GAS TURBINE POWER STATIONS

Dramatic increases in unit size and performance, coupled with significant price reductions and very low emissions, have made the large gas turbine the dominant form of electric power generation.

In the period between 1970 and 1993 worldwide orders for pure steam (Rankine cycle) power plants decreased continually while those for combined cycle and single cycle gas turbine (Brayton cycle) power plants increased. These strong sales trends have carried over into the past few years, and show every sign of continuing into the next century.

Heavy frame industrial gas turbines, designed specifically for central and distributed power plants, and aeroderivative gas turbines—modified jet engines—are in fact playing an increasingly important role in the generation of electricity.

Key factors in the success of gas turbine electricity generation have been very low emissions coupled with low values of the unit cost of fuel brought about by burning natural gas. This value, coupled with high gas turbine efficiencies, tends to minimize the unit cost of the electricity produced, creating the most significant boost for gas turbine sales.

To summarize the main positive factors of gas turbine power plants:

1. Low specific cost ($/kWe)
2. High specific power (kWe/kg/s; kWe/m^3)
3. Flexibility, availability, and reliability
4. Short build-up time (preassembled module)
5. Short start-up time
6. Remote control (i.e., pipelines)
7. Water cooling not necessary
8. Wide power range commercially available (100 kWe to 300 MWe)
9. Very high efficiency in the cogeneration configuration
10. Waste gas temperature well suited for combined cycle (Rankine bottoming cycle solution)
11. Very low CO, unburned hydrocarbon (UHC), and NO_X emissions

while, as a negative factor, we can consider the need for clean fuel (kerosene, distillate oil, natural gas).

In this article, after a simplified thermodynamic analysis of the Brayton gas turbine cycle, the data (efficiency, specific work, turbine outlet temperature, pressure ratio, costs) of existing gas turbines are presented and discussed.

A detailed analysis of the main components of gas turbines and the technological innovations, particularly regarding materials, blade cooling, turbomachinery design, and combustion chamber emissions is also presented.

In conclusion, the cycle innovations under investigation (regenerative cycle, intercooling, reheating, steam injection, humid air gas turbine, etc.) are presented, including a comparison with actual and short term combined cycle performance.

THERMODYNAMIC ANALYSIS OF THE GAS TURBINE POWER CYCLE

The gas turbine cycle refers to the Brayton (or Juole) cycle (1,2), composed of an adiabatic compression, a constant pressure heating, an adiabatic expansion, a constant pressure cooling. This is a description of a closed Brayton cycle (CBC) (3), however, gas turbines for power generation operate with an open Brayton cycle (OBC), see Fig. 1. In the OBC the working fluid is air, while in the CBC various working fluids can be used (helium, nitrogen, CO_2, air, etc.); the OBC compressor inlet conditions are coincident with ambient conditions (ambient pressure and temperature), while CBC compressor inlet pressure can be higher than ambient pressure (pressurized CBC system). In the OBC, the heating phase is realized through the combustion of fuel inside the system (combustion chamber), and the gases are directly utilized as working fluid in the expander. In the CBC, the expander working fluid is clean because the heating is realized with a heat exchanger (external combustion cycle). Finally, in the OBC, the heat is directly wasted to the environment without the need for a gas cooler as in the CBC.

Therefore, the OBC does not need heat exchangers and the cost, weight, and dimensions are greatly reduced compared to CBC systems. Indeed, in the CBC constraints are applied to maximum cycle temperature, due to the use of a heat exchanger and heating system—see Fig. 2—(CBC maximum temperature 850–900°C) (4). While in the OBC, higher maximum temperatures (1400–1450°C) are possible thanks to the use in the hot section (combustor and turbine blades) of advanced cooling techniques. CBC systems can also utilize dirty fuel thanks to the external combustion solution, while for the OBC the fuel must be clean because the combustion gases are utilized inside the expander thus increasing the high temperature corrosion/erosion effects.

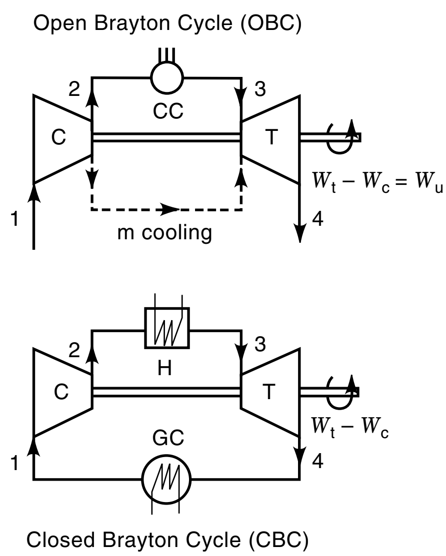


Figure 1. Gas turbine cycle lay-out. C: compressor; T: turbine; CC: combustion chamber; H: heater; GC: gas cooler; W_c: compressor work; W_t: turbine work; W_u: useful work.

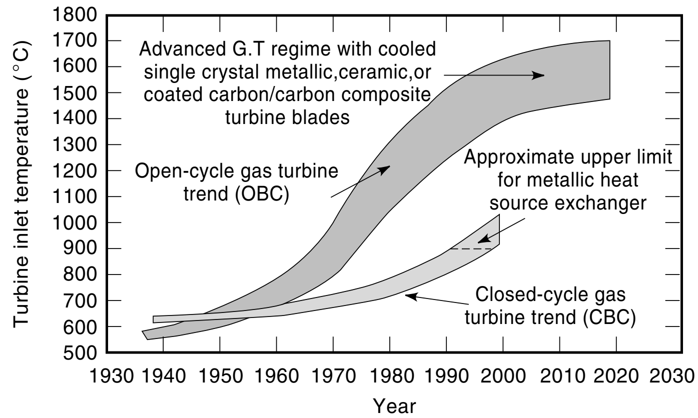


Figure 2. Open and closed cycle turbine inlet temperature trend. This parameter is one of the major factors in OBC efficiency advancement. It has been continuously increased in the last 40 years.

For electricity generation an OBC is usually utilized, while a CBC is used for special applications: space and underwater power generation; gas-cooled nuclear reactor power systems; cryogenic fluid (LNG, LH2) applications (5,6,7,8).

As an example, Fig. 3 shows the performance of an ideal OBC utilizing the perfect gas hypothesis or considering the specific heat c_p as a function of the temperature. Efficiency and specific work (the ratio between the power generated and inlet air mass flow rate) are represented versus cycle pressure ratio β, while maximum cycle temperature T_3, is considered as a parameter. The efficiency is always an increasing function of β and the influence of T_3 is negligible; while specific work curves always present a maximum value, and the effect of T_3 is quite evident. When the gas perfect hypothesis is utilized, the specific work is lower than for $c_p = f(T)$ conditions, while regarding efficiency, the difference is quite reduced.

The next step to improve the thermodynamic analysis is the introduction of the irreversibilities (nonideal cycle). The irreversibilities can be associated to (see Fig. 4):

1. Nonisentropic compression and expansion: compressor and expander efficiency lower than unit
2. Pressure losses in the inlet section-filter, inlet pipes, in the combustion chamber, in the connecting pipes between the compressor, combustion chamber and expander, in the silencer, in the waste heat recovering system, if present
3. Thermal losses in the gas turbine hot section
4. Chemical losses correlated to incomplete fuel combustion
5. Compressed air leakages

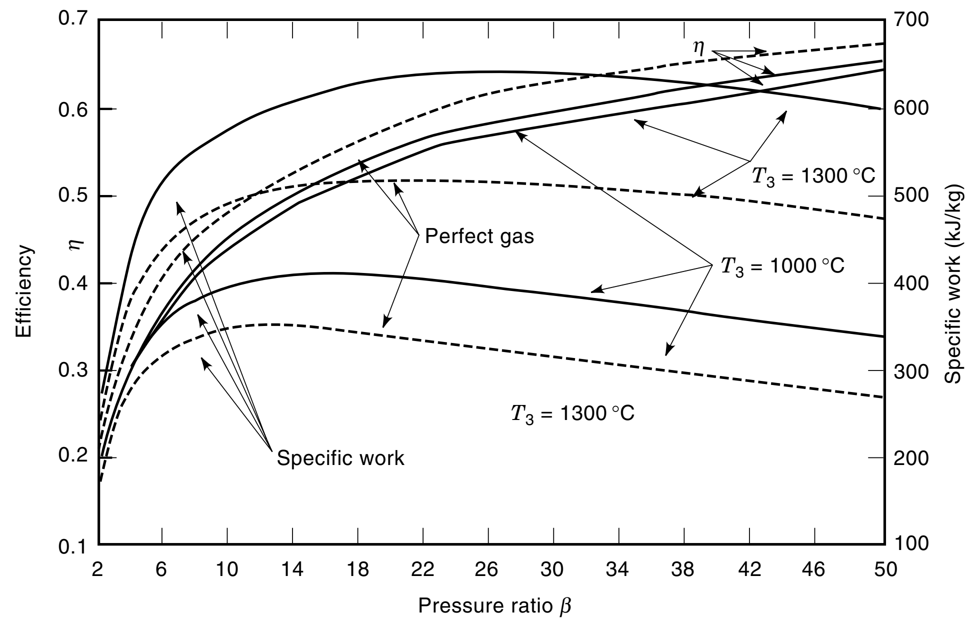


Figure 3. Ideal open Brayton cycle (OBC) performance: perfect gas hypothesis and maximum cycle temperature influence is evident.

6. Hot section-combustion chamber, nozzle, and rotor blades-cooling
7. Mechanical losses (ventilation, bearings, auxiliary systems, etc.)
8. Electrical losses

Regarding the irreversibility associated with blade cooling, it is necessary to point out that it has always been the practice to pass a quantity of cooling air over the turbine disks and blade roots, and a substantial quantity of coolant to the nozzles and the rotor blades. Indeed, to obtain a very high gas temperature coupled with a satisfactory blade material temperature, cooling must be utilized (combustor gas exit temperature 1100–1300°C, material temperature 800–950°C).

When cooling is utilized, the expansion is no longer adiabatic, and taking into account the mixing between the main flow and the cooling air, the specific entropy can be reduced, while the mixing is highly irreversible, and this produces an efficiency reduction.

The influence of nonunitary turbomachinery efficiency is one of the most important differences between the ideal and nonideal cycle; if the network of the cycle is considered:

$$w_\mathrm{u} = w_\mathrm{t} - w_\mathrm{c} = w_\mathrm{is,t} \cdot \eta_\mathrm{is,t} - \frac{w_\mathrm{is,c}}{\eta_\mathrm{is,c}}$$

and taking into account that compressor and turbine adiabatic work are quite similar, the effect of turbomachinery inefficiency is very clear. If the efficiency versus pressure ratio

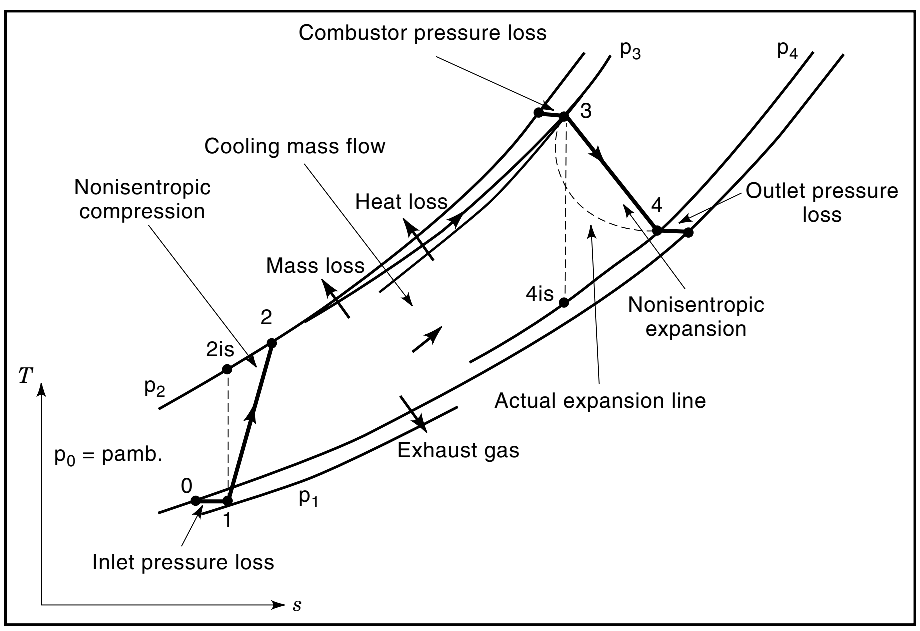


Figure 4. Entropy diagram (T, s) of a real open Brayton cycle (OBC): cycle irreversibilities are shown.

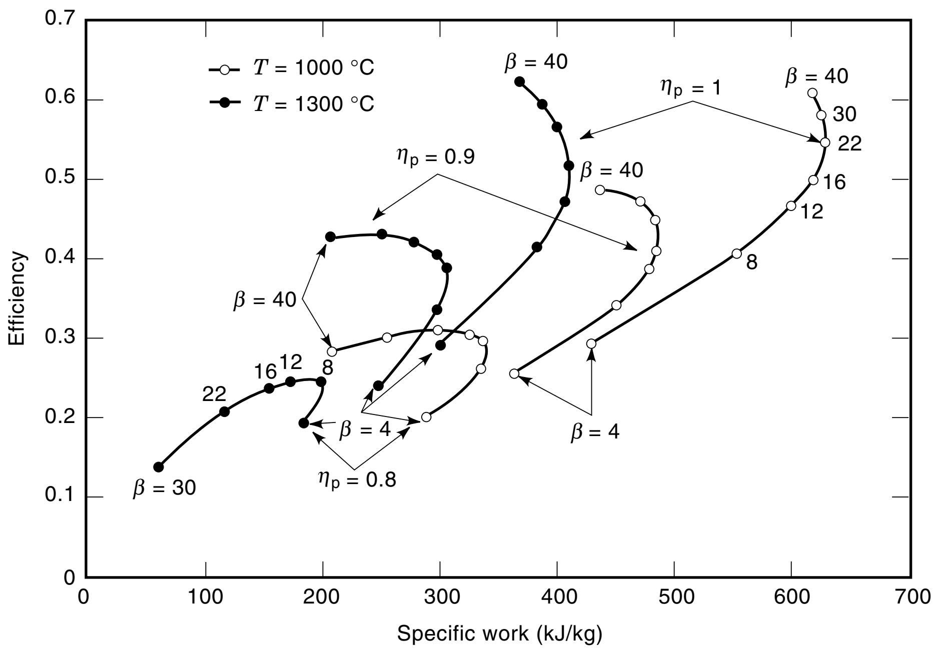


Figure 5. Efficiency vs. specific work of OBC: compressor and expander efficiency and maximum cycle temperature influence is evident.

is analyzed in the nonideal OBC, a different behavior can be found compared to the results illustrated in Fig. 2. In the nonideal cycle, the efficiency is not a monotonic function with β, but it presents a maximum. The pressure ratio value where η is maximum depends on maximum temperature value T_3.

Figure 5 shows the nonideal OBC performance-efficiency versus specific work where the pressure ratio values, the maximum temperature, and the turbomachinery efficiencies are those normally utilized in OBC systems. When ideal turbomachinery are considered ($\eta_p = 1$), the influence of T_3 is present only for specific work data, while when $\eta_p = 0.8$, the OBC efficiency is greatly reduced and optimum β value is around 12 ÷ 13, while the T_3 influence on η is quite evident. If $\eta_p = 0.9$, typical value for large size industrial gas turbines, a large difference between the pressure ratio value for optimum efficiency ($\beta > 30$ depending on T_3 value) and pressure ratio value for optimum specific work ($16 < \beta < 22$) exists. In this way a design parameter as pressure ratio shows a very wide range, and this is very important for the gas tubine designer. Indeed, maximum efficiency (high β) corresponds to minimum fuel consumption (aeroengine solutions), while the maximum specific work condition (low β) minimizes the system costs. More detail about this aspect will be provided in the next section.

EXISTING GAS TURBINES

The calculation of the thermodynamic cycle of a real OBC gas turbine, as already stated, is a very complicated matter. Just to mention some of the phenomena to be accounted for, it must be considered that: air flows bled from various points of the compressor are reinjected in the main stream after cooling turbine nozzles, rotor blades, and disks; the compression and expansion process in turbomachinery is neither adiabatic nor at constant polytropic efficiency; leakages, pressure, and heat losses occur in various parts of the machine. Accurate evaluation of such complex phenomena can be completely performed only by engine manufacturers, who can resort to the information provided by exhaustive measurements to calibrate their computer programs and verify the accuracy of their design methods. While for general thermodynamic analyses it is useful to use much simpler computing procedures, concentrating the previously mentioned loss sources in few parameters.

Another useful strategy is the use of performance data about gas turbines currently produced around the world, which can be found either in handbooks and catalogues issued by specialized publishers, or in technical literature, or in company publications. As an example, Fig. 6, based on the Gas Turbine World Handbook (9), shows the efficiency values versus power for most of the world's existing gas turbines. The output power data shown are for base load at ISO conditions (15°C, 60% relative humidity, sea level, burning natural gas). The efficiency increase versus power is quite evident—for power under 3 MW the efficiency is under 25%, while for aeroderivative turbines, in the range between 10 to 40 MW, the efficiency is in the range 35–40%. Large industrial (heavy-duty) turbines show efficiency values in the range 35–37%. It is important to point out that aeroderivative gas turbines are based on aeroengine gas turbines, while heavy-duty gas turbines are directly designed for electricity generation purposes. The difference between these two different designs is shown more clearly in Fig. 7, where the existing gas turbine pressure ratio is plotted versus power. Aeroderivative systems, taking also into account the very high T_3 values utilized in aeroengines, show very high pressure ratio values-$\beta > 20$ (maximum efficiency design—see Fig. 5—minimum fuel consumption), while lower values for the pressure ratio are utilized for industrial gas turbines (maximum specific work-minimum plant cost; high outlet turbine temperature T_4 value-combined cycle application), considering also the reduced T_3 values utilized in heavy-duty gas turbines.

For small gas turbines (P < 10 MW) β is between 5 and 15, due to the impossibility, at commercial costs, of realizing small, high pressure and high efficiency turbomachinery. Fi-

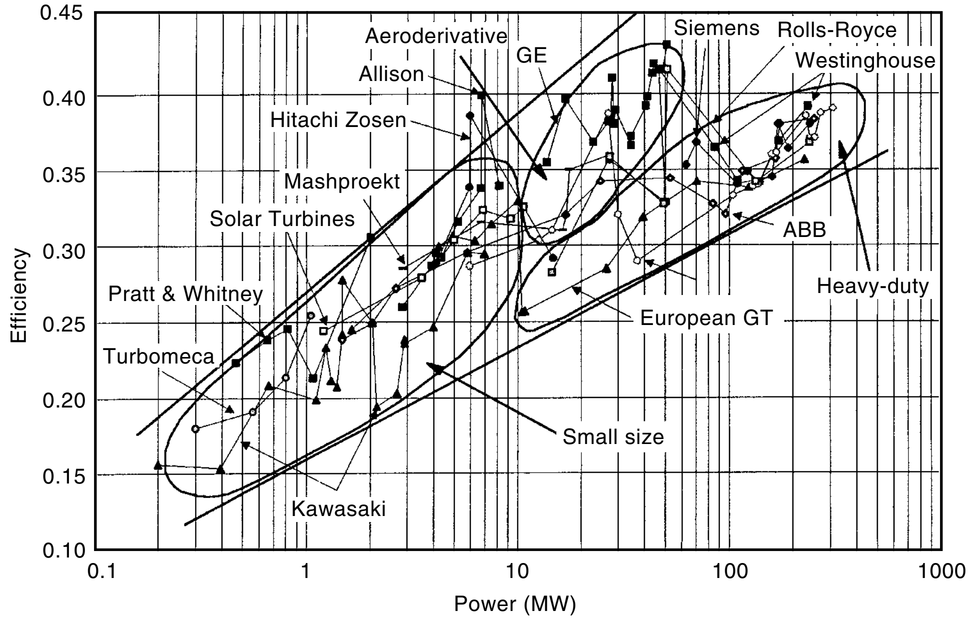


Figure 6. Efficiency vs. power of existing gas turbines (ISO conditions). Three different groups are evident: small size, aeroderivative, and heavy-duty gas turbines.

nally, it is worth noting that only two large gas turbines present a very high pressure ratio ($\beta = 30$): this value is correlated to the nonsimple OBC configuration (reheating cycle), analyzed in the final part of this paper.

Figure 8 shows a simultaneous increase in specific work and efficiency of the existing gas turbines. At fixed specific work, aeroderivative gas turbines show higher efficiency, while at fixed efficiency industrial gas turbines show higher specific work values (see also Fig. 5).

Another thermodynamic feature of existing gas turbines is reported in Fig. 9, where turbine outlet temperature (TOT) is shown. This parameter is particularly important when a gas turbine is utilized as a topping system in combined cycle configurations (the bottoming system is a Rankine cycle heated with the heat recovered from the gas turbine exhaust gas). All the TOT's data are in the range 400–600°C, and the heavy-duty values are higher than aeroderivatives, but this difference is not so evident as in the case of the previous data analyzed (Figs. 7 and 8). Large industrial gas turbines ($P >$ 50 MW) show TOT values higher than 500–520°C, which are particularly well-suited for a combined cycle solution.

To complete the analysis of existing gas turbines Fig. 10 shows the average prices listed only for equipment (not turnkey). Prices per electrical kilowatt generated shown include a basic gas turbine generating package with a single-fuel gas turbine, an air cooled electric generator, standard starting system, skid enclosure, standard inlet and exhaust ducts, exhaust silencer, and standard controls. Not included are any financial or debt service costs, contingency and insurance costs, legal and environmental permitting costs, and so on.

The price per kilowatt drops as the unit size (output) of the gas turbine increases as a function of the economy of scale.

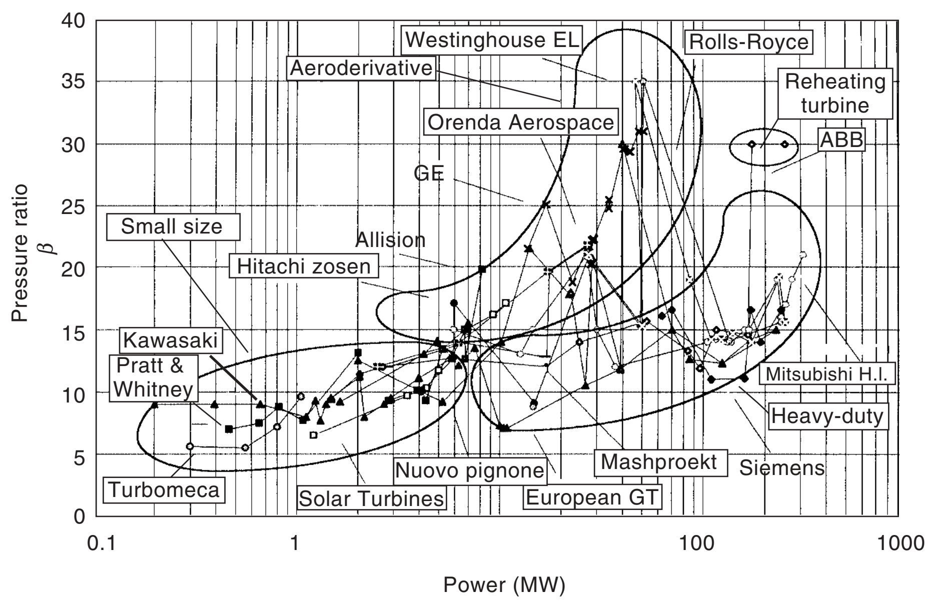


Figure 7. Pressure ratio vs. power of existing gas turbines. Three different groups are evident: small size, aeroderivative, and heavy-duty gas turbines.

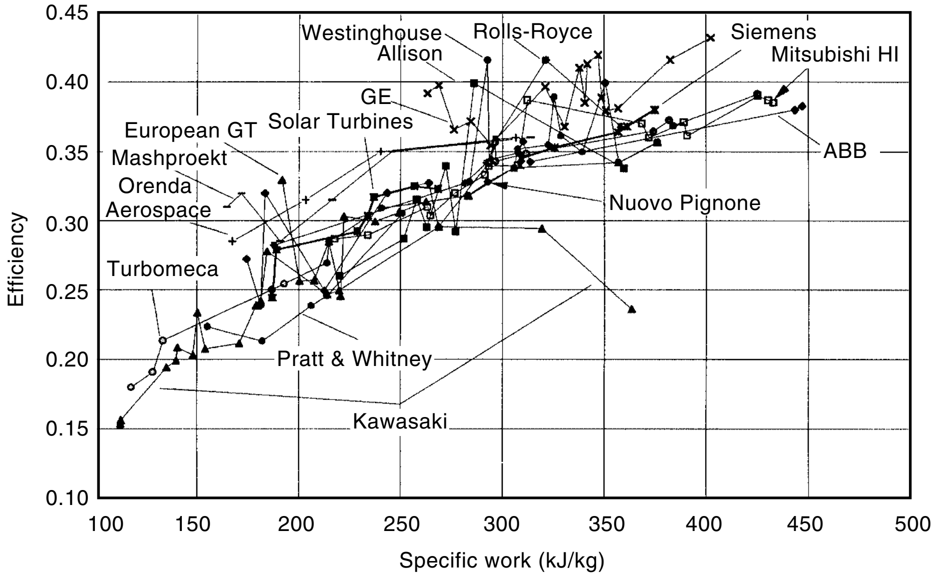


Figure 8. Efficiency vs. specific work of existing gas turbines. In this case the three different groups (small size, aeroderivative, and heavy-duty gas turbines) are not evident.

Obviously prices can vary significantly depending upon the purpose of the plant equipment, geographical area, special site requirements, and competitive market conditions.

GAS TURBINE COMPONENTS AND TECHNOLOGICAL INNOVATIONS

The main components of OBC gas turbines are (see Fig. 1): an axial flow compressor, combustion chamber, and axial flow turbine (expander). Only for small systems are the compressor and/or expander of the radial type (centrifugal compressor; centripetal turbine) instead of the axial type.

To design more effective components, research and development activities have been carried out mainly in the following areas:

- Analytical and design methodologies (computational fluid dynamic, optimization techniques)
- Combustion and pollution control
- Compressor and turbine aerodynamic
- Materials technology and blade cooling techniques
- Fabrication methods
- Instrumentation and diagnostic systems
- Testing approaches

As previously shown one of the major factors in OBC efficiency advancement has been the continuously increasing level of turbine inlet temperature (Fig. 2), made possible by materials technology and turbine blade cooling techniques.

There are several approaches to the problem associated with high gas temperature; in general they can be categorized as developing suitable materials and cooling systems.

Materials

The components that suffer most from a combination of high temperatures, high stress, and chemical attack are those of

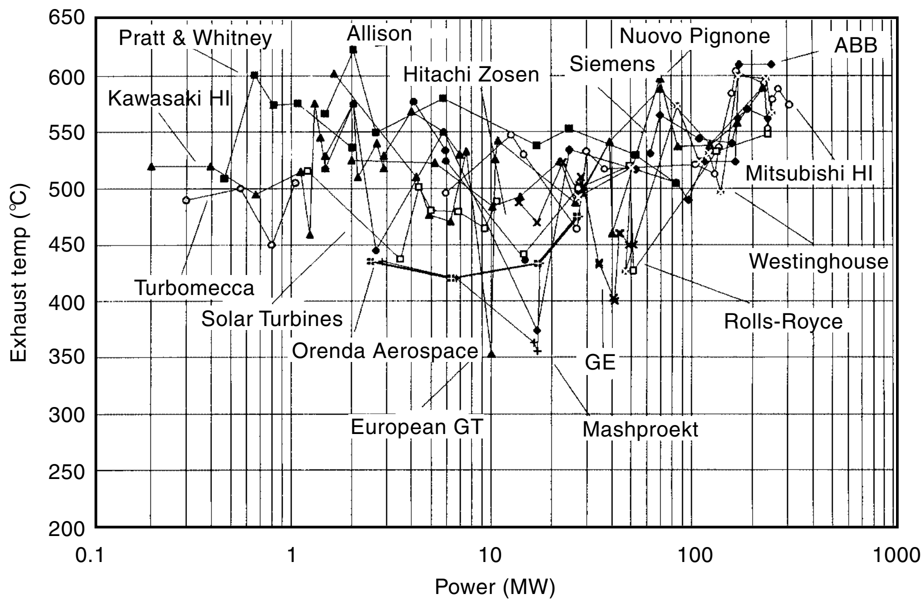


Figure 9. Turbine outlet temperature of existing gas turbines. In this case the three different groups (small size, aeroderivative, and heavy-duty gas turbines) are not evident.

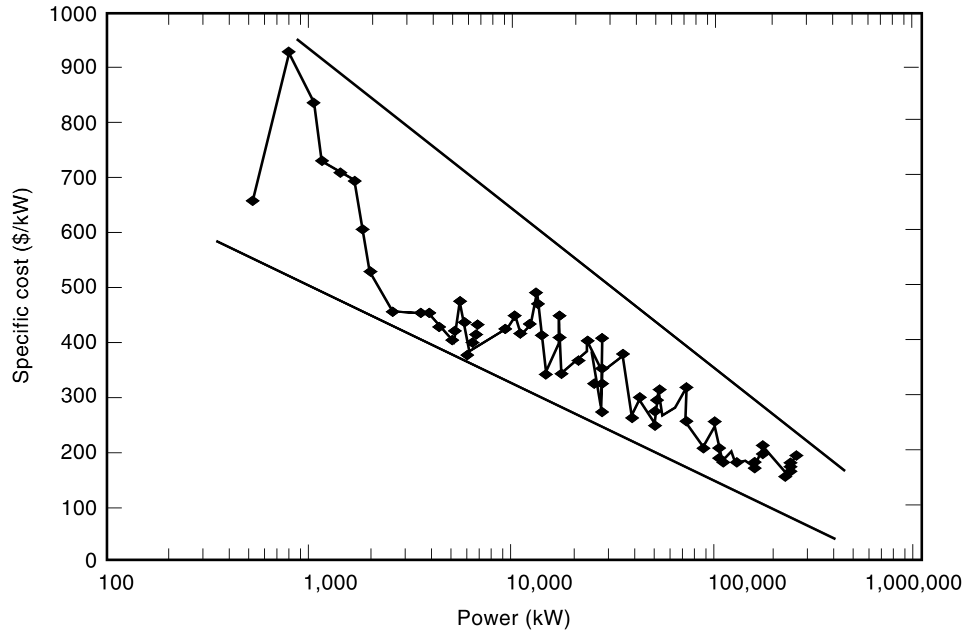


Figure 10. Specific cost vs. power of some existing gas turbines. Not included are any financial or debt service costs, contingency and insurance costs, legal and environmental permitting costs, etc.

the turbine first stage fixed blades (nozzles) and moving blades. Heat resistent materials and precision casting are two recent advances, largely attributable to developments in aircraft engines (Fig. 11). Cobalt based alloys have been used for the first stage fixed blades. These alloys are now being supplemented by vacuum-cast nickel-based alloys that are strengthened through solution and precipitation hardened heat treatment. For the moving blades, cobalt based alloys with high cromium content are now used (4). Ceramic materials are also being developed, especially for combustor and turbine inlet fixed blades. A problem here is inherent brittleness, which causes fabrication problems and raises uncertainties about the mechanical properties of ceramic materials (10).

To ensure the integrity of the blades against chemical attack and increasingly higher operating temperatures, protective coatings are utilized (11). Indeed, the intensity of chemical attack is dependent on the temperature of the blade's surface as well as the purity and temperature of the hot gas. For surface temperatures up to around 800°C, chromium based diffusion coatings have proved to be quite effective and are still being used today in the rear blade turbine stages that run at lower temperatures. Blades operating at higher temperatures require coatings that are thicker and more complex in composition. Here MCrAlY coatings (M comprises Co and/or Ni) are used worldwide for this purpose. The coatings are applied using the vacuum plasma-spray process. All the coatings mentioned are consumed while the turbine is in use. At present, the service life is still much shorter than actual blade material and, consequently, the coatings need refurbishing after some 25,000 h of operation.

A different type of coating is the thermal barrier coating (TBC) (11,12); this has been used for several years in high performance aero-engines. The importance of such coatings will increase significantly over the coming years because their

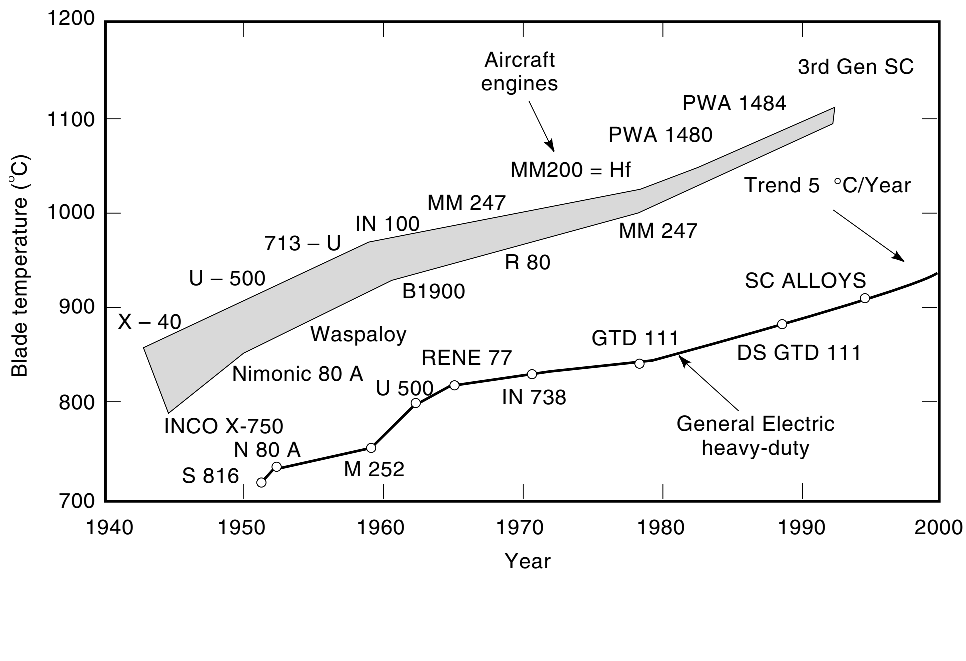


Figure 11. Aeroengine and heavy-duty gas turbine blade temperature trend. The blade temperature increase is largely attributable to development in aircraft engines.

use will allow the operating temperature to be increased by approximately 100 K. A TBC comprises at least two layers: an outer ceramic layer that limits the flow of heat into the blade, and a metallic bonding layer that is usually a member of the MCrAlY family. Zirconia (ZrO_2) is used as the ceramic layer because it has a thermal expansion coefficient similar to that of the materials. The ZrO_2 is stabilized with yttria to avoid detrimental phase transformation. For TBC two dominant failure mechanisms must be taken into account: flaking (incompatible transient stresses between the inner and outer surfaces of the ceramic materials), and peeling (oxide growth on the bonding layer as a result of oxygen diffusion through ZrO_2). A satisfactory service life for a TBC is, nevertheless, achieved by:

- Limiting the thickness of the ceramic layer
- Increasing the strain tolerance of the ceramic material to improve resistance to expansion and thermal shock
- Selecting a bounding layer with maximum oxidation resistance

There are two principal methods for applying these coatings: thermal spraying and physical vapour deposition. The second method is much more costly than the first, but yields better results.

Cooling

The contribution of maximum cycle temperature to the increase in the cycle thermodynamic efficiency of gas turbines was described earlier. However, to obtain very high temperature coupled with satisfactory blade material temperature, cooling must be utilized. The most widespread cooling technique is air cooling, which includes: convection, film, or transpirational cooling (1). It is believed that combined convection and film cooling offers the most promise for air-cooling approaches. At current levels of turbine inlet temperature, three or four stages of a turbine rotor may be cooled, and air would be bled from earlier stages of the compressor to cool the later stages of the turbine. Bleeding air from earlier stages reduces the work input required to pressurize the cooling air, with beneficial effects on the net output.

As already stated, cooling is absolutely necessary to operate gas turbines at very high gas temperature, however, it also has some negative influences on cycle efficiency. The sources of losses are as follows: loss of turbine work due to the reduction in mass flow; nonadiabatic expansion with a negative reheat effect in multistage turbines; mixing of spent cooling air with the main stream; pumping work done by the blades on the cooling air.

To minimize these negative aspects, the cooling mass flow rate must be minimized at fixed T_3 and blade temperature. To obtain this result improvements must be carried out in the following areas:

1. Knowledge of the external (main flow) flow distribution (velocity, pressure, turbulence) and of the heat transfer coefficient distribution on the blade surface. Particular attention must be devoted to the transition point (or zone) position and tip clearance/endwall effects.
2. The internal flow (cooling passage) distribution must be designed to optimize the internal heat transfer coefficient distribution.

Both aspects are very complex and a lot of work has been carried out utilizing theoretical and experimental analysis. In the case of internal flow detailed data have been obtained utilizing the naphtalene sublimation technique (13) and, more recently, the thermocromic liquid crystals technique (14). Both are useful to study the very complex internal passage of cooled blades, including pins, fins, and others.

As shown in Fig. 2, by improving both materials and cooling the turbine inlet, temperature has been steadily increased in the last few decades. However, to further improve the cooling efficiency, closed loop steam cooling in the heavy industrial gas turbines designed for combined cycle operation has been under development (15), and it can be expected to enter service in the late 1990s. The higher heat capacity and heat transfer capability of water permit lower metal temperatures (for the same gas temperature) and hence reduced hot corrosion and deposition from contaminated fuels. Water cooling also eliminates the need for air passage through the blades as in film cooling, which would be subject to plugging combustion gases.

This approach is suitable for use with a combined cycle, where steam is readily available, but requires the use of sophisticated sealing technology to prevent loss of steam. However, the losses due to bleeding high pressure air from the compressor for use in an air cooled turbine are eliminated.

Turbomachinery

Significant improvements in existing gas turbines have also been obtained through improving the performance of turbomachinery. For example, the general trends seen in axial compressor design over the past 30 years are: higher speeds, higher spool pressure ratio, higher pressure ratio per stage, higher aerodynamic loading, lower blade aspect ratios, higher blade row solidities, and improved configurations and blade shapes (16). The trend toward higher tip speeds, higher pressure rise per stage, and lower aspect ratios is seen for the first three compressors in Table 1, each of which has a pressure ratio between 12 : 1 to 13 : 1. A time of about 20 years is covered by these three machines while the Ge-NASA E^3 compressor is a recent design that continues the trend illustrated by the first three compressors. In this case the ratio 23 : 1 is developed in 10 stages!

Core compressor pressure ratio values, for a single spool, have been moving steadily upward as a result of the efficient use of improved materials and advanced mechanical design technique (Fig. 12). Pressure ratios per stage have been increasing partially because higher speeds are being used, but also because the nondimensional loading has been increased.

Table 1

Compressor Name	Pressure Ratio	Corrected Tip Speed (m/s)	Stages
CJ805	12.5	291	17
CF6-50	13	359	14
CFM56	12	396	9
E^3	23	455	10

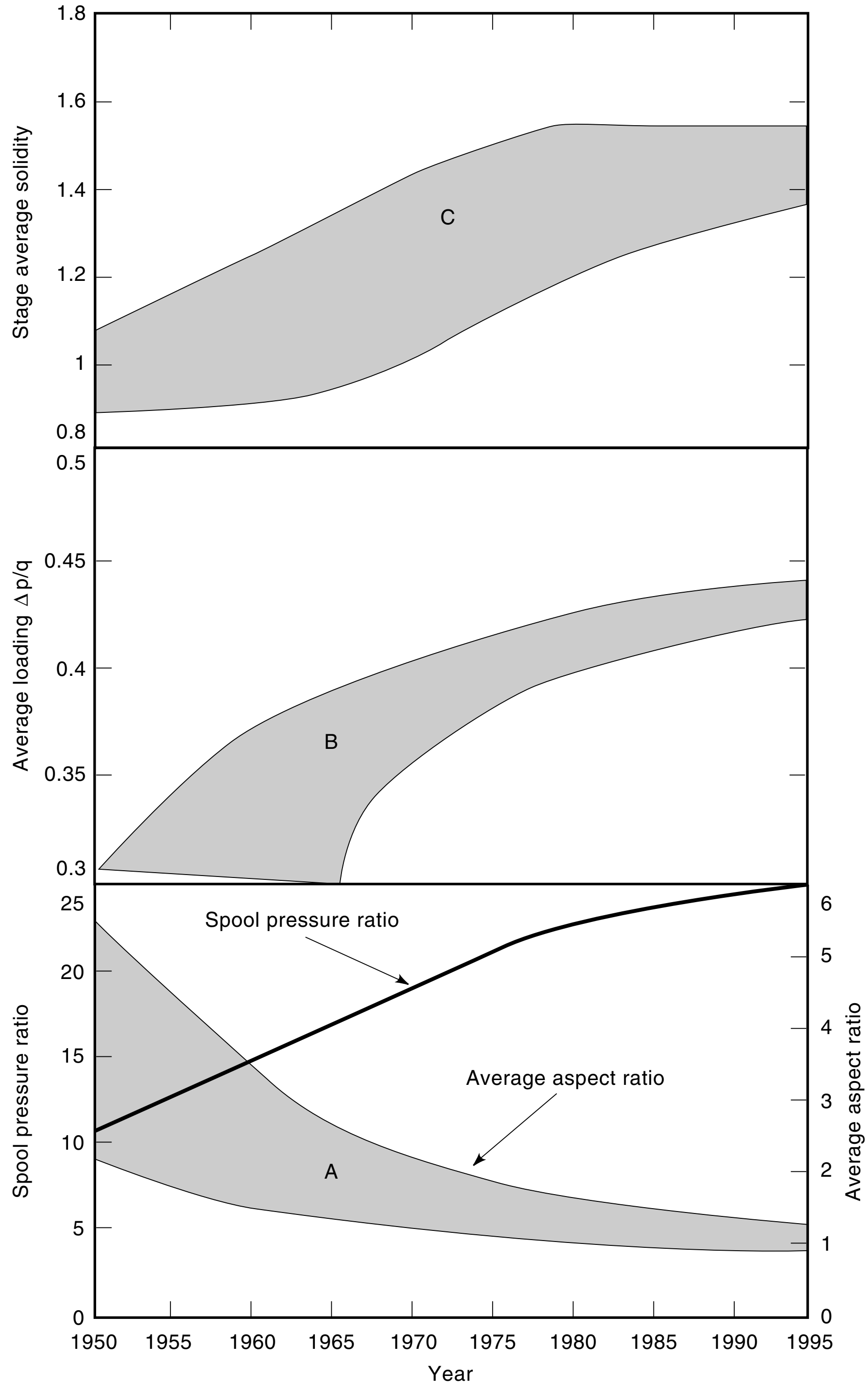


Figure 12. Axial flow compressor design parameters trend: higher speed, higher spool pressure ratio, higher aerodynamic load, lower blade aspect ratios, higher blade row solidities (see also Table 1).

Lower aspect ratios, plus higher solidity and higher stagger blading, are the major design advancements that make this possible. The trend toward lower blading aspect ratios in multistage compressors over the past 30 years is clearly shown in the figures. Average aspect ratios of about 1.3 to 1.4 are now common. Similar considerations can be carried out for solidity values.

In conclusion, the application of advanced design methods and research and development results has led to the compressor system having: improved efficiency and stall-margin, compactness and ruggedness, blading arrangements, and higher stage loading with good performance.

Similar considerations can be carried out for the gas turbine expander, particularly for blade stages, transonic and supersonic design, tip clearance, and endwall effects.

Combustion Chamber

Figure 13 shows a simplified representation of a gas turbine combustion chamber: the diffuser, the swirler, the liner, the

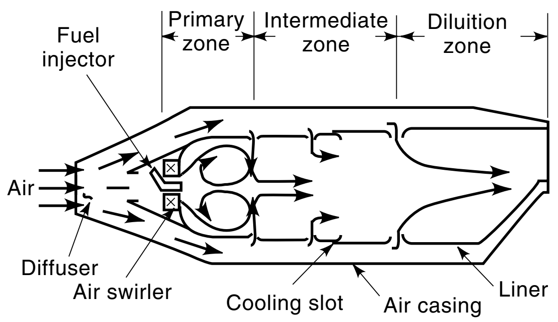


Figure 13. Simplified lay-out of a gas turbine combustion chamber.

primary, secondary, and diluition zones, and the fuel nozzle are evident. More complex configurations can be found in the literature (17). Usually combustors are classified according to their geometry as: single can, multi-can, annular, can annular. Different types of classification are based on fuel systems, air direction inside the chamber, and others.

The main factor of importance in assessing combustion chamber performance are: pressure loss, combustion efficiency, outlet temperature distribution, combustion stability limits, and combustion intensity. As permissible turbine inlet temperatures have increased a problem which has assumed greater importance is that of cooling the flame tube. A recent technique is the use of transpiration cooling, allowing cooling air to enter a network of passages within the flame tube wall before exiting to form an insulating film of air. Another important aspect is correlated to the higher turbine inlet temperatures: they imply the use of lower air fuel ratios, with consequently less air available for film cooling. Furthermore, a higher cycle temperature is usually accompanied by the use of a higher cycle pressure ratio (see Fig. 5). Thus the temperature of the air leaving the compressor is increased and its cooling potential is reduced.

Gas turbine combustion is essentially a clean and efficient process and for many years there was no concern about emissions, with the exception of the need to eliminate smoke from the exhaust. Recently, however, control of emissions has become the most important factor in the design of industrial gas turbines.

A modern well-developed combustor will produce negligible carbon monoxide (CO) and hydrocarbons (UHC), while sulfur emissions are practically absent (virtually no sulfur is present in natural gas). The principal pollutants emitted are nitrogen oxides NO_x. The most important factor affecting the formation of NO_x is the flame temperature (Fig. 14). The rate of formation of NO_x varies exponentially with the flame temperature, so the key to reduce NO_x is reduction of flame temperature; NO_x is also slightly dependent on the residence time of the fluid in the combustor. Unfortunately, the reduction of flame temperature corresponds to an increase in CO and UHC emission, and the residence time has a similar effect.

There are basically three major methods of minimizing NO_x emissions:

- Water or steam injection into the combustor
- Selective catalytic reduction (SCR)
- Dry low NO_x design

The purpose of water (or steam) injection is to provide a substantial decrease in flame temperature; SCR is a system for exhaust clean-up, where a catalyst is used together with injection of controlled amounts of ammonia resulting in the conversion of NO_x to N_2 and H_2O. SCR has been used in situations where extremely low (<10 ppm) limits of NO_x have been specified. For dry low NO_x systems three different designs exist, and they depend on the gas turbine type: heavy-duty, aero-derivative, and aeroengine.

ADVANCED CYCLES

As already stated, the simple OBC gas turbine performances are good, due mainly to the high efficiency and reliability of the components. However, several different thermodynamic cycles can be utilized to improve simple OBC performance. Unfortunately, they usually lose the compactness and other characteristics of simplicity of simple cycle gas turbines.

As an example, three modifications of the simple cycle are presented here: regeneration, intercooling, and reheating (see Fig. 15).

Regeneration

It consists in the heat recovery of exhaust gas to heat the compressor outlet air before the combustion chamber (in the CBC regeneration it is absolutely necessary to obtain high efficiency). In this way, a heat exchanger between the compressor and combustion chamber must be utilized. A complete analysis of the regenerative cycle is presented in Ref. 2; here in Fig. 16 the efficiency versus specific work of such a cycle is compared to simple cycle data. The efficiency of the regenerative cycle is less dependent on compressor pressure ratio, particularly for low β values; for simple cycle β has the opposite influence; the efficiency improvement is not very high for high β values, but is evident for low β values; efficiency is around 43–44%. This value, for industrial power generation, does not seem to be decisive for the use of the regenerative cycle instead of the simple cycle. As for specific work it decreases a lot due mainly to the regenerator pressure losses. Regenerative solutions are utilized only for terrestrial and marine propulsion systems (18) and for small gas turbines employed to generate electricity; for large industrial gas turbines regenerative cycles are not utilized due to the increment of cost and size of the plant, while there is not great improvement in efficiency.

Intercooling

In this case, the compression phase of the OBC is obtained in more than one part (max 3) and between every compression phase the air is cooled (usually until ambient temperature) utilizing a water cooling heat exchanger. This allows the compression work to be minimized (approximation of the isothermal compression). Net OBC work increases, due to the reduction of compression work, but at the same time the cycle efficiency decreases due to the higher fuel rate necessary to heat the air until turbine inlet temperature. Indeed, the outlet compressor temperature with intercooling is lower than the corresponding value in simple cycle configuration. Nevertheless, the effect of the colder air in the last stages of the compressor, which is very useful for turbine cooling purposes, at the same β values as for simple cycle must be considered.

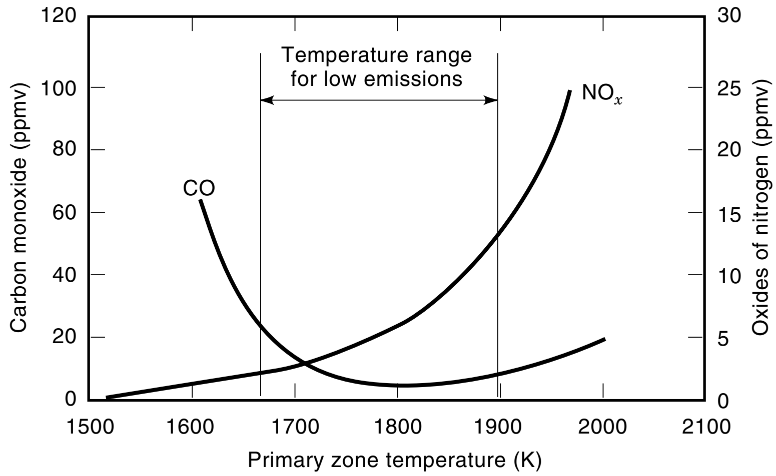


Figure 14. Carbon monoxide and oxides of nitrogen vs. combustion chamber primary zone temperature. The rate of oxides of nitrogen varies exponentially with the flame temperature.

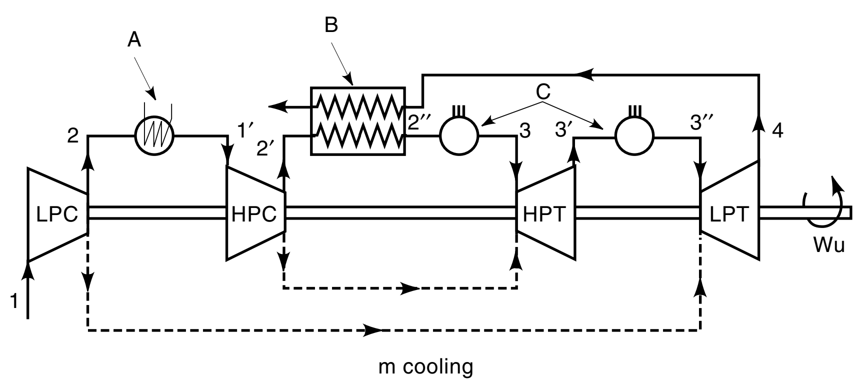
Figure 16 also shows intercooled cycle efficiency versus specific work. At fixed β values, efficiency is practically constant, while specific work increase is quite evident (corresponding to a high increase in power generated). It is possible to note that with the available technology, the intercooled cycle pressure ratio can arrive at $\beta = 80$ instead of 30 as for the simple cycle. Intercooling is very useful to improve the performance of small OBC gas turbines, while for large systems it is less so. While specific work increases (positive economic aspect), the cost of the system increases due to the heat exchangers, connecting pipes, and compressor modifications. Cooling water must be available, while for the simple OBC this constraint is not present (see for example gas turbine installations in deserts).

Reheating

In this case the expansion phase is divided into two parts and after the first, a second combustion chamber is introduced to heat the working fluid (recombustion is possible due to the high excess of air in the first combustor). In this way, the expansion work is greatly increased; however, the negative thermodynamic aspect of this solution is the increase in turbine outlet temperature. Nevertheless, the influence of this defect is greatly reduced, or it vanishes, if the waste heat is recovered in a Rankine bottoming cycle (combined cycle).

Figure 16 shows the performances of the reheating cycle, which are greatly dependent on the expansion ratio in the first (high pressure) and second (low pressure) turbine. The increase in specific work is evident in both the cases presented here, while the efficiency is similar to simple OBC data only when the expansion ratio of the high pressure turbine is equal to 2.

One of the most important gas turbine companies launched a large industrial reheating cycle gas turbine (19), a revolutionary proposal in a market where the simple cycle gas turbine is absolutely dominant. The efficiency of this machine is



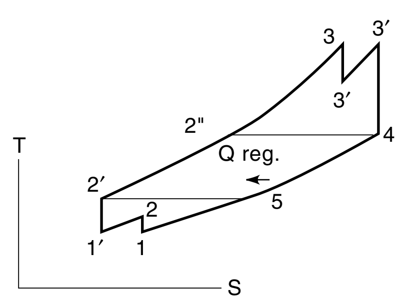


Figure 15. Three modifications of OBC simple cycle are presented: regeneration, intercooling, and reheating (A: intercooler, B: regenerator, C: combustion chamber, LPC and HPC low and high pressure compressor, LPT and HPT low and high pressure turbine.

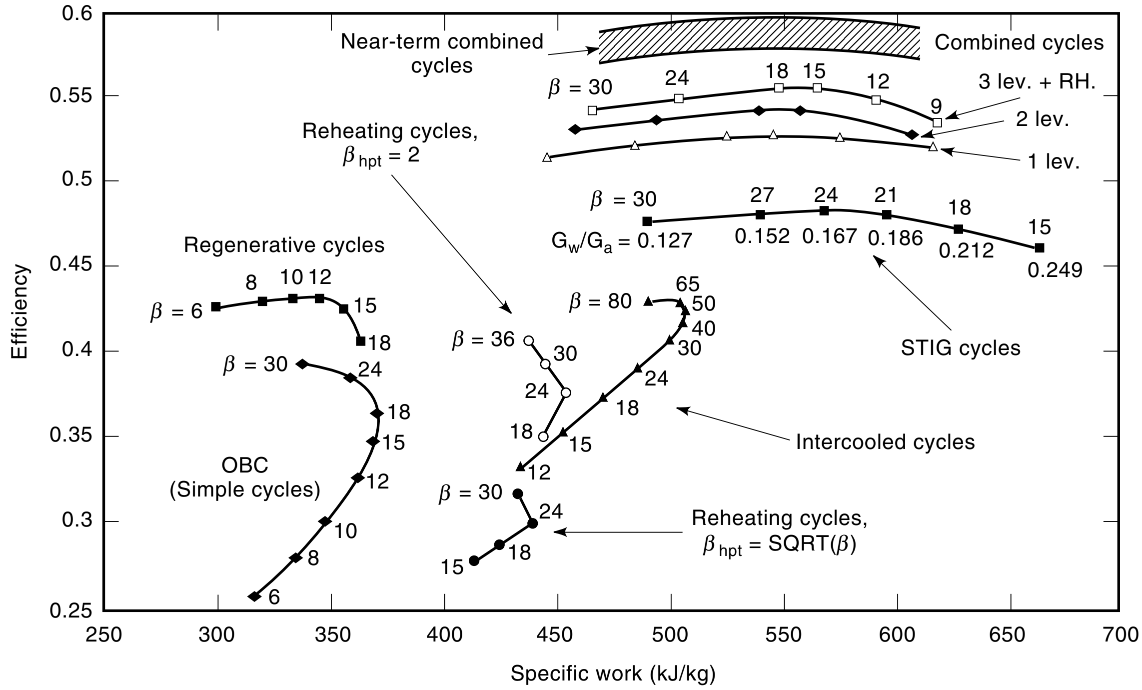


Figure 16. Efficiency vs. specific work for several advanced gas turbine-based cycles and combined cycles (simple OBC cycle, regenerative cycles, intercooled cycles, re-heating cycles, steam injection cycles, existing combined cycles, near-term combined cycles).

similar to the simple cycle gas turbine, but not so high as to justify the technological difficulties associated with the recombustion process. However, the associated combined cycle performance is very high, about 58%, thanks to the high turbine outlet temperature values (>600°C) and the economic results also seem to be very interesting.

Obviously, several combinations of nonsimple solutions can be utilized, with intriguing performances. An interesting aspect of the regeneration and intercooled or reheated cycle is that regeneration avoids the negative thermodynamic aspect of intercooling or reheating previously discussed (more heat entering in the cycle).

Other systems based on OBC technology are steam injection turbines (STIG (20), CHENG (21)), humid air turbines (HAT (22)), cascaded humid air turbine (CHAT (23)) or more complex solutions.

The most utilized is the steam injection cycle, where the steam is generated by recovering the waste heat of the exhaust gas at the expander exit, through the use of a heat recovery steam generator. Normally the steam pressure is 1.25–1.30 times the maximum OBC pressure to allow steam injection in the combustion chamber and in the power turbine stages. The high c_p value of steam or water increases the output of the gas turbine for fixed temperature drop (the expander working fluid is a mixture of combustion gases and steam) and decreases the combustion flame allowing a reduction of No_x emissions.

Figure 16 shows a comparison between the simple OBC system and steam injection cycle performance: utilizing the available technology, an interesting improvement in efficiency and specific work is quite evident. In this way, it is possible to understand the potential cost reduction of the steam injection solution, particularly for small gas turbines and aeroderivative systems. Another interesting aspect of the steam injection cycle is the very high performance when electricity is generated together with heat—cogeneration power plant—and also when water recovery is utilized (24). However, the steam injection solution requires very clean water (demineralized water; careful control of the compressor surge margin limit; new compressor-expander matchline; mechanical stress limitation due to the high power increase (see Fig. 16). An offshoot of steam injection in OBC systems is the coupling with the intercooling steam injection gas turbine cycle (ISTIG); in this case the efficiency can be higher than 50%.

New cycles are also under investigation all around the world, and, at this moment, the most promising is the HAT (humid air turbine) or CHAT (cascaded humid air turbine). The HAT cycle is a regenerative cycle with a complex lay-out: the high temperature heat at the expander outlet is transferred to the hot and water saturated air at the compressor outlet. Some preliminary performance evaluations for the HAT show a cycle efficiency value around 55% and very high specific work values (600–650 kW/kg/s), and higher for the CHAT solution.

A last very promising solution is the possible coupling between the gas turbine simple-cycle with solid oxide fuel cells (SOFC) (25). In this case an efficiency value higher than 70% seems to be possible.

SUMMARY

The success of gas turbine power plants for generating electricity is dependent on the technological development of the components of the plant (axial flow compressor, combustion chamber, expander, materials, cooling, control systems) that has taken place mainly in the last twenty years, and correlates closely to the research and development activities in the aeroengine field.

However, at this moment, very important reasons for the gas turbine power plant's success are also the low cost of energy generation utilizing natural gas, the reduced environmental impact, the wide availability and reliability, and finally the possibility to be utilized in the combined cycle configuration (the heat of the high temperature exhaust gas—see Fig. 9—is recovered in a bottoming steam Rankine cycle). The combined cycle solution shows efficiency greater than 55%, and in the short term period close to 60% (see Fig. 16), while an increase in the specific work of about 60% is evident compared to the OBC simple cycle.

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