

Air Filter		***Fuel Heater***	
$\Delta p/p$	0.007402	$\Delta T_{hot\ end}$, °C	30
Compressors		ε_{th}	0.03
$\Delta h_{is,\ max\ stage}$ kJ/kg	25	$T_{fuel,\ max}$, °C	220
lk	0.004	$\Delta p/p$	0.000
η_{org}	0.99865	***Intercooler***	
Combustor		$T_{out,\ IC}$, °C	30

η_{comb}	*0.997*	*$\Delta p/p$*	*0.02*
$\Delta p/p$	*0.03*	*No heat losses*	
HP Turbine		***LP Turbine***	
N. stages	*2*	*N. stages*	*3*
N. cooled stages	*2*	*N. cooled stages*	*1*
TIT, K	*1823.15*	*p_{out}, Pa*	*101925*
Stanton coefficient	*0.28*	*Mach number at turbine exit*	*0.55*
Polytropic efficiency	*0.93*	*Stanton coefficient*	*0.28*
HP Shaft		*Polytropic efficiency*	*0.93*
Velocity of rotation, rpm	*4100*		
LP Shaft and Generator			
Velocity of rotation, rpm	*3000*		
Electrical efficiency	*0.987*		

Table 4.3 – Design conditions relative to hydrogen twin shaft ICGT plant

Most of the convergence variables are imposed to the LP turbine, because if the flow at the exit of the HP turbine has still a high energy content, it can be recuperated producing power in the LP's one.

Heat losses at LP turbine exit are imposed in component M through a temperature drop, as an independent variable. They are equal to one third of the thermal losses of the combustor.

Peripheral velocity at the first rotor of the LP turbine is varied to reach the pressure imposed at turbine exit.

Temperature at point 11, which is the TIT, has to converge to the assigned value.

Flow coefficient at LP turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage of the LP turbine is varied to reach the desired h/D_m of 0.3 at LP turbine exit.

The overall compression ratio is imposed as a pressure ratio between point 9 and 2, equal to 60.

In the HP shaft a power balancing is imposed varying the LP compressor compression ratio.

In Table 4.4 are reported the flow properties of each point of the cycle as results of the GS simulations.

Point	T	P	G	Q	Composition (% mol)					
	°C	bar	kg/s	kmol/s	Ar	CO_2	H_2	H_2O	N_2	O_2
1	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
2	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
3	125.0	2.89	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
4	112.4	2.89	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
5	30.0	2.83	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
6	30.0	2.83	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
7	30.0	2.83	967.9	33.55	0.92	0.03	0.0	1.03	77.28	20.73
8	30.0	2.83	913.8	31.67	0.92	0.03	0.0	1.03	77.28	20.73
9	484.3	60.34	909.8	31.53	0.92	0.03	0.0	1.03	77.28	20.73
10	484.3	60.34	805.6	27.92	0.92	0.03	0.0	1.03	77.28	20.73
11	1621.9	58.53	816.4	30.61	0.84	0.027	0.0	18.49	70.50	10.14
12	1116.4	17.58	974.7	36.09	0.85	0.027	0.0	15.83	71.53	11.75
13	466.8	1.02	1006.8	37.21	0.85	0.028	0.0	15.39	71.71	12.02
14	465.8	1.02	1010.8	37.34	0.85	0.028	0.0	15.34	71.73	12.05
15	30.0	2.83	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
16	30.0	2.83	32.1	1.11	0.92	0.03	0.0	1.03	77.28	20.73

17	30.0	2.83	54.1	1.88	0.92	0.03	0.0	1.03	77.28	20.73
18	484.3	60.34	4.0	0.14	0.92	0.03	0.0	1.03	77.28	20.73
19	484.3	60.34	104.2	3.61	0.92	0.03	0.0	1.03	77.28	20.73
20	15.0	70.0	10.8	5.37	0.0	0.0	100	0.0	0.0	0.0
21	95.0	70.0	10.8	5.37	0.0	0.0	100	0.0	0.0	0.0

Table 4.4 – Hydrogen twin shaft ICGT plant flows properties

A little increase in the cooling flows and fuel consumption have been noticed with respect to a single shaft plant, but anyway much lower values than simple cycle configuration, but the most important effect is in the increase of efficiency of almost 5% with respect to the highest-pressure ratios investigated in single shaft configuration. Therefore, this configuration should be interesting with respect to the single shaft one. In table 4.5 are reported the performances of the hydrogen intercooled twin shaft GT plant as results of the simulation in the GS Software.

Net electric power, MW	*679.32*
Mass flow, kg/s	*1000.0*
Electric specific work, kJ/kg	*679.3*
GT electric LHV efficiency, %	*52.32*

Table 4.5 – Hydrogen twin shaft ICGT plant performances

5. Regeneration

A little increase of efficiency has been obtained through intercooling, but there is still an important difference in respect of the one of a CCGT. For this reason, in this chapter the addition of a regenerator has been analyzed.

5.1 Thermodynamics

One of the most relevant losses in gas turbine plants are those related to the energy content of exhaust gasses released in atmosphere. One of the possibilities to reduce this loss and increase the efficiency of the plant is to add a heat exchanger that transfers a part of this heat to the compressor discharge air before combustion takes place, as schematized in Figure 5.1.

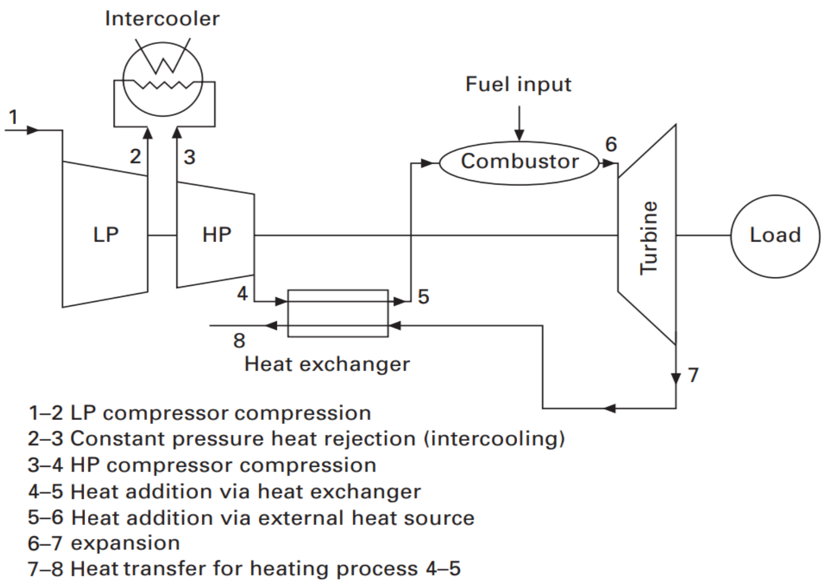

Figure 5.1 – Schematic representation of an ICRGT plant

The heat exchanger must be placed after the compression section because the transferal of heat from the turbine outlet to the compressor inlet would reduce efficiency, as hotter inlet air means more volume, thus more work for the compressor.

In this configuration the goal of the intercooler is to minimize compression work and maximize power output, therefore the best idea is to have the intercooling pressure set at $\sqrt{overall\ pressure\ ratio}$. On the other side, the heat exchanger is used to pre-heat the air coming from the HP compressor, as shown in Figure 5.2, therefore reducing fuel consumption and having the most relevant effect in the increase of efficiency.

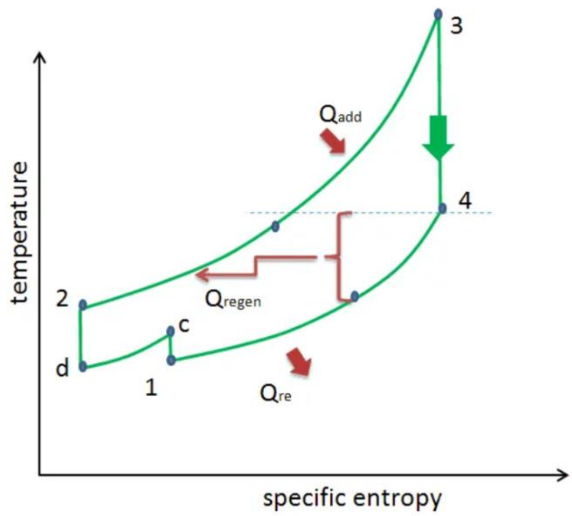

Figure 5.2 – (T,s) diagram of an ICRGT cycle

The presence of regeneration is more effective at low or moderate overall pressure ratios and so twin shaft configuration is no more necessary. In fact, higher overall pressure ratios decrease the TOT (Turbine Outlet Temperature) and increase air temperature at the exit of the HP compressor, thus reducing the quantity of rescuable heat. Therefore, this temperature difference is crucial to increase the efficiency of the plant and the presence of the intercooler increases both the positive effect of regeneration and the quote of the rescued heat.

The addition of another heat exchanger increases the cost of the plant. Moreover, both intercooler and regenerator are connected to pressure losses that must be considered in the plant evaluation.

5.2 Intercooled regenerative hydrogen gas turbine

As for the previous cases the intercooled regenerative cycle has been implemented in the GS Software, to highlight the performances and to evaluate the optimal overall pressure ratio of the new configuration. The main design conditions have been maintained the same of the hydrogen intercooled gas turbine in single shaft configuration, while some have been added due to the presence of the new heat exchanger, and they are reported in Table 5.1. The GS scheme is presented in Figure 5.3.

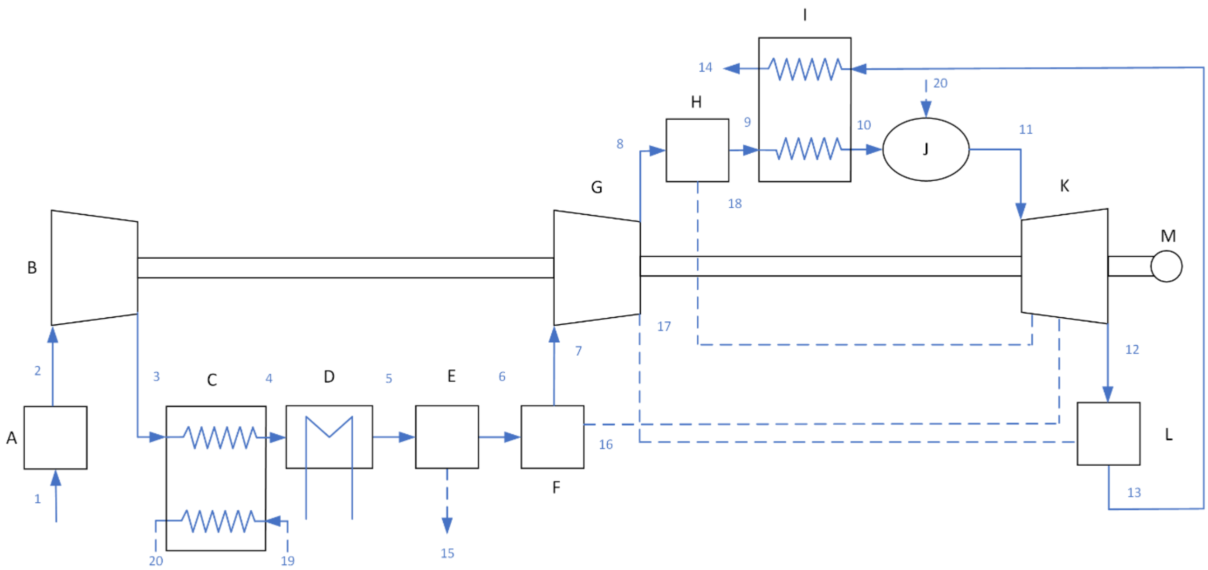

Figure 5.3 – GS scheme of ICRGT plant

In respect of the intercooled configuration the only difference is represented by the addition of the regenerator, a heat exchanger that has been placed after the HP splitter and before the combustor, because cooling flows must be at lower temperature. In this component pressure losses of both hot side and cold side and thermal losses have been imposed. The pressure at the exit of the turbine is now increased due to the presence of the regenerator, which imposes a pressure loss of 3500 Pa, and now a new pressure loss is present between the HP compressor and the combustor, a relative $\Delta p/p$ of 2%. The main design condition is the temperature difference between the combustor inlet temperature (T_{10}) and the temperature of the hot fluid entering the heat exchanger (T_{13}), this difference is $\Delta T_{hot\ end}$, with a maximum temperature acceptable of 750°C for the cold flow, was never reached. A 4 stages turbine was chosen to be sure not to have too hot exhaust gasses at the entrance of the regenerator.

Air Filter		***Fuel Heater***	
$\Delta p/p$	*0.007402*	$\Delta T_{hot\ end}$, °C	*30*
Compressor		ε_{th}	*0.03*
$\Delta h_{is,\ max\ stage}$ kJ/kg	*25*	$T_{fuel,\ max}$, °C	*220*
lk	*0.004*	$\Delta p/p$	*0.000*
η_{org}	*0.99865*	***Shaft and Generator***	

Combustor	
η_{comb}	*0.997*
$\Delta p/p$	*0.03*
Turbine	
N. stages	*4*
Cooled stages	*Yes*
p_{out}*, Pa*	*101925*
TIT, K	*1823.15*
Mach number at turbine exit	*0.55*
Stanton coefficient	*0.28*
Polytropic efficiency	*0.93*
Velocity of rotation, rpm	*3000*
Electrical efficiency	*0.987*
Intercooler	
$T_{out, IC}$*, °C*	*30*
$\Delta p/p$	*0.02*
No heat losses	
Regenerator	
Δp *hot side, Pa*	*3500*
$\Delta p/p$ *cold side*	*0.03*
Heat loss	*0.005*
$\Delta T_{hot\ end}$*, °C*	*40*

Table 5.1 – Design conditions relative to hydrogen ICRGT plant

Also for the intercooled regenerative gas turbine plant, the convergence variables have been maintained in the design case similar to the hydrogen SCGT.

Heat losses at turbine exit are imposed in component L through a temperature drop, as an independent variable. They are equal to one third of the thermal losses of the combustor.

Peripheral velocity at the first rotor of the turbine is varied to reach the pressure imposed at turbine exit.

Temperature at point 11, which is the TIT, has to converge to the assigned value.

Flow coefficient at turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage is varied to reach the desired h/D_m of 0.3 at turbine exit.

In this cycle the configuration of both the intercooling advantages and the regeneration should be presented and highlighted.

Although this is a secondary and less relevant effect, the choice of the intercooling pressure ($\sqrt{overall\ pressure\ ratio}$) brings to higher temperatures at the exit of the LP compressor respect ICGT, with a little increase of the heat recuperated in the fuel pre-heater.

In this configuration the presence of the intercooler brings also to a reduction of the compression work, with a consequent increase in power output, as presented in Figure 5.4. Moreover, the reduction is more relevant in the ICR gas turbine due to the choice of the intercooling pressure that minimizes the compression work. This positive effect is more important by increasing the overall compression ratio.

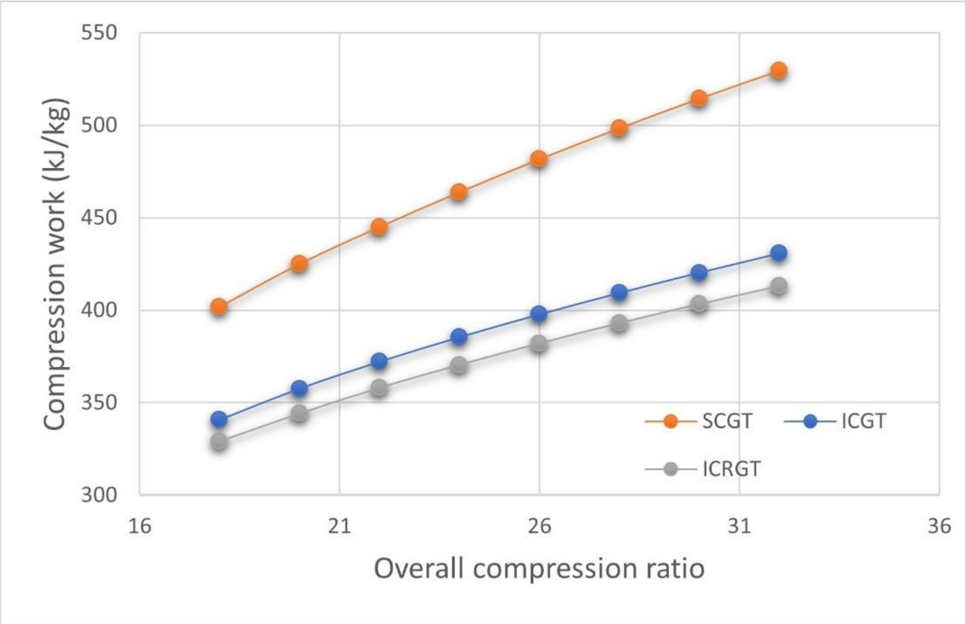

Figure 5.4 – Compression work, overall pressure ratio chart of ICRGT, ICGT and SCGT

The second positive effect of intercooling is in the reduction of the cooling mass flows, and as expected also in the ICRGT simulations this effect has been demonstrated, as Figure 5.5 justifies. In simple cycle configuration the growth of cooling flows is more important due to higher discharge temperatures at the compressor. In the other configurations between overall compression ratio of 18 and 20 there is a little reduction, because at these overall pressure ratios also the stator of the firth stage is cooled, instead, increasing it, only three stages are cooled. Cooling flows remain almost constant, increasing overall compression ratio in ICR configuration. Moreover, in ICR configuration cooling flows are even lower than in the cycle only with intercooling. This is due to the choice of the intercooling pressure. In fact, the temperature of the cooling flow of the LP splitter doesn't change between the two configurations, but we have a reduction of the temperature at the exit of the HP compressor (T_8), where we have the HP splitter used to take the second cooling flow, as Figure 5.6 shows. The difference of T_8 between the two cycles and the reduction of cooling flows between them and SCGT becomes more relevant at higher overall compression ratios.

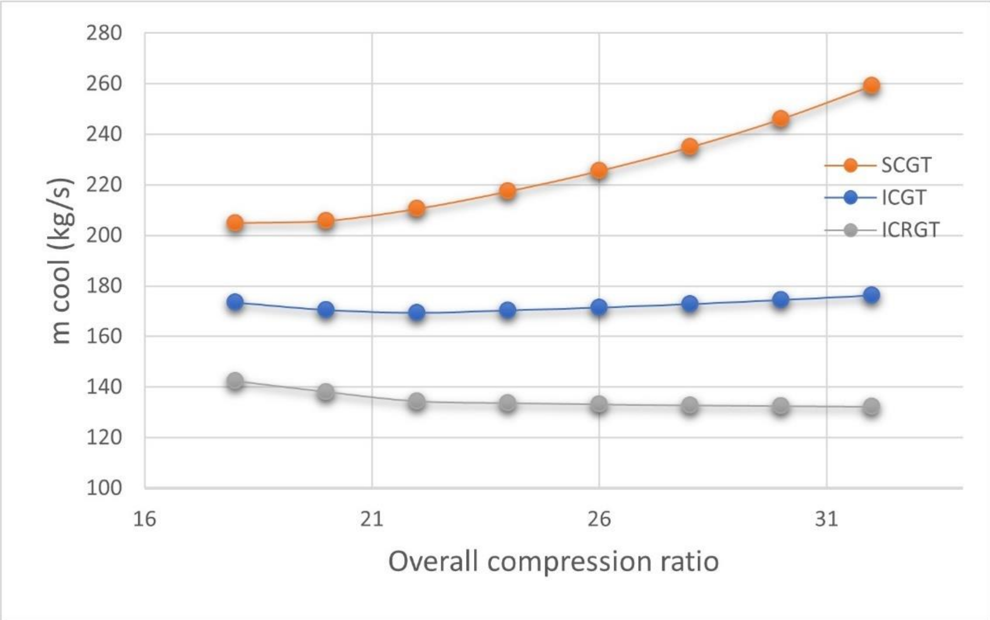

Figure 5.5 – Cooling mass flows, overall pressure ratio chart of ICRGT, ICGT and SCGT

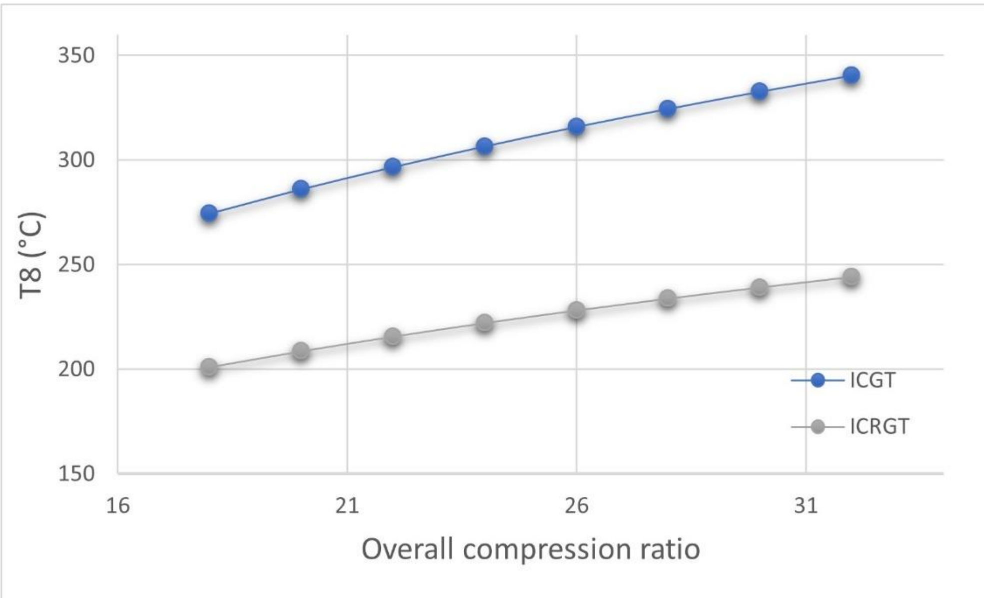

Figure 5.6 – Temperature exit HP compressor, overall pressure ratio chart of ICRGT and ICGT

The presence of the regenerator should mitigate the main negative effect of intercooling, which is the increased need of fuel to obtain the same TIT, caused by the decrease of discharge temperature of the HP compressor. As shown in Figure 5.7, as result of the GS simulations, the regeneration has its higher effects at lower overall compression ratios, with an important decrease of fuel with respect to both ICGT and SCGT. Instead increasing the overall compression ratio increases the fuel consumption, which becomes higher than the simple cycle configuration only at higher values but is anyway lower with respect to ICGT. This is caused by the fact that increasing the overall compression ratio increases the HP compressor discharge temperature (entering cold flow temperature of the regenerator), as previously shown in Figure 5.6, and decreases the TOT (Turbine Outlet Temperature) (entering hot flow temperature of the regenerator), thus the quote of heat recuperated, as Figure 5.8 demonstrate.

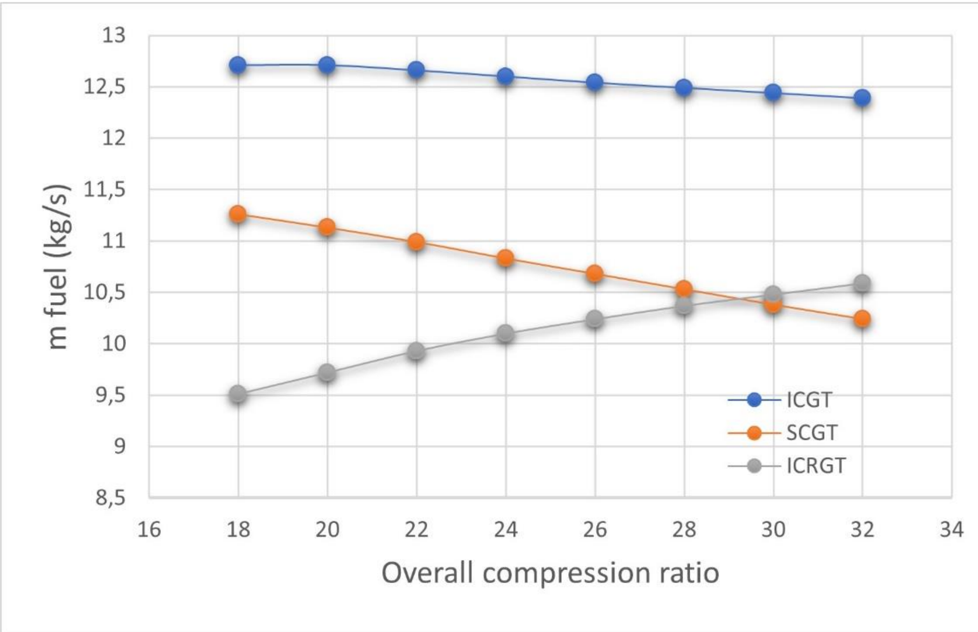

Figure 5.7 – Fuel mass flow, overall pressure ratio chart of ICRGT, ICGT and SCGT

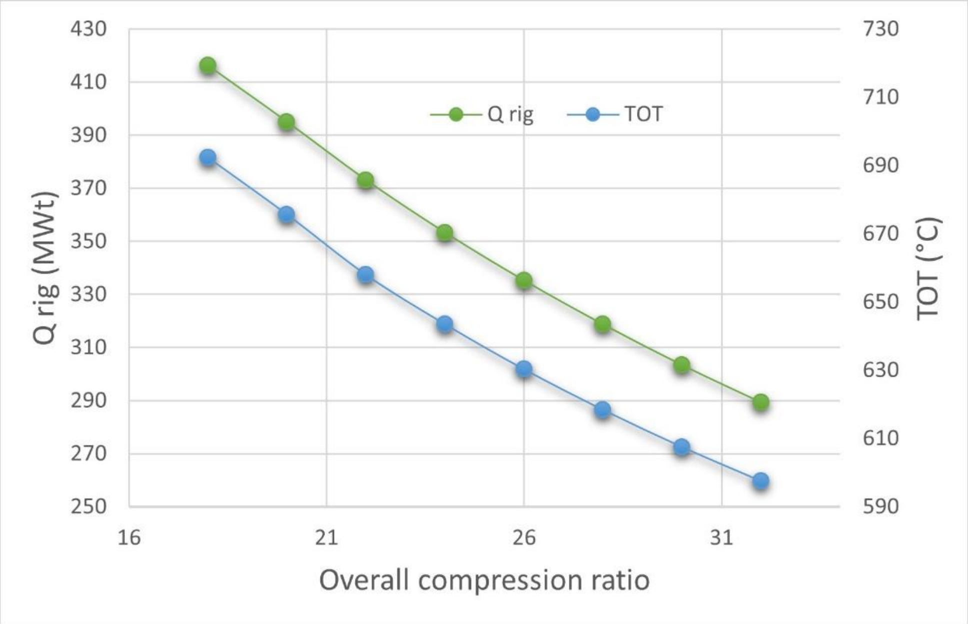

Figure 5.8 – Recuperated heat, TOT, overall pressure ratio chart of ICRGT

In conclusion, in Figures 5.9 and 5.10, the trends of specific work and efficiency are reported to highlight the differences between ICRGT, ICGT and SCGT. The specific work growth is evident in both ICRGT and ICGT with respect to simple cycle configuration. At almost all compression ratios studied, the increase of pressure losses from 600 Pa to 3500 Pa, with the consequent reduction of the expansion process, due to the presence of the regenerator, is stronger than the choice of intercooling pressure that minimize compression work and cooling flows, and this brings to higher specific work for the ICGT. The same value is almost obtained at overall compression ratios higher than 30. On the efficiency side, the growth is relevant at every overall pressure ratio, and it keeps almost constant for values higher than 20. It is slightly decreased for values lower than 20 due to the cooling of the stator of the firth stage, which increases cooling flows and losses. At low pressures the reduction in the positive effects of intercooling is balanced by the reduction in fuel consumption thanks to the regeneration and vice versa at higher pressures. For this reason, it would probably be the best idea to design the machine at a lower design overall compression ratio, at the expense of a lower specific work, to reduce the number of stages of the compressor, thus costs that have grown with the addition of the regenerator and reducing fuel costs.

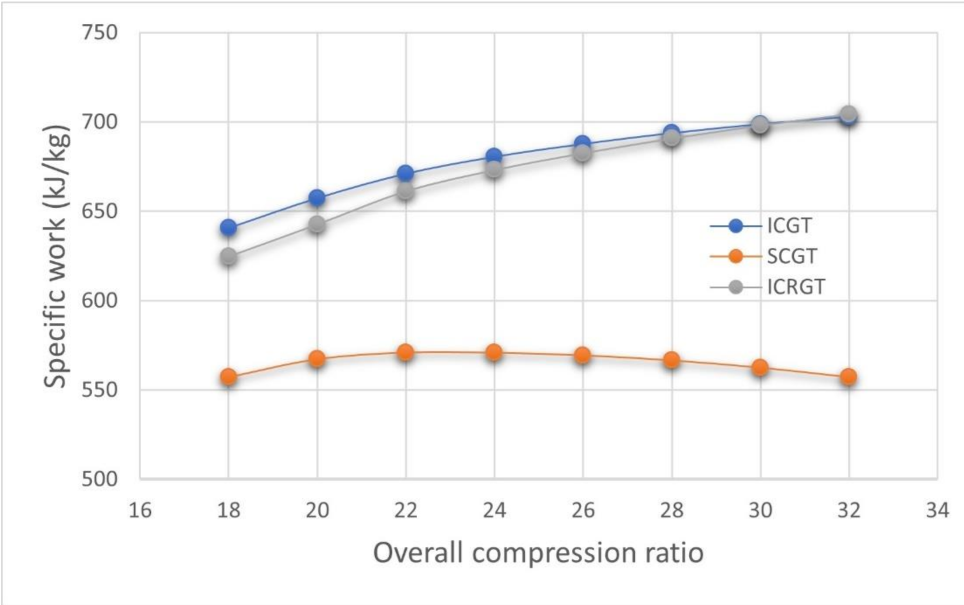

Figure 5.9 – Specific work, overall pressure ratio chart of ICRGT, ICGT and SCGT

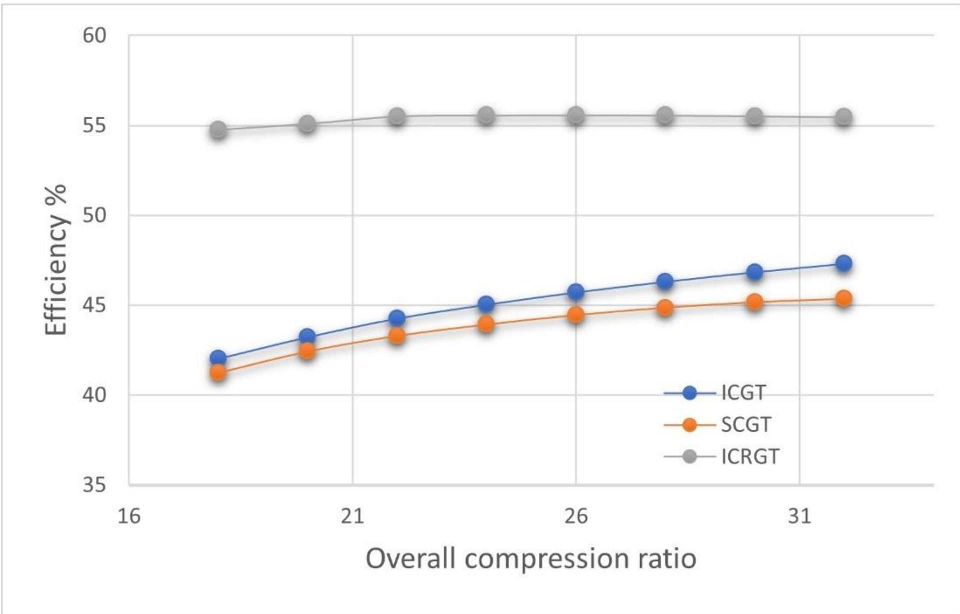

Figure 5.10 – Efficiency, overall pressure ratio chart of ICRGT, ICGT and SCGT

In Table 5.2 are reported the flow properties of each point of the cycle for the optimal overall pressure ratio of 26, as results of the GS simulations.

Point	T	P	G	Q	Composition (% mol)					
	°C	bar	kg/s	kmol/s	Ar	CO_2	H_2	H_2O	N_2	O_2
1	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
2	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
3	200.7	5.13	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
4	177.7	5.13	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
5	30.0	5.03	1000.0	34.66	0.92	0.03	0.0	0.84	77.28	20.73
6	30.0	5.03	998.8	34.59	0.92	0.03	0.0	0.84	77.43	20.73
7	30.0	5.03	908.3	31.46	0.92	0.03	0.0	0.84	77.43	20.73
8	228.1	26.15	904.3	31.32	0.92	0.03	0.0	0.84	77.43	20.73
9	228.1	26.15	861.7	29.85	0.92	0.03	0.0	0.84	77.43	20.73
10	587.8	25.37	861.7	29.85	0.92	0.03	0.0	0.84	77.43	20.73
11	1602.9	24.60	872.0	32.39	0.85	0.027	0.0	16.46	71.36	11.30
12	630.2	1.05	1005.0	37.00	0.86	0.028	0.0	14.51	72.11	12.48
13	627.8	1.05	1009.0	37.13	0.86	0.028	0.0	14.46	72.13	12.51
14	349.2	1.01	1009.0	37.13	0.86	0.028	0.0	14.46	72.13	12.51
15	30.0	5.03	1.2	0.66	0.0	0.0	0.0	100	0.0	0.0
16	30.0	5.03	90.5	3.13	0.92	0.03	0.0	0.84	77.43	20.77
17	228.1	26.15	4.0	0.13	0.92	0.03	0.0	0.84	77.43	20.77
18	228.1	26.15	42.6	1.47	0.92	0.03	0.0	0.84	77.43	20.77
19	15.0	70.0	10.2	5.08	0.0	0.0	100	0.0	0.0	0.0
20	170.7	70.0	10.2	5.08	0.0	0.0	100	0.0	0.0	0.0

Table 5.2 – Hydrogen ICRGT plant flows properties

6. Water injection

The efficiency and power output reached with the addition of intercooling and regeneration are considerably higher than the ones of a simple cycle configuration and close to the values of a combined cycle plant. Anyway, there is still the possibility to modify the cycle to increase efficiency and power output through water injection, although the complexity of the plant will inevitably grow, and so costs.

6.1 Effects

From the previous chapter simulations exhaust gasses were emitted in atmosphere at a temperature around 350°C. The injection of water into the gaseous stream helps to reduce further this temperature, thus heat losses at stack, and to accomplish an efficient and balanced regeneration. In fact, water injection is used to increase the thermal capacity of the stream that will be heated in the regenerator, to balance the thermal capacity of the turbine exhaust gasses and to increase the quote of heat recuperated respect simple regeneration. At the same time, it must be pointed out that the presence in the exhaust gasses of large quantities of steam, which release their latent heat of condensation during the cooling process, still keep stack losses higher respect combined cycle configuration. The reduction of stack losses, the more efficient regeneration, the increase of working mass flow due to addition of water, thus the power output, contribute to a further increase of efficiency. Usually, before injection, water is also pre-heated, before exhaust gasses are released, to increase further the quote of heat recuperated. This increases investment and design costs due to the presence of a new heat exchanger.

In addition, humid air cycles have been demonstrated to reduce nitic oxide (NOx) formation, mitigating pollutant emissions, thanks to a reduction of temperature peaks into the combustion chamber. In natural gas turbines water injection increases the percentage of hydrogen that can be added in the fuel maintaining combustion safe. In a full hydrogen combustion, a percentage of steam in the flow can improve combustion stability and mitigate some of the problems anticipated in Chapter 3, like flashbacks and autoignition, mainly thanks to a flame speed reduction.

Moreover, water injection is possible if water is previously pumped to the adequate pressure, which means an increase in auxiliary consumption and investment costs, due to the addition of pump and control systems. In the injection point a new loss is present, this configuration will now also face some mixing losses.

6.2 Intercooled regenerative hydrogen gas turbine with water injection

As for the previous cases the intercooled regenerative cycle with water injection has been implemented in the GS Software, to highlight the performances and to evaluate the optimal overall pressure ratio of the new configuration. The main design conditions have been maintained the same of the hydrogen intercooled regenerative gas turbine, while some have been added due to the modification of the first heat exchanger (component D), the addition of the pump (component C) and of the mixer (component J), and they are reported in Table 6.1. The GS scheme is presented in Figure 6.1.

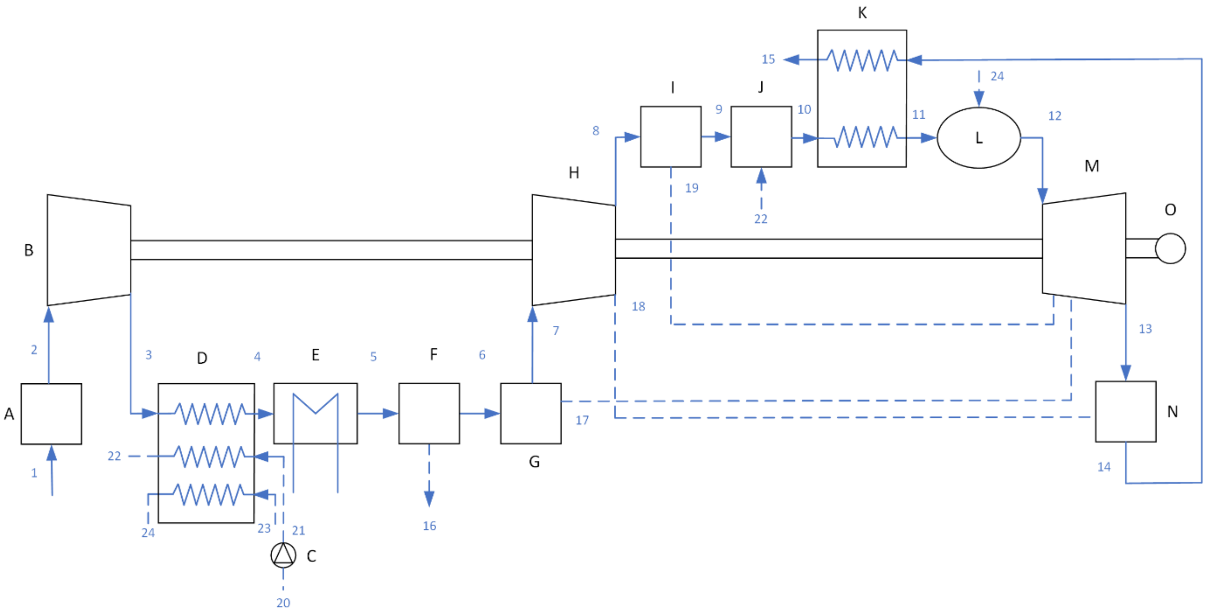

Figure 6.1 – GS scheme of ICRWIGT plant

Respect the intercooled regenerative configuration some changes have been effectuated to the plant. The water before being injected need to be pumped to a pressure higher than the introduction point. For this reason, a pump has been added, component C, characterized by a compression ratio that ensure a pressure of water 10% higher respect the mixing point, with an hydraulic efficiency of 80%. Moreover, the first heat exchanger (component D) is now a multi-flows heat exchanger, where the three flows are: the air coming from the LP compressor, that must be cooled down, the fuel, that must be pre-heated before combustion, and water, that in this way is heated up before injection. The realization and design of this component for the real plant would be for sure challenging. In the simulation no pressure losses have been considered in this component, while the pressure losses on the air side due to intercooling are concentrated, as in the previous cases, on component E. The main design conditions of this multi-flows heat exchanger are the $\Delta T_{hot\ end}$ between hot air coming from the first compressor (T_3) and the exit temperatures of water (T_{22}) and hydrogen (T_{24}), differences respectively imposed to 15°C for water and to 30°C of for hydrogen. As in the previous configuration, a 4 stages turbine was chosen to be sure of not having too hot exhaust gasses at the entrance of the regenerator.

Air Filter		***Fuel and Water Heater***	
$\Delta p/p$	*0.007402*	*$\Delta T_{hot\ end}$, °C H_2*	*30*
Compressor		*$\Delta T_{hot\ end}$, °C H_2O*	*15*
$\Delta h_{is,\ max\ stage}$ kJ/kg	*25*	*ε_{th}*	*0.03*
lk	*0.004*	*$T_{fuel,\ max}$, °C*	*220*
η_{org}	*0.99865*	*$\Delta p/p$*	*0.000*
Combustor		***Shaft and Generator***	
η_{comb}	*0.997*	*Velocity of rotation, rpm*	*3000*
$\Delta p/p$	*0.03*	*Electrical efficiency*	*0.987*
Turbine		***Intercooler***	
N. stages	*4*	*$T_{out,\ IC}$, °C*	*30*
Cooled stages	*Yes*	*$\Delta p/p$*	*0.02*
p_{out}, Pa	*101925*	*No heat losses*	
TIT, K	*1823.15*	***Regenerator***	
Mach number at turbine exit	*0.55*	*Δp hot side, Pa*	*3500*
Stanton coefficient	*0.28*	*$\Delta p/p$ cold side*	*0.03*

Polytropic efficiency	*0.93*	*Heat loss*	*0.005*
Pump		*$\Delta T_{hot\ end}$, °C*	*40*
Compression ratio (respect point)	*+10%*		
$\eta_{hydraulic}$	*0.8*		

Table 6.1 – Design conditions relative to hydrogen ICRWIGT plant

Also for the intercooled regenerative gas turbine plant with water injection, the convergence variables have been maintained the same of the hydrogen SCGT in the design case, except for the addition of the last one.

Heat losses at turbine exit are imposed in component N through a temperature drop, as an independent variable. They're equal to one third of the thermal losses of the combustor.

Peripheral velocity at the first rotor of the turbine is varied to reach the pressure imposed at turbine exit.

Temperature at point 12, which is the TIT, has to converge to the assigned value.

Flow coefficient at turbine exit is varied to reach the imposed axial Mach in the same point of the machine.

Reaction degree of the first stage is varied to reach the desired h/D_m of 0.3 at turbine exit.

Mass flow at point 20, which is water flow, is varied to balance the regenerator and to obtain the desired $\Delta T_{hot\ end}$ of 40°C.

Water injection doesn't affect in any way the compression work, so its reduction due to intercooling is still present. The choice of the intercooling pressure has been maintained at $\sqrt{overall\ pressure\ ratio}$ to minimize compression work, therefore no differences have been noticed respect ICRGT plant, so the same trend of Figure 5.4 was obtained from the simulations, with a further increase of the reduction of compression work at higher overall compression ratios.

A relevant increase of the heat recuperated in the first heat exchanger has been highlighted in Figure 6.2 from the simulations. In the ICRGT plant it was used only as a fuel pre-heater, while now it is a multi-flows heat exchanger used to heat up both fuel and water. The difference is relevant, with the thermal power exchanged becoming more than 4 times the regenerative case, mainly thanks to the higher specific heat capacity of water, that causes a different slope for the two trends. In fact, the effect is stronger at higher pressure ratios.

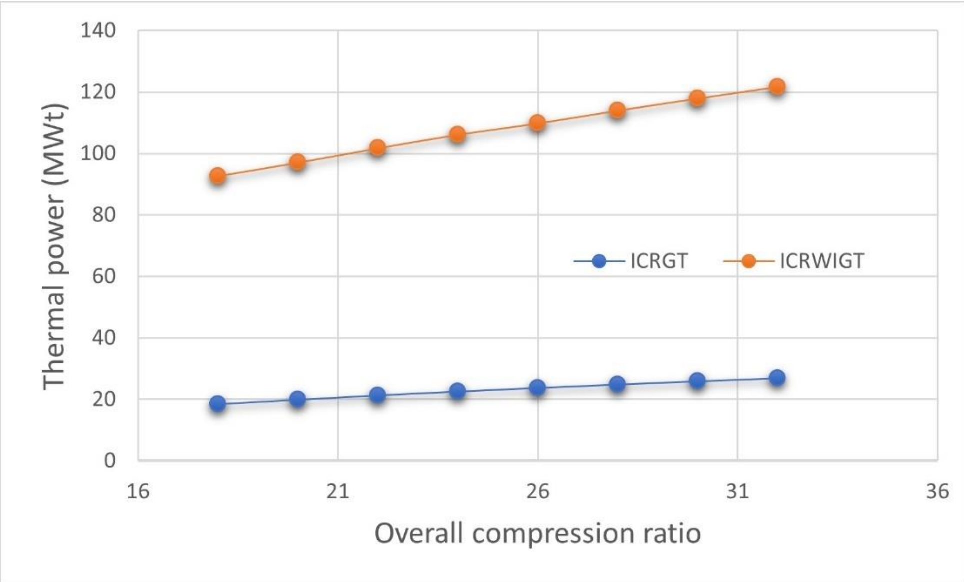

Figure 6.2 – Thermal power fuel pre-heater, overall pressure ratio chart of ICRWIGT and ICRGT

Instead, water injection affects heavily cooling mass flows. First of all, the injection of water increases the working fluid of the turbine and also the cooling flows needed to keep the admissible temperature for the metal, if the TIT is kept constant. Moreover, respect ICRGT part of the flow is now composed of molecules of three atoms, with lower Cp/Cv respect molecules with two atoms. The consequence using the same turbine design and the same number of stages, is a higher TOT, as Figure 6.4 shows. The increase is of around 20°C constant increasing overall pressure ratio. At a pressure ratio of 18 all the turbine stages of the turbine are cooled down, bringing cooling flows almost at the same value of SCGT. Between 20 and 24, turbine, expansion brings to a lower TOT, and cooling is needed until the firth stator, with a decreasing trend. From 26, only three stages are cooled down, with cooling flows found almost at the same value, close to ICGT. The increase respect ICRGT is very important, although the intercooling pressure is the same, which means that the temperature of the cooling flows is the same.

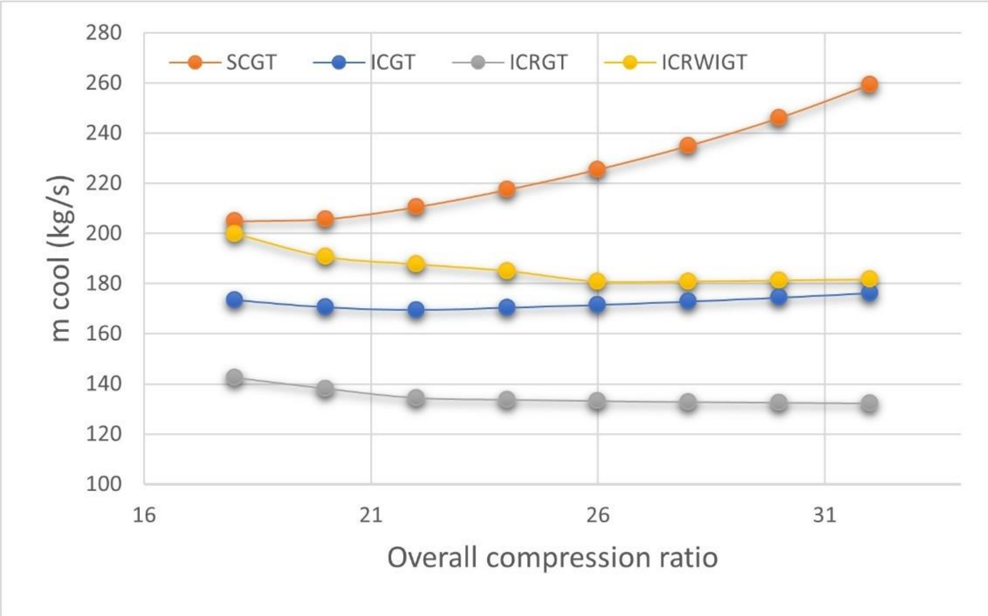

Figure 6.3 – Cooling mass flows, overall pressure ratio chart of ICRWIGT, ICRGT, ICGT and SCGT

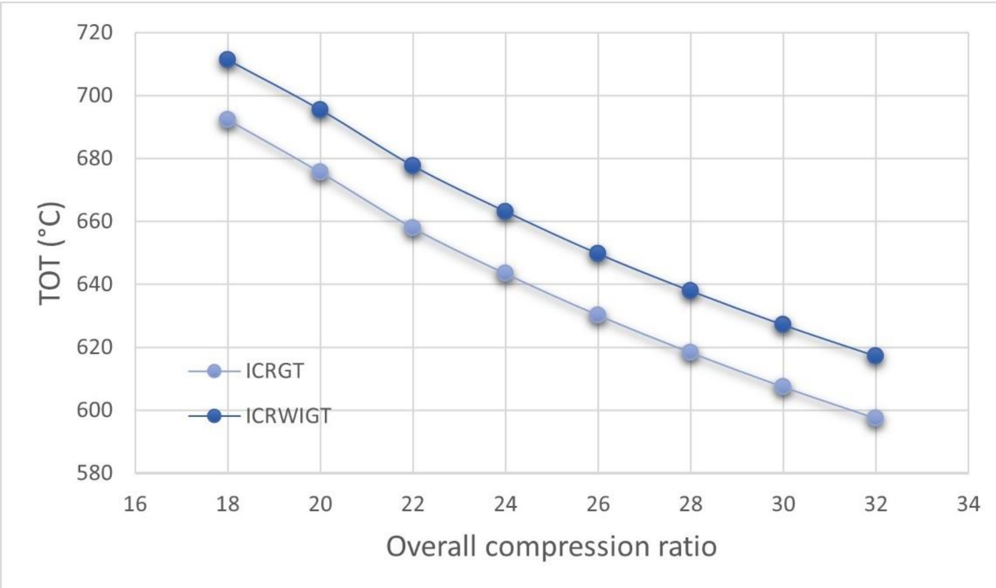

Figure 6.4 – TOT, overall pressure ratio chart of ICRWIGT and ICRGT

The increase of mass flow due to water injection is associated to an increase of fuel consumption respect ICRGT, as Figure 6.5 shows. Fuel consumption is even higher than SCGT, and higher than ICGT at overall pressure ratios greater than 26. On the other side, the heat recuperated thanks to the balancing of the regenerator becomes double respect ICRGT (Figure 6.6), also thanks to the increase of the TOT (Figure 6.4) and the decrease of the temperature at the inlet of the regenerator (Figure 6.7). Increasing the overall pressure ratio decreases the TOT and increases the temperature of the cold flow entering the regenerator, decreasing the heat recuperated. The consequence is that going to higher pressure ratios increases fuel consumption, even due to an increase of the water injected. In conclusion the increase of heat recuperated is a weaker effect than the increase of mass flow on fuel consumption, also because of the presence of steam and not only diatomic molecules.

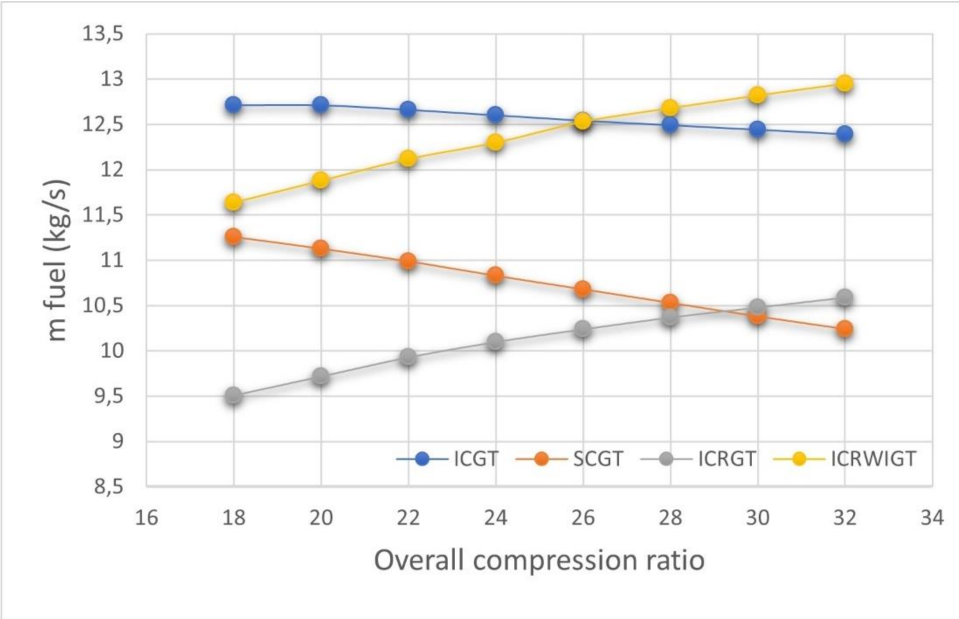

Figure 6.5 – Fuel mass flow, overall pressure ratio chart of ICRWIGT, ICRGT, ICGT and SCGT

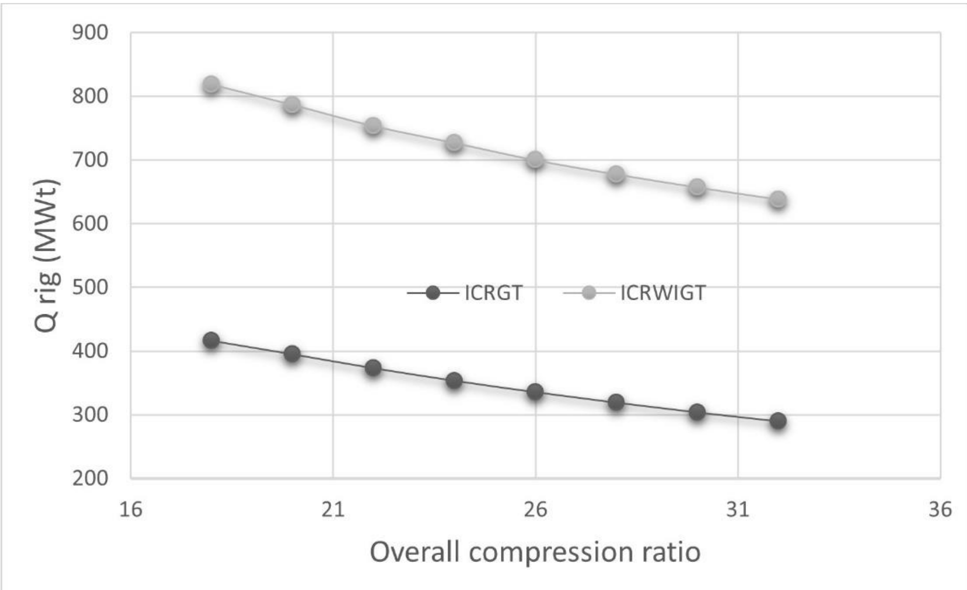

Figure 6.6 – Recuperated heat, overall pressure ratio chart of ICRWIGT and ICRGT

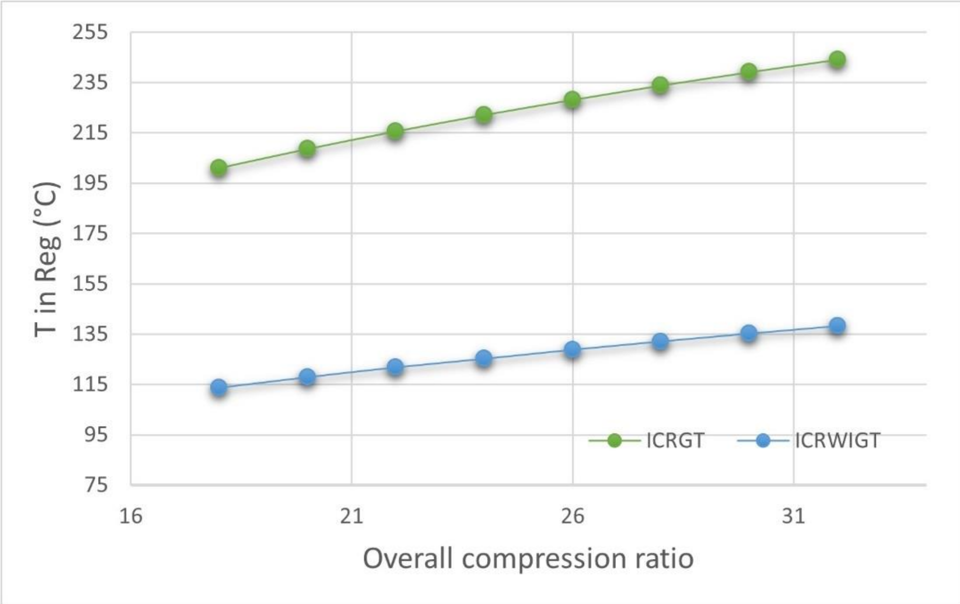

Figure 6.7 – Temperature cold flow entering regenerator, overall pressure ratio chart of ICRWIGT and ICRGT

In conclusion, in Figures 6.8 and 6.9, the trends of specific work and efficiency. A very important growth of specific work is evident, gaining importance at higher overall compression ratios, mainly due to the reduction of compression work thanks to intercooling. The increase of auxiliary consumption due to the addition of the pump have a weak and not relevant effect con specific work. Increasing the overall pressure ratio also the water injected grows, and this is another key that explain the growth of specific work. In this configuration values equal or even higher to the specific work of the hydrogen CCGT (Table 3.2) have been obtained.

On the efficiency side, the growth is relevant at every overall pressure ratio, and it keeps almost constant for values higher than 26. The decrease for lower values is explained by the increase of cooling flows, because as mentioned before also the firth stage of the turbine needs cooling, decreasing working fluid in the turbine and increasing losses. At low pressures the reduction in the positive effects of intercooling is balanced by the reduction in fuel consumption thanks to the regeneration and vice versa at higher pressures, as for the ICRGT. In addition, as before increasing too much pressure ratios, for example with a twin shaft configuration, decreases efficiency because of the higher fuel consumption and of the lower heat recuperated. Compression ratios around 26 would probably be adequate, with an efficiency of 60%, almost the same value of the hydrogen CCGT (Table3.2).

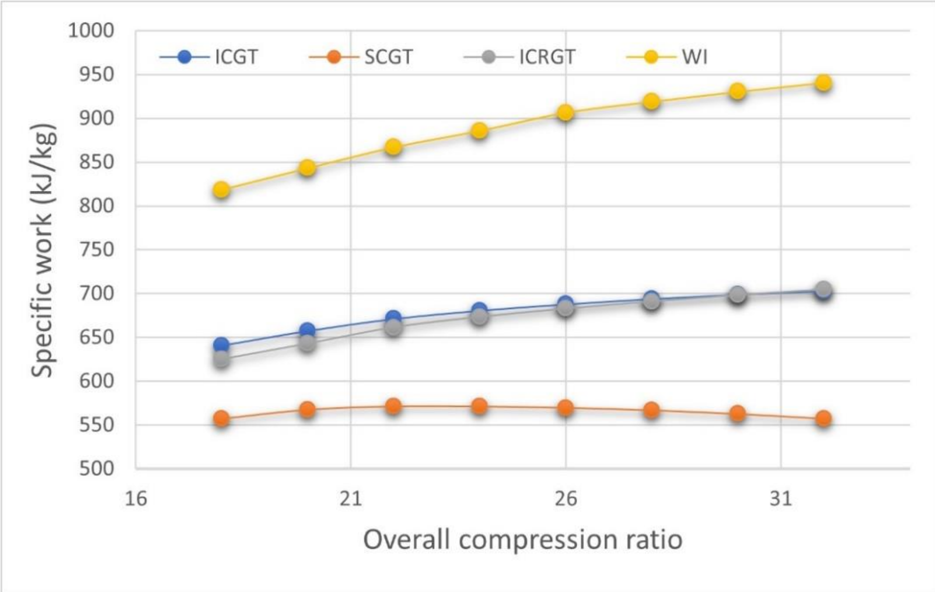

Figure 6.8 – Specific work, overall pressure ratio chart of ICRWIGT, ICRGT, ICGT and SCGT

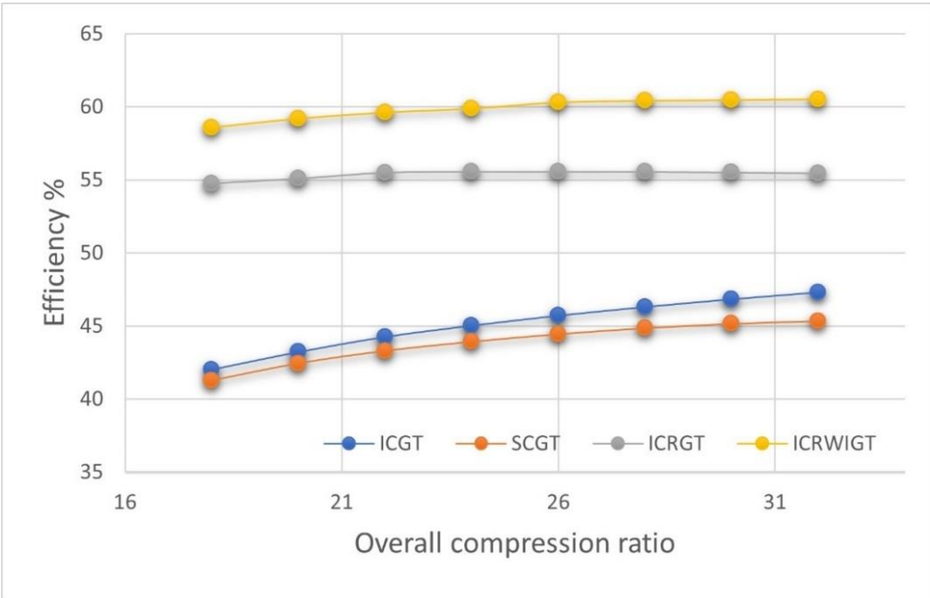

Figure 6.9 – Efficiency, overall pressure ratio chart of ICRWIGT, ICRGT, ICGT and SCGT

In Table 6.2 are reported the flow properties of each point of the cycle for the overall pressure ratio of 26, at which the same specific work of CCGT has been obtained, as results of the GS simulations.

Point	T	P	G	Q	Composition (% mol)					
	°C	bar	kg/s	kmol/s	Ar	CO_2	H_2	H_2O	N_2	O_2
1	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
2	15.0	1.01	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
3	200.7	5.13	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
4	92.7	5.13	1000.0	34.66	0.92	0.03	0.0	1.03	77.28	20.73
5	30.0	5.03	1000.0	34.66	0.92	0.03	0.0	0.84	77.28	20.73
6	30.0	5.03	998.8	34.59	0.92	0.03	0.0	0.84	77.43	20.77
7	30.0	5.03	870.3	30.14	0.92	0.03	0.0	0.84	77.43	20.77
8	228.2	26.15	866.3	30.01	0.92	0.03	0.0	0.84	77.43	20.77
9	228.2	26.15	814.1	28.20	0.92	0.03	0.0	0.84	77.43	20.77
10	128.8	26.15	937.9	35.07	0.74	0.02	0.0	8.84	62.25	16.70
11	607.5	25.37	937.9	35.07	0.74	0.02	0.0	20.28	62.25	16.70
12	1601.6	24.60	950.5	38.18	0.68	0.02	0.0	34.91	57.18	7.20
13	649.7	1.05	1131.2	44.44	0.71	0.02	0.0	30.11	60.04	9.11
14	647.5	1.05	1135.2	44.58	0.72	0.02	0.0	30.02	60.09	9.15
15	168.8	1.01	1135.2	44.58	0.72	0.02	0.0	30.02	60.09	9.15
16	30.0	5.03	1.2	0.066	0.0	0.0	0.0	100	0.0	0.0
17	30.0	5.03	128.5	4.45	0.92	0.03	0.0	0.84	77.43	20.77
18	228.2	26.15	4.0	0.14	0.92	0.03	0.0	0.84	77.43	20.77
19	228.2	26.15	52.3	1.81	0.92	0.03	0.0	0.84	77.43	20.77
20	30.0	1.01	123.9	6.87	0.0	0.0	0.0	100	0.0	0.0
21	30.2	28.37	123.9	6.87	0.0	0.0	0.0	100	0.0	0.0

22	185.7	28.37	123.9	6.87	0.0	0.0	0.0	100	0.0	0.0
23	15.0	70.0	12.5	6.21	0.0	0.0	100	0.0	0.0	0.0
24	170.7	70.0	12.5	6.21	0.0	0.0	100	0.0	0.0	0.0

Table 5.2 – Hydrogen ICRGT plant flows properties

7. Conclusions

Peak generation is one of the most important challenges in the power sector. As highlighted in the first part of this work, balancing the grid and coupling electricity demand and supply is becoming harder and harder, mostly because of the growth of renewable energy. In the future, combined cycles gas turbines will not be anyway the most suitable solution for peak generation, for the multiple reasons well described in this work.

Hydrogen-based gas turbines may be the key for the energy transition. To reach high efficiencies and power outputs of combined cycles, some possible modifications have been analyzed in this work and applied to the simple cycle. The analysis has been divided in phases to have the possibility of highlighting the advantages of each modification.

Intercooling is more effective on power output than on efficiency, although important growth of efficiency can be achieved passing to a twin shaft configuration. The twin shaft configuration, characterized by the power balancing of the HP shaft, would need very advanced control systems for the transient and important investment costs for the design of the turbomachines of both HP and LP shafts.

The combination of intercooling and regeneration have relevant effects on both power output and efficiency. At the same time the presence of two heat exchangers would strongly increase the cost of the plant. The relevant increase of efficiency highlighted in design conditions would be much smoother on a real plant because rarely a gas plant for peak generation would work in design condition, and a little modification of the working point can cause strong reduction in the efficiencies of the heat exchangers.

In conclusion, water injection contributes to balance the regenerator, increases the recuperation of the heat and both power output and efficiency, to values close to the reference combined cycle run on natural gas. This modification would need sophisticated control systems which would increase investment costs and auxiliary consumption. Such an overall complex system would obviously be much more difficult to control in transient, therefore when turning the plant on and off and partially loaded. Higher number of components also means lower reliability, higher maintenance costs and failures.

Possible future studies may be related to the off-design conditions and balancing of the final configuration, aiming to highlight its flexibility or even to evaluate the possibility of adding a re-heating phase to further increase efficiency or even highlight the reduction of NOx linked to the presence of steam in the flow before a combustion based on hydrogen as well as to also evaluate the economic analysis of the plant.

Hydrogen gas turbines can be thermodynamically competitive in respect of combined cycles. Today gas turbine completely running on hydrogen are an important challenge potentially able to change the future of the power generation and may help to reach decarbonization. The hope is that this type of machines could enter as soon as possible the market as well as in real plants for the production of electricity

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