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Title

Comparative Study Between Aero-derivative Turbine and Heavy-duty Turbine

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DEDICACE

I dedicate this work to my dear family, to my parents who have supported me all the way through my school years and to my brothers who guided me through my life.

I dedicate this also to my little brother Youcef and my little sister Nour who brought up smiles to our life and throughout stress that came along this work.

LAGREB Aimen

I dedicate this work to my wife who have supported me all the time and to my little son who didn't let me sleep and do my work comfortably, without forgetting Mr.Guergadj who help us since we were L2, I dedicate this also to my family specially my mother and sister.

AMARA Mohamed Iskander

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Abbreviation	Meaning
GT	Gas Turbine
HDGT	Heavy Duty Gas Turbine
ADGT	Aero-Derivative Gas Turbine
HP	Horse Power
IGV	Inlet Guide Vanes
EGV	Exit Guide Vanes
CC	Combustion Chamber
HPT	High Pressure Turbine
HSPT	Hot Section Power Turbine
PT	Power Turbine
NACA	National Advisory Committee for Aeronautics
TP	Transition Piece
HPS	High Pressure Stages
LPS	Low Pressure Stages
FS	Final Stages
CI	Combustor Inspection
HGPI	Hot Gas Past Inspection
LTPI	Liner Transition Piece Inspection
MI	Major Inspection
MTBF	Mean Time Between Failure
MTTR	Mean Time To Repair
W	Work
Q_{in}	Heat in
Q_{out}	Heat out

General Introduction:

The use of gas turbines is widespread in many industries that require power. The power is used to generate electricity or drive equipment such as pumps and process compressors. Gas turbines are also used extensively in naval propulsion, railways and aviation.

Aero-derivative and heavy-duty gas turbines have demonstrated their suitability for heavy duty, continuous, base load operation in power generation, pump and compressor applications. While they share many similarities, there are times when their differences make them uniquely more suitable for a specific application. These differences are not always adequately considered during the equipment selection phase. As a result, operations and maintenance personnel must deal with them throughout their useful life.

Our study aims to demonstrate the similarities and differences of each category, from construction, maintenance and performance, taking into consideration each turbine maintainability, reliability and availability, and compare which is more suitable in different cases. The study is split into three chapters as follows:

-In the first chapter: we are going to talk first about the company GE, then general information on gas turbine and its applications, the main components, operating principle, thermodynamic cycle, and single /double shaft turbines.

-In the second chapter: we are going to talk about the main components of each gas turbine's section (compressor, combustion chamber, turbine) and accessory systems.

-In the third chapter: we are going to do the comparison of heavy-duty and aero-derivative gas turbines.

A) GENERAL ELECTRIC:

General Electric Company is an American multinational conglomerate incorporated in New York City and headquartered in Boston.
General Electric is a company present in several activity sectors, as numerous as varied, ranging from the fields of energy, aviation to the fields of finance, digital and transport. From aircraft engines and power generation to financial services, television shows and healthcare solutions, GE operates in more than 100 countries and employs approximately 300,000 people worldwide. The main GE subsidiaries around the world are: GE Aviation, GE Capital, GE Healthcare, GE Lightning, GE Power, GE Renewable Energy, GE Transportation, GE Digital, Baker Hughes a GE company, NUOVO PIGNONE and ALGESCO. Present in Algeria for more than 40 years, General Electric supports the Algerian oil and gas industry and its fleet installed in Algeria includes approximately 4,000 gas turbines, 340 compressors, 200 centrifugal pumps, 50 steam turbines and 35,000 kilometers of pipelines.

B) BAKER HUGHES:

Baker Hughes Company is an international industrial service company and one of the world's largest oil field services companies. The company provides the oil and gas industry with products and services for oil drilling, formation evaluation, completion, production and reservoir consulting. Baker Hughes is organized in Delaware and headquartered in Houston. The company was originally known as Baker Hughes Incorporated until 2017 when it was merged with GE Oil and Gas to become Baker Hughes, a GE Company (BHGE). Baker Hughes operates in over 120 countries across the globe. Its headquarters are divided between Texas in the United States and the United Kingdom, with research and manufacturing facilities in Australia, Singapore, Malaysia, India, Dubai, Saudi Arabia, Italy, Germany, Norway, Louisiana and Missouri. The company provide a wide range of products and services, covering all areas of oil and gas exploration and production.

- **Oilfield Services -** provides products and services for onshore and offshore operations across the lifecycle of a well, including drilling, evaluation, completion, production, and intervention.

- **Oilfield Equipment** - provides products and services required to facilitate the safe and reliable flow of hydrocarbons from the wellhead to the production facility, including Deepwater drilling equipment, subsea production systems, onshore wellheads, and related services.

- **Turbomachinery & Process Solutions** - provides equipment and related services for mechanical-drive, compression and power-generation applications across the petroleum industry as well as products and services to serve the downstream segments of the industry.

- **Digital Solutions -** provides operating technologies helping to improve the health, productivity, and safety of asset intensive industries and enable the Industrial Internet of things, including measurement & controls and software application development platform.

C) NUOVO PIGNONE:

Nuovo Pignone is now a Baker Hughes company, a leader in advanced technology equipment and services for all segments of the oil and gas industry in more than 100 countries. The Italian company manufactures quality machinery for customers engaged in drilling offshore or subsea, enhanced oil recovery (EOR), industrial power generation, petrochemical refineries, LNG solutions and unconventional resources.

Nuovo Pignone has built gas turbines of “heavy duty” type for industrial applications since 1961. These were made in the Florence workshop under a Manufacturing Agreement with General Electric, Schenectady - N.Y. - USA, which, in time, has led to the acquisition of complete licenses (MS5002 gas turbine) and to the complete execution of some gas turbine models (turbines of the PGT range), starting from engineering and on to all construction phases. As a complement to their main activities, Nuovo Pignone converts gas turbines intended originally for the aircraft industry into packages for industrial applications which use the originary gas generator in conjunction with power turbines made by General Electric (LM range), or by Nuovo Pignone (PGT16 and PGT25 ranges). Since 1962 up to the present time, Nuovo Pignone has built about 1000 turbines, complete with all auxiliaries required for their operation.

D) ALGESCO:

ALGESCO is a joint venture between "Baker Hughes, a GE Company", "SONATRACH" and "SONELGAZ". Originally, ALGESCO was created in 1993 in Hassi R'mel, then transferred to its new industrial site in Boufarik in 2010. It is a center specializing in the provision of maintenance services and optimization of turbomachinery equipment, which contributes to the increase of operating and production capacities in the field of the Algerian oil and gas industry. This 18,300 m^2 workshop offers the most advanced technologies for the inspection, maintenance, repair and performance improvement of turbine equipment (rotors, compressors, blades, etc.). It is the largest General Electric site of its kind in the world.

The main services provided by ALGESCO are: maintenance and repair (engineering service), technical training and the purchase and resale of spare parts for turbines. The expanded services of this service center include the refurbishment of gas turbine components under their original conditions, restoring the original physical characteristics of the materials and extending the expected life of the gas turbine components. Technology areas covered include LNG, pipeline compression, pipeline inspection and integrity, gas and CO_2 storage, refinery and petrochemical applications, and remote monitoring and diagnostics of GE’s installed fleet.

I.1 Introduction [1]

A gas turbine is an internal combustion motive machine. From all points of view, it can be considered as a self-sufficient system: in fact, it is capable to aspirate and compress ambient air via its own compressor, to enrich the energetic power of air in its own combustion chamber and to convert this power into useful mechanical energy during the expanding process that takes place in its own turbine section. The resulting mechanical energy is transmitted via a coupling to a driven operating machine, which produces work useful for the industrial process in which the gas turbine belongs.

I.1.1 Stationary applications:

Stationary applications are intended for the following industrial uses:

- Generator drive, in order to produce electric energy by an open cycle.
- Generator drive, to produce electric energy by a combined cycle.
- Generator drive, to produce electric energy by co-generation.
- Compressor drive.
- Pump drive.
- Pipeline compressor drive.
- Pipeline pump drive.
- Particular industrial processes.

I.1.2 Mobile applications:

These applications were the first ones to be introduced in terms of time. They include the following fields:

- Railways.
- marine propulsion.
- aviation.
- road traction.

I.1.3 Historical Notes:

The first gas turbines to be used in operating applications appeared on the market at the end of the Forties; they were generally used in railways and presented the advantage of burning liquid fuel, even of poor quality. In this regard, we will mention the MS3001 turbine built by General Electric, with a power of 4500 HP, which was used just for this purpose. Successive achievements in material technology and extensive research into combustion resulted in rapid improvements in performance, in terms of specific power and efficiency, obtained by increasing maximum temperatures in the thermodynamic cycle. In this matter, three generations of evolution can be defined, distinguished by the maximum temperature (°C) ranges of gases entering the first rotor stage of the turbine:

- First generation 760°C < Tmax. < 955°C
- Second generation 955°C < Tmax. < 1124°C
- Third generation 1149°C < Tmax.

Obviously, to an increase in temperature there corresponded an increase in thermodynamic efficiency, which passed from values lower than 20% on the first machines to current values higher than 40% (LM6000 gas turbine).

I.1.4 Operating principle:

A gas turbine works in the following way:

• it aspirates air from the surrounding environment;
• it compresses it to a higher pressure;
• it increases the energy level of compressed air by the addition of fuel gas which undergoes combustion in a combustion chamber;
• it directs high pressure and high temperature air to a turbine section, which converts thermal energy into mechanical energy that makes the shaft revolve; this serves, on the one hand, to supply useful energy to the driven machine, coupled to the machine by means of a coupling and, on the other hand, to supply energy necessary for air compression, which takes place in a compressor connected directly with the turbine section itself;
• it expels low pressure and low temperature gases resulting from the abovementioned converting process into the atmosphere.

Fig.1 overleaf shows the behavioral pattern of pressures and temperatures, in terms of quality, inside the different machine sections corresponding to the abovementioned operating phases.

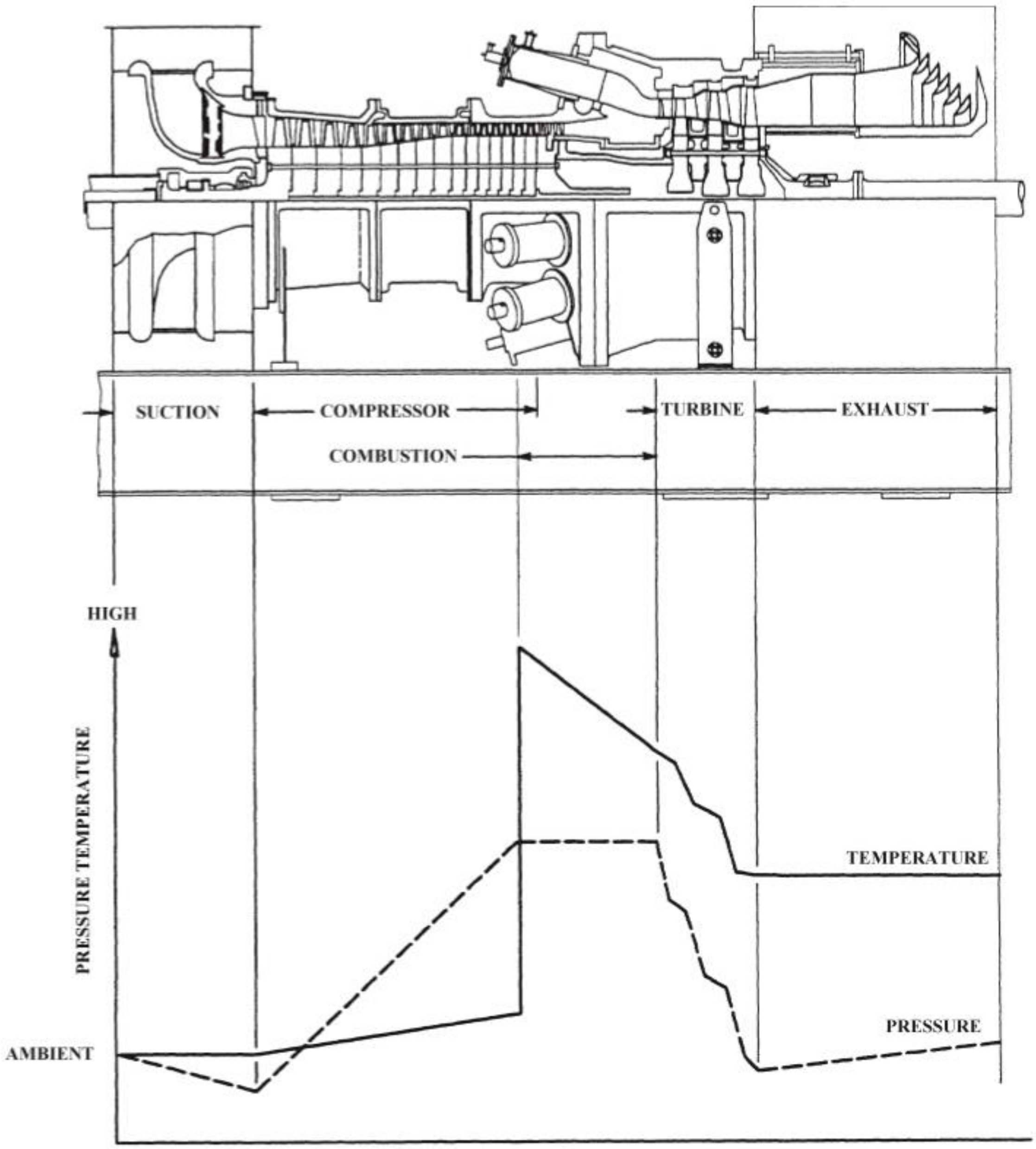


Fig.1 pressures and temperatures behavioral pattern inside the different machine sections

Fig.1 enhances the fact that combustion takes place under almost constant pressure conditions. Unlike alternative motors, compression and expansion take place on a continual basis, as happens for power generation. On the contrary, in an alternative motor (for ex., a four-stroke, eight cycle motor), though power is generated in the expansion phase, like in a turbine, this process takes only 1/4 of the complete cycle, whereas in a gas turbine expansion takes place continually all through the cycle. The same applies to compression

I.1.5 Brayton cycle: [2]

The thermodynamic cycle according to which a gas turbine works is known as the Brayton cycle. Where the working fluid is air, the cycle is composed by:

- 1-2 compression;
- 2-3 heat addition at constant pressure;
- 3-4 expansion;
- 4-1 heat rejection at constant pressure.

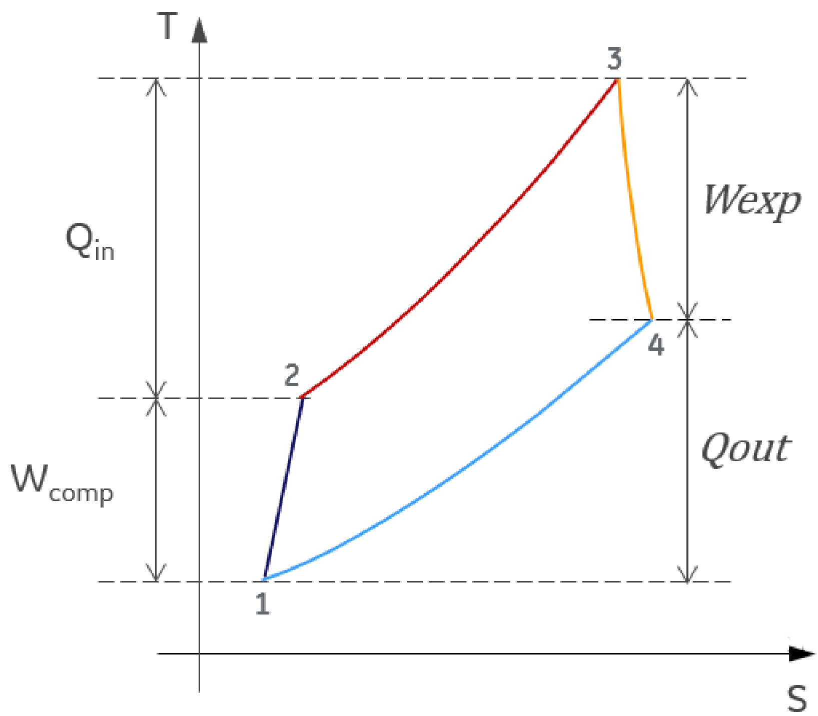


Fig.2 Typical Brayton cycle

Useful Specific Work is:

$$W_{out} = W_{exp} - W_{comp}$$

where:

$$W_{exp} \approx h_3 - h_4 = c_p\,(T_3 - T_4)$$

$$W_{comp} \approx h_2 - h_1 = c_p\,(T_2 - T_1)$$

The Useful Power is defined as:

$$P_u = P_t - P_c = G_{gas}\,W_t - G_{air}\,W_c$$

where G is mass flow and:

$$G_{gas} = G_{air} + G_{fuel}$$

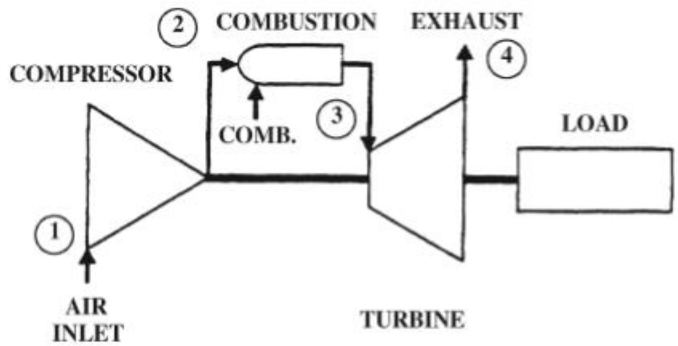


Fig.3 MS6001 single shaft turbine

Air enters the compressor at point (1), which represents ambient air conditions. These conditions are classified according to pressure, temperature and relative humidity values. It was agreed that standard design conditions be classified as ISO Conditions, to which there correspond the following reference values:

ISO CONDITIONS

Ambient temperature (°C) 15
Ambient pressure (mbar) 1013 = 101325 (Pa)
Relative humidity (%) 60

Afterwards, air is compressed inside the compressor, and exits in the condition indicated at point (2). During the converting process from (1) to (2), no heat is released outside. However, air temperature increases, due to polytropic compression, up to a value variable depending on gas turbine model and ambient temperature. After leaving the compressor, air enters the combustion area, practically under the same pressure and temperature conditions as at point (2) (except for losses undergone on the way from the compressor delivery side to the combustion chamber inlet, which amount to about 3 to 4% of the absolute value of delivery pressure). Fuel is injected into the combustion chamber via a burner, and combustion takes place at practically constant pressure.

Conversion between points (2) and (3) represents not only combustion. In fact, the temperature of the combustion process properly said, which takes place under virtually stochiometric conditions, reaches excessively high values (around 2000 °C) locally in the combustion area next to the burner, due to the resistance of materials downstream. Therefore, the conversion final temperature, relative to point (3), is lower, as it is the result of primary gases mixing with cooling and dilution air. The following transformation, comprised between points (3) and (4), represents the expansion of gases through the turbine section, which converts thermal energy and pressure into kinetic energy and, thanks to the revolutions of the power shaft, into work used for compression (internal, not useable) and external useful work, thanks to the connection with an operating machine. Over 50% of the energy developed by expansion in the gas turbine is absorbed by the axial compressor for its compressing work. Downstream of section (4), gases are exhausted into the atmosphere. The thermodynamic representation of the events described so far is visible in Fig.4 (pressure diagrams - volume P-V and temperature - entropy T-S).

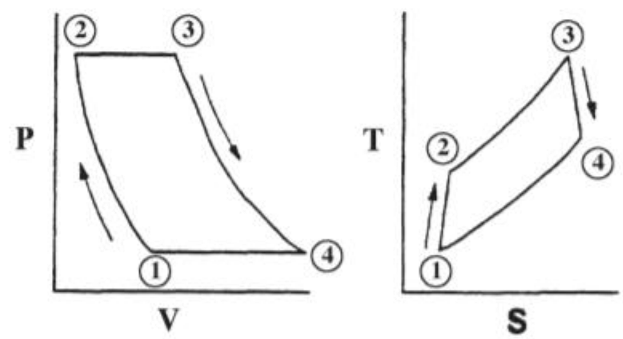


Fig.4 pressure diagrams - volume P-V and temperature - entropy T-S

In the cycle illustrated in the above figure, the 4 points correspond to the same described before. In particular, note the compression and expansion transformations, obviously these are not isentropic. In this aspect:

the specific compression work W_c, from (1) to (2), is expressed with good approximation by the following ratio:

$$W_c = C_{pm(T2-T1)} \bullet (T2-T1)\ (Kj/Kg_{asp.\ Air})$$

the specific expansion work Wt , from (3) through (4), is expressed by:

$$W_t = C_{pm(T3-T4)} \bullet (T3-T4)\ (Kj/Kg_{gas})$$

Heat Q_1, supplied to the combustion chamber from (2) to (3), is expressed by:

$$Q_1 = C_{pm(T3-T2)} \bullet (T3-T2)\ (Kj/Kg_{gas})$$

The gas turbine cycle "closes" ideally with the transformation from (4) to (1), which corresponds to the cooling of exhaust gases, in that heat Q_2 is aspirated out into the atmosphere, as though the latter were a refrigerant of infinite capability. The thermodynamic ratio that describes the cooling process of exhaust gases is the following:

$$Q_2 = C_{pm(T4-T1)} \bullet (T4-T1)\ (Kj/Kg_{gas})$$

The various values for C_{pm}, expressed in the preceding ratios, represent the average specific heat at constant pressure between the extreme temperature values in the interval examined. For a more rigorous evaluation, it would be necessary to proceed by means of integral calculation. Once Q_1, Q_2, W_c and W_t, are known, you can obtain the values for the following significant parameters:

Thermodynamic efficiency $\eta = (Q_1 - Q_2)/Q_1$ This ratio means that, by parity of heat Q_1, introduced into the combustion chamber by fuel, efficiency will increase as heat Q_2 decreases, “dissipated” into the atmosphere.

Useful work N_u supplied to the driven machine = $G_{gas} W_t - G_{aria} W_c$

In the latter ratio, G_{gas} and G_{aria} correspond respectively the weight of gases delivered to the turbine inlet section, and to the air aspirated by the compressor, necessary to pass from specific to global values. So far, all descriptions and examples refer to a single shaft turbine like MS 6001. In fact, in the diagram illustrated in Fig.3, the entire turbine section is connected mechanically with the axial compressor. Such types of single shaft turbines are suitable for driving operating machines that run at constant speed, such as alternators and, for this reason, are used typically in the generation of electric energy. In applications, in which power regulation is achieved by means of speed variation in the driven machine, two-shaft gas turbines are usually employed; in this case, the turbine is divided into two mechanically separate sections:

• A high-pressure section, which runs at constant speed within a wide range of powers, and drives exclusively an axial compressor.

• A low-pressure section, connected with the operating machine via a coupling; this section can vary its running speed independent of the high-pressure turbine section.

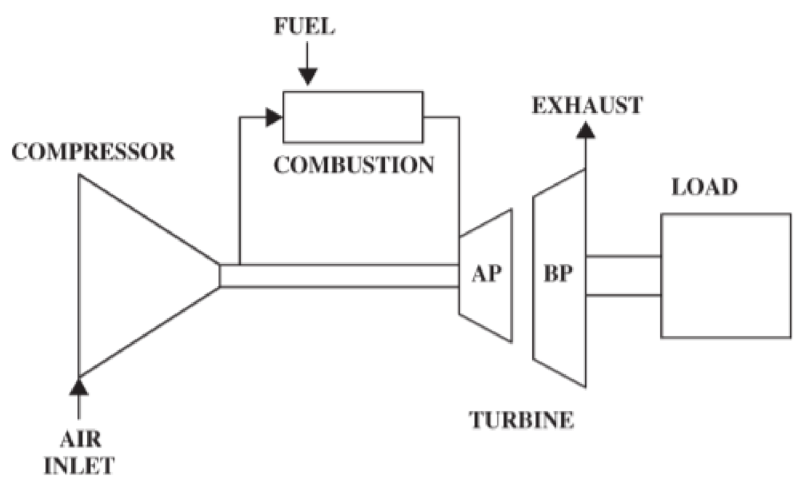


Fig.5 Two Shaft Gas Turbine Diagram

I.2 Main component parts of a gas turbine: [1]

A gas turbine, also called a combustion turbine, is a type of continuous and internal combustion engine. The main elements common to all gas turbine engines are:

- An upstream rotating gas compressor.
- A combustor.
- A downstream turbine on the same shaft as the compressor.

A fourth component is often used to increase efficiency, to convert power into mechanical or electric form.

I.2.1 Compressor:

The compressor shall be of the axial type. The choice of this type of compressor depends on the fact that this compressor is capable to deliver high air output ratings, necessary to obtain high values of useful power in a reduced size. A compressor consists of a series of stages of rotating blades, which increase air speed in terms of kinetic energy, followed alternately by stages of stator blades, which convert kinetic energy into higher pressure. The number of compression stages is related to the structure of the gas turbine and specifically to the compression ratio to be obtained. At the compressor inlet side, there are Inlet Guide Vanes (or, IGV), whose primary purpose is to direct air, delivered by the suction system, towards the first stage of rotating blades. Another important function of IGVs is to ensure a correct behavior by the compressor, in terms of fluid dynamics, under different transient operating conditions (for example, during start-up and shut down) when, due to different running speeds as apposed to normal operating speed, the opening angle of IGVs is changed: this serves to vary the air delivery rate and to restore ideal speed triangles in transient phases. Finally, in combined cycles and in the co-generation process, the capability to change the geometrical position of IGVs permits to optimize temperatures at the turbine exhaust side and, thus, to increase the efficiency of the recovery cycle by varying the flow rate of the air entering the compressor. At the compressor output side there are a few stages of Exit Guide Vanes or EGV, necessary to obtain maximum pressure recovery before air enters the combustion chamber. The compressor serves also to supply a source of air needed to cool the walls of nozzles, buckets and turbine disks, which are reached via channels inside the gas turbine, and via external connecting piping. Additionally, the compressor supplies sealing air to bearing labyrinth seals.

I.2.2 Combustion section:

In the case of "heavy duty" gas turbines, the combustion section consists of a system of one or more tubular combustion chambers arranged symmetrically and evenly in a circumference; these chambers receive and burn fuel by means of an equal number of burners (one per combustion chamber). Air enters each chamber with a flow direction inverse to that of the hot gases inside (this method of air distribution is called "reverse flow"). This external flow, which marginally touches the various chambers, serves to cool them. In addition, the part of air that does not take part in the combustion process is used for cooling the combustion products, it is introduced into the chambers through "diluition" holes until optimal temperature conditions are established to allow the gas and air mixture into the turbine section. Air passage from the combustion section properly said to the gas turbine inlet takes place inside manifolds called "transition pieces"; here, the gases flowing out of the single combustion chambers are led to form a continuous annular profile, equal to the one that leads into the first stage nozzles ring. Initially, the combustion process is ignited by one or more spark plugs. Once ignited, combustion proceeds in a self-sufficient way without the help of spark plugs, as long as the delivery conditions of fuel and combustion air are fulfilled. In the case of gas turbines built for the aviation industry, the combustion section consists of a single chamber of toroidal shape, with direct and not "reverse flow" cooling, this helped reduce external diametral sizes, since a smaller frontal section was needed in order to offer as little resistance as possible to aircraft motion. For the same reason, this combustion chamber does not need any separate transition pieces. The other operating principles are the same as those described for tubular chambers.

I.2.3 Turbine section:

In the case of "heavy duty" turbines, the turbine section comprises a certain number of stages (in this specific case, three stages), each one of them consisting of one stator stage (distributor nozzle). In this stage, high-temperature and high-pressure gases delivered by the transition piece are accelerated and directed towards the rotor stage of buckets mounted on a disk connected with the power shaft. The conversion of thermal energy and pressure into kinetic energy takes place in the stator stage. The rotor stage completes this conversion, as here kinetic energy is transformed into energy that drives the shaft, thus generating the power required to drive the compressor (internal compression work, cannot be used as externally useful work) and to operate the driven machine (generator, compressor, etc.) connected to the gas turbine by means of a coupling. The energy of gases supplied by the combustion system can be varied by changing the delivery rate of fuel. In this way you can regulate the useful power values needed for the technological process of which the gas turbine is the motor.

I.2.4 Single shaft and two shaft gas turbines:

A) Single shaft gas turbine:

the compressor and turbine sections on these machines are made up of a single rotor, in which the energy absorbed by compression is expanded in the turbine section to produce energy useful for load drive.

Fig.6 overleaf shows schematically the operation of a single shaft gas turbine, in which there is a continual mechanical connection between the entire turbine section T, compressor C and, on the opposite side, load G.

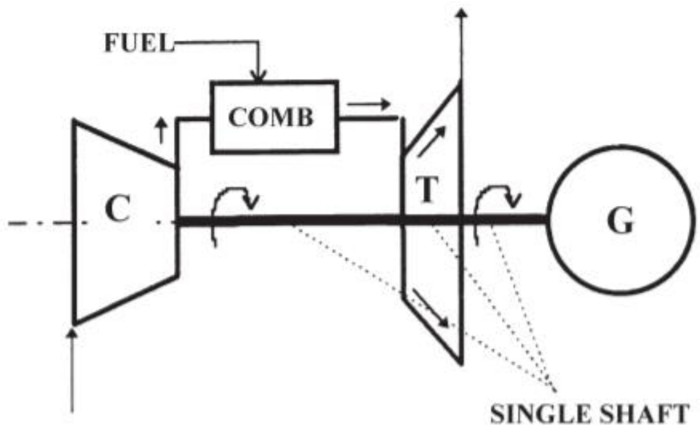


Fig.6 Single shaft turbine diagram

<u>B) Two shaft gas turbines</u>:

Unlike single shaft gas turbines, on two shaft gas turbines only one part of the turbine section (called "high pressure turbine" Thp) is mechanically connected with the compressor to build the so-called "gas generator", while the remaining energy obtained by combustion is processed by a second turbine section (called "low pressure turbine"), mechanically separated from the former, to build the so-called "power turbine" Tp, which is connected with load L by a coupling.

Fig.7 shows a schematic view of the operating cycle of a two-shaft gas turbine, with the two sections ("modules"), gas generator and power turbines, mechanically separated.

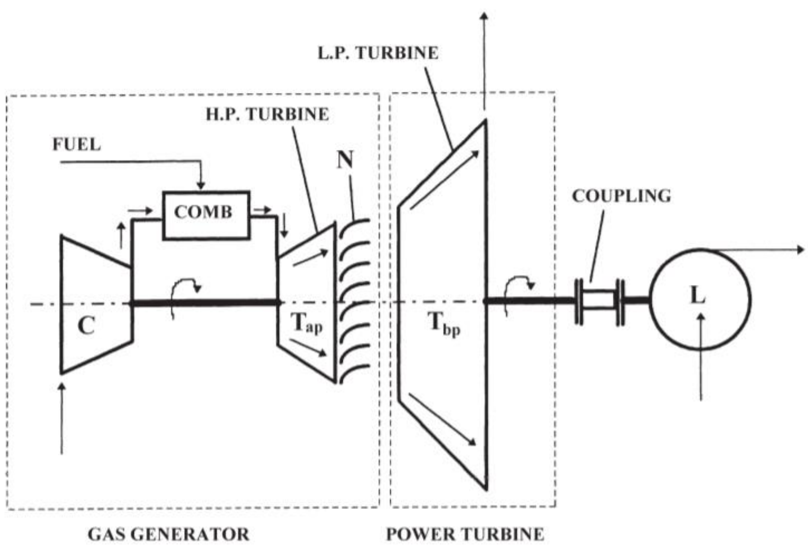


Fig.7 Two shaft gas turbine diagram

I.3 Conclusion:

In this chapter, we explained what a gas turbine is, its applications, operating principle, Brayton cycle and the main sections of the gas turbine. In the next chapter we are going to be talking in-depth about each section of the turbine.

II.1 Introduction:

The working principle of gas turbine is to convert the chemical energy of fuel into mechanical energy through combustion process, and then the mechanical energy is converted by a generator into electrical energy. The simplest gas turbine system consists of 3 main sections: compressor, combustor and turbine.

In this chapter, an in-depth study of each section's components, design and materials used, citing similarities and differences of the components in heavy-duties and aero-derivatives.

II.2 Compressor: [1] [7] [8] [9]

Axial compressors that are employed on industrial, heavy duty gas turbines were initially designed on account of the experience acquired with compressors employed in aviation. Successively, in the early '50s, in parallel with growing demands by the electric power generation market and following the introduction of NACA blade profiles, notable advancements were made concerning air delivery capacity and compression efficiency. In this phase, the "air extraction" system was introduced to avoid air pumping during start-up and shut down. Over the years, the compressor inlet section was enlarged progressively, thus augmenting both the air delivery capacity and the compression ratio; inlet guide vanes (IGV) were introduced to optimize compressor stability throughout the startup and shut-down phases, in addition to one or more stages for air extraction. Later, the IGV profiles were modified on some gas turbine models to further increment the air delivery rate and turbine performance. Another process still in use for increasing the air delivered by the compressor consists in the addition of one stage upstream of the first one (known as "stage zero"). In the case of a PGT5 gas turbine, for example, stage zero resulted in a 18% increase in air delivery, with a power increase of the same magnitude. The choice of a compressor of axial type is motivated by the need to reach high air delivery rates with moderate compression ratios and by a historical reason, according to which compressors heavy duty turbines were made on the basis of the experience acquired with compressors employed in aviation; these were always of axial type in order to offer minimum aerodynamic resistance to aircraft motion.

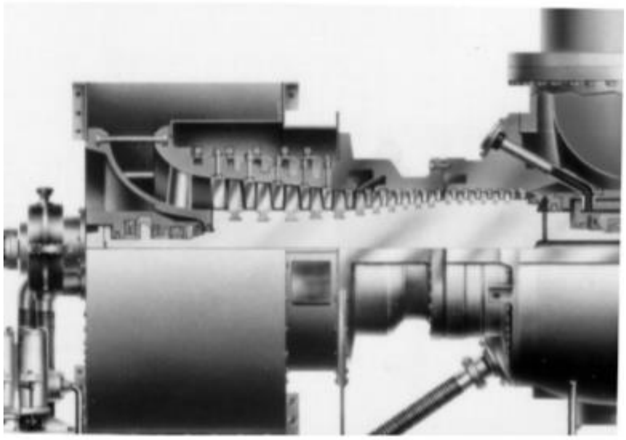


Fig.8 shows the axial compressor of the PGT10 gas turbine

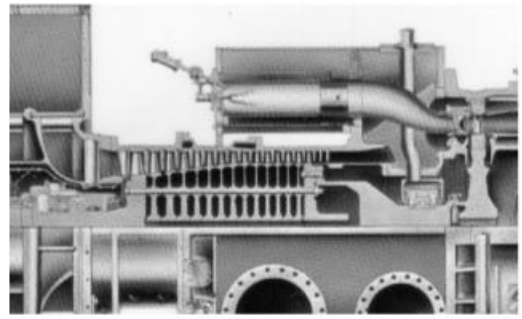


Fig.9 axial compressor of the MS5002 gas turbine

Fig.10 shows a two-stage centrifugal compressor of the PGT2 gas turbine

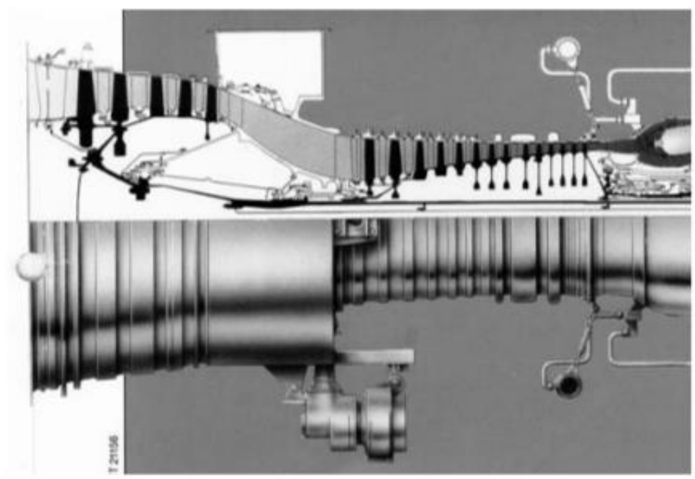


Fig.11 that of an aeronautical gas turbine (LM6000)

In PGT10 (fig.8), the compressor shaft was made entirely from a single forged piece, machined in order to insert the various rotor buckets.

In MS5002 (fig.9), instead, the compressor is composed of discs contoured in a way to reach high peripheral speeds and joined together axially by perimetral tie-rods (Fig.12) overleaf, or by a bolted structure made lighter for turbines derived from aeronautical models (Figs. 13 and14) overleaf.

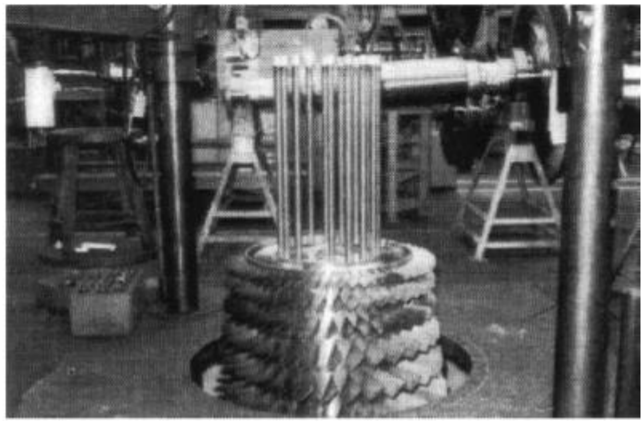


Fig.12

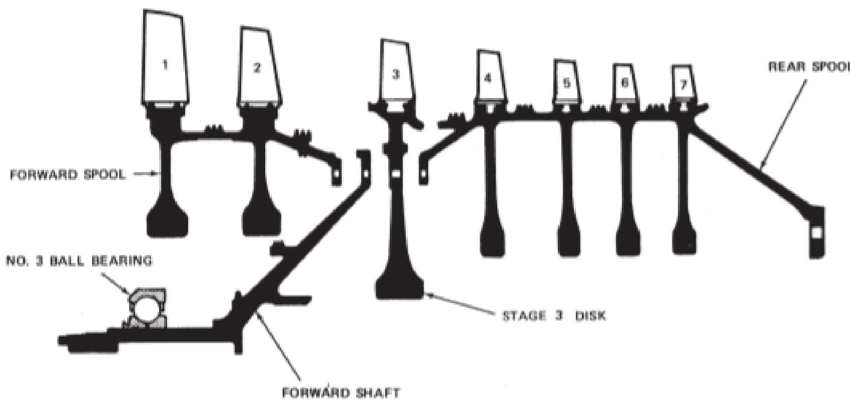


Fig.13

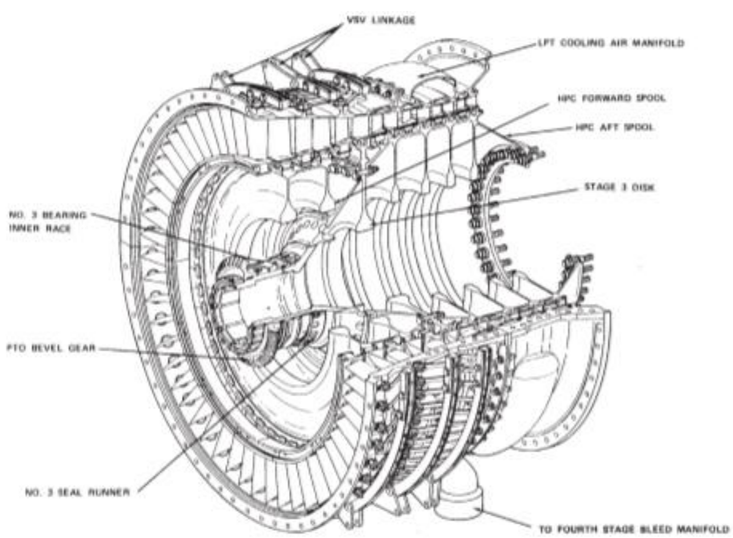


Fig.14

The reason for these choices resides in the different dimensions of the two compressors; in fact, dimensions are related to the output power that both machines must generate. If the compressor on the MS5002 gas turbine was made entirely from one forged piece like the one on the PGT10 gas turbine, it would weigh so much as would cause practically unsolvable problems to find bearings of suitable dimensions; furthermore, a much more powerful and costly starting system would be needed in order to accelerate such a bigger mass during start-up. All Nuovo Pignone gas turbines are equipped with compressors whose rotors have multiple discs, except for PGT10, PGT5 and PGT2 gas turbines. The rotor in fig.11 is composed of various forged sections, of the type indicated in fig.13 (PGT16 compressor). This solution greatly reduces its weight, as necessary for aeronautical applications.

II.2.1 Stator casings:

On heavy duty gas turbines:

Compressor stators casings are built of high resistance cast iron, in which the following elements are made:

• compressor inlet section (called “inlet bell mouth”).
• stator blade housings.
• support of bearing n°1 and of stator labyrinth seals.
• control system for variable inlet guide vanes.
• air apertures for cooling hot parts and sealing bearings.
• blow-off air bleeding holes to prevent phenomena such as compressor surging or instability during the gas turbine start-up and shut-down phases.

On aeroderivative gas turbine:

Casings are made in light alloy. Figures 15-16 overleaf show a compressor closeup, with stator casings open for installation/inspection purposes, and the support of bearing n° 1. Casings are generally composed of a number of housings split along the horizontal centerline and coupled to one another by vertical, bolted and pinned flanges that serve to maintain casings correctly aligned with one another. This is necessary in order to observe correct concentricity tolerance limits, and thus to maintain correct clearances between rotating and stator parts; this is of capital importance in order to avoid rubbing and, in the opposite case, excessive clearances, with consequent loss of efficiency.

On machines of aeronautical type, most of the frontal casings are built in light alloy (generally, aluminum-based), while the casings that house high pressure stages are made of austenitic steel or nickel-based super alloys, because of the high temperatures caused by the high compression ratios at work.

Fig.14 shows in detail the outside of casings on the H.P compressor of the LM1600-PGT16 gas turbine.

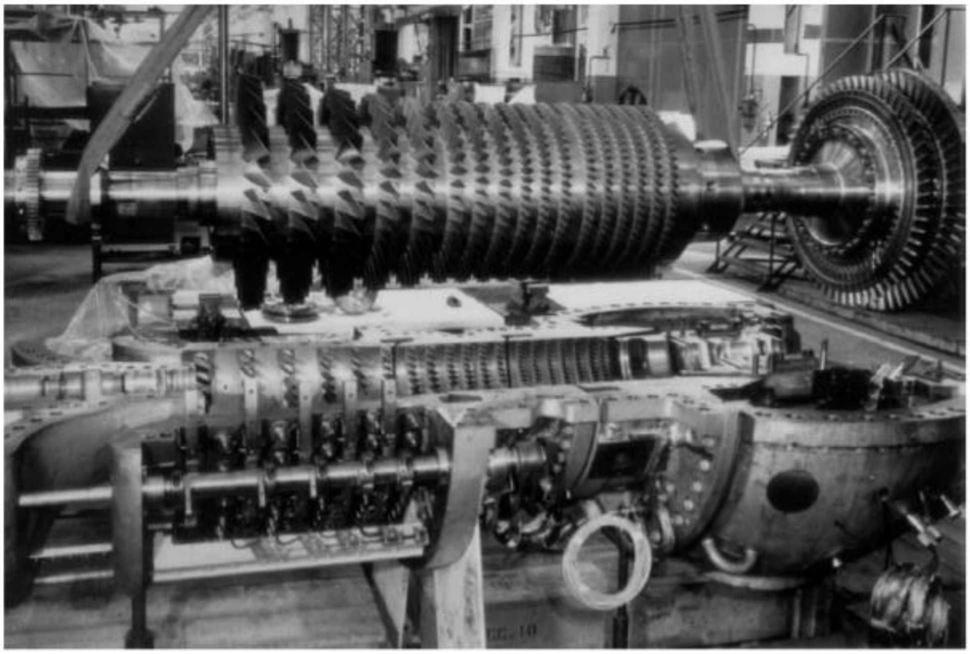


Fig.15

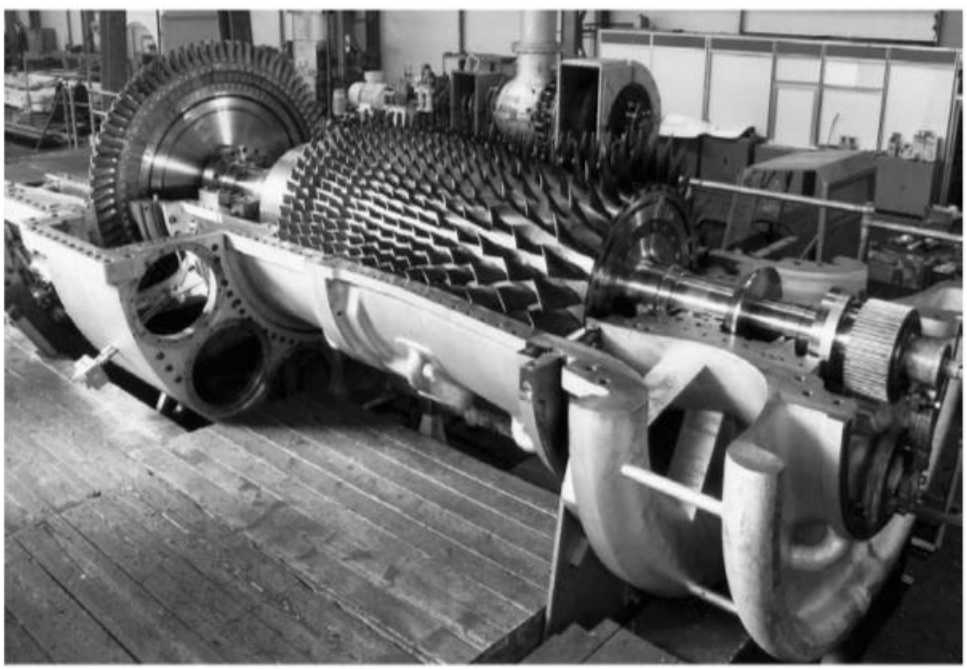


Fig.16

II.2.2 Stator blades:

On heavy duty gas turbines:

The material of which stator blades are made is of the X15Cr13 type or equivalent, capable to withstand atmospheric agents contained in the air aspirated by the compressor.

On aeroderivative gas turbine:

The material of which stator blades on compressors installed on gas turbines of aeronautical are built type is titanium-based for low pressure stages, and austenitic steel (stainless steel) or super alloys for final stages, where temperatures reach the highest values.

Fig.17 shows an example of stator blades fitted on the PGT10 turbine inlet casing.

II.2.3 Inlet Guide Vanes (IGV):

IGVs are moved by a toothed ring that engages into pinions fitted on each pin of the IGVs themselves, in a way to produce even angular movement, as required to obtain correct air flow to the inlet of downstream stages.

Fig.18 shows a view of inlet guide vanes IGV fitted on the compressor inlet casing.

Fig.19 shows a detail of the blade, with the characteristic rotating pin that fits into the casing bushing.

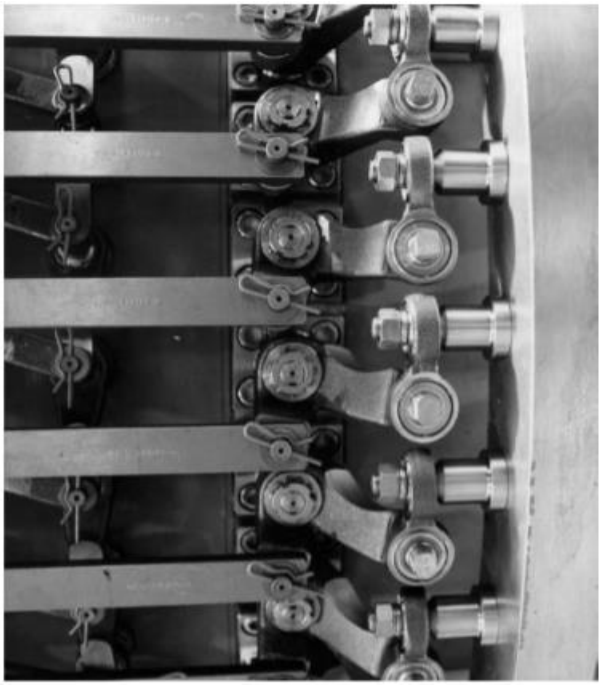


Fig.18

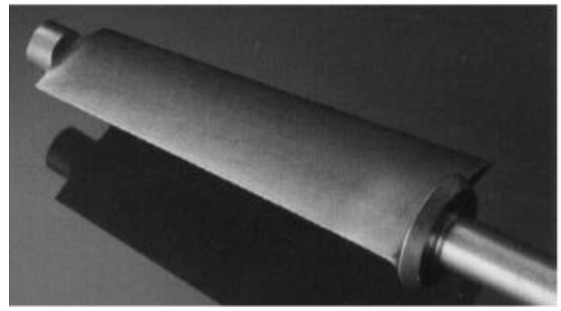


Fig.19

On heavy duty gas turbines:
The material of which heavy duty inlet guide vanes IGV are made is GTD 450 (or C 450), a type of stainless steel particularly resistant to corrosion.

On aeroderivative gas turbine:
On aeronautical turbines titanium-based materials, stainless steel or super alloys are used, according to the temperatures reached inside stages, which depend on pressure.

II.2.4 Rotor blades:

Rotor blades are built in the same material as stator blades for each turbine, and are provided with the same additional protection.

Fig.20 shows a view of rotor blades, with the typical dovetail root, which fits into the relative cavities present on the peripheral surface of the rotor discs.

Fig.20

II.3 Combustion:

It is composed of multiple, "reverse flow" combustion chambers of tubular shape. This system is the fruit of years of intense development, which was obtained by taking into account the excellent performance results achieved in various fields of application, using different types of fuel gas and under different duty conditions, that is peak, base, regenerative, etc

The choice of this design solution has following advantages:

• Possibility to carry out laboratory tests in a single, 100% scale combustion chamber, these characteristic permits to reproduce and, thus, to foresee the actual behavior of the combustion process in terms of chemical reactions, heat transmission, fluid dynamic coefficients and emissions such as Nox, Co, etc., in which respect precise legislative prescriptions are in force. The possibility to run trial tests with a single combustion chamber permits to limit the quantity of air and fuel employed, and thus to simplify the combustion laboratory equipment.

• Flexibility due to the possibility to vary the diameter and/or length of the liner in order to adapt it to a large variety of fuels.

• Smaller dimensions of component parts permit easier cooling, reduce thermal and mechanical stresses and maintenance work.

In addition to the above advantages it must be remarked that the diameters of combustion chambers, unlike other primary component parts of a gas turbine no "scale" criterion is applied to the various machine models, except a different principle: keeping diameters equal, the number of combustion chambers is varied in proportion to the amount of air delivered divided by the compression ratio. In this way, for example, on the MS 3002, 5001, 5002 and 6001 gas turbines there are respectively 6, 10, 12 and 10 combustion chambers whose rated diameter is 268 mm, whereas on the MS 9001E and MS 7001E gas turbines there are 14 and 10 combustion chambers, whose rated diameter is 358 mm, according to the above-mentioned rule of amount of air delivered and compression ratios on the different machines. A clear advantage of this rule is that it is possible to standardize and accumulate precious data from operating experiences.

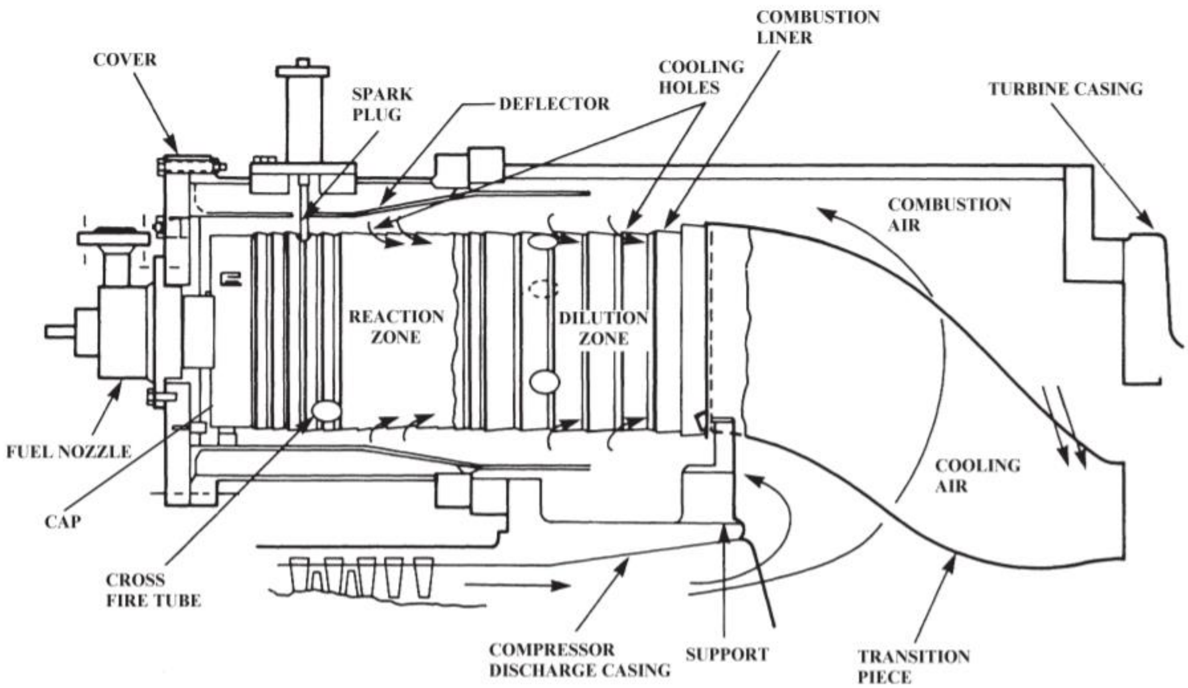


Fig.21 shows the main component parts of the combustion chamber

On aeroderivative gas turbine:

On machines built for the aviation industry, the need for smaller radial dimensions in order to limit aerodynamic resistance to aircraft motion was faced by constructing an "annular" combustion chamber. (as shown in Fig.22 and Fig.23)

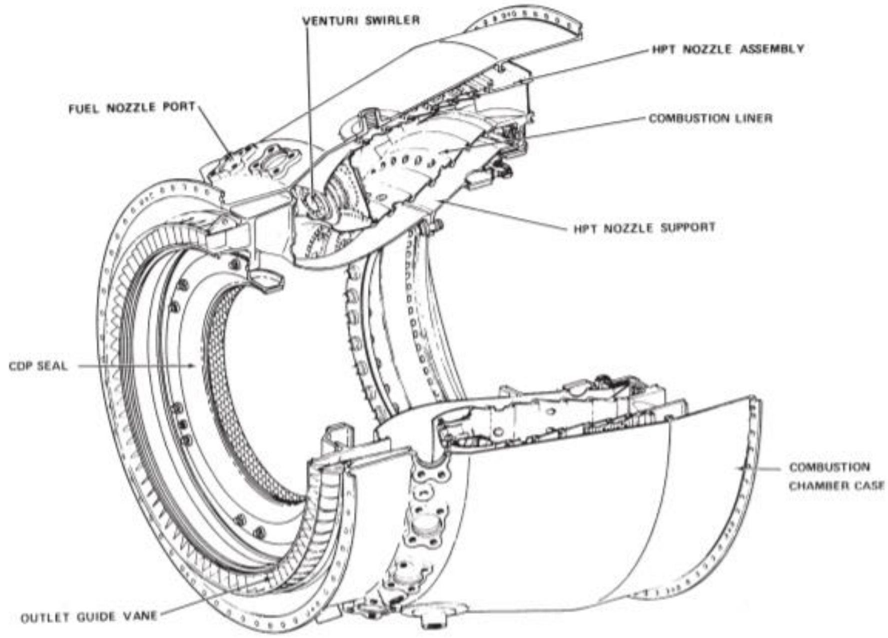

VENTURI SWIRLER
HPT NOZZLE ASSEMBLY
FUEL NOZZLE PORT
COMBUSTION LINER
HPT NOZZLE SUPPORT
CDP SEAL
COMBUSTION CHAMBER CASE
OUTLET GUIDE VANE

Fig.22

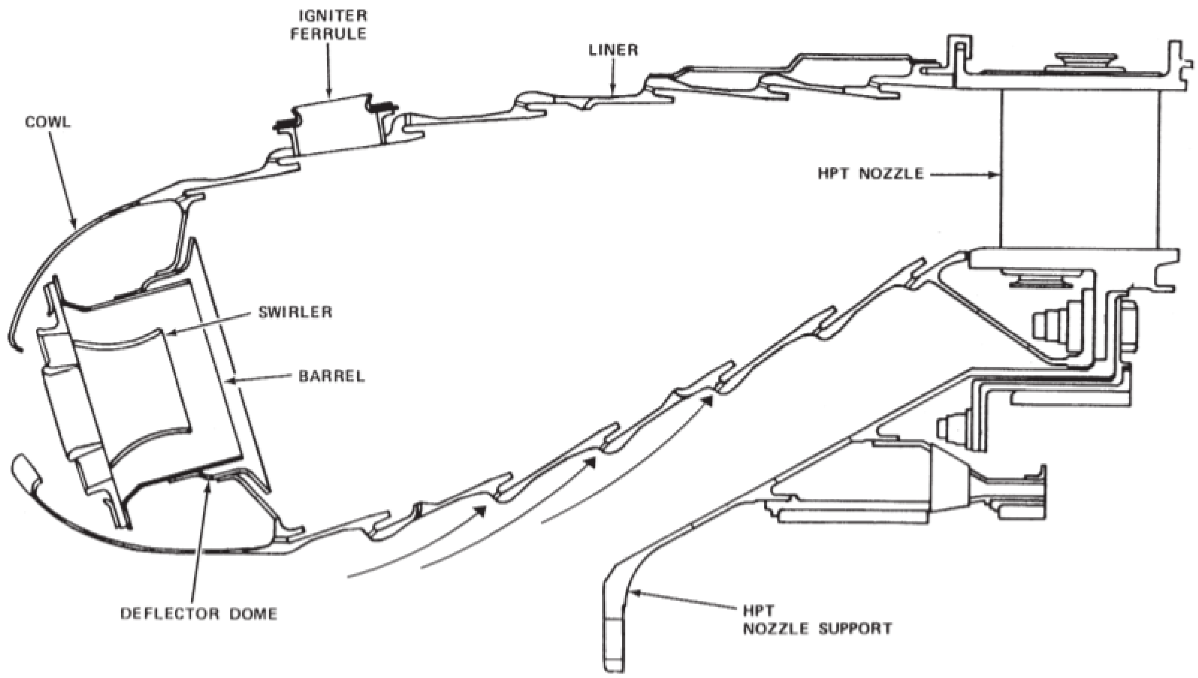

IGNITER FERRULE
LINER
COWL
HPT NOZZLE
SWIRLER
BARREL
DEFLECTOR DOME
HPT NOZZLE SUPPORT

Fig.23

On heavy duty gas turbines:

On heavy duty machines, the flame is initially ignited by two spark plugs, positioned in two diametrically opposed combustion chambers, then it propagates to the other combustion chambers via “cross fire tubes”. Meanwhile the spark plugs retract, contrasted by a spring, by effect of growing pressure in the combustion chamber as the gas turbine start-up sequence goes on. In this way, the spark plug is subtracted from direct action by combustion gas. On gas turbines with a single combustion chamber (PGT2, PGT5 e PGT10) there is only one spark plug, and no cross-fire tubes. On machines of aeronautical derivation, the annular shape of the combustion chamber does not require any cross-fire tubes. Hot gases produced by combustion proceed inside the liner into the transition piece, whose shape permits to transform the gas flow from cylindrical into annular, before it reaches the gas turbine first stage. In the case of an annular chamber, there is no transition piece, as hot gases already follow an annular path.
Air enters the combustion system from axial compressor discharge casing to implement the following functions:

• it becomes combustion air (in the combustion area adjacent to the burner);

• to cool the transition piece walls in inverse direction to that of the hot gases inside (“reverse flow”);

• it cools the combustion chamber liner by flowing in inverse direction to that of the hot gases inside the chamber ("reverse flow"), whereas flow is direct in the case of machines of aeronautical derivation;

• it dilutes high temperature gases originated by combustion, in order to reduce their temperature to a value acceptable by the gas turbine materials before gases enter the turbine. The following are the constructive details for the single component parts of the combustion system.

II.3.1 Combustion cap & liner

The primary purpose of the combustion chamber is to ensure flame stability throughout all operating phases, such as ignition, start-up, maximum load, unexpected variations in load conditions (load rejection), etc. As firing temperatures were increased to achieve ever better performance results, an ever-greater quantity of air was needed in order to restore the correct stochiometric ratio to the increased quantity of fuel, therefore the amount of air available to cool the combustion chamber liner diminished on a percentage basis. To obviate the difficulty of providing a sufficient quantity of cooling air, “slot cooled” combustion chambers have been in use for several years, they are shorter so a smaller area needs to be cooled.
The “slot cooling” method serves to reduce liner temperatures by about 140 °C, if compared to a louver cooled liner. Like many other improvements introduced over the years, also this one derives from experiences acquired with aeronautical applications. The length of a combustion chamber is a function of the time required to have a complete combustion reaction, which varies according to the type of fuel, and to have the necessary dilution and radial and circumferential distribution of temperatures acceptable for the component parts downstream. The latter characteristic is very important, because correct temperatures along the radial centerlines of blades and nozzles greatly prolong their operating life. This optimization is obtained by

appropriately arranging dilution holes, in a way that minimum temperatures are obtained at the roots of blades (in which the effect of the centrifugal force is at its maximum) and at the outer diameter of nozzles, where flexional stress is at its maximum. The material used for combustion chambers is a nickel-based super alloy (Hastelloy X, INCO 625 or equivalent).
This type of combustion chamber (Fig.24) has now been adopted on most gas turbines instead of the "louvered" version (Fig.25).

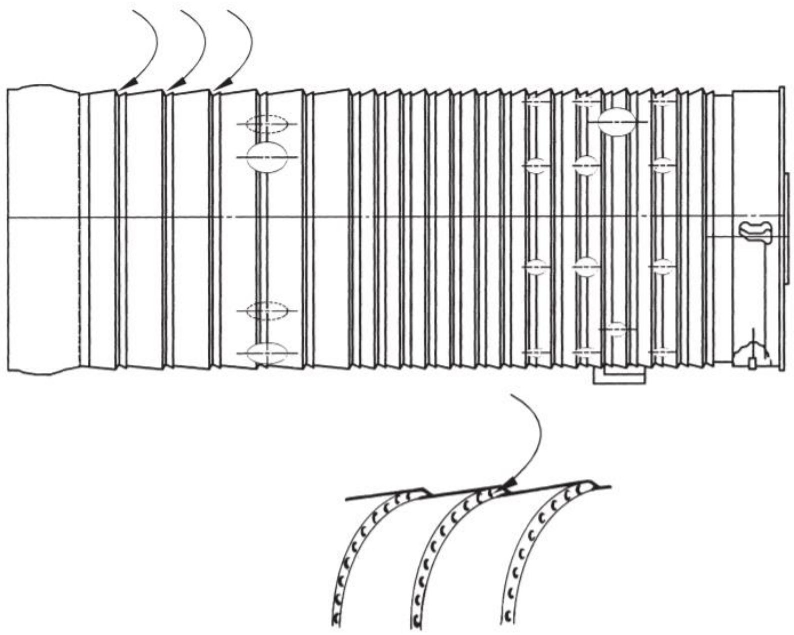
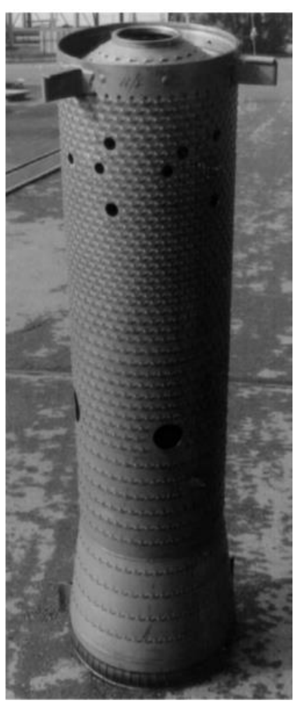


Fig.24



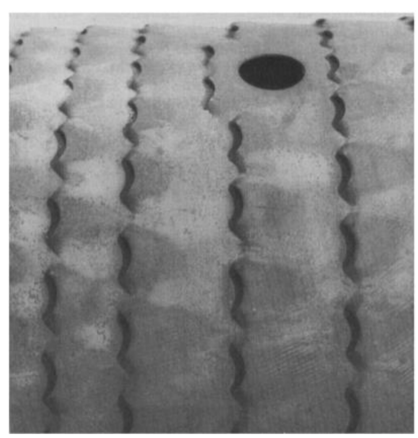


Fig.25

II.3.2 Transition piece:

the purpose of the transition piece is to transform the cylindrical flow of gases upstream of the combustion chamber of heavy-duty gas turbines into an annular flow at the inlet to the 1st stage nozzle of the gas turbine downstream. On gas turbines which have only one combustion chamber (PGT2, PGT5 and PGT10), the transition piece is built of two parts divided at the centerline for assembling reasons.

Fig.26 transition piece

As for other machine models, which have more than one combustion chamber, the transition piece is built of as many parts as are the combustion chambers joined to one another. Each one of these parts transforms the cylindrical flow of gases from the various combustion chamber into one annular flow segment.

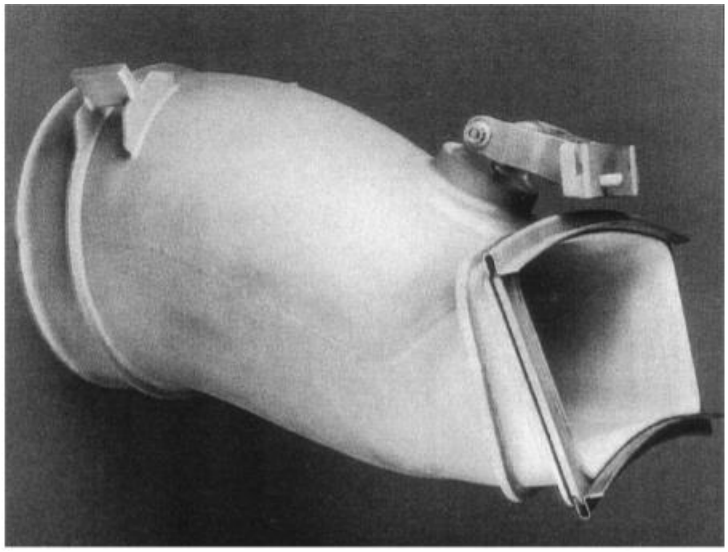


Fig.27 one annular flow segment

The transition piece is built in the same material as the combustion liner. The combustion liner fits into the transition piece mouth by means of springs arranged circumferentially, which allow the two component parts to dilate both in radial and in longitudinal sense (Fig.28). The transition piece is connected with the gas turbine 1st stage nozzle by means of flexible metal strips and, in some cases, by means of hinges, which serve to limit stresses caused by thermal transients or induced by the combustion process itself.

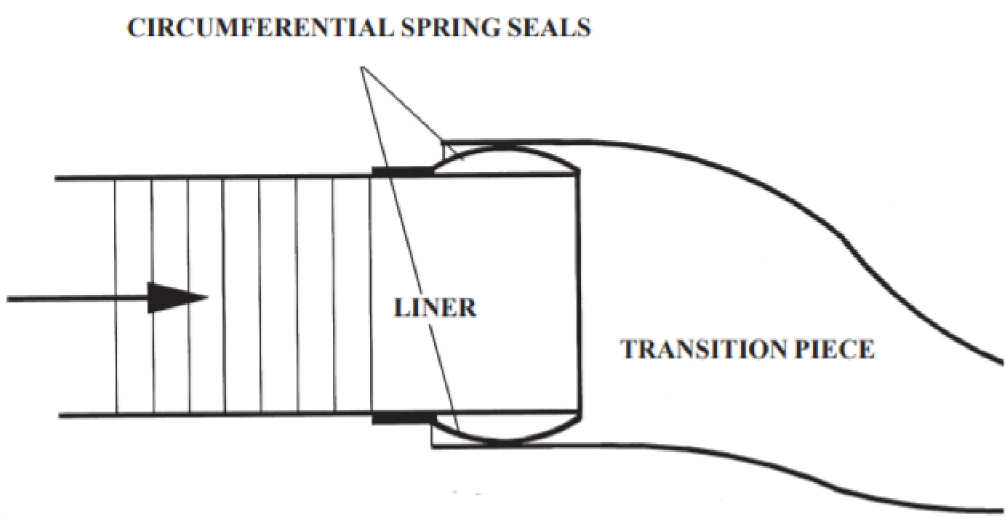


Fig.28

II.3.3 Burner:

Each combustion chamber is provided with a burner, connected with the combustion chamber outer housing aligned radially with a central hole in the cap in a way to leave a certain clearance. There exist three types of burner:

- for fuel gas
- for liquid fuels
- Dual fuel

Fig.29 shows the terminal part of a dual fuel burner for natural gas and gas oil, and a sectional view typical of a complete dual fuel burner. The part that concerns the gas burner is the one with external holes arranged circumferentially. The latter are inclined, in order to generate a swirl that helps fuel gas mix with air in a correct way. The dimensions of flow passage areas in gas inlet holes play a very important role in order to ensure flame stability and correct shape under all operating conditions. These dimensions depend on several factors, such as the fuel calorific value, molecular weight and the rated power the gas turbine must generate. In the case gas oil is used, this liquid will enter the combustion chamber through the central hole. Gas oil supply pressure must be regulated as needed to help liquid fuel be atomized correctly, which is a condition necessary for the combustion process to take place correctly and to produce few fumes. To favor atomization, compressed air is taken from the compressor and sent into the burner, where it mixes with the liquid fuel in the terminal section in a swirling motion induced by a series of swirlers.

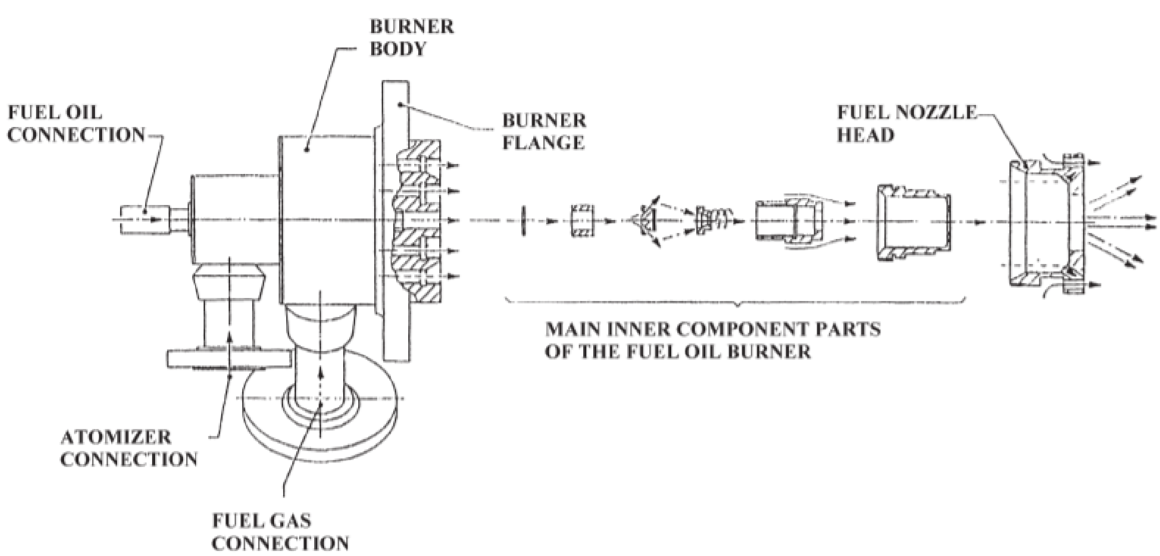


Fig.29

II.4 Turbine:

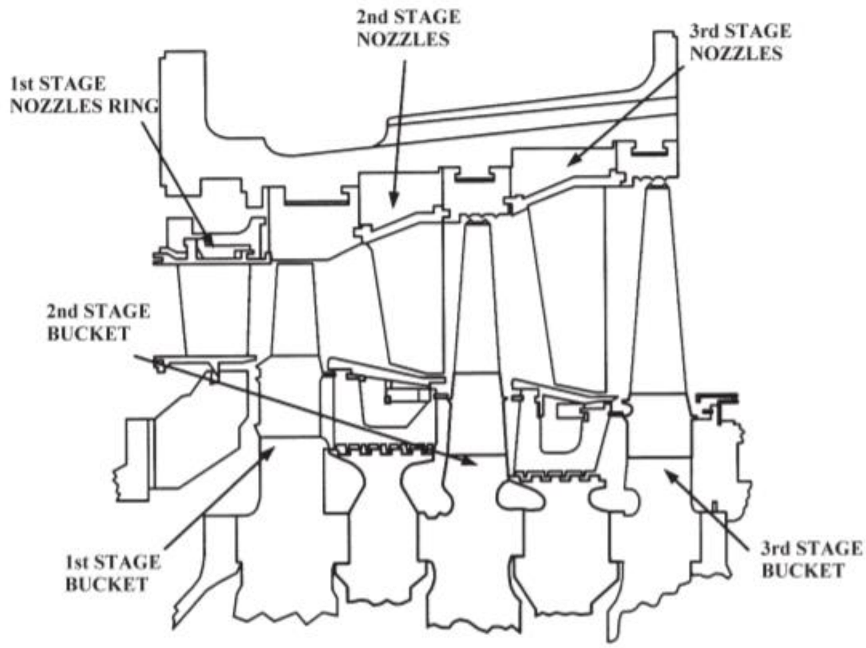


Fig.30

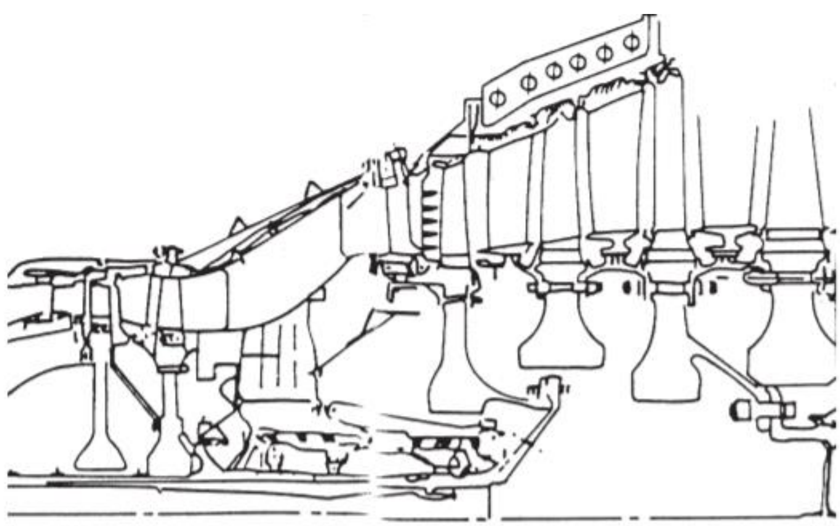


Fig.31

Fig.30 illustrates the part that is properly defined as "turbine section" in a heavy-duty gas turbine where the main component parts are highlighted.

Fig.31 shows a sectional view typical of a gas turbine of aeronautical derivation.

On heavy duty gas turbines:

A characteristic peculiar to heavy-duty gas turbines is that they have high energy stages or, more correctly, a low degree of reaction. As a consequence, the conversion of energy from the combustion chamber to the exhaust section can be carried out with a reduced number of turbine stages. Heavy-duty gas turbines have two or maximum three stages. This configuration presents several advantages:

• A reduced number of stages calls for a reduction in the number of component parts, thus simplifying construction and increasing machine reliability.

• A low reaction degree at equal firing temperature accounts for lower temperatures on the rotor metal blades (in the 1st stage, the difference in temperature amounts to about 50-60 °C), and for a smaller quantity of cooling air, which contributes to increase thermodynamic performance. Similarly, keeping the quantity of cooling air equal, the temperatures reached by metal blades will be lower, which will prolong the life of the blades.

On aeroderivative gas turbine:

On jet derivative gas turbines, the high compression ratio calls for a larger number of expansion stages, so the considerations expressed above cannot be applied to this case, due to different construction needs.

Fig.32 shows a sectional view of a gas turbine section opened at its horizontal centerline during assembling operations.

II.4.1 Gas turbine cooling process

Cooling critical parts of a gas turbine is necessary to prevent the same from deteriorating as an effect of the high temperatures involved in the operating process. Firing temperatures were progressively increased to values closer and closer to the melting temperature of materials (super alloys), therefore, to keep gas firing temperature high (which provides the required thermodynamic efficiency) without damaging or drastically shorten the life of hot parts, it was necessary to cool the metal of which said parts are made. The air needed for this purpose is obtained by bleeding one or more stages of the axial compressor depending on machine model and on the levels of pressure/air flow required to cool the various turbine stages.

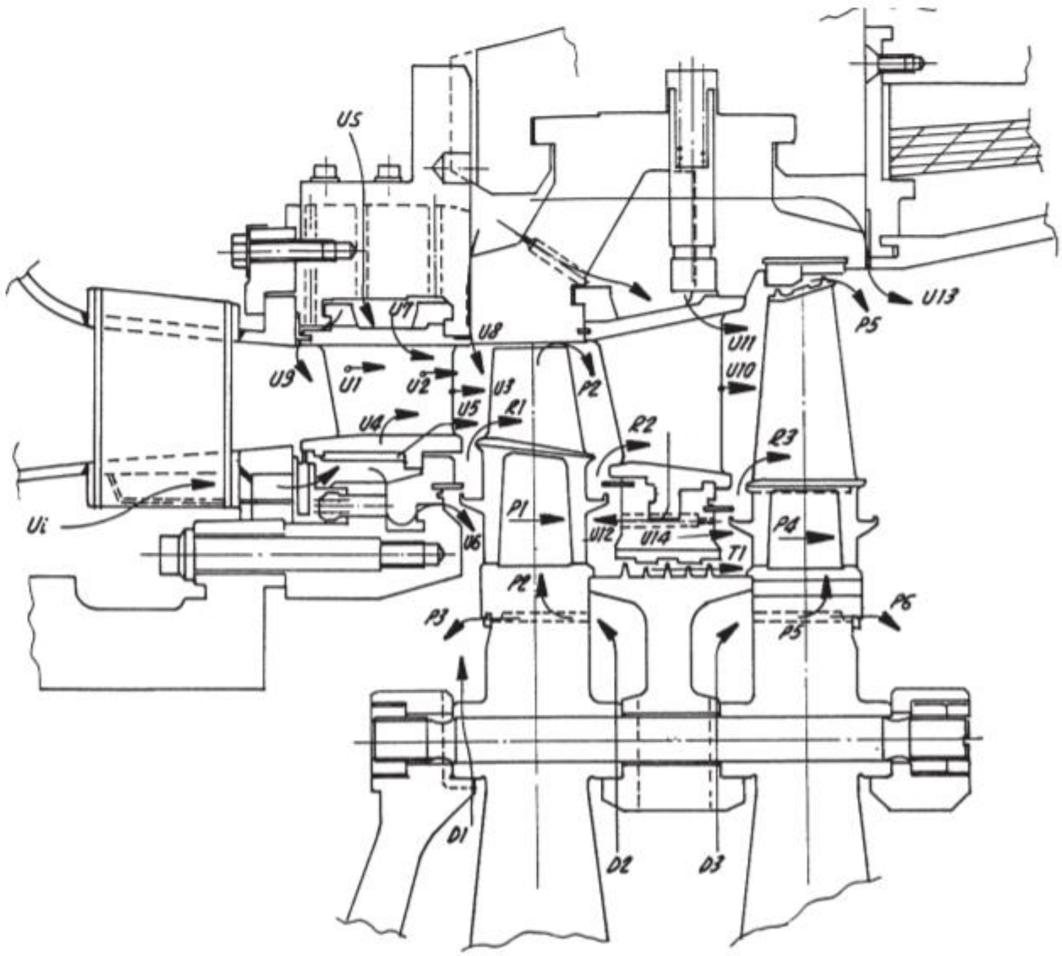


Fig.33 Gas Turbine Cooling Diagram

Fig.33 shows an example of how the various component parts are cooled, along with the origin and path followed by the air bled from the compressor. It must be added, finally, that optimizing the flow rates of fluids delivered to each part of the turbine is one of the most critical problems to be faced during the designing and testing phases of a prototype. In fact, a "conservative" exorbitance of cooling air would be positive for the duration of machine life, but would be negative for its efficiency, and vice-versa. In fact, cooling air does not take part in the combustion process and by-passes the thermodynamic cycle, in this way compression work (absorbed) being equal, useful work is reduced.

II.4.2 Nozzle:

The purpose of nozzles is to transform the energy contained in gases proceeding from the combustion system (combustion chamber + transition piece) by augmenting their kinetic component and reducing their pressure and temperature (expansion). The high kinetic energy of gases is converted into mechanical energy by way of changing the gas direction when these pass through the blades in the successive rotor stage, which results in a series of tangential forces that produce the shaft revolving torque (fig.34).

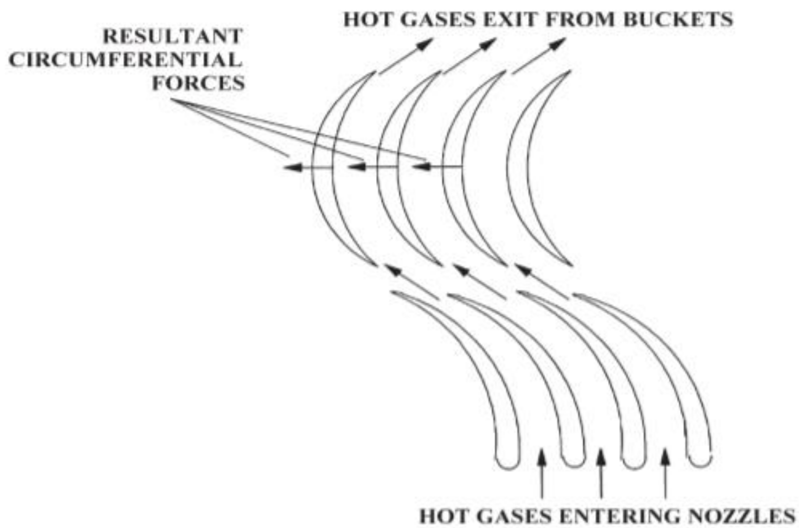


Fig.34 Energy transformation through a turbine stage (plan section)

A nozzle, in the current definition of the term, consists of a series of contiguous segments, each one composed of one or more compartments, all together they build a ring that couples itself at its inlet to an equal ring, which corresponds to the distribution mouth of the various transition pieces.

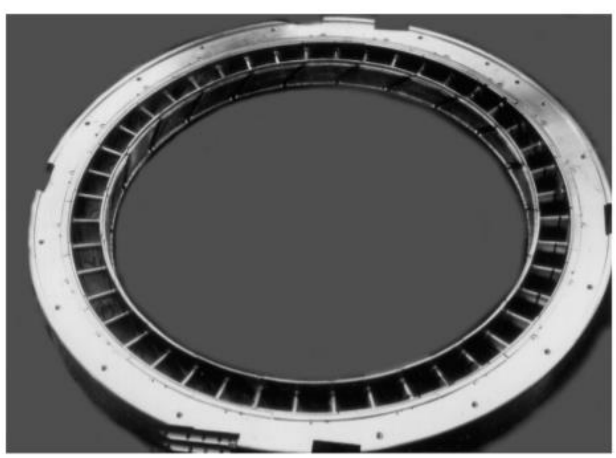


Fig.35 PGT10 1st stage nozzles ring

Fig.36 Segment of PGT10 2nd stage nozzles

They are made of materials with a high cobalt content, listed in the following table, which undergo precision investment casting:

Table 1: Nozzle materials:

Nominal percentage composition

Alloy	C	Cr	Ni	Co	W	Fe	B
X40	0.50	25	10	Diff.	8	1	0.01
X45	0.25	25	10	Diff.	8	1	0.01
FSX 414	0.25	29	10	Diff.	7	1	0.01
N155	0.20	21	20	20	2.5	Diff.	-
GTD222	0.10	22.5	Diff.	19	2	-	0.008

GTD222 contains also Ti, Cb and Al, to increase resistance to creep and corrosion.
N155 is used for nozzles in stages following the first one.
The choice of this class of materials is due to the need to meet the following requirements:
- good melting properties, as required by the particularly complex casting procedure of segments.
- High mechanical characteristics at high temperatures (the highest in the entire turbine section), and above all, resistance to the various thermal cycles that are created in start-up, stop transient, during shut-down and abrupt load variations.

Such kind of stress, due to thermal transients, is commonly called "thermal fatigue" and is the main cause of stress for nozzles. For these reasons, 1st stage nozzles are cooled by air sent by the compressor.
Air enters the nozzle inner cavity, reaches the inner surfaces of the same through a hollow space and cools it by means of film cooling, convection and impingement cooling (fig.37 and 38).
At completion of the cooling cycle, air flows out through a series of openings near the nozzle output edge.

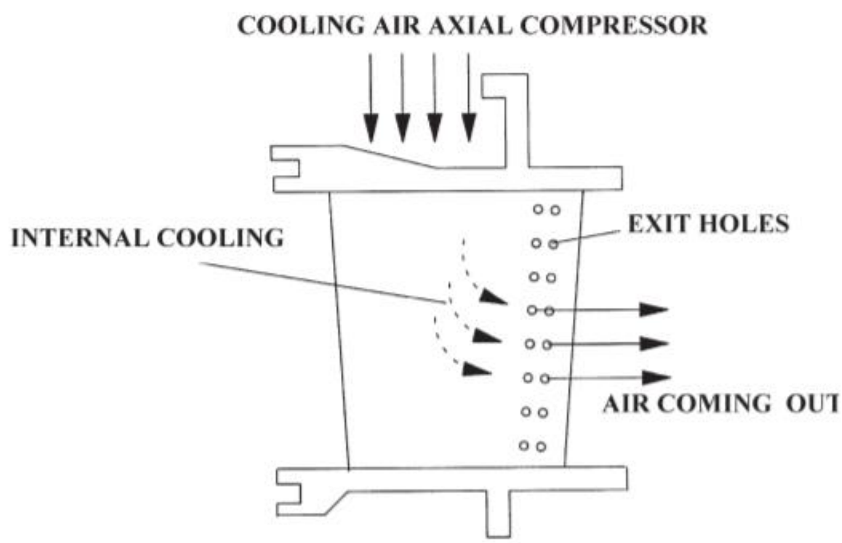


Fig.37

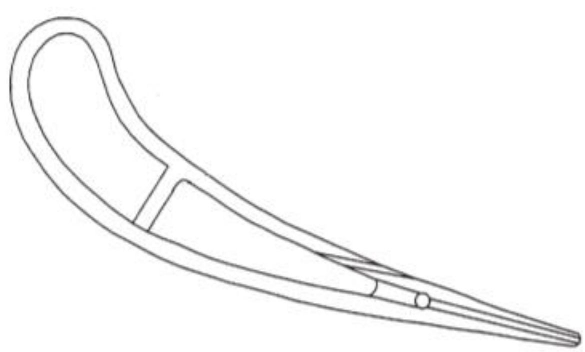


Fig.38
Fig.37-38 Cooling diagram from the inside of a nozzle

II.4.3 Turbine buckets:

The aim of the turbine buckets is to convert the kinetic energy of gases proceeding from the nozzles into mechanical energy, by varying the quantity of motion obtained by changing the direction of gases when they pass through vanes between contiguous buckets (fig.34). For this reason, blades are subjected to forces that produce the rotor torque.

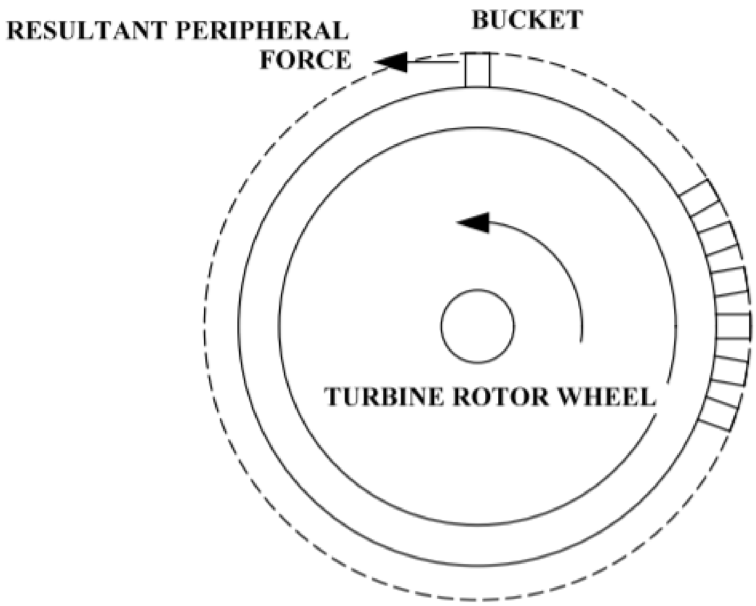


Fig.39 Mechanical operating diagram of the buckets-turbine wheel

Turbine buckets are made of materials with a high nickel content, listed in the following table:

Table 2: Materials typical of heavy-duty gas turbine buckets:

Nominal percentage composition

Alloy	C	Cr	Ni	Co	W	Fe	Mo	Ti	Al	Cb	Ta	B
U500*	0.07	18.5	Diff.	18.5	-	-	4	3	3	-	-	0.01
IN738	0.10	16	Diff.	8.3	2.6	0.2	1.75	3.4	3.4	0.9	1.75	0.01
GTD111	0.10	14	Diff.	9.5	3.8	-	1.5	4.9	3	-	2.8	0.01

* U500 is used for stages following the first one (that is, at lower temperatures).

As for jet derivative gas turbines, the materials employed are always nickel based, DSR 142 type, for first stages, and René 80 or 77 for successive stages. The main characteristics that must be conferred to turbine buckets are the following:
- Resistance to break, ductility
- Resistance to creep (combined thermo-mechanical stress)
- Resistance to corrosion and oxidation
- Resistance to thermal and mechanical fatigue

*Thermal fatigue consists in thermal transients that originate during start-up, shutdown, load rejection, etc., in which buckets are subjected to alternate heating and cooling cycles.
*Resistance to break, creep and ductility depend on intergranular bonds in the internal structure of the bucket material. In this respect, 1st stage buckets with directional solidification have long been in use. The latter property is obtained via a casting process that produces oriented solidification, in which the grain structure orients itself parallel to the bucket main centerline, which coincides with the centerline on which the centrifugal force acts during rotation. In this way, intergranular structures placed at 90° from the main centerline are eliminated, which limit resistance to centrifugal traction. This results in increased resistance to traction and greater resistance to fatigue, due to increased material elasticity.
Another measure that permits to increase firing temperatures to ever higher values without increasing thermal stress is derived from aeronautical applications and consists in cooling bucket walls by channeling air from the compressor into the bucket through distribution holes located at the base.

Fig.40 shows a diagram of the channeling system around the bucket.
Fig.41 shows a sectional view of a 1st stage blade cooled via its inner cooling channels, made by casting.
Fig.42 shows a detail of air inlet holes on the base.

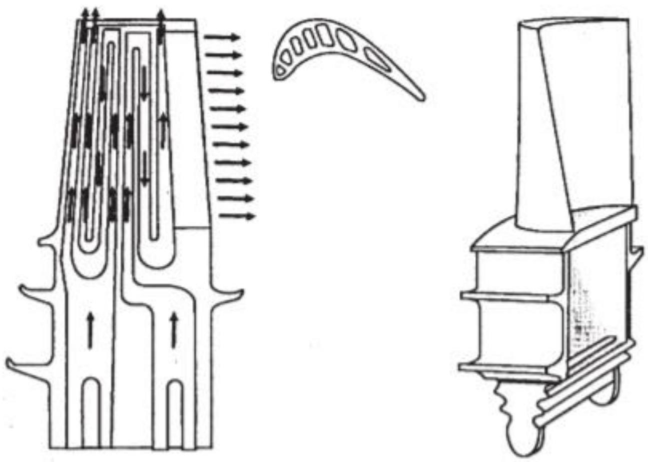
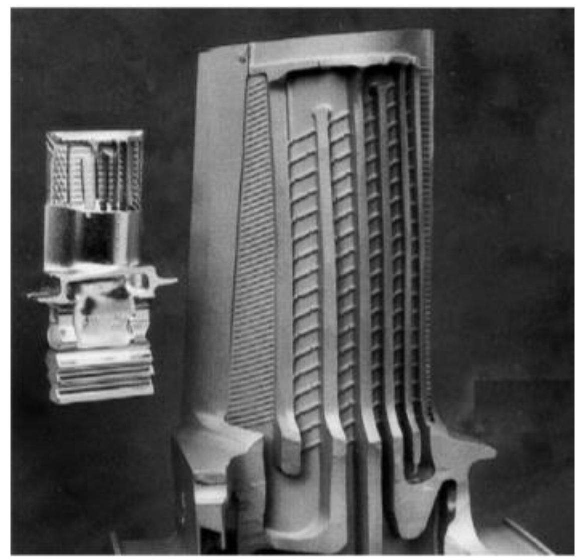


Fig.40

Fig.41



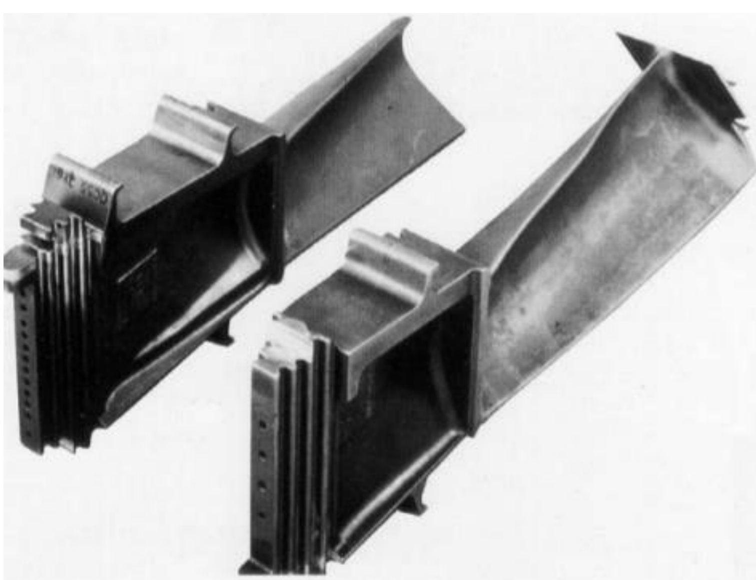


Fig.42

Fig.40, 41, 42 Bucket cooling systems

Resistance to corrosion and to oxidization is a property that is conferred to buckets both by adopting materials of appropriate composition, and by applying a protective coating to the outer surface of the bucket.
Hot corrosion is associated with the presence of alcaline metals (sodium and potassium) in fuel. When these metals exceed a certain concentration limit, they form salts that attack the bucket base metal.
Oxidization is caused by the combination of oxygen atoms with metal atoms, which originates scales and incrustations. As is well known, the higher the temperature, the quicker is the oxidizing process. It became all the more indispensable to apply a protective coating, especially to the first stages, because the need to increase resistance to break and to creep made it necessary to employ ever more complex super alloys, which have not proved resistant enough against corrosion and oxidation.
Considering higher firing temperatures to which gas turbines have been subjected in the course of their technological developments, coatings have now become a standard procedure for all gas turbine models currently produced.
The function of all types of coating is to create a superficial layer composed of chemical elements (generally, pt and Al) which, under the action of thermal agents, build a protective oxide layer (such as Al2 O3) to defend the underlying base material against corrosion and oxidization.
Fig.43 shows, as a function of the sodium percentage contained in fuel, the life curve of buckets in uncoated U700, in uncoated IN738 and in IN738, coated with protective Pt-A1.

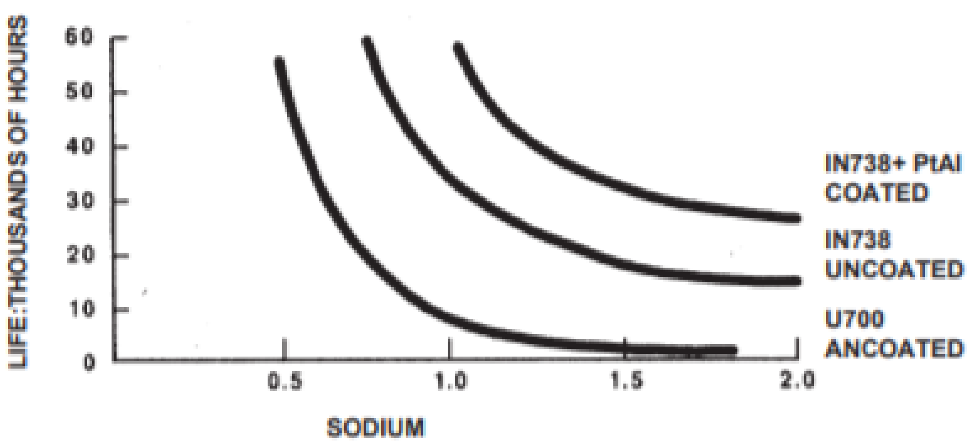


Fig.43 Effect of sodium (ppm) on bucket life

II.4.4 Turbine wheels:

Turbine wheel is one of the most critical components from the standpoint of temperatures and mechanical stress when in operation. Turbine wheels are designed under the principle of high-speed revolving discs. This principle takes into account the following factors:

– stresses (due substantially to centrifugal force)
– temperature reached by the wheel metal
– design life, which is estimated at 100,000 hours.

Like buckets and nozzles, also turbine wheels are designed with the help of diagrams known as Larson-Miller diagram, which represents the combined effect of the two factors (temperature and design life) in a logarithmic formula. The same diagram shows also curves for admissible stresses as a function of the Larson-Miller parameter (at 100,000 hours' design life), for 4 different types of materials.

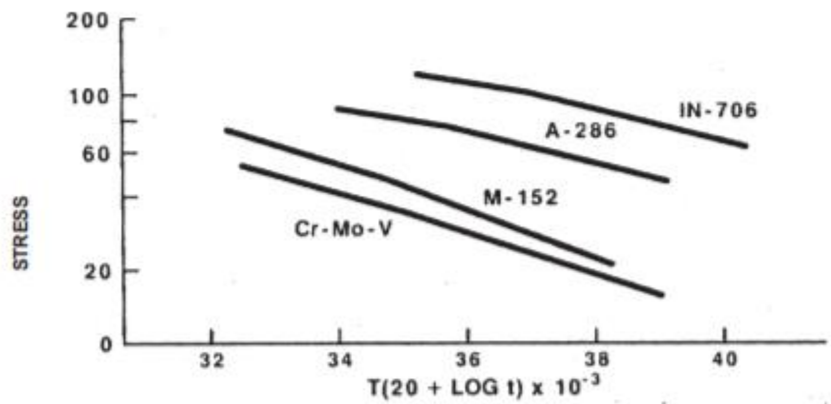


Fig.44 Larson-Miller diagram for turbine wheels

Turbine wheels are made from forging. Typical materials are listed in Table 3:

Table 3: Turbine wheel typical materials

Nominal percentage composition

Alloy	C	Cr	Ni	Fe	Mo	Ti	Al	Cb	V	B
Cr-Mo-V	0.30	1	0.50	Diff.	1.25	-	-	-	0.25	-
M152	0.12	12	2.50	Diff.	1.70	-	-	-	0.30	-
A286	0.08	15	25	Diff.	1.20	2	0.3	-	0.25	0.006
IN706	0.06	16	Diff.	37	-	1.8	-	2.9	-	0.006

Nickel base materials (like, for example IN706 and others similar) were introduced when it became necessary to withstand ever increasing operating temperatures. The fundamental characteristics taken into consideration when choosing materials and the production process of turbine wheels are the following:

- High mechanical and creep characteristics
- Material stability
- Ductility
- Good forgeability

Fig.45 shows a sectional view of the wheel; yielding areas, marked in gray, are compressed by external areas which have maintained themselves within the elastic field. The diagram on the left shows the path of stresses along the wheel radius during the hot spinning test; the abscissa represents the area adjacent to the central hole, subjected to a stress higher than the yielding limit.

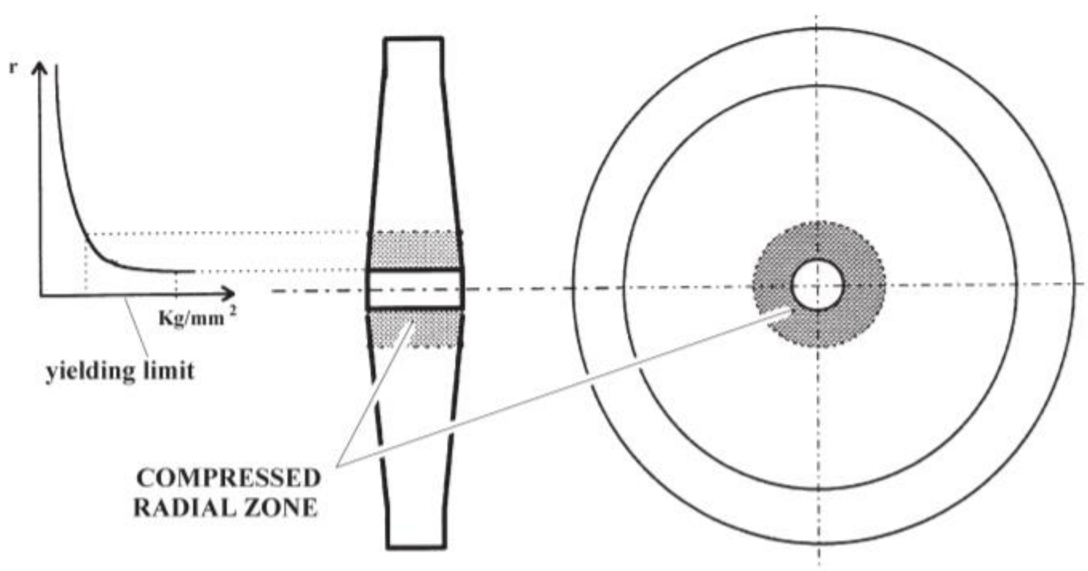


Fig.45 Turbine wheel - Differentiated radial areas after the hot spinning test

Wheels built in M152 (for ex. PGT5 gas turbine) and Cr-Mo-V (for ex., MS9001 gas turbine), unlike the other two materials in the table, need higher temperatures to pass from a ductile to a fragile structure, are tested at high speed and at a temperature lower than transition temperature (around -30°C), to make sure there are no internal defects of critical proportions, which would become dangerous during operation. This test procedure is commonly known as "cold stretching".

One of the main sources of heat is the heat transmitted by buckets, which are constantly assaulted by hot gases.

To limit heat transmission from the bucket to the wheel, acting on the factors that determine heat conductivity from the bucket foot to the wheel itself is necessary, by taking two preventive measures:

– Reduce the flow section from the bucket platform to its base;

– Give the bucket base a dove-tail shape, in order to reduce the contact surface obtained during operation by effect of centrifugal action on the buckets.

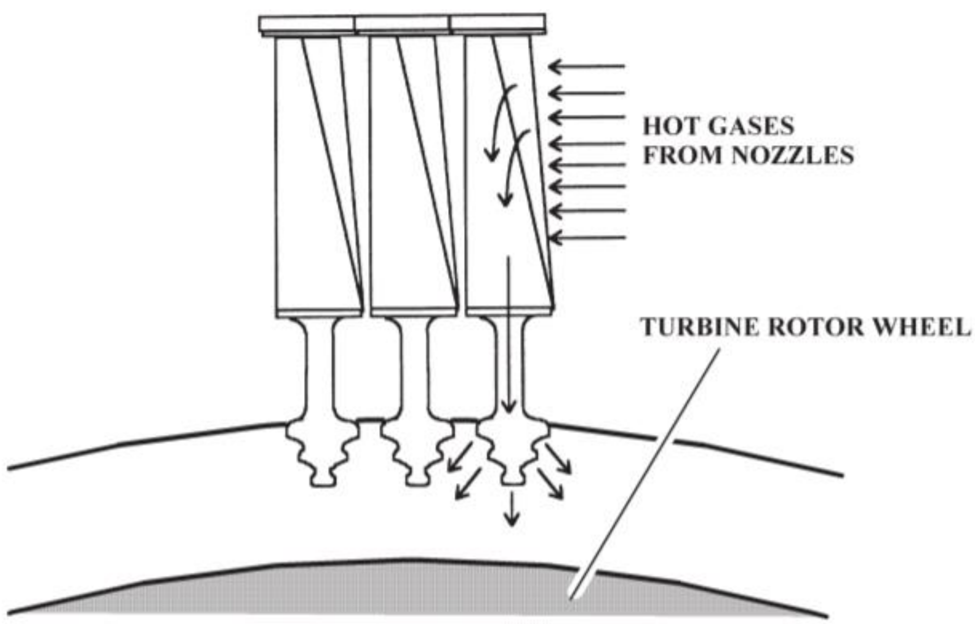


Fig.46 Heat transmission from buckets to turbine rotor wheel

Fig.47 Detail of "dove-tailed" profiles between buckets and wheel

II.4.5 Turbine casings:

Heavy duty gas turbine stators are made in high resistance cast-iron of spheroidal type for working temperatures up to 350-400 °C. Some parts of casing are made of weld steel for constructive reasons. Like compressor stator casings, also gas turbine casings contain housings for the various internal parts, such as for example, stator rings, which are near the external diameter of rotor buckets. These rings are made in separate segments whose clearance is regulated when segments are cold, to prevent stress during start-up transients.

As concerns heavy duty turbine casings in the PGT range, there apply the same considerations expressed above. In the case of the LM6000 gas turbine, casings in the turbine section are made of super alloy INCO 718, with antioxidant coating (Waspaloy).

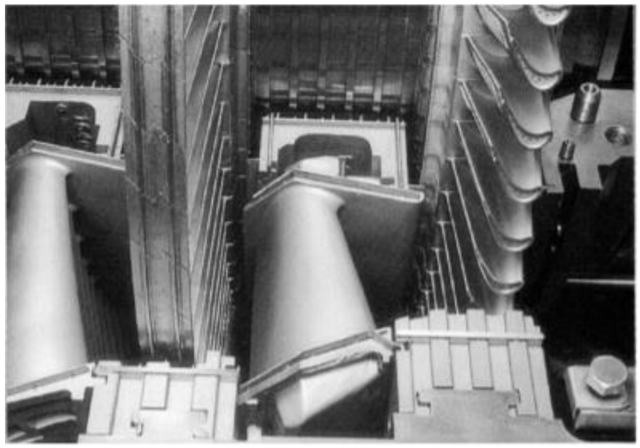


Fig.48 A detail view of turbine stator rings

II.5 Gas turbine main accessory systems:

II.5.1 Suction system:

The suction system of a gas turbine has the function to convey combustion air to the axial compressor inlet section, so as to guarantee:

• the required degree of filtration for correct operation of the compressor and the turbine under the existing environmental conditions of installation.

• compliance with noise limits in proximity of the system and at a distance from it as agreed upon in the contract.

• compliance with the contractually agreed upon power and fuel consumption values.

• regularity of air flow to the compressor inlet section, and consequent regular fluid dynamic operation of the latter.

• levels of reliability and availability not lower than in the rest of the plant.

The gas turbine suction system consists of the following main component parts: Suction filter, Duct, Silencer, Elbow, Inlet plenum, Accessories (such as anti-icing system).

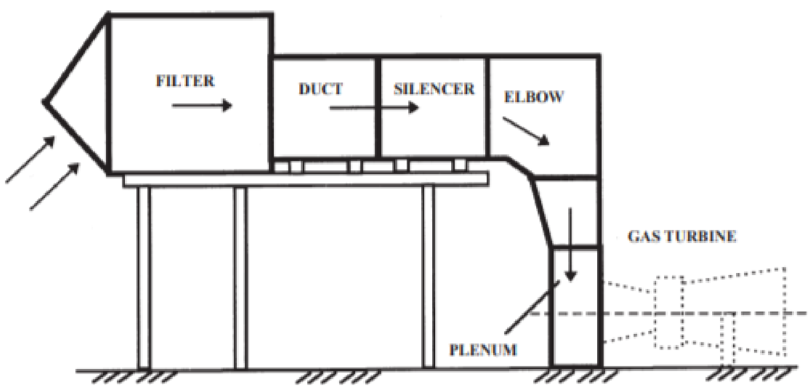


Fig.49 Typical suction system

II.5.2 Exhaust system:

The exhaust system has the function of discharging into the atmosphere the exhaust gases originating from the expansion process that takes place in the gas turbine. This system must be designed and built in a way to meet the following requirements:

• compliance with contractual noise levels in proximity to the system and at a distance from it;

• compliance with contractual power and fuel consumption levels;

• compliance with contractual personnel safety requirements (heat insulation);

• gas flow even, optimal speed distribution from the exhaust flange to vent into the atmosphere, so as to reduce turbulence, leakage and abnormal stress on structures to a minimum;

• optimum "isostaticity" levels on every part of the system, to prevent conditions of self-stress from developing during thermal transients;

• reliability and availability levels not lower than those of the rest of the plant;

• the inspection ability level required for critical areas of the system.

The exhaust system of a gas turbine consists of the following main elements:
• Diffuser or exhaust plenum (according to gas turbine model);
• Connecting ducts;
• Exhaust silencer;
• Stack;
• Variators;
• Thermal and acoustic insulation;
• Service stairways and walkways;
• Supporting frames.

II.5.3 Lube and control oil system:

Lube oil system on heavy duty gas turbines:

This system has the task of supplying oil to the gas turbine bearings, to the driven machine, to integrated auxiliaries (auxiliary gear, starting motor with torque converter, joints, etc.), to the control oil and to the hydraulic oil systems.
The oil supplied must meet all pureness (filtration), pressure and temperature required by the different systems it is fed to. The main component parts of a lube system are:
Oil tank It can be either housed in the turbine baseplate or separate. Tank capacity is calculated on the criterion that it must be capable of being emptied within a definite minimum time, with the oil pump running and a hypothetical loss in the circuit that prevents oil from flowing back into the tank. In gas turbine designing norms (API), this time is also defined as "retention time".
Lube oil pumps
Oil is taken from the tank and sent into circulation by a main pump. The pump can be driven by the auxiliary gear, the load gear (ex. single-shaft PGT5 and PGT10), or by an electric motor in the case of an oil system separate from the turbine baseplate. Downstream of the main pump a stand-by pump is generally installed, driven by an AC motor, which serves during start-up phases, when the main pump, driven by the auxiliary gear, has not reached sufficient rpms to deliver the quantity of oil required by the system, or in case of failure of the main pump.
In both cases, however, the stand-by pump acts as a reserve in the case of main pump failure during gas turbine operation, and makes it possible to manage a "controlled" stop of the turbine, in order to replace the damaged main pump. The stand-by pump serves also to supply oil to the gas turbine bearings (after a normal or emergency stop) in order to disperse heat due to irradiation inside the gas turbine.
Parallel to these two pumps, there is an emergency pump, whose aim is to deliver the oil required to stop the gas turbine in case of emergency due to an AC power failure.

Lube oil system on gas turbines with aeroderivative gas generator

Turbines with aeroderivative gas generator (PGT range) have two separate lube oil systems:
• Mineral oil system similar to the one for lubrication of heavy-duty power turbine and, in case, load gear and driven machine (alternator, compressor, pump, etc...)

• Synthetic oil system, at all similar to the one used on aeronautical turbines in order to lubricate gas generator bearings
• The oil tank has a greater capacity than its aeronautical version, because in industrial applications there is less need for smaller dimensions.
• The use of synthetic oil is derived from the aeronautical criterion of ensuring maximum safety against fire.
• The system includes a number of main and suction ("scavenge") pumps driven mechanically by an auxiliary gear. The main pumps deliver oil to the bearing sprayers, while the scavenging pumps extract oil from the leakproof "box" of each bearing and send it to the tank.
• The system is completed by a cooler and a system of 10µ duplex filters.

II.5.4 Fuel system:

One of the peculiar characteristics which have distinguished gas turbines in the course of their history is their flexibility in the use of different types of fuel, as especially heavy-duty gas turbines can burn a large variety of fuels. Furthermore, combinations of these fuels can be used on the same turbine, with dual fuel applications, thus increasing its flexibility and availability of use. The most widespread fuels are the followings:

1. Natural gas (mixtures of methane, ethane and hydrocarbons $C_n H_{2n+2}$ with progressively growing molecular weight, with the addition of other components such as CO, H_2, etc.).
2. Mixtures of propane and butane, also called LPG.
3. Refinery gases with a high hydrogen content.
4. Gases deriving from coal gasification and other gases with a low calorific value.
5. Conventional liquid fuels, such as fuel oil, crude oil and residual oils.
6. Less conventional liquid fuels (Kerosene).
7. Unconventional liquid fuels such as naphta and gasolines.
8. Liquid fuels such as process residuals.
9. Methanol.

Natural gas and fuel oil are, among else, the only fuels (along with Kerosene) that may also be used on aeroderivative gas turbines (PGT 16, PGT 25, LM2500, LM 5000, LM 6000), whereas on heavy duty gas turbines it is possible to use even less refined fuels provided they are treated.

II.5.5 Starting system:

The main purpose of the starting system is to accelerate the gas turbine up to self-sustaining speed, gas turbine reaches a self-sustaining condition when the energy available in the combustion chamber is at least equal to the sum of energy required by compression and mechanical losses in the gas turbine and the driven machine. Based on this, as long as the compressor revolving speed (that is, the amount of air aspirated) and the relative amount of fuel delivered do not produce the above mentioned self-sustaining energy, there is the need for a starting motor connected with the compressor shaft, which compensates for deficient energy by accelerating the gas turbine up to self-sustaining speed. The starting motor may consist of an engine, a hydraulic or diesel motor, a natural gas expanding turbine or steam turbine, according to plant characteristics. As for gas turbines that generate electric power, the starting means often consists of the alternator itself, which acts temporarily as an electric motor, whose torque characteristics and rpms are controlled by inverter systems (static starters).

II.5.6 Auxiliary gear:

The auxiliary gear has the function to:

• connect the starting means with the turbine shaft for startup.

• drive a number of user elements (pumps, etc.) mounted on shafts connected with one another via gears, which receive input energy from the axial compressor shaft of the gas turbine in running condition, via an auxiliary coupling.

II.5.7 Load gear:

The load gear has the function to adapt the gas turbine speed value to that of the driven machine in all cases in which it is impossible for them to be equal. This need derives from the fact that, in many cases, the useful power produced by the driving machine (gas turbine) requires rpms higher or lower than those that fall in the ideal operating speed range of the driven machine.

II.5.8 Couplings:

A gas turbine, as seen in previous paragraphs, drives many types of machines via an auxiliary gear (pumps, barring gear, etc.), and a load gear (alternator, compressor, pumps, etc.). Connection between the gas turbine and these apparatuses is made by using couplings, which can be of two types:

• toothed couplings, continually lubricated

• couplings with dry, elastic diaphragms.

II.5.9 Electric system:

The electric system on a gas turbine, which is described in this chapter, refers to the equipment related to the machine and its auxiliaries, including the relative junction boxes. It serves following purposes:

1. connect wires which carry signals coming from the instrumentation on the machine body and from the apparatuses inside the enclosure (lighting, firefighting system, dangerous atmosphere, etc.) to the junction boxes located at the baseplate border;
2. Connect power cables for the various user elements (electric motors, etc.) to their power supply sources.

Connections can be made by using cables passing through rigid conduits or external cables on cable trays, according to plant requirements and the standards to be complied with.

II.5.10 Oil cooling system:

The gas turbine oil heats in its passage through the bearings and on account of irradiation from hot casings and from pipings. Therefore, it is necessary to cool down oil in a continuative way,

in order to limit its temperature and obtain correct lubrication and adequate cooling of bearings and shafts. The oil cooling process can take place in the following ways:
•with water/oil coolers in an open circuit
•with water/air coolers with water in a closed circuit
•with oil/air coolers

II.5.11 Control, regulation and protection system:

The “control, regulation and protection” system runs a variety of functions aimed at ensuring correct operation of a gas turbine, in compliance with its operative needs. To carry out these functions, the system "manages" a series of parameters in the form of input/output signals received from the turbine and sent to the same via "interface" elements (junction boxes and relative terminal boxes) , to which there converge control signals sent by the machine instrumentation (pressure transducers, position transducers of fuel valves, thermocouples, flame detectors, etc.), and signals arriving from the control and protection system and from the plant. Parameter management is the task of the local control panel SPEEDTRONICTM.
From an operative standpoint, the control panel is divided into the following subsystems:
- Control and sequential system
- Protection system

Within these systems, the control panel plays the following primary functions:
• it controls fuel flow and operation of auxiliaries during the starting, stopping, emergency stop and cool-down sequences
• it synchronizes and turns on the voltage supply to the network alternator
• it controls fuel gas and exhaust gases during operation
• it commands all protective measures in case of faults and failures.
• it records all functions of the gas turbine and of its auxiliaries with historic record functions

II.6 Conclusion:

In this chapter, we reviewed both HD and AD gas turbines design, constructive characteristics and materials used and we were able to conclude that:
-HD gas turbines are generally made of high resistance cast-iron, that explains their heavy weight and their cheap spare-parts price and the price of maintenance in general.
-AD gas turbines are generally made of alloys and super alloys that are light-weight and resistant to high temperatures, which explains the difference in weight comparing to HD gas turbines. Alloys and super alloys are expensive which means higher price of maintenance and spare-parts.

Introduction:

In this chapter, comparison between the HDGT and the ADGT is going to take place citing main differences and characteristics of both turbines.

III.1 Construction wise:

In this section, construction comparison of the three main component of each gas turbine.

III.1.1 Compressor section: Table 4: Compressor section comparison

	HDGT	ADGT
Stator casings	Built of high resistance cast iron	1- Frontal casings: built in light alloy (generally, aluminum-based) 2-HPS casings: made of austenitic steel or nickel-based super alloys
Stator blades	Made of the X15Cr13 type or equivalent	1-LPS: titanium-based 2-FS: made of austenitic steel (stainless steel) or super alloys
IGV	Made of GTD 450 (or C 450), a type of stainless steel particularly resistant to corrosion	Super alloys, stainless steel and titanium-based materials
Rotor blades	Same material as stator blades	Same material as stator blades

*HPS: High Pressure Stages

*LPS: Low Pressure Stages

*FS: Final Stages

III.1.2 Combustion section: Table 5 – Combustion section comparison

	HDGT	ADGT
Combustion cap & liner	Made of nickel-based super alloy (Hastelloy X, INCO 625 or equivalent)	Made of nickel-based super alloy (Hastelloy X, INCO 625 or equivalent)
Transition piece	1-Made of the same material as the combustion liner 2-On GT with 1 CC: TP is built of 2 parts 3-On GT with more than 1 CC: TP is built of as many parts as are the combustion chambers joined to one another to form a circumference	No TP in ADGT
Burner	12 burners	32 burners

*GT: gas turbine

*CC: combustion chamber

*TP: transition piece

Notes:
-In the case of "heavy duty" gas turbines, the combustion section consists of a system of one or more tubular combustion chambers.
- On gas turbines with a single combustion chamber there is only one spark plug, and no cross-fire tubes.
-On heavy duty machines with multiple combustion chambers, the flame is initially ignited by spark plugs, then it propagates to the other combustion chambers via “cross fire tubes”.
-On ADGT no transition piece required.
-On ADGT air flow is annular, while on HDGT air flow is transformed from cylindrical into annular by the transition piece which causes pressure drops and therefor efficiency drop.

III.1.3 Turbine section: Table 6: Turbine section comparison

	HDGT	ADGT
Turbine buckets	Made of materials with a high nickel content	-Materials employed are always nickel based - DSR 142 type for first stages - René 80 or 77 for the successive stages
Turbine casings	high resistance cast iron	In the case of the LM6000 gas turbine, casings in the turbine section are made of super alloy INCO 718, with antioxidant coating (Waspaloy).

Notes:
-In the case of "heavy duty" turbines, the turbine section comprises a certain number of stages.
-A characteristic peculiar to heavy-duty gas turbines is that they have high energy stages or, more correctly, a low degree of reaction. As a consequence, the conversion of energy from the combustion chamber to the exhaust section can be carried out with a reduced number of turbine stages.
-Heavy-duty gas turbines (made by Nuovo Pignone) have two or maximum three stages.
-On jet derivative gas turbines, the high compression ratio calls for a larger number of expansion stages.

III.2. Maintenance wise: [6]**Table 7: Shuttle maintenance for capital parts**

Components	HDGT	Down-time for maintenance	ADGT	Down-time for maintenance
Axial compressor	48000 Fired hours	45 days	In request / 50000 fired hours	30 days
Burners	12000 hours	10 days	8000 hours	2 days
Liners and fuel nozzle /CC	12000 fired hours	10 days	25000 fired hours	10 days
Transition pieces	24000 hours	21 days	No TP in ADGT	N/A
1st stage nozzle	24000 hours	21 days	25000 hours	10 days
2nd stage nozzle	24000 hours	21 days	25000 hours	10 days
HSPT (high pressure)	24000 hours	45 days	25000 hours	10 days
PT (Low pressure)	48000 hours (200000 hours live limit)	45 days	100000 hours (live limit)	30 days
Mobility	Hard		Easy / can be trailed	
On-site maintenance	Easy		Must be moved to workshop/requires special tools	
Installation time	1 month (on double shift teams)		1 week (on double shift teams)	

*HSPT: Hot Section Power Turbine
*PT: Power turbine

III.2.1 Maintenance policy for both HDGT and ADGT:

ADGT: [8] [9]

Based on 3 levels of maintenance: (transportation time and intervention time are not considered in level 2 and level 3 which may lead to longer down-time to transport from the location of ADGT to the workshop)

Level 1: inspection borescope 8000hrs fired hours or 1 year since last maintenance.

Level 2: divided by 2 sections

•cold section: in request "based on borescope inspection" consist on maintaining axial compressor or else in 50000 fired hours;

•hot section: mandatory on 25000 fired hrs. consist on maintaining HSPT.

Level 3: major inspection (or MI), done in 2 workshops over the world: in Huston, Texas, USA and in Massa, Italy. Major inspection done each 50000hr

HDGT:

CI (combustor inspection): each 12000 hours; [3]

HGPI (hot gas past inspection): 24000 hours; [4]

MI (major inspection): 48000 hours. [5]

Notes:

-On ADGT maintenance policy consist of levels (3 levels), while on HDGT it consists of inspections (3 inspections).

-HDGT can be maintained on location, while ADGT must be moved to work shops as maintaining it requires special tools.

-in some cases, on HDGT, instead of HGPI (hot gas past inspection) LTPI (liner transition piece inspection) is conducted to gain time. (done on customer's demand)

III.3. Performance wise:

in this section, a comparison between the aero-derivative gas turbine PGT25 DLE and the heavy-duty gas turbine MS5002C at ISO conditions with natural gas fuel, ambient temperature 15°C,

III.3.1. ENERGY AND THERMODYNAMICS STUDY OF THE PGT25: [10]

WORKING CONDITIONS

The PGT25 gas turbine operates under the ambient conditions of pressure and temperature. For our study, we took the parameters mentioned in Table below with an ambient temperature of 15°C. The fuel used is natural gas CH_4. Air is considered an ideal gas.

Parameter	Value	Unit
Compressor inlet temperature (T1)	288	K
Compressor inlet pressure (P1)	1,013	bar
Total compression ratio (β)	13.03	---
Mass air flow ($\dot{m}_a$)	45	Kg / s
Air speed in the compressor (υ_c)	135	m / s
Compressor outlet air speed (υ_2)	40	m / s
Density of ISG gas (ρ_g)	0.68	kg/m^3
Lower calorific value of ISG gas (Pci)	44	Mj / kg
Combustion chamber efficiency (η_{cc})	98 %	---
HP turbine manometric coefficient (μ)	0,5~4,0	---
HP shaft rotation speed (N)	8676	Rd / min
Isentropic efficiency (η_{is})	85 %	---
Mechanical efficiency (η_{me})	95 %	---

Table 8: PGT25 parameters under ambient conditions

The entire energy study of the PGT25 DLE will be based on the indices in **Fig.50** as index 1 designates the compressor inlet or air intake, 2 for the outlet of the compressor / combustion chamber inlet, 3 for the combustion chamber outlet combustion / HP turbine inlet, 4 for the HP turbine outlet / LP turbine inlet and the index 5 designates the outlet of the LP turbine or the exhaust.

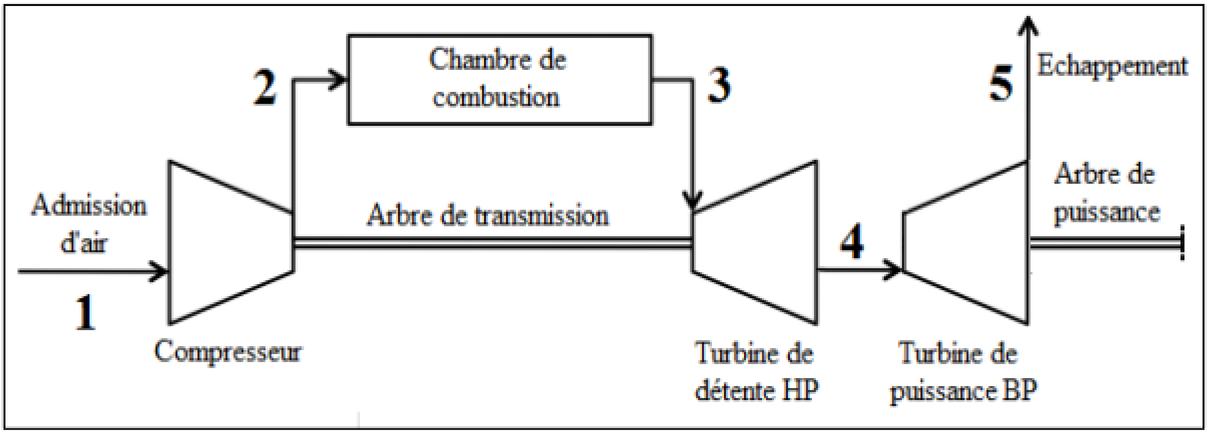


Fig.50: Block diagram of the PGT25 DLE with indices for calculations

•CALCULATION OF THE COMPRESSOR POWER

Knowing the power absorbed by the compressor helps us determine the parameters of the HP turbine or expansion because the latter is intended only to drive compressor rotation.

•Compressor outlet temperature

The total compression ratio β is the ratio of the outlet and inlet pressure of the compressor, therefore:

$$\beta = \frac{P_1}{P_2} \Rightarrow P_2 = \beta P_1 = 13.03 \times 1.013 \Rightarrow P_2 = 13.2 \text{ [bar]}$$

If the compression was isentropic, then according to Poisson's law we will have:

$$\frac{Tis_2}{T_1} = \left({}^{P_2}/_{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow Tis_2 = T_1 \left({}^{P_2}/_{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

As compressed fluid is air which is made up of diatomic molecules ($O2et\ N2$), we take γ = 1.41:

$$Tis_2 = 288 \left({}^{13.2}/_{1.013}\right)^{\frac{1.41-1}{1.41}}$$

$$Tis_2 = 607.5\ [K]$$

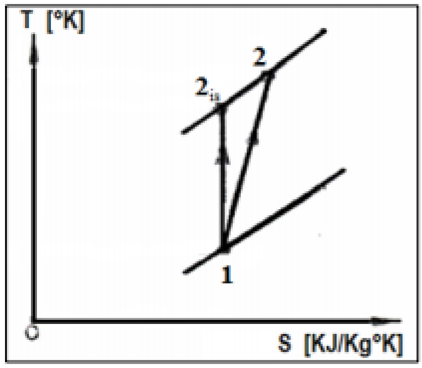


Fig.51 TS diagram of isentropic and real compression

By introducing the notion of isentropic efficiency ηis, the temperature $T2$:

$$\eta \text{is} = \frac{\text{Tis}_2 - \text{T}_1}{\text{T}_2 - \text{T}_1} \Longrightarrow \text{T}_2 = \text{T}_1 + \frac{\text{Tis}_2 - \text{T}_1}{\eta \text{is}} = 288 + \frac{607.5 - 288}{0.85}$$

$$T_2 = 663.88\ [K]$$

•Increased enthalpy

Actual increase in enthalpy throughout the compressor is given by:

$$\Delta hc = Cp\ (T2 - T1) \qquad \textbf{(II.1)}$$

We take Cp = 1,039 $[Kj.Kg^{-1}.K^{-1}]$and we consider it constant. It is obtained by linear interpolation from Table A-1 in Annex A for an average temperature between the compressor inlet and outlet.

So the enthalpy increase will be:

$$\Delta hc = 1{,}039(663.88 - 293)$$

$$\Delta hc = 385{,}34\ [KJ.Kg{-}1\ ]$$

•Useful work

The useful work received by the mass of one kilogram of air is obtained by the equation of energy conservation:

$$Wuc + Qc = \Delta(h + Ec + Ep)\ c \qquad \textbf{(II.2)}$$

$$Wuc = \Delta(h + Ec + Ep)\ c - Q\text{c}$$

The compression is carried out in an axial horizontal manner (no change in level therefore ΔEpc = 0) and isentropic (i.e. reversible adiabatic, no heat exchange Qc = 0), we will have:

$$Ec_0 = \frac{1}{2}v_0^2\ \ et\ Ec_2 = \frac{1}{2}v_2^2 \qquad \textbf{(II.3)}$$

So :

$$\text{Wuc} = \Delta\text{hc} + \frac{1}{2}(\text{v}_2{}^2 - \text{v}_0{}^2) = 385344 + \frac{1}{2}(40^2 - 0^2)$$

$$\text{Wuc} = 386{,}144[\text{KJ/Kg}\ ]$$

•Absorbed power

The useful power supplied by the compressor is:

$$Puc = \dot{m}a.Wuc = 45 \times 386{,}144 \quad \textbf{(II.4)}$$

$$Puc = 17376{,}48\ [KJ/\ \ ]\ ou\ [KW]$$

If we admit the mechanical efficiency ηm, the power actual compressor absorption will be:

$$\eta mé = \frac{Puc}{Pc} \implies Pc = \frac{Puc}{\eta mé} = \frac{17376{,}48}{0{,}95}$$

$$Pc = 18290{,}52\ [KW] \quad \textbf{(II.5)}$$

•DETERMINATION OF THE CC OUTPUT PARAMETERS

The energy study of the combustion chamber is based on the assumptions following:

- Combustion takes place at constant pressure;
- The transformation is adiabatic;
- The flow is permanent;
- In reality, combustion is accompanied by pressure drops, therefore an efficiency ηcc was assigned to it.

•Mass determination and molar mass of the air / gas mixture

The natural gas in the field contains a high percentage of methane in the order of 97%, the remainder being hydrocarbons and internal gases, we have from the station chromatograph compression for one mole of fuel:

Percentage molar x_i	constitute	Chemical formula	Molar mass [$\boldsymbol{g}$ / $\boldsymbol{mol}$]	Mass $\boldsymbol{Mi}$ [$\boldsymbol{g}$]
0,5947	Nitrogen	N_2	28	0,166516
0,0101	Carbon dioxide	CO_2	46	0,004646
97,0526	Methane	CH_4	16	15,528416
1,7403	Ethane	C_2H_6	30	0,52290
0,4018	Propane	C_3H_8	44	0,176792
0,1187	Butane	C_4H_{10}	58	0,068846
0,0392	Pentane	C_5H_{12}	72	0,028224
0,0204	Hexane	C_6H_{14}	86	0,017544
0,0130	Heptane	C_7H_{16}	100	0,01300
0,0074	Octane	C_8H_{18}	114	0,008436
0,0009	Nonane	C_9H_{20}	128	0,001152
0,0009	Decane	$C_{10}H_{22}$	144	0,001295

Table 9: Composition of the TEG fuel gas

The molar mass of the fuel is the sum of the respective percentages of mass molar of each component:

$$Mg = \sum_{i=1}^{12} Mi$$

$$Mg = 16{,}5368\ [Kgg.Kmolg\ -1\ ]$$

To have the number of moles of stoichiometric oxygen, you must know all the equations of stoichiometric reactions of fuel gas constituents.

Reagents		Products
$CH_4 + 2(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$CO_2 + 2\ H_2O + 7{,}52\ N_2$
$C_2H_6 + 3{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$1\ CO_2 + 3\ H_2O + 13{,}16\ N_2$
$C_3H_8 + 5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$2\ CO_2 + 4\ H_2O + 18{,}80\ N_2$
$C_4H_{10} + 6{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$3\ CO_2 + 5\ H_2O + 24{,}44\ N_2$
$C_5H_{12} + 8(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$4\ CO_2 + 6\ H_2O + 30{,}08\ N_2$
$C_6H_{14} + 9{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$5\ CO_2 + 7\ H_2O + 35{,}72\ N_2$
$C_7H_{16} + 11(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$6\ CO_2 + 8\ H_2O + 41{,}36\ N_2$
$C_8H_{18} + 12{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$7\ CO_2 + 9\ H_2O + 47{,}00\ N_2$
$C_9H_{20} + 14(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$8\ CO_2 + 10\ H_2O + 52{,}64\ N_2$
$C_{10}H_{22} + 15{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$9\ CO_2 + 11\ H_2O + 58{,}28\ N_2$

Table 10: Stoichiometric combustion equations of CmHn fuels

The number of stoichiometric moles of oxygen for one mole of fuel is sum of the products of elementary oxygen mole numbers with percentages respective presence of each constituent:

$$n_{o2} = \sum_{i=1}^{10} xi \times n_{o2}i \qquad \textbf{(II.6)}$$

$$n_{o2} = 2{,}03526\ [kmol_{o2}/kmolg]$$

Considering that air is made up of 21% oxygen and 79% nitrogen, the number of stoichiometric mole of air for one mole of fuel will be:

$$\mathrm{n_a} = \frac{n_{o2}}{\mathrm{x_{o2}}} = \frac{2{,}03526}{0{,}21}$$

$$\mathrm{n_a} = 9{,}6914[\mathrm{Kmol_a/Kmolg}]$$

The molar mass of air is generally given by:

$$Ma = 0{,}79MN2 + 0{,}21MO2 = 0{,}79 \times 28 + 0{,}21 \times 32$$

$$Ma = 28{,}85\ [Kg_a/Kmol_a]$$

The excess air of Nuovo Pignone machines for an ambient temperature of 15° C is $\lambda = 4$, we will thus have the mass dosage of the premix:

$$\varphi = \frac{\dot{m}g}{\dot{m}a} = \frac{Mg}{\lambda naMa} = \frac{16{,}5368}{4 \times 9{,}6914 \times 28{,}85} \quad \textbf{(II.7)}$$

$$\varphi = \frac{1}{67{,}63}$$

For one kilogram of air, the mass of the mixture:

$$m_m = ma + mg = 1 + \frac{1}{67{,}63}$$

$$m_m = 1{,}01478\ [kg]$$

The number of moles of the mixture is:

$$n_m = \frac{m_a}{M_a} + \frac{mg}{Mg} = \frac{1}{28{,}85} + \frac{1}{67{,}63 \times 16{,}5368}$$

$$n_m = 35{,}556\ [mol]$$

So the molar mass of the mixture will be:

$$M_m = \frac{m_m}{n_m} = \frac{1{,}01478}{0{,}035556}$$

$$M_m = 28{,}54\ [kg/kmol]$$

•**Amount of heat supplied by combustion**

$$T_3 = T_2 + \frac{\eta cc \times \varphi \times Qi}{Cp} \quad \textbf{(II.8)}$$

The heat released per unit mass of the gaseous fuel Qi is obtained by lower calorific value Pci such that: $Qi = Pci\ /\ \rho g$.

From the entrance of the combustion chamber to the exhaust, it is assumed that Cp is constant and equal to Cp = 1150 $[J/Kg.K]$, determined in the same way as that of compressor.

The temperature at the end of combustion would then be:

$$T_3 = T_2 + \frac{\eta cc \times \varphi \times Qi}{Cp\ \rho g} = 663.88 + \frac{0{,}98 \times \frac{1}{67{,}63} \times 44 \times 10^6}{1150 \times 0{,}68}$$

$$T_3 = 1479{,}20\ [K]$$

Taking into account the efficiency of the combustion chamber and after the expression, the heat that combustion provides is:

$$Qcc = \eta cc\dot{m}\ mCp\ (T3 - T2) = 0{,}98 \times 45{,}8739 \times 1150(1479{,}20 - 663{,}88) \quad \textbf{(II.9)}$$

$$Qcc = 42{,}151985\ [MW]$$

•**Molar constant of produced gases**

Natural gas contains a strong percentage of methane, so we will change the ratio $H\ /\ C$ taking into account only CH_4:

$$\frac{H}{C} = \frac{4\mathrm{m_H}}{m_C} = \frac{4 \times 1{,}008}{12{,}010}$$
$$\frac{H}{C} = 0{,}3357$$

We will thus have the value of the molar constant of the combustion products:

$$\mathrm{r_3} = \frac{1}{M_m}\left(9238{,}7 + \varphi\left(\frac{66543}{1 + \frac{\mathrm{H}}{\mathrm{C}}}\right)\right)\frac{1}{1+\varphi} \quad \textbf{(II.10)}$$
$$= \frac{1}{28{,}54}\left(9238{,}7 + \frac{1}{67{,}63}\left(\frac{66543}{1 + 0{,}3357}\right)\right)\frac{1}{1 + \frac{1}{67{,}63}}$$
$$\mathrm{r_3} = 345{,}98[J/kg.K]$$

•**Mach number CC output**

The Mach number at the exit of the combustion chamber $M3$:

$$\frac{\gamma_3^{\frac{1}{2}} Ma_3 \left(1 + \left(\frac{\gamma_3 - 1}{2}\right) Ma_3^2\right)^{\frac{1}{2}}}{1 + \gamma_3 Ma_3^2} = \frac{\gamma_2^{\frac{1}{2}} Ma_2 \left(1 + \left(\frac{\gamma_2 - 1}{2}\right) Ma_2^2\right)^{\frac{1}{2}}}{1 + \gamma_2 Ma_2^2}\left(\frac{r_3 T_3}{r_2 T_2}\right)^{\frac{1}{2}}(1+\varphi) \quad \textbf{(II.11)}$$

First, we need to calculate the Mach number $Ma2$ at the compressor outlet. We know that the air flow velocity in the compressor is vc = 135 [m / s], then using the expression of Mach (III-25) we find:

$$M_{a_2} = \frac{\nu_c}{\sqrt{\gamma_2 \mathrm{r_2} \mathrm{T_2}}} = \frac{135}{\sqrt{1{,}41 \times 287 \times 663{,}88}}$$
$$M_{a_2} = 0{,}26.$$

Now coming back to solving equation (III-40):

$$\frac{1{,}35^{\frac{1}{2}} M_{a_3}\left(1 + \left(\frac{1{,}35 - 1}{2}\right) M^2{}_{a_3}\right)^{\frac{1}{2}}}{1 + 1{,}35 M^2{}_{a_3}} = \frac{1{,}41^{\frac{1}{2}} \times 0{,}26 \left(1 + \left(\frac{1{,}41 - 1}{2}\right) 0{,}26^2\right)^{\frac{1}{2}}}{1 + (1{,}41 \times 0{,}26^{2\,2}}\left(\left(\frac{345{,}98 \times 1479{,}20}{287 \times 663.88}\right)\right)^{\frac{1}{2}}\left(1 + \frac{1}{67{,}63}\right)$$
$$\frac{\sqrt{1{,}35} M_{a_3}\sqrt{1 + 0{,}175 M^2{}_{a_3}}}{1 + 1{,}35 M^2{}_{a_3}} = 0{,}5239$$

We raise the two sides of the equation to the power of 2, we obtain:

$$\frac{1{,}35 M^2{}_{a_3}\left(1 + 1{,}175 M^2{}_{a_3}\right)}{1{,}8225 M_{a_3}^4 + 2{,}70 M_{a_3}^2} = 0{,}2735$$
$$M_{a_3}^4 - 4{,}5078 M_{a_3}^2 + 1.1458 = 0$$

To solve this equation, we bring it back to a 2nd degree equation in $M_{a_3}^2$, what gives us 4 solutions in total.The Mach number is always positive $M_{a_3} \geq 0$ and we have a subsonic flow M_{a_3} <1, so we admit as a positive solution: $M_{a_3} = 0.52$

•**Relative pressure loss**

The pressure loss in the combustion chamber ΔP will be calculated relative to the inlet pressure P_2, such as:

$$\zeta_{cc} = \frac{\Delta P}{P_2} = \frac{P_2 - P_3}{P_2} \quad \textbf{(II.12)}$$

We calculate in the first place the output pressure P_3:

$$\frac{P_{s_3}}{P_{s_2}} = \frac{1 + \gamma_2 M_{a_2}^2}{1 + \gamma_3 M_{a_3}^2} \quad \Longrightarrow \quad P_{s_3} = P_{s_2}\left(\frac{1 + \gamma_2 M_{a_2}^2}{1 + \gamma_3 M_{a_3}^2}\right)$$

The static inlet pressure P_{s2} is obtained by the relation of Saint Venant:

$$\frac{P_2}{P_{s_2}} = \left(1 + \frac{\gamma_2 - 1}{2} M_{a_2}^2\right)^{\frac{\gamma_2}{\gamma_2 - 1}} \quad \Longrightarrow \quad P_{s_2} = P_2\left(1 + \frac{\gamma_2 - 1}{2} M_{a_2}^2\right)^{\frac{\gamma_2}{1-\gamma_2}} \quad \textbf{(II.13)}$$

$$P_{s_2} = 13{,}2\left(1 + \frac{1{,}41 - 1}{2} 0{,}26^2\right)^{\frac{1{,}41}{1-1{,}41}}$$

$$P_{s_2} = 12{,}6\ [bar]$$

So:

$$P_{s_3} = 12{,}6\left(\frac{1 + 1{,}41 \times 0{,}26^2}{1 + 1{,}35 \times 0{,}52^2}\right)$$

$$P_{s_3} = 10{,}11\ [bar]$$

Using for a second time the relation of Barré de Saint Venant, the outlet pressure P_3 will be:

$$P_3 = Ps_3\left(1 + \frac{\gamma_3 - 1}{2} M_{a_3}^2\right)^{\frac{\gamma_3}{\gamma_3 - 1}} = 10{,}11\left(1 + \frac{1{,}35 - 1}{2} 0{,}52^2\right)^{\frac{1{,}35}{1{,}35-1}}$$

$$P_3 = 12{,}08\ [bar]$$

And so we can determine the relative pressure loss:

$$\zeta_{cc} = \frac{P_2 - P_3}{P_2} = \frac{13{,}2 - 12{,}08}{13{,}20} = 0{,}0848$$

$$\zeta_{cc} = 8{,}48\%$$

•**OVERALL POWER AND PERFORMANCE RATING**

To get there, we have to first go through the determination of the parameters thermodynamics of the high pressure expansion turbine and those of the power turbine at low pressure, **Fig.52** below illustrates the two expansion phases in transformations isentropic and real.

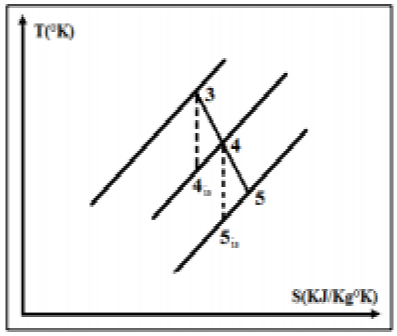


Fig.52 TS Diagram of Isentropic and Actual Detents

•**Temperature and pressure at the outlet of the HP turbine**

The work done by the high-pressure turbine is used exclusively to turn the axial compressor, so the powers of the latter are equal to the nearest sign, we write so:

$$P_{THP} = P_c$$

According to the expression of the power we will have:

$$P_{THP} = \dot{m}_m W_{THP} = P_c$$

If we introduce the notion of mechanical efficiency ηm for the turbine:

$$\dot{m}_{mé} = \frac{W_{THP}}{W_{U_{THP}}} \Rightarrow W_{THP} = \eta_{mé} . W_{U_{THP}}$$

We obtain:

$$\dot{m}_m \eta_{mé} W_{U_{THP}} = P_c$$

As the relaxation takes place axially ($\Delta EpTHP = 0$) in an isentropic manner ($QTHP = 0$), the energy conservation equation is established:

$$W_{U_{THP}} = \Delta(h + E_C)_{THP}$$

With the expression of kinetic energy $Ec = 1 / 2\ v^2$, we will have:

$$W_{U_{THP}} = \left(h_3 + \frac{1}{2} v_3^2\right) - \left(h_4 + \frac{1}{2} v_4^2\right) = H_3 - H_4 = C_p(T_3 - T_2)$$

From the expression of the powers, the temperature at the outlet of the HP turbine is determined:

$$\dot{m}_m \eta_{mé} C_p(T_3 - T_2) = P_c$$

$$T_4 = T_3 - \frac{P_c}{\dot{m}_{\ m}\eta_{mé}C_p} = 1479{,}20 - \frac{18290520}{45{,}8739 \times 0{,}95 \times 1150} \tag{II.14}$$

$$T_4 = 1114{,}5\ [K]$$

Now going to calculate the outlet pressure $P4$. For isentropic relaxation, it is given according to Poisson's law by:

$$\frac{T_{is_4}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{P_4}{P_3} = \left(\frac{T_{is_4}}{T_3}\right)^{\frac{\gamma}{\gamma-1}} \tag{II.15}$$

$$P_4 = P_3\left(\frac{T_{is_4}}{T_3}\right)^{\frac{\gamma}{\gamma-1}}$$

The inlet isentropic temperature T_{is_4} is obtained by the expression of the yield isentropic of the turbine:

$$\eta_{is} = \frac{T_4 - T_3}{T_{is_4} - T_3} \Rightarrow T_{is_4} = T_3 + \frac{T_4 - T_3}{\eta_{is}} = 1479{,}20 + \frac{1114{,}5 - 1479{,}20}{0{,}85} \tag{II.16}$$

$$T_{is_4} = 1050{,}14\ [\mathrm{K}]$$

So the outlet pressure will be:

$$P_4 = P_3\left(\frac{T_{is_4}}{T_3}\right)^{\frac{\gamma}{\gamma-1}} = 12{,}08\left(\frac{1050{,}14}{1479{,}2}\right)^{\frac{1{,}35}{1{,}35-1}} \tag{II.17}$$

$$P_4 = 3{,}28\ [bar]$$

•Temperature at the outlet of the LP power turbine

It is considered that the pressure at the outlet of the power or exhaust turbine is atmospheric, i.e. $P5 = 1.013\ [bar]$. For an isentropic transformation, we use always Poisson's law:

$$\frac{T_{is_5}}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{is_5} = T_4\left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = 1114{,}5\ \left(\frac{1{,}013}{3{,}28}\right)^{\frac{1{,}35-1}{1{,}35}} \tag{II.18}$$

$$T_{is_5} = 821{,}84\ [K]$$

By introducing the notion of isentropic efficiency we find:

$$\eta_{is} = \frac{T_5 - T_4}{T_{is_5} - T_4} \Rightarrow T_5 = T_4 + \eta_{is}(T_{is_5} - T_4) \tag{II.19}$$

$$T_5 = 1114{,}5 + 0{,}85(821{,}84 - 1114{,}5)$$

$$T_5 = 865{,}73\ [K]$$

•Power supplied by the gas turbine

The power supplied by the gas turbine is equal to the power of the free turbine (low pressure), we therefore write:

$$P_{TAG} = P_{TBP}$$

As for the HP turbine, the expression of the power will be:

$$P_{TAG} = \dot{m}_{m} W_{TBP}$$

We introduce the mechanical efficiency ηm, we obtain

$$P_{TAG} = \dot{m}_{m} \eta_{mé} C_p (T_4 - T_5)$$

With:

$$\eta_{mé} = \frac{W_{TBP}}{W_{u_{TBP}}} \Longrightarrow W_{TBP} = \eta_{mé} W_{u_{TBP}} \quad \text{et} \quad W_{u_{TBP}} = C_p (T_4 - T_5) \qquad \textbf{(II.20)}$$

We will finally have the real power supplied by the gas turbine:

$$P_{TAG} = 45{,}8739 \times 0{,}95 \times 1150(1114{,}5 - 865{,}73)$$
$$P_{TAG} = 12{,}46766 \; [MW]$$

•Overall efficiency of the gas turbine

The actual efficiency of the gas turbine is given by the expression (III-22), it is the ratio between the power delivered and the quantity of heat released by combustion:

$$\eta_{TAG} = \frac{P_{TAG}}{Q_{cc}} = \frac{12{,}46766}{42{,}151985} \qquad \textbf{(II.21)}$$
$$\eta_{TAG} = 29{,}57 \; \%$$

The theoretical overall efficiency of the gas turbine can also be determined by using the relation (III-23):

$$\eta_{TAG} = \eta_{is} \times \eta_{mé} \times \eta_{th\,TAG} \Longrightarrow \eta_{th\,TAG} = \frac{\eta_{TAG}}{\eta_{is} \times \eta_{mé}} = \frac{29{,}57}{0{,}85 \times 0{,}95}$$
$$\eta_{th\,TAG} = 36{,}62 \; \%$$

III.3.2. ENERGY AND THERMODYNAMICS STUDY OF THE MS5002C: [11]

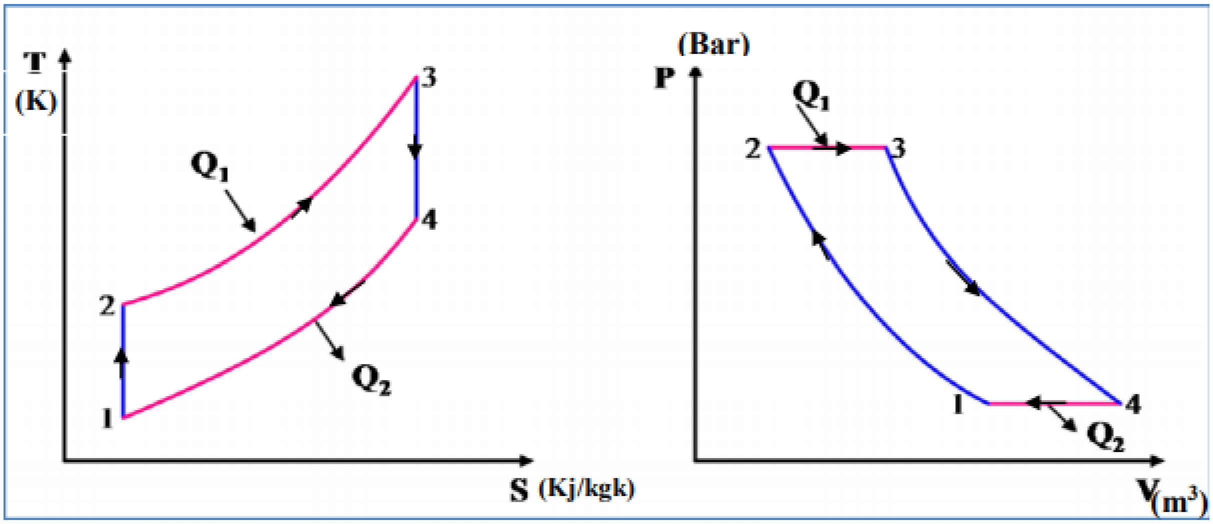


Fig.53 Theoretical cycle of the turbine

•Energy equations

For the calculation of work, power and efficiency we will use some equations and some hypotheses:

- For an ideal gas, the pressure, the volume and the temperature are related by the relation:

PV = n r T (III.1)

With:

P: Pressure [bars]

V: Volume [m^3]

n: number of moles [mole]

r: constant of an ideal gas [J / kg.k]

T: Temperature [k]

$$C_{p\,a} = aT^4 - bT^3 + cT^2 - dT + e \quad \textbf{(II.22)}$$

With

a = 1.9327.10-10

b = 7.9999.10-7

c = 1,1407.10-3

d = 4.4890.10-1

e = 1.0575.103

$$C_{pa_{(T_i T_j)}} = \frac{C_P(T_i) + C_p(T_j)}{2} \quad \textbf{(II.23)}$$

According to the first principle of thermodynamics we have:

$$Q + W = \Delta H + \Delta E_P + \Delta E_C$$

W: Compressor work [kJ / kg]

Q: The amount of heat [kJ / kg]

ΔH: Enthalpy [kJ / kg]
ΔE_P: Potentille energy [J]
ΔE_C: Kinetic energy [J]
By neglecting the variations of kinetic and potential energy, and considering that the transformation in the compressor and the turbine is adiabatic => Q = 0.
- Air is assimilated to an ideal gas.

•The compressor

We apply the equation of the first principle between the input and the output obtains:

$$W_{1-2} = C_{p_{(2-1)}}\left(T_{t_2} - T_{t_1}\right)$$

$$W_{1-2} = C_{p_{(2-1)}}\left(\left(T_2 + \frac{v_2^2}{2}\right) - \left(T_1 + \frac{v_1^2}{2}\right)\right)$$

$$\text{With:} \quad v_1 = v_2 \tag{II.24}$$

$$W_{1-2} = h_2 - h_1 = C_{p_{(2-1)}}(T_2 - T_1)$$

$$\text{With:} \quad h = C_{p_{(2-1)}}.T$$

W_{1-2} = Wc: the mass work of air according to the isentropy (1-2). [Kj / kg] air
h_l: enthalpy at the compressor inlet [kj / kg]
h_2: enthalpy at the compressor outlet [kj / kg]
$C_{P_{(2-1)}}$: specific heat of air at constant pressure [kj / kg.k]
T_l: Temperature at the compressor inlet [k]
T_2: Temperature at the compressor outlet [k]

By introducing the isentropic relation $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

The compression ratio: $\tau = \frac{P_2}{P_1}$ and $\gamma = \frac{C_p}{C_v}$

P_l: pressure at the compressor inlet (atmospheric); [bars]
P_2: pressure at the compressor outlet [bars]
So the power absorbed by the compressor is:

$$\dot{W}_{1-2} = \dot{m}_a W_{1-2} = \dot{m}_a(h_2 - h_1) \tag{II.25}$$

$\dot{m}_a$: The mass air flow [kg / s]
$\dot{W}_{1-2}$: The compressor power [MW]

•The combustion chamber section

In the combustion chamber, there is no work, so the heat Q2−3 supplied by the combustion is as follows:

$$Q_{2-3} = h_3 - h_2 = C_{p(2-3)} \tag{II.26}$$

h_3: The enthalpy at the outlet of the combustion chamber [kj / kg]
T_3: Temperature at the outlet of the combustion chamber [k]

•The turbine section

By reasoning in the same way as for the compressor (adiabatic expansion, with an ideal gas) we obtain:

$$W_{THP} = W_{3-4} = h_3 - h_4$$
$$W_{TLP} = W_{4-5} = h_4 - h_5 \quad \textbf{(II.27)}$$

$W_{3-4} = W_{THP}$: the work of the HP turbine [kj / kg]
h_4: the enthalpy at the outlet of the high-pressure turbine [kj / kg]
$W_{4-5} = W_{TLP}$: The work of the LP turbine [kj / kg]
h_5: the enthalpy of the outlet of the low-pressure turbine (exhaust) [kj / kg]
we have complete combustion in the combustion chamber so we have:
The relaxation rate: $\frac{P_2}{P_1} = \frac{P_4}{P_3}$ From where $(P_2 + P_3 \; and \; P_1 = P_4)$
P_3: the pressure at the outlet of the combustion chamber [bars]
P_4: the pressure at the outlet of the high-pressure turbine [bars]

The power provided by the relaxation:

$$\dot{W}_{3-4} = \dot{m}_{3-4} W_{3-4} = \dot{m}_{3-4}(h_3 - h_4) \quad \textbf{(II.28)}$$
$$\dot{m}_g = \dot{m}_f + \dot{m}_{1-2}$$

With:
$\dot{W}_{3-4}$: The power of the HP turbine [kW]
$(\dot{m}_g)$: Gas flow $[m^3 / s]$
$(\dot{m}_f)$: Fuel flow rate $[m^3 / s]$
The useful work is given by the following relation:

$$W_u = W_t - W_c = C_P(T_3 - T_4) - C_P(T_2 - T_1) \quad \textbf{(II.29)}$$

The useful power is given by the following relation:

$$\dot{W}_u = \dot{W}_t - \dot{W}_c \quad \textbf{(II.30)}$$

•Real Brayton Cycle

The real cycles differ by the incorporation of irreversibility in the transformations real. Compression and rebound are practically adiabatic and irreversible, which increases compressor power and reduces turbine power. The result is an increase in compressor and turbine outlet temperature. So the power generated by the turbine will be lower than that of the ideal cycle, whereas that required by the compressor will be greater.

•Representation of the real Brayton cycle

The MS5002C gas turbine operates on the joule thermodynamic cycle as it is shown in **fig.54**:

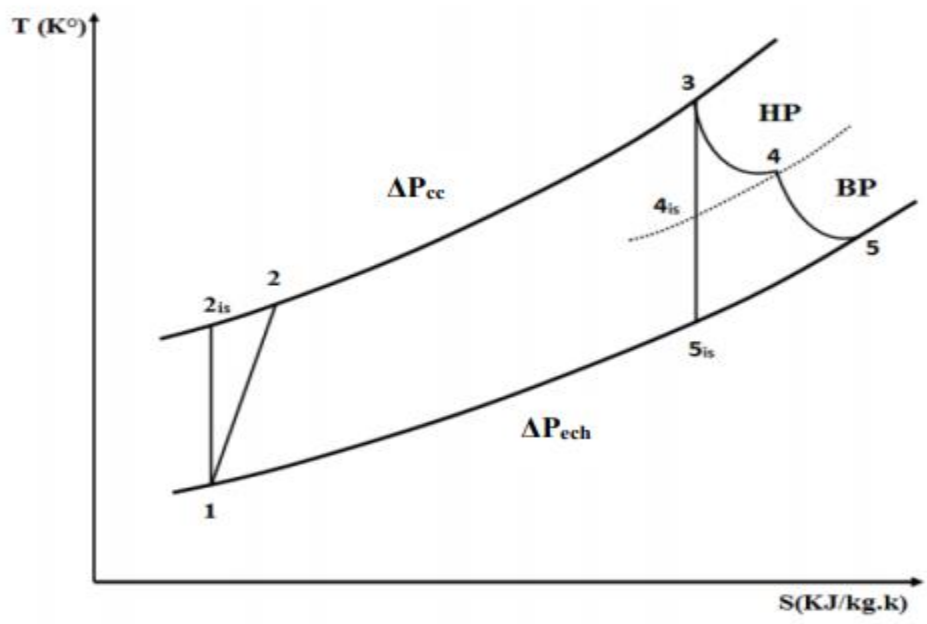


Fig.54 Real cycle of an MS5002C gas turbine

- the segment (1-2) represents a real compression with a temperature T2 higher than T2is that would have been obtained from isentropic compression.
- the segment (2-3) represents the combustion which takes place almost at constant pressure, a pressure drop ΔPch in the combustion chamber.
- the segment (3-4) represents the expansion of the turbine (HP) it dissipates energy at through the walls, the temperature T4 and higher than T4is that would have given a relaxation isentropic.
- the segment (4-5) represents the turbine expansion (LP), the final expansion temperature T5 is greater than T5is.
- the pressure P5 is slightly higher than the atmospheric one, which pushes the combustion gas to atmosphere (exhaust).

•Energy balance of the MS5002C turbine

•Compression phase

The air is compressed in the compressor from pressure P1 to outlet pressure P2. Compression is accompanied by a rise in temperature from T1 to T2.
The actual process is accompanied by losses, which result in an increase entropy
$\Delta S = S2\text{-}S1$
So for the compression from P1 to P2, the increase in temperature across the compressor is larger for the actual cycle, and consequently the work and power absorbed are more important than for the ideal cycle.
We call the isentropic efficiency of compression η_{isc}, the ratio between the work changed in an isentropic transformation and actual work.

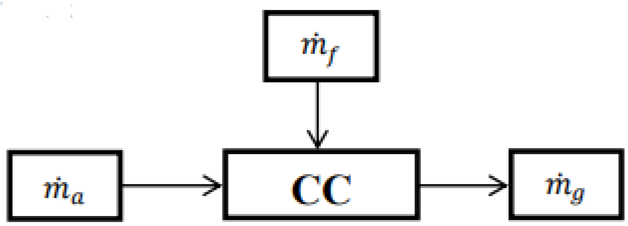
$$\eta_{is} = \frac{Isentropic\ work}{Real\ work} = \frac{W_{is}}{W_{real}} = \frac{C_{P1-2}(T_{2is} - T_1)}{C_{P1-2}(T_2 - T_1)} = \frac{(T_{2is} - T_1)}{(T_2 - T_1)} \quad \textbf{(II.31)}$$

η_{is}: The isentropic efficiency of axial compressor [%]
W_{is}: The isentropic work of axial compressor [kJ / kg]
W_{real}: The real work of compressor [kJ / kg]
T_{2is}: Isentropic temperature at the compressor outlet [k]
With:

$$\frac{T_{2is}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} ==> T_{2is} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \textbf{(II.32)}$$

$$T_2 = \frac{T_{2is} - T_1}{\eta_{is}} + T_1$$

•Combustion phase



The amount of heat supplied to the combustion chamber is:

$$Q_{2-3} = C_{p2-3}(T_3 - T_2) \quad \textbf{(II.33)}$$

The calorific power due to combustion is as follows:

$$\dot{Q}_{2-3} = \dot{m}_{2-3}C_{P2-3}(T_3 - T_2) = (\dot{m}_{1-2} + \dot{m}_f)C_{p2-3}(T_3 - T_2) \quad \textbf{(II.34)}$$

$$\dot{m}_{1-2} = \dot{m}_f\lambda L_0 \quad ==> \dot{m}_f = \frac{\dot{m}_{1-2}}{\lambda L_0}$$

$$PCI_{\eta cc} = \lambda L_0 C_{p2-3}(T_3 - T_2) \quad ===> \lambda = \frac{PCI_{\eta cc}}{L_0 C_{p2-3}(T_3 - T_2)} \quad \textbf{(II.35)}$$

λ: Coefficient of excess air
Determination of air fuel ratio (L_0):

$$L_0 = \frac{\dot{m}_{1-2}}{\dot{m}_f}$$

The stoichiometric mass of the combustion air L_0 is determined by reactions of different fuel components.
The fuel used in the gas turbine is natural gas from the plant of HASSI Messaoud gas, the composition of which is presented in the following table Chemical composition of gas in the Hassi-Messaoud region.

Gas	Symbol	Chemical formula	Partial molar mass M_i (kg/kmol)	Molar fraction N_i (%)
Nitrogen	N_2	N_2	28.01	5.13
Carbon dioxide	CO_2	CO_2	44.01	0.23
Methane	C_1	CH_4	16.04	83.93
Ethane	C_2	C_2H_6	30.07	7.61
Propane	C_3	C_3H_8	44.09	2.14
I-butane	I-C_4	I-C_4H_{10}	58.12	0.29
N-butane	I-C_4	N-C_4H_{10}	58.12	0.43
I-pentane	N-C_4	I-C_5H_{12}	72.15	0.095
N-pentane	N-C_5	N-C_5H_{12}	72.15	0.095
Hexane	C_6	C_6H_{14}	86.17	0.05
Total	-	-	-	100

Reagents		Products	
$CH_4 + 2(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$CO_2 + 2\ H_2O + 2(3{,}76\ N_2)$	x83.93%
$C_2H_6 + 3{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$2\ CO_2 + 3\ H_2O + 3{,}5\ (3{,}76\ N_2)$	x 7.61%
$C_3H_8 + 5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$3CO_2 + 4\ H_2O + 5\ (3{,}76\ N_2)$	x 2.14%
i-$C_4H_{10} + 6{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$4\ CO_2 + 5\ H_2O + 6{,}5\ (3{,}76\ N_2)$	x 0.29%
n-$C_4H_{10} + 6{,}5(O_2 + C_5H_{12}$	$\longrightarrow$	$4\ CO_2 + 5\ H_2O + 6{,}5\ (3{,}76\ N_2)$	x0.43%
i-$C_5H_{12} + 8(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$5\ CO_2 + 6\ H_2O + 8\ (3{,}76\ N_2)$	x0.095%
n- $C_5H_{12} + 8(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$5\ CO_2 + 6\ H_2O + 8\ (3{,}76\ N_2)$	x0.095%
$C_6H_{14} + 8{,}5(O_2 + 3{,}76\ N_2)$	$\longrightarrow$	$7\ CO_2 + 9\ H_2O + 9{,}5\ (3{,}76\ N_2)$	x0.05%
N_2	$\longrightarrow$	N_2	x5.13%
CO_2	$\longrightarrow$	CO_2	x0.23%

•**Molar amount of air required is obtained by:**

The amount of oxygen for the combustion of a CmHn component is Ni:

Ni = (m + n / 4) .Xi **(II.36)**

Ni: molar amount of oxygen required.

Xi: Mole fraction of component i of gas.

•**Total reaction:**

1 mole natural gas (NG) + 2, 1187 (O_2 + 3.76N_2) 1.097 CO_2 + 2.041 H_2O + 8.017 N_2 Oxygen makes up 21% of the molar mass of air.

$$N_a = \frac{N_{(O2)}}{0{,}21}$$

•**Molar mass of air:**

Ma = 0.79M_{N_2} + 0.21M_{O_2}

•**Mass of air required to burn 1 kg of fuel:**

ma = Na. Ma

•**Corresponding fuel mass is:** **(II.37)**

$mf = \sum Xi.\ Mi$

Xi: Mole fraction of component i of gas

Mi: The molar masses of component i of the gas

•**Turbine phase**

At the entrance of the turbine:

$$P_3 = P_{2c}(1-\lambda) \quad \textbf{(II.38)}$$

$$P_{2c} = P_2 * \Delta P_{cc}$$

With:

P_{2c} : the pressure at the entrance to the combustion chamber [bar]

ΔPcc: Coefficient of pressure losses in the diffuser [bar]

λ: Coefficient of pressure losses in the combustion chamber

The expansion of the burnt gases in the turbine, from pressure P_3 to outlet pressure P_4 is accompanied by a decrease in temperature from T_3 to T_4

$$W_{THP} = W_{3-4} = \frac{W_C}{\eta_{mc}} \quad \textbf{(II.39)}$$

$$W_{THP} = Cp_{3-4}(T_3 - T_4) \quad \textbf{(II.40)}$$

$$\dot{W}_{THP} = \dot{m}_{3-4}Cp_{3-4}(T_3 - T_4) \quad \textbf{(II.41)}$$

$$\eta_{is\,THP} = \frac{T_3 - T_4}{T_3 - T_{4is}} \quad \textbf{(II.42)}$$

$$T_4 = T_3 - \eta_{is\,THP}(T_3 - T_{4is}) \quad \textbf{(II.43)}$$

T4: Temperature at the outlet of the HP turbine

$$T_{4is} = \frac{T_3}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}} \quad \textbf{(II.44)}$$

From the energy balance of a gas generator (axial turbine compressor) we have:

$$\dot{W}_c = \dot{W}_{THP} \qquad ==> \dot{W}_c = \dot{m}_{1-2}\frac{W_c}{\eta_{mc}}$$

$$\dot{W}_{THP} = \dot{m}_g W_{THP}\eta_{mcTHP}$$

$$\dot{m}_{1-2}\frac{W_c}{\eta_{mc}} = \dot{m}_g W_{THP}\eta_{mcTHP}$$

$$\Rightarrow\ W_c = \eta_{mc} W_{THP}\eta_{mcTHP}$$

η_{mc}: Mechanical efficiency with mechanical losses

$$W_c = Cp_{1-2}(T_2 - T_1)$$

$$\frac{W_c}{\eta_{mc}} = \frac{1}{\eta_{mc}}Cp_{1-2}(T_2 - T_1) = \frac{1}{\eta_{mc}}T_1 Cp_a\left[\left(\frac{T_2}{T_1}\right) - 1\right]$$

On another part:

$$\frac{T_{2is}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$W_c = Cp_{1-2}\frac{1}{\eta_{mc}}T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad \textbf{(II.45)}$$

The same thing for W_{THP} :

$$W_{THP} = Cp_{3-4}(T_3 - T_4)$$

$$W_{THP} = Cp_{3-4}\frac{1}{\eta_m}T_1\left[1 - \left(\frac{P_3}{P_4}\right)^{-\left(\frac{\gamma-1}{\gamma}\right)}\right] \qquad \textbf{(II.46)}$$

So

$$Cp_{1-2}\frac{T_1}{\eta_{mc}}\left[\left(\frac{P_2}{P_1}\right)^{0,28}\right] = Cp_{3-4}\frac{1}{\eta_{mc}}\eta_{mc}\eta_{mTHP}\left[1 - \left(\frac{P_3}{P_4}\right)^{-0,28}\right]$$

$$P_4 = \frac{P_3}{\left[\frac{Cp_{1-2}\frac{T_1}{\eta_{mc}}\left[\left(\frac{P_2}{P_1}\right)^{0,28}\right]}{\eta_{mTHP}Cp_g T_3}\right]^{\left(\frac{1}{-0,.28}\right)}} \qquad \textbf{(II.47)}$$

- The work and the real power supplied by the trigger is: **(II.48)**

$$W_{4-5} = W_{TBP} = Cp_{4-5}(T_4 - T_5)$$

With: **(II.49)**

$$T_5 = T_4 - \eta_{isLP}(T_4 - T_{5is})$$

And

$$T_{5is} = T_4\left(\frac{T_5}{T_4}\right)^{\frac{\gamma-1}{\gamma}} \qquad \textbf{(II.50)}$$

$$\dot{W}_{4-5} = (\dot{m}_{1-2} + \dot{m}_f)Cp_{4-5}(T_4 - T_5) \qquad \textbf{(II.51)}$$

The efficiency of the installation

$$W_{net} = W_{3-4} - W_{1-2} \qquad \textbf{(II.52)}$$

The net power, is defined as the difference between the real power supplied by the turbine, and that actually absorbed by the compressor.

$$\dot{W}_{net} = \dot{W}_{3-4} - \dot{W}_{1-2} \tag{II.53}$$

The net power is defined as the difference, the thermal efficiency is defined as the ratio between the net power supplied by the turbine and the amount of heat supplied by the combustion.

$$\eta_{Th} = \frac{\dot{W}_u}{\dot{Q}_{2-3}} \tag{II.54}$$

With:

$$\dot{W}_u = \left(\dot{W}_{THP} + \dot{W}_{TLP}\right) - \dot{W}_c \tag{II.55}$$

•Effective efficiency

$$\eta_{eff} = \eta_{mc}.\eta_{Th}.\eta_{cc} \tag{II.56}$$

•**Calculation part**

•**Initial data**

- Ambient air temperature: $T_1 = 288K$
- Ambient air pressure: $P_1 = 1bar$ $P_2 = 7$bar

➢ Characteristic of generator operation:

- Axial compressor rate: $\tau = 7$
- Axial compressor efficiency: $\eta_{isc} = 0.87$
- Efficiency of the combustion chamber: $\eta_{cc}= 0.97$

T_3: The temperature of the combustion chamber outlet is given by the manufacturer between 900 and 1500 ° k, and in our case, it is fixed at 1200 ° K

- Coefficient of pressure losses in the diffuser $\Delta Pcc = 0.98$
- Coefficient of pressure losses in the combustion chamber $\lambda = 0.01$
- Mechanical efficiency of the compressor-turbine transmission: $\eta_{mc}= 0.97$
- Isentropic efficiency of the HP turbine: $\eta_{THP}= 0.88$
- HP turbine speed: $N = 5100\ tr / mn$

➢ Operating characteristics of the LP power turbine

- Isentropic efficiency of the turbine: $\eta_{TLP} = 0.88$
- Useful power = 26 MW
- Mechanical efficiency of turbine-compressor transmission: $\eta_m = 0.98$
- LP turbine speed: N = 4670 tr / mn

- Characteristic of the motor fluid

- Calorific value of the fuel: PCI = 45119 j / kg.k = 10790.20 kcal / kg
- Thermal efficiency of the gas turbine: η_{Th} = 28%
- Mass air flow: $\dot{m}_{\mathrm{a}}$ = 129.9 kg / s

Results table:

T_{2is} =496 k	$Cp_{\{\mathrm{T4,T5}\}}$ = 1.11587 kJ / kg.k
T_2 = 527k	W_{TBP} = 204.44 K J/kg
Cp_1 = 1.00505 kJ / kg.k	$\dot{W}_{BP}$ = 26.9 MW
Cp_2 = 1.03555 kJ / kg.k	$\dot{W}$u = 26.9 MW
Cp_{1-2}= 1.0203 kJ / kg.k	η_{th} = 0.286 = 28.6%
W_C = 243.85 kJ / kg	η_{eff} = 27.18%
$\dot{W}_{1-2}$ = 31.6 MW	
Cp_3 = 1.17980 kJ / kg.k	
Cp_{2-3} = 1.10767 kJ / kg.k	
N_{O2}= 2.1187 moles	
Na = 10.04 kmoles air / kmole fuel	
M_{a} = 28.85 kg / k moles	
m_{a} = 289.65 kg of air / kmole fuel	
m_{C} = 18.83 kg / kmole	
L_0 = 15.34 kgair / kgcomb	
α = 3.9	
$\dot{m}_f$ = 1.98 kg / s	
$\dot{m}_g$ = 131.88 kg / s	
Q_{2-3}= 718.65 kJ / kg	
$\dot{Q}_{2-3}$= 94 MW	
$P_{2\mathrm{c}}$ = 6.86 bars	
P_3 = 6.69 bars	
P_4 = 2.64 bars	
$T_{4\mathrm{is}}$ = 983.8k	
T_4 = 1009.7k	
Cp_{T_4}= 1.5668 kJ / kg.k	
$Cp_{\{\mathrm{T3-T4}\}}$ = 1.37330 kJ / kg	
W_{THP} = 261.3 kJ / kg	
$\dot{W}_{THP}$= 31.6 MW	
$T_{5\mathrm{is}}$ = 775k	
T_5 = 826.8k	
$Cp_{(T_5)}$ = 0.66494 kJ / kg.k	

Table 11: Parameters result table

Notes:
-ADGT are more efficient than the HDGT due to the direct flow in the combustion section, while HDGT are less efficient due to air flow transition which causes pressure drops.

III.4 Reliability, Maintainability and Availability: [12] [13]

This table represents intervention times and the down-time for maintenance for 10 machines of the same category (HD or AD) in 10 years of time operating 20h per day.

Component	HDGT	Down-time for maintenance (HDGT)	ADGT	Down-time for maintenance (ADGT)
Axial compressor	19	69 days	10	44 days
Liners and fuel nozzle /CC	760	61 days	30	29 days
Transition pieces	380	64 days	/	N/A
1st stage nozzle	380	64 days	30	29 days
2nd stage nozzle	380	64 days	30	29 days
HSPT (high pressure)	19	69 days	30	29 days
PT (Low pressure)	19	137 days	0	22 days
Total intervention time / down-time for maintenance	1957	528 days	130	182 days

Table 12: intervention times and the down-time for maintenance for 10 machines of the same category

*10 years = 3650 days
*Uptime (HDGT) = 3650 – 528 = 3122 days = 62440 hours
*Uptime (ADGT) = 3650 – 182 = 3468 days = 69360 hours

III.4.1 Reliability:

Reliability is the ability of accomplishing a required function under given time.

•Calculating reliability:

$$MTBF = \frac{Time\ Between\ Failure(TBF)}{Numbre\ of\ failures}$$

*More the MTBF more the reliability
•Failure rate λ:

$$\lambda = \frac{1}{MTBF}$$

•Calculating R:

$$R = 1 - \lambda = 100 - \lambda(\%) = \mathrm{R}(\%)$$

A) Reliability for heavy-duty gas turbines:

Component	$MTBF(days)$	$\lambda(\%)$	$R(\%)$
Axial compressor	$\frac{3122}{19} \approx 164$	$\frac{1 \times 100}{164} \approx 0.61\%$	$100 - 0.61 = 99.39\%$
Liners and fuel nozzle /CC	$\frac{3122}{760} \approx 4$	$\frac{1 \times 100}{4} \approx 25\%$	$100 - 25 = 75\%$
Transition pieces	$\frac{3122}{380} \approx 8$	$\frac{1 \times 100}{8} \approx 12.5\%$	$100 - 12.5 = 87.5\%$
1st stage nozzle	$\frac{3122}{380} \approx 8$	$\frac{1 \times 100}{8} \approx 12.5\%$	$100 - 12.5 = 87.5\%$
2^{nd} stage nozzle	$\frac{3122}{380} \approx 8$	$\frac{1 \times 100}{8} \approx 12.5\%$	$100 - 12.5 = 87.5\%$
HSPT (high pressure)	$\frac{3122}{19} \approx 164$	$\frac{1 \times 100}{164} \approx 0.61\%$	$100 - 0.61 = 99.39\%$
PT (Low pressure)	$\frac{3122}{19} \approx 164$	$\frac{1 \times 100}{164} \approx 0.61\%$	$100 - 0.61 = 99.39\%$

Notes:

-For 10 HD gas turbines and for 10 years of operation, the possibility to have a failure on the axial compressor, HSPT. PT is very low (0.61%), which make them the most reliable parts on the HDGT.

-For the same turbines and the same time of operation, 1^{st} and 2^{nd} stage nozzles and the transition pieces have low failure rate (12.5%), and reliability of (87.5%) which makes them acceptable to operate.

-The most vulnerable components in the HDGT are the liners and fuel nozzle with a failure rate of (25%) and a reliability of (75%), the reason behind this high failure rate is due to the exposer of these parts to very high temperatures inside the combustion chamber, this is also the reason why HDGT combustion chambers designers placed the CC on the outer side of the turbine, to give the staff easy access to these parts in order to repair/replace them.

B) Reliability for aero-derivative gas turbines:

Component	$MTBF$	$\lambda(\%)$	$R(\%)$
Axial compressor	$\frac{3468}{10} \approx 347$	$\frac{1 \times 100}{347} \approx 0.3\%$	$100 - 0.3 = 99.7\%$
Liners and fuel nozzle /CC	$\frac{3468}{30} \approx 116$	$\frac{1 \times 100}{116} \approx 1\%$	$100 - 1 = 99\%$
Transition pieces	/	/	/
1st stage nozzle	$\frac{3468}{30} \approx 116$	$\frac{1 \times 100}{116} \approx 1\%$	$100 - 1 = 99\%$
2^{nd} stage nozzle	$\frac{3468}{30} \approx 116$	$\frac{1 \times 100}{116} \approx 1\%$	$100 - 1 = 99\%$
HSPT (high pressure)	$\frac{3468}{30} \approx 116$	$\frac{1 \times 100}{116} \approx 1\%$	$100 - 1 = 99\%$
PT (Low pressure)	/	0%	$100 - 0 \approx 99.99\%$

Notes:

-For the same turbines and the same time of operation PT has a 0% failure rate, PT does not require any maintenance and must be replaced after reaching the live limit of 100000 hours which makes it the most reliable part in ADGT.

-Other components have very low failure rate which makes them reliable.

III.4.2 Maintainability:

Maintainability is the ability of being maintained or restored in a state in which it can perform a required function under a given condition of use for which it was designed.

N.B: Maintainability can only be applied to maintainable material (repairable).

•Calculating maintainability:

$$MTTR = \frac{Intervention\ time\ for\ "n"\ failures}{Numbre\ of\ failure}$$

*MTTR: Mean Time To Repair

•Repair rate μ:

$$\mu = \frac{1}{MTTR} = \frac{Numbre\ of\ failure}{Intervention\ time\ for\ "n"\ failures}$$

*Shorter MTTR means higher maintainability

A) Maintainability for heavy-duty gas turbines:

Component	$MTTR$	μ
Axial compressor	$\frac{528}{19} \approx 28$	$\frac{1}{28} \approx 0.04$
Liners and fuel nozzle /CC	$\frac{528}{760} \approx 0.7$	$\frac{1}{0.7} \approx 1.43$
Transition pieces	$\frac{528}{380} \approx 1.39$	$\frac{1}{1.39} = 0.72$
1st stage nozzle	$\frac{528}{380} \approx 1.39$	$\frac{1}{1.39} = 0.72$
2nd stage nozzle	$\frac{528}{380} \approx 1.39$	$\frac{1}{1.39} = 0.72$
HSPT (high pressure)	$\frac{528}{19} \approx 28$	$\frac{1}{28} \approx 0.04$
PT (Low pressure)	$\frac{528}{19} \approx 28$	$\frac{1}{28} \approx 0.04$

B) Maintainability for aero-derivative gas turbines:

Component	$MTTR$	μ
Axial compressor	$\frac{182}{10} \approx 18.2$	$\frac{1}{18.2} = 0.05$
Liners and fuel nozzle /CC	$\frac{182}{30} \approx 6.07$	$\frac{1}{6.07} = 0.16$
Transition pieces	/	/
1st stage nozzle	$\frac{182}{30} \approx 6.07$	$\frac{1}{6.07} = 0.16$
2nd stage nozzle	$\frac{182}{30} \approx 6.07$	$\frac{1}{6.07} = 0.16$
HSPT (high pressure)	$\frac{182}{30} \approx 6.07$	$\frac{1}{6.07} \approx 0.16$
PT (Low pressure)	/	0

Notes:

-From the results of the two tables, heavy-duty have more maintainable parts than the aero-derivative.

III.4.3 Availability:

Availability is the ability of accomplishing a required function under given instant or given interval of time. This ability depends on the combination of reliability, maintainability and maintenance logistics.

•Calculating availability:

$$A = \frac{MTBF}{MTBF + MTTR}$$

*A: equipment availability

A) Availability for heavy-duty gas turbines:

Component	$A(\%)$
Axial compressor	$\frac{164}{164+28} \approx 85.41(\%)$
Liners and fuel nozzle /CC	$\frac{4}{4+0.7} \approx 98.28(\%)$
Transition pieces	$\frac{8}{8+1.39} \approx 85.2(\%)$
1st stage nozzle	$\frac{8}{8+1.39} \approx 85.2(\%)$
2nd stage nozzle	$\frac{8}{8+1.39} \approx 85.2(\%)$
HSPT (high pressure)	$\frac{164}{164+28} \approx 85.41(\%)$
PT (Low pressure)	$\frac{164}{164+28} \approx 85.41(\%)$

B) Availability for aero-derivative gas turbines:

Component	$A(\%)$
Axial compressor	$\frac{347}{347+18.2} \approx 95.02(\%)$
Liners and fuel nozzle /CC	$\frac{116}{116+6.07} \approx 95.03(\%)$
Transition pieces	/
1st stage nozzle	$\frac{116}{116+6.07} \approx 95.03(\%)$
2nd stage nozzle	$\frac{116}{116+6.07} \approx 95.03(\%)$
HSPT (high pressure)	$\frac{116}{116+6.07} \approx 95.03(\%)$
PT (Low pressure)	$100(\%)$

Notes:

-AD gas turbines are more available than the HD gas turbines.

III.4.4 Conclusion:

In this chapter we were able to see the similarities and differences of both heavy-duty and aero-derivate gas turbines, and we were able to conclude that:

-Aero-derivatives components are generally made from light alloys, AD have higher availability, higher efficiency at iso conditions, reliable and maintenance policy that consist of levels.

-Heavy-duty gas turbines components are generally made from high resistance cast iron, HD are reliable as well, have high maintainability, slightly lower efficiency at iso conditions than the aero-derivative and have maintenance policy that consist on inspections.

General conclusion:

Both heavy-duty and aero-derivative gas turbines are widely used in the world, they have different uses, range of power, efficiency, operating cost, maintenance ... etc.

The main differences concluded from this study are:

-AD are more efficient more reliable and more available than the HD gas turbines, while the HD gas turbines are efficient as well, more robust, reliable as well, easier and cheaper to maintain and can be maintained on site.

-HD gas turbines have cheaper operating cost and have a higher life expectancy than the AD gas turbines.

-HD is designed that its exhaust is meant to raise steam and the hotter the steam the more useful it is in industrial application so they are designed to have hotter exhaust and are less efficient at turning heat into mechanical work.

-AD are designed for maximum simple cycle efficiency (without secondary regeneration) so they are useful for simple cycle gas power plant that does not want the complexity of combined cycle or steam generation. AD therefore is also useful in navy ships as well since ships do not need the complexity of combined cycle GT and requires a compact engine.

-HDGT is easier to operate and to maintain.

-AD is smaller in size and light weight than the HDGT, which can be helpful in situations where moving and relocating the gas turbines is required and it is easier, faster to install and can be operational in less time than the HDGT.

-AD gas turbines are generally made from alloys and superalloys, which explains the higher price comparing to the HD gas turbines.

In the case of power generation in our country Algeria, the use of HD is more common and preferable in both northern and southern regions, this is mainly due to:

High life expectancy, easier to operate and maintain, different range of power, cheaper maintenance and spare parts, reliable, efficient and have cheaper operating cost.

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N.B:
All books have been authorized to be read and used in this thesis by Mr. Rabah Yasser.

Chapter I:

GENERAL INFORMATION ON GAS TURBINES

Chapter II:

DESIGN AND CONSTRUCTIVE CHARACTERISTICS

Chapter III:

HDGT AND ADGT COMPARISON

General conclusion

ABSTRACT:

Gas turbines are the backbone of the majority of industries, they are built to be efficient, versatile and reliable. Gas turbines are divided into two categories: Heavy-duty and Aeroderivative, and they are proven to perform in simple and combined-cycle operation for power generation, cogeneration and mechanical drive.

Our study aims to define and cite similarities and differences of each category in general.

Keywords: Heavy-duty, Aero-derivative, Gas turbine.

ملخص:

التوربينات الغازية هي العمود الفقري لمعظم الصناعات ، وقد صممت لتكون فعالة ، ومتعددة الاستخدامات ويمكن الاعتماد عليها. تنقسم توربينات الغاز إلى فئتين: توربينات الخدمة الشاقة و التوربينات المشتقة من الطائرات, و قد اثبتت كفائتها في التشغيل البسيط و و الدورة المدمجة لتوليد الطاقة, التوليد المشترك و التحريك الميكانيكي.

و تهدف دراستنا الى تعريف و تحديد اوجه التشابه و الاختلاف لكل فئة على العموم.

الكلمات المفتاحية: توربينات الخدمة الشاقة, التوربينات المشتقة من الطائرات, التوربينات الغازية.

RESUME :

Les turbines à gaz sont le pilier de la majorité des industries, elles sont conçues pour être efficaces, polyvalentes et fiables. Les turbines à gaz sont divisées en deux catégories : Heavy-duty et Aero-dérivative, et il est prouvé qu'elles fonctionnent en fonctionnement simple et à cycle combiné pour la production d'énergie, la cogénération et l'entraînement mécanique.

Notre étude vise à définir et à citer les similitudes et les différences de chaque catégorie en général.

Mots-clés: Heavy-duty, Aero-derivative, Gas turbine.