

GAS TURBINE PERFORMANCE

by

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ABSTRACT

Industrial gas turbines show performance characteristics that distinctly depend on ambient and operating conditions. They are not only influenced by site elevation, ambient temperature, and relative humidity, but also by the speed of the driven equipment, the fuel, and the load conditions. Proper application of gas turbines requires consideration of these factors. This tutorial explains these characteristics based on the performance of the engine compressor, the combustor and the turbine section, and certain control strategies. It introduces fundamental concepts that help to understand the flow of energy between the components. Additionally, methods are introduced that allow the use of data for trending and comparison purposes. The impact of component degradation on individual component performance, as well as overall engine performance is discussed, together with strategies to reduce the impact of degradation.

INTRODUCTION

Gas turbines have been used for many aerospace and industrial applications for many years. They are used successfully to power aircraft as well as in industrial applications (Figure 1).

Gas turbines for industrial applications consist either of an air compressor driven by a gas generator turbine with a separate power turbine (two shaft engine, Figure 2) or of an air compressor and a turbine on one shaft, where the turbine provides both power for the air compressor and the load (single shaft engine, Figure 2). The power and efficiency characteristics of a gas turbine are therefore the result of a complex interaction of different turbomachines and a combustion system.

The topics presented should enhance the understanding of the principles that are reflected in performance maps for gas turbines, or, in other words, explain the operation principles of a gas turbine in industrial applications.

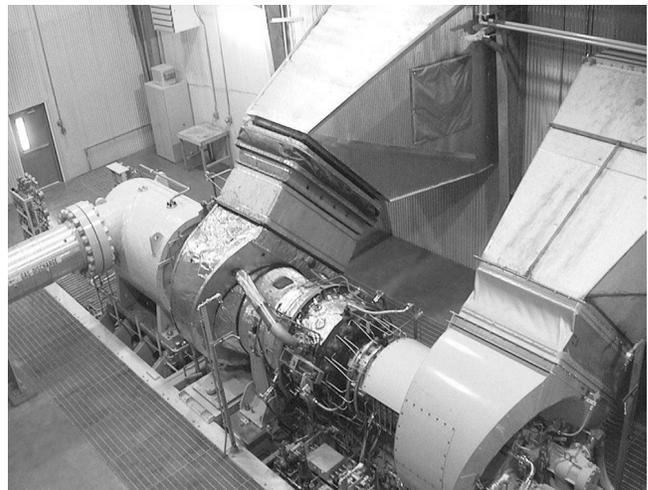


Figure 1. Gas Turbine Applications.

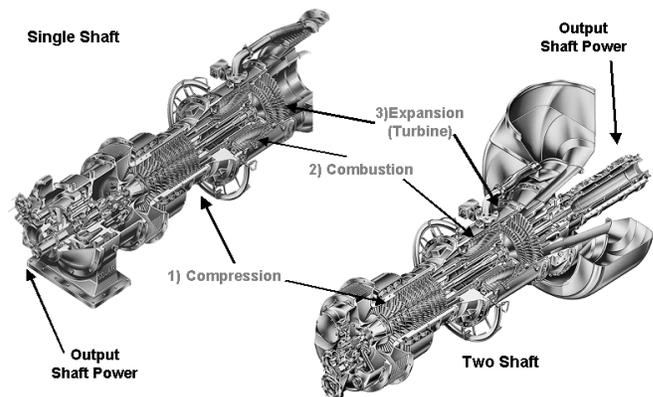


Figure 2. Single Shaft (Cold End Drive) and Two Shaft (Hot End Drive) Gas Turbines.

THERMODYNAMICS OF THE GAS TURBINE CYCLE (BRAYTON CYCLE)

The conversion of heat released by burning fuel into mechanical energy in a gas turbine is achieved by first compressing air in an air compressor, then injecting and burning fuel at (ideally) constant pressure, and then expanding the hot gas in the turbine (Brayton Cycle, Figure 3). The turbine provides the necessary power to operate the compressor. Whatever power is left is used as the mechanical output of the engine. This thermodynamic cycle can be displayed in an enthalpy-entropy (*h-s*) diagram (Figure 3). The air is compressed in the engine compressor from state 1 to state 2. The heat added in the combustor brings the cycle from 2 to 3. The hot gas is then expanded. In a single shaft turbine, the expansion is from 3 to 7, while in a two shaft engine, the gas is expanded from 3 to 5 in the gas generator turbine and afterwards from 5 to 7 in the power turbine. The difference between Lines 1-2 and 3-7 describes the work output of the turbine, i.e., most of the work generated by the expansion 3-7 is used to provide the work 1-2 to drive the compressor.

In a two shaft engine, the distances from 1 to 2 and from 3 to 5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 5-7 describes the work output of the power turbine.

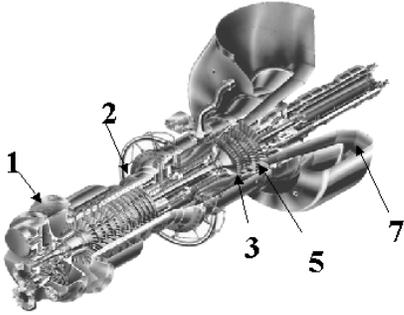
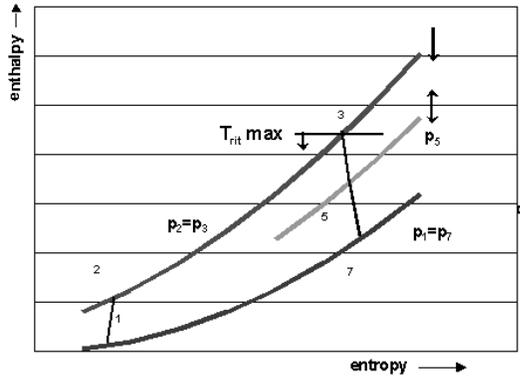


Figure 3. Enthalpy-Entropy Diagram for the Brayton Cycle.

For a perfect gas, enthalpy and temperature are related by:

$$\Delta h = c_p \Delta T \quad (1)$$

For the actual process, the enthalpy change Δh for any step can be related to a temperature rise ΔT by a suitable choice of a heat capacity c_p for each of the steps.

The entire process can be described (assuming that the mass flow is the same in the entire machine, i.e., neglecting the fuel mass flow and bleed flows, and further assuming that the respective heat capacities c_p , c_{pe} , and c_{pa} are suitable averages):

$$\begin{aligned} -c_{pa}(T_2 - T_1) + c_{pe}(T_3 - T_7) &= P / W \\ c_{pe}(T_3 - T_2) &= E_f W_f / W \end{aligned} \quad (2)$$

In this equation, the first term is the work input by the compressor, and the third term describes the work extracted by the turbine section. The second term is the temperature increase from burning the fuel in the combustor.

For two shaft engines, where the gas generator turbine has to balance the power requirements of the compressor, and the useful power output is generated by the power turbine, the equation above can be rearranged to find:

$$\begin{aligned} -c_{pa}(T_2 - T_1) &= c_{pe}(T_3 - T_5) \\ c_{pe}(T_5 - T_7) &= P / W \end{aligned} \quad (3)$$

This relationship neglects mechanical losses and the difference between the gas flow into the compressor and into the turbine due to the addition of fuel mass flow. However, the resulting inaccuracies are small, and do not add to the understanding of the general principles.

The compressor and the turbine sections of the engine follow the thermodynamic relationships between pressure increase and work input, which are for the compressor:

$$P = \frac{\Delta h}{W} = \frac{c_p}{W}(T_2 - T_1) = \frac{c_p T_1}{W \eta_c} \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (4)$$

and the turbine:

$$P = \frac{\Delta h}{W} = \frac{c_p}{W}(T_3 - T_7) = \frac{c_p}{W} \cdot \eta_t \cdot T_3 \left[1 - \left(\frac{P_7}{P_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (5)$$

The efficiency of a gas turbine is defined by comparing the amount of power contained in the fuel fed into the engine with the amount of power yielded. The thermal efficiency is thus:

$$\eta_{th} = \frac{P}{W_f E_f} \quad (6)$$

and the heat rate is:

$$HR = \frac{1}{\eta_{th}} = \frac{W_f E_f}{P} \quad (7)$$

In this paper, T_3 , turbine inlet temperature (TIT) and turbine rotor inlet temperature (TRIT) will be (loosely) referenced as firing temperatures. The differences, which lie simply in the fact that temperatures upstream of the first turbine nozzle (TIT) are different from the temperatures downstream of the first nozzle (TRIT) due to the cooling of the nozzles, are not important for the understanding of the topic of this paper. APPENDIX A shows an example for a typical gas turbine (GT) cycle.

A QUICK EXCURSION TO AERODYNAMICS

Any gas turbine consists of several turbomachines. First, there is an air compressor, and after the combustion has taken place, there is a turbine section. Depending on the design of the gas turbine, the turbine section may consist either of a gas generator turbine, which operates on the same shaft as the air compressor, and a power turbine, which is on a separate shaft.

The task of the compressor is to bring the inlet air from ambient pressure to an elevated pressure. To do this, power is necessary, i.e., the compressor imparts mechanical power into the air. The same relationships that apply to the compressor can also be applied to the turbine, except that the turbine extracts work from the flow. The transfer of energy is accomplished with rotating rows of blades, while the stationary rows allow conversion of kinetic flow energy (i.e., velocity) into pressure or vice versa.

The fundamental law describing the conversion of mechanical energy into pressure in a turbomachine is Euler's Law. Euler's law connects thermodynamic properties (head) with aerodynamic properties (i.e., velocities u and c , Figure 4):

$$\Delta h = u_2 c_{u2} - u_1 c_{u1} \quad (8)$$

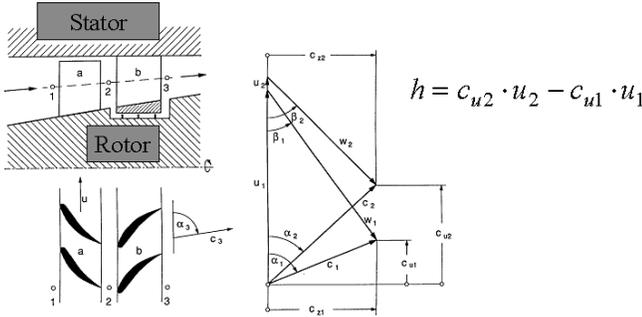


Figure 4. Velocities in a Typical Compressor Stage.

or, for axial flow machines, where the rotational speed u is about the same for inlet and exit of a stage:

$$\Delta h = u(c_{u2} - c_{u1}) \quad (9)$$

This correlation expresses the fact that the force on the rotating blade in direction of the rotation is proportional to the deflection of the flow in circumferential direction, i.e.:

$$F_u = W(c_{u1} - c_{u2}) \quad (10)$$

and therefore that power introduced into the flow is the angular velocity ω times the torque generated by F_u :

$$P = \omega r \cdot F_u = \omega r \cdot W(c_{u1} - c_{u2}) = W \cdot [u(c_{u1} - c_{u2})] = W \cdot \Delta h \quad (11)$$

A rotating compressor blade passage (stage) imparts energy on the fluid (air) by increasing the fluid's angular momentum (torque).

Mach Number

The aerodynamic behavior of a turbine or compressor is significantly influenced by the Mach number of the flow. The same turbine or compressor will show significant differences in operating range (flow range between stall and choke), pressure ratio, and efficiency.

The Mach number increases with increasing flow velocity, and decreasing temperature T . It also depends on the gas composition, which determines the ratio of specific heats γ and the gas constant R . To characterize the level of the Mach number of a turbomachine, the "machine" Mach number M_n is frequently used. M_n does not refer to a gas velocity, but to the circumferential speed u of a component, for example a blade tip at the diameter D :

$$M_n = \frac{u}{\sqrt{\gamma RT}} = \frac{2\pi DN}{\sqrt{\gamma RT}} \quad (12)$$

This points to the fact that the Mach number of the component in question will increase once the speed N is increased. The consequences for the operation of the gas turbine are that:

- The engine compressor Mach number depends on its speed, the ambient temperature, and the relative humidity.
- The gas generator turbine Mach number depends on its speed, the firing temperature, and the exhaust gas composition (thus, the load, the fuel, and the relative humidity).
- The power turbine Mach number depends on its speed, the power turbine inlet temperature, and the exhaust gas composition.

For a given geometry, the reference diameter will always be the same. Thus, the machine Mach number can be defined also in terms of a speed, for example, the gas generator speed, and get the so-called corrected gas generator speed:

$$N_{corr} = \frac{N}{\sqrt{T/T_{ref}}} \quad (13)$$

Even though N_{corr} is not dimensionless, it is a convenient way of writing the machine Mach number of the component. In the following text, the author will also use this simplified expression N/\sqrt{T} , which is based on the above explanations.

In a modern gas turbine, the compressor front stages are transonic, which means that the relative flow velocity into the rotor is higher than the speed of sound, while the flow velocity leaving the rotor is below the speed of sound (Figure 5). Turbine stages usually see subsonic inlet velocities, but the velocities within the blade channels can be locally supersonic (Figure 6).

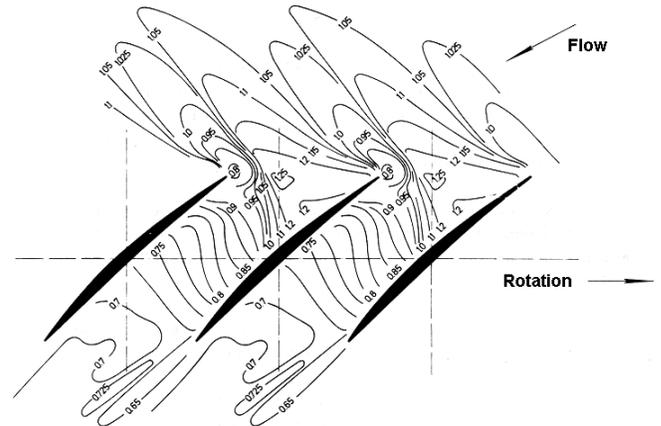


Figure 5. Mach Number Distribution for Typical Transonic Compressor Blades. The Flow Enters at Supersonic Speeds, and Is Decelerated to Subsonic Speeds at the Exit. (Courtesy Schodl, 1977)

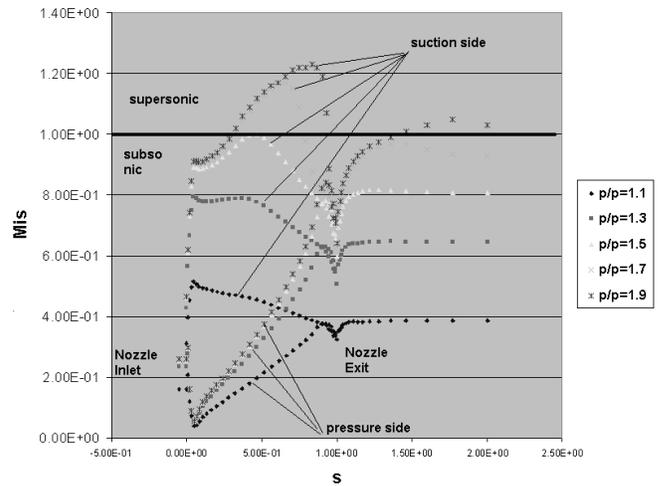


Figure 6. Velocity Distribution in a Turbine Nozzle at Different Pressure Ratios. As Soon as the Maximum Local Flow Velocity Exceeds Mach 1 (At a Pressure Ratio of 1.5 in this Example), the Inlet Flow Can No Longer Be Increased. (Courtesy Kurz, 1991)

Component performance maps show a significant sensitivity to changes in Mach numbers. There is a strong dependency of losses, enthalpy rise or decrease, and flow range for a given blade row on the characteristic Mach number. Figures 7 and 8, with the com-

pressor maps for typical gas turbine compressors, show in particular the narrowing of the operating range with an increase in Mach number, which in this case is due to increasing compressor speed N_{GP} (Cohen, et al., 1996).

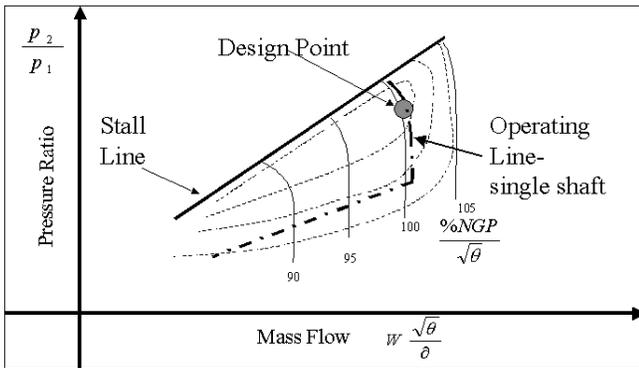


Figure 7. Typical Compressor Performance Map with Operating Lines for a Single Shaft Engine.

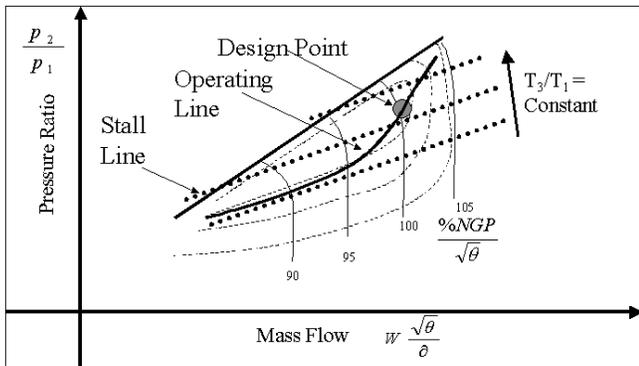


Figure 8. Typical Compressor Performance Map with Operating Lines for a Two Shaft Engine.

For turbine nozzles, one of the effects connected with the Mach number is the limit to the maximum flow that can pass through a nozzle. Beyond a certain pressure ratio, the amount of actual flow Q that can pass through the nozzle can no longer be increased by increasing the pressure ratio. As demonstrated with Figure 6, which shows the flow velocities in a turbine nozzle for increasing pressure ratios, the velocity or Mach number levels in the nozzle become higher and higher, until the speed of sound ($=$ Mach 1) is reached in the throat (for a pressure ratio of 1.7 in the example). A further increase of the pressure ratio yields higher velocities downstream of the throat, but the through-flow (which is proportional to the velocity at the inlet into the nozzle) can no longer be increased.

Because each gas turbine consists of several aerodynamic components, the Mach number of each of these components would have to be kept constant in order to achieve a similar operating condition for the overall machine. While the characteristic temperature for the engine compressor is the ambient temperature, the characteristic temperature for the gas generator turbine and the power turbine is the firing temperature T_3 and the power turbine inlet temperature T_5 , respectively. Therefore, if two operating points (op1 and op2) yield the same machine Mach numbers for the gas compressor and the gas generator turbine, and both operating points are at the respective optimum power turbine speed, then the thermal efficiencies for both operating points will be the same—as long as second order effects, such as Reynolds number variations, effects of gaps and clearances, etc., are not considered.

The requirement to maintain the machine Mach number for the compressor and gas generator turbine can be expressed by N_{GPcorr}

$=$ constant (which leads to identical Mach numbers for the compressor):

$$\frac{N_{GP,op1}}{\sqrt{T_{1,op1}}} = \frac{N_{GP,op2}}{\sqrt{T_{1,op2}}} \quad (14)$$

and, in order to maintain at the same time the same Mach number for the gas generator turbine, which rotates at the same speed as the compressor, required for the firing temperature is:

$$\frac{T_{3,op1}}{T_{1,op1}} = \frac{T_{3,op2}}{T_{1,op2}} \quad (15)$$

In this case, the fact that the volumetric flow through the turbine section is determined by the nozzle geometry also enforces (approximately) identical head and flow coefficients for compressor and turbine.

Therefore, the engine heat rate will remain constant, while the engine power will be changed proportional to the change in inlet density. This approach does not take effects like Reynolds number changes, changes in clearances with temperature, changes in gas characteristics, or the effect of accessory loads into account. This approach also finds its limitations in mechanical and temperature limits of an actual engine that restrict actual speeds and firing temperatures (Kurz, et al., 1999).

Reynolds Number

While the Mach number essentially accounts for the compressibility effects of the working gas, the Reynolds number describes the relative importance of friction effects. In industrial gas turbines, where neither the working temperatures nor the working pressures change as dramatically as in the operation of aircraft engines, the effects of changes in the Reynolds number are typically not very pronounced. A change in the ambient temperature from 0°F to 100°F changes the Reynolds number of the first compressor stage by about 40 percent. The typical operating Reynolds numbers of compressor blades and turbine blades are above the levels where the effect of changing the Reynolds number is significant.

Blade Cooling

The temperature in the hot section of gas turbines requires the cooling of nozzles and blades (as well as cooling for the combustor liner). Pressurized air from the engine compressor is brought to the blade and nozzle internals. In some designs, steam is used rather than air. There are a number of different ways the cooling is accomplished (Figure 9). The air is pushed through the inside of the blade with the goal of removing as much heat from the blade surface as possible. To this end, ribs are used to increase the turbulence and thus the heat transfer (convection cooling), and jets of air are blown through small holes to impinge on the blade inside (impingement cooling). Another design brings cold air from the inside of the blade through small holes to the outer blade surface, this generating a thin layer of cooler air between the blade surface and the hot gas (film cooling). The amount of air used impacts the performance of the gas turbine, because some (but not all) of the work to compress the air is lost if air is used for cooling purposes.

COMBUSTION

The engine combustor is the place where fuel is injected (through fuel injectors) into the air previously compressed in the engine compressor. The released fuel energy causes the temperature to rise:

$$\tilde{c}_{pe}(T_3 - T_2) = E_f W_f / W \quad (16)$$

The heat capacity \tilde{c}_{pe} in the equation above is a suitable average heat capacity. Modern combustors convert the energy stored in the

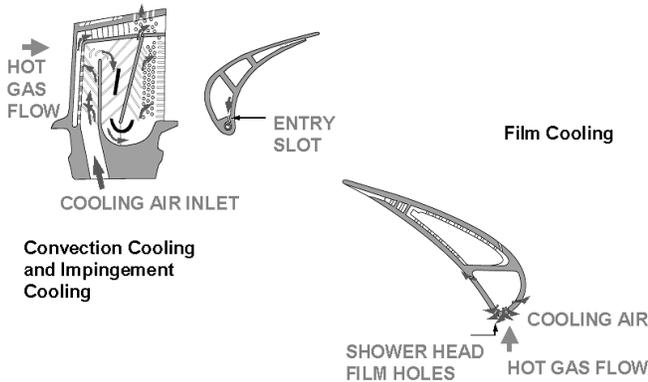


Figure 9. Typical Cooling Arrangements: Convection and Impingement Cooling (Left) and Film Cooling (Right).

fuel almost completely into heat (typical combustion efficiencies for natural gas burning engines range above 99.9 percent). This is also evident from the fact that the results of incomplete combustion, namely carbon monoxide (CO) and unburned hydrocarbons (UHC), are emitted only in the parts per million level.

Only some part of the compressed air participates directly in the combustion, while the remaining air is later mixed into the gas stream for cooling purposes. The temperature profile in a typical combustor is shown in Figure 10. The local temperatures are highest in the flame zone. Cooling of the combustor liner and subsequent addition of air bring the gas temperature down to an acceptable combustor exit temperature. The flow will also incur a pressure loss due to friction and mixing.

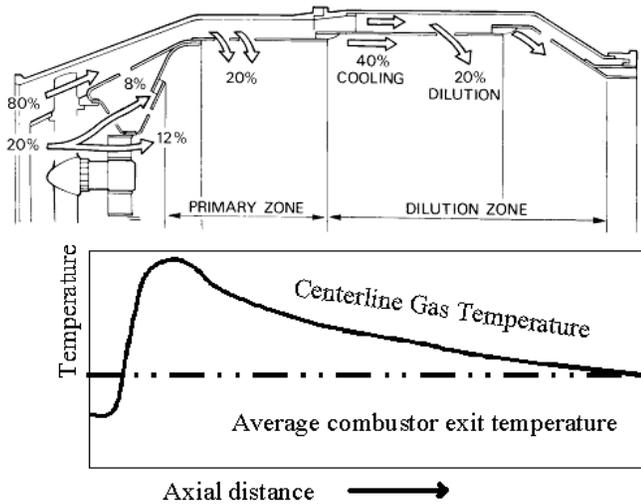


Figure 10. Axial Temperature Distribution in a Combustor.

Dry Low NOx

The general idea behind any dry low NOx (DLN) combustor currently in service is to generate a thoroughly mixed lean fuel and air mixture prior to entering the combustor of the gas turbine (Greenwood, 2000). The lean mixture is responsible for a low flame temperature, which in turn yields lower rates of nitrous oxide (NOx) production (Figure 11). Because the mixture is very lean, in fact fairly close to the lean extinction limit, the fuel-to-air ratio has to be kept constant within fairly narrow limits. This is also necessary due to another constraint: the lower combustion temperatures tend to lead to a higher amount of products related to incomplete combustion, such as CO and UHC. The necessity to control the fuel-to-air ratio closely yields different part-load behavior when comparing gas turbines with conventional combustors and DLN engines. (Note—Regarding the requirements for

DLN engines, multispool engines show no fundamental differences from single-spool engines.) At certain levels of part load, DLN engines usually bleed a certain amount of air from the compressor exit directly into the exhaust duct. Therefore, while the airflow for any two shaft engines is reduced at part load, the reduction in airflow is greater for a conventional combustion engine than for a DLN engine. This sounds paradoxical because the amount of air available at the combustor in part-load operation has to be less for a DLN engine (to maintain the fuel-to-air ratio) than for an engine with conventional combustion. However, due to the bleeding of air in a DLN engine, the flow capacity of the turbine section is artificially increased by the bleeding duct. The combustor exit temperature at part load drops significantly for engines with conventional combustion, while it stays high for DLN engines. Once the bleed valve opens, the part-load efficiency of a DLN engine drops faster than for an engine with conventional combustion. Since the opening of the bleed valve is driven by emissions considerations, it is not directly influenced by the load. Regarding emissions, the drop in combustor temperature in engines with conventional combustion, leading to a leaner fuel-to-air ratio, automatically leads to NOx emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel-to-air ratio. However, the NOx emissions levels of DLN engines are always lower than for engines with conventional combustion.

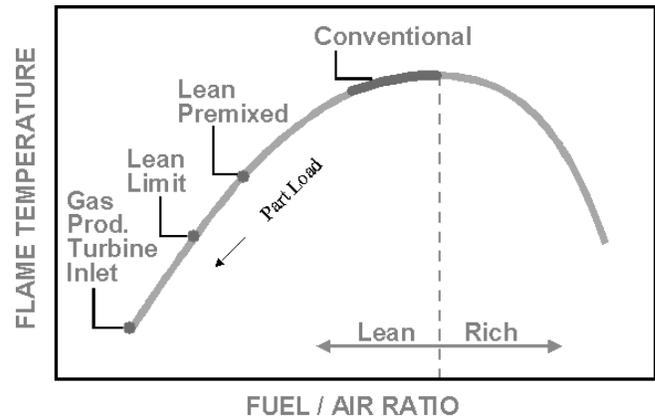


Figure 11. Flame Temperature as a Function to Fuel-to-Air Ratio.

The impact of design considerations on NOx emissions needs to be considered. Fortunately, there is no evidence that pressure ratio influences NOx production rate [on a parts per million (ppm) basis] in DLN systems. There might be some compromises necessary for engines with high firing temperatures regarding the cooling air usage. But this is a secondary effect at best, because the combustor exit temperature and the flame temperature are not directly related. Some aeroderivative engines, which tend to have high pressure ratios, have space limits for the combustion system, thus might be at a disadvantage. This is not primarily due to the high pressure ratio, but rather due to the specific design of the engine. A limiting factor for lowering NOx emissions is often driven by the onset of combustor oscillations. Again, there is no evidence that the operating windows that allow operation without oscillations are influenced by operating pressure or firing temperature. They rather seem to depend far more on the specific engine design.

**COMPONENT INTERACTION—
DESIGN AND OFF-DESIGN BEHAVIOR**

When the compressor, the gas generator turbine, and the power turbine (if applicable) are combined in a gas turbine, the operation of each component experiences certain operating constraints,

which are caused by the interaction between the components (Figure 12).

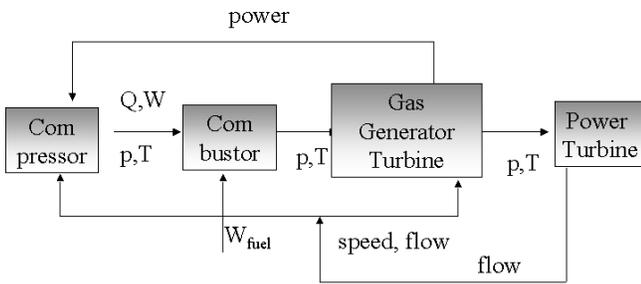


Figure 12. Interaction Between the Gas Turbine Components.

For example, the engine compressor will compress a certain mass flow, which in turn dictates the compressor discharge pressure necessary to force the mass flow through the turbine section. On the other hand, the gas generator turbine has to produce sufficient power to drive the generator. The firing temperature influences the power that the turbines can produce, but it also impacts the discharge pressure necessary from the compressor.

The components are designed to work together at their highest efficiencies at a design point, but the operation of the components at any other than the design point must also be considered. The constraints and requirements are different for single shaft and two shaft engines, hence they are treated separately. In the following section, the author will look into the interaction between the engine components, because it is this interaction that generates the typical behavior of gas turbines.

Single Shaft Engines

A single shaft engine consists of an air compressor, a combustor, and a turbine. The air compressor generates air at a high pressure, which is fed into the combustor where the fuel is burned. The combustion products and excess air leave the combustor at high pressure and high temperature. This gas is expanded in the gas generator turbine, which provides the power to turn the air compressor. The excess power is used to drive the load. Most single shaft turbines are used to drive electric generators at constant speeds. The author will not consider the rare case of single shaft turbines driving mechanical loads at varying speeds. The operation of the components requires the following compatibility conditions:

- Compressor speed = gas generator turbine speed
- Mass flow through turbine = mass flow through compressor – bleed flows + fuel mass flow
- Compressor power < turbine power

Typical compressor and turbine maps are shown in Figures 7 and 8, respectively.

The fact that the gas turbine operates at constant speed means any operating point of the engine compressor (for given ambient conditions) lies on a single speed line. Load increases are initiated by increasing the fuel flow, which in turn increases the firing temperature. Due to the fact that the first turbine nozzle is usually choked, the compressor operating point moves to a higher pressure ratio to compensate for the reduced density (from the higher firing temperature). The possible operating points of the compressor depending on the load running are also shown in the compressor maps (Figures 7 and 8).

In the case of the single shaft engine driving a generator, reduction in output power results in only minute changes in compressor mass flow as well as some reduction in compressor pressure ratio. A single shaft engine has no unique matching temperature. Used as a generator drive, it will operate at a single speed and can be temperature topped at any ambient temperature as long as the load is large enough.

Two Shaft Engines

A two shaft gas turbine (Figure 2) consists of an air compressor, a combustor, a gas generator turbine, and a power turbine. [Note—Some engines are configured as multispool engines. In this case, the gas generator has a low-pressure compressor driven by a low-pressure turbine and a high-pressure compressor driven by a high-pressure turbine. For this configuration, the shaft connecting the low-pressure (LP) compressor and turbine rotates inside the shaft connecting the high-pressure (HP) compressor and turbine. In general, all the operating characteristics described above also apply to these engines.] The air compressor generates air at a high pressure, which is fed into the combustor where the fuel is burned. The combustion products and excess air leave the combustor at high pressure and high temperature. This gas is expanded in the gas generator turbine, which has the sole task of providing power to turn the air compressor. After leaving the gas generator turbine, the gas still has a high pressure and a high temperature. It is now further expanded in the power turbine. The power turbine is connected to the driven equipment. It must be noted at this point that the power turbine (together with the driven equipment) can and will run at a speed that is independent of the speed of the gas generator portion of the gas turbine (i.e., the air compressor and the gas generator turbine).

The gas generator is controlled by the amount of fuel that is supplied to the combustor. Its two operating constraints are the firing temperature and the maximum gas generator speed (on some engines, torque limits may also constrain the operation at low ambient temperatures). If the fuel flow is increased, both firing temperature and gas generator speed increase, until one of the two operating limits is reached. Variable stator vanes at the engine compressor are frequently used, however, not for the purpose of controlling the airflow, but rather to optimize the gas producer speed. In two shaft engines, the airflow is controlled by the flow capacities of the gas generator turbine and power turbine nozzles.

Increasing the speed and temperature of the gas generator provides the power turbine with gas at a higher energy (i.e., higher pressure, higher temperature, and higher mass flow), which allows the power turbine to produce more power. If the power supplied by the power turbine is greater than the power absorbed by the load, the power turbine together with the driven compressor will accelerate until equilibrium is reached.

The operation of the components requires the following compatibility conditions:

- Compressor speed = gas generator turbine speed
- Mass flow through turbine = mass flow through compressor – bleed flows + fuel mass flow
- Compressor power = gas generator turbine power (– mechanical losses)
- The subsequent free power turbine adds the requirement that the pressure after the gas producer (GP) turbine has to be high enough to force the flow through the power turbine.

Typical compressor and turbine maps are shown in Figures 8 and 13, respectively. The gas generator for a two shaft engine adapts to different load requirements (and, accordingly, different fuel flow) by changing both speed and firing temperature. Note that the compressor operating points are very different between a single shaft and a two shaft engine.

Two shaft engines operate with the gas generator turbine and the power turbine in series. The power turbine pressure ratio p_5/p_a is thus related to the compressor pressure ratio p_2/p_a by the identity:

$$\frac{p_5}{p_a} = \frac{p_2}{p_a} \frac{p_3}{p_2} \frac{p_5}{p_3} \quad (17)$$

The pressure drop in the gas generator turbine p_5/p_3 and the pressure increase in the compressor p_2/p_a are related insofar as the

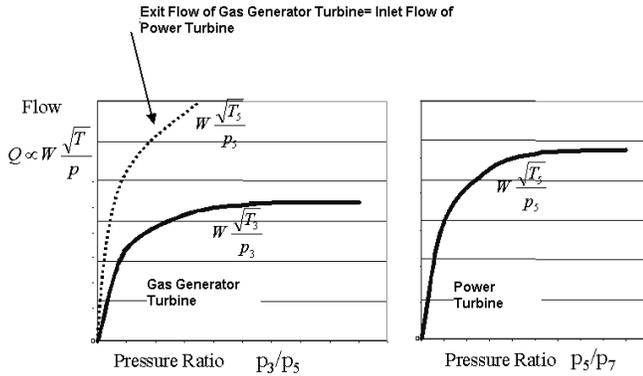


Figure 13. Schematic Turbine Performance Map for Two Turbines (Gas Generator and Power Turbine) in Series.

gas generator turbine has to provide enough power to drive the compressor.

The maximum possible pressure ratio p_5/p_a is controlled by the flow capacity Q_5 of the power turbine. In particular if the power turbine is choked, it will cause the gas generator turbine to operate at one fixed point (Figure 13). In many cases, both the gas generator turbine first-stage nozzle and the power turbine nozzle operate at or near choked flow conditions. In this case, the actual flow Q_3 through the gas generator turbine nozzle is practically constant. The mass flow is then only dependent on the combustor exit pressure p_3 , the firing temperature T_3 , the gas composition (which determines γ and thus the volume increase during the expansion), and the geometry of the nozzle, which determines the through-flow area (in reality, it is determined by the critical nozzle area, the clearance area, and the effective bleed valve area).

The above relationship has the following consequences:

- Increasing the firing temperature (without changing speed or geometry) will lead to a lower mass flow.
- Increasing the gas generator speed, thus increasing p_2 and p_3 , will allow for a larger mass flow.
- Pressure ratio, speed, and firing temperature are all related, and cannot be changed independently of each other. The turbine geometry determines both flow capacities Q_3 and Q_5 , as well as the gas generator turbine efficiency. The compressor geometry and speed set the airflow.

- With variable inlet guide vanes (IGVs) the airflow can be altered without changing the gas generator speed, thus also setting a new T_3 and a different compressor pressure ratio p_2/p_1 . The relationship between p_2/p_1 and T_3 remains, however, unchanged. The turbine flow capacities alone determine the gas generator match, not the IGV setting. Closing the IGVs will raise the speed of a temperature topped gas generator, but since the temperature remains constant, the airflow tends to remain unchanged (because the flow through the gas generator turbine nozzle Q_3 remains constant). If, however, η_{ggt} increases due to the change in speed, T_3 has to drop, leading to an increase in compressor mass flow. The gas generator pumps more airflow with the IGV closed and the higher speed than with the IGV open and the lower speed. The IGVs thus allow trimming of the engine such that the rated T_3 is always reached at full corrected gas generator speed (N_{GP}). Therefore, at high ambient temperatures, when the gas generator would normally slow down, IGVs can be used to keep the speed at a higher level, thus avoiding efficiency penalties in the gas generator turbine. Reducing the net power output in a two shaft engine involves a reduction in compressor speed and hence in air flow, pressure ratio, and temperature rise. From a comparison of the maps (Figure 14) one can see that the compressor in a two shaft engine operates for most of the load points close to its best efficiency.

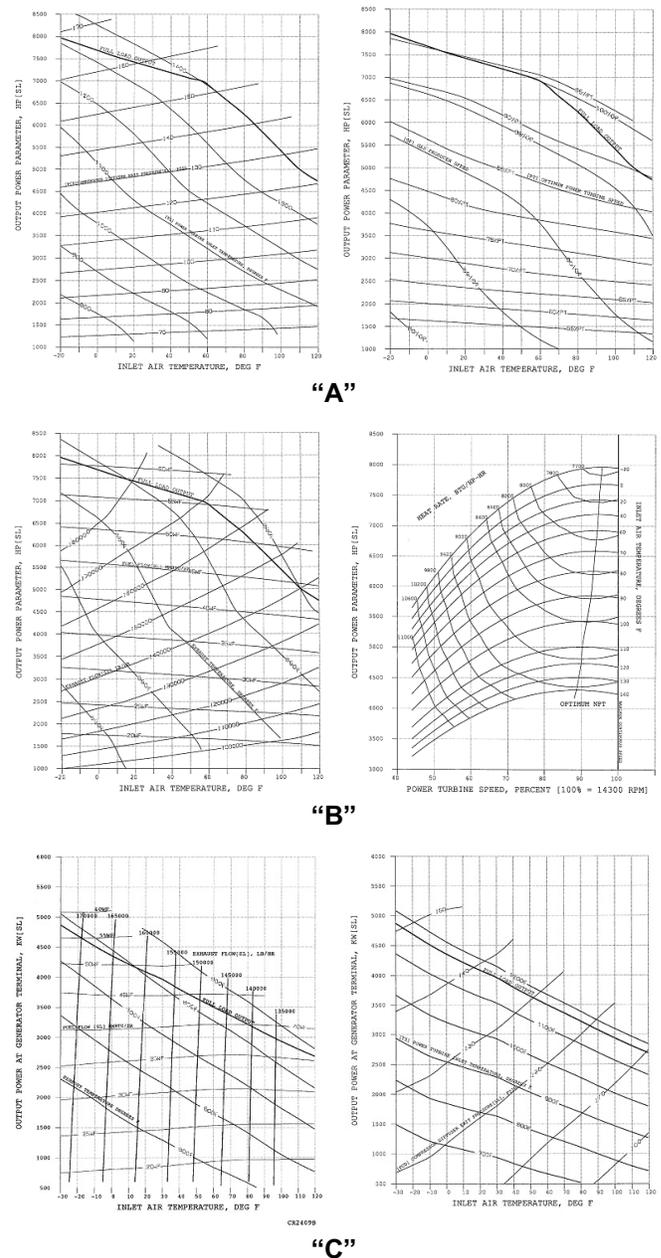


Figure 14. Gas Turbine Performance Maps—A and B: Two Shaft; C: Single Shaft.

Two shaft engines have a power turbine where the shaft is not mechanically coupled with the gas generator shaft. These components need to be “matched,” such that the overall performance of the gas turbine is optimized for a defined operating ambient temperature. The speed of the gas generator is therefore not controlled by the speed of the driven equipment (such as in single shaft generator set applications). The gas generator speed only depends on the load applied to the engine. If the power turbine output has to be increased, the fuel control valve allows more fuel to enter the combustor. This will lead to an increase both in gas generator speed and in firing temperature, thus making more power available at the power turbine.

The setting of the flow capacity of the power turbine has obviously a great influence on the possible operating points of the gas generator. For a high resistance of the power turbine (i.e., low mass flow W for a given p_5/p_1), the gas generator reaches its limiting firing temperature at lower ambient temperatures than with a low resistance of the power turbine. By altering the exit flow

angle of the first stage power turbine nozzle the required pressure ratio for a certain flow can be modified (i.e., the flow capacity). This effect is used to match the power turbine with the gas generator for different ambient temperatures.

Due to mechanical constraints, both the gas generator speed and the firing temperature have upper limits that cannot be exceeded without damaging the engine or reducing its life. Depending on:

- The ambient temperature,
- The accessory load, and
- The engine geometry (in particular the first power turbine nozzle),

the engine will reach one of the two limits first. At ambient temperatures below the match temperature, the engine will be operating at its maximum gas generator speed, but below its maximum firing temperature (speed topping). At ambient temperatures above the match temperature, the engine will operate at its maximum firing temperature, but not at its maximum gas generator speed (temperature topping). The match temperature is thus the ambient temperature at which the engine reaches both limits at the same time (Figure 14a and b).

Because the first power turbine nozzle determines the amount of pressure ratio needed by the power turbine to allow a certain gas flow it also determines the available pressure ratio for the gas generator turbine. If the pressure ratio available for the gas generator does not allow balancing of the power requirement of the engine compressor (refer to the enthalpy-entropy diagram), the gas generator will have to slow down, thus reducing the gas flow through the power turbine. This will reduce the pressure ratio necessary over the power turbine, thus leaving more head for the gas generator to satisfy the compressor power requirements.

Some effects can cause the gas turbine to exhibit an altered match temperature. Gas fuel with a low heating value or water injection increase the mass flow through the turbine relative to the compressor mass flow. The temperature topping will thus be shifted to higher ambient temperatures. Dual fuel engines that are matched on gas will top early on liquid fuel. This is caused by the change in the thermodynamic properties of the combustion product due to the different carbon to hydrogen ratio of the fuels. The matching equations indicate that a reduction in compressor efficiency (due to fouling, inlet distortions) or turbine efficiency (increased tip clearance, excessive internal leaks, corrosion) will also cause early topping. Accessory loads also have the effect of leading to earlier topping.

Single Shaft Versus Two Shaft Engines

The choice of whether to use a single shaft or two shaft power plant is largely determined by the characteristics of the driven load. If the load speed is constant, as in the case of an electric generator, a single shaft unit is often specified; an engine specifically designed for electric power generation would make use of a single shaft configuration. An alternative, however, is the use of a two shaft engine. If the load needs to be driven with varying speeds (compressors, pumps), two shaft engines are advantageous.

The two types have different characteristics regarding the supply of exhaust heat to a cogeneration or combined cycle plant, primarily due to the differences in exhaust flow as load is reduced; the essentially constant air flow and compressor power in a single shaft unit results in a larger decrease of exhaust temperature for a given reduction in power, which might necessitate the burning of supplementary fuel in the waste heat boiler under operating conditions where it would be unnecessary with a two shaft. In both cases, the exhaust temperature may be increased by the use of variable inlet guide vanes. Cogeneration systems have been successfully built using both single shaft and two shaft units.

The torque characteristics are very different, and the variation of torque with output speed at a given power may well determine the engine's suitability for certain applications. The compressor of a

single shaft engine is constrained to turn at some multiple of the load speed, fixed by the transmission gear ratio, so that a reduction in load speed implies a reduction in compressor speed. This results in a reduction in mass flow hence of output of torque. This type of turbine is only of limited use for mechanical drive purposes. The two shaft unit, having a free power turbine, however, has a very favorable torque characteristic. For a constant fuel flow, and constant gas generator speed, the free power turbine can provide relatively constant power for a wide speed range. This is due to the fact that the compressor can supply an essentially constant flow at a given compressor speed, irrespective of the free turbine speed. Also, at fixed gas generator operating conditions, reduction in output speed results in an increase in torque. It is quite possible to obtain a stall torque of twice the torque delivered at full speed.

The actual range of speed over which the torque conversion is efficient depends on the efficiency characteristic of the power turbine. The typical turbine efficiency characteristic shown in Figure 14 suggests that the efficiency penalty will not be greater than about 5 or 6 percent over a speed range from half to full speed.

Load

Any gas turbine will experience a reduced efficiency at part load (Figure 15). The reduction in efficiency with part load differs from design to design. In particular, DLN engines show different part-load efficiencies than their conventional combustion counterparts. The necessity to control the fuel to air ratio closely yields different part load behavior when comparing gas turbines with DLN combustors and with conventional combustors. A typical way of controlling these engines is by controlling the airflow into the combustor, thus keeping the combustor primary zone temperature within narrow limits. The part load behavior of single shaft and two shaft, standard combustion, and dry low NOx concepts are fundamentally different. This is both due to the different aerodynamic configuration and the requirements of keeping the fuel to air ratio within a narrow window for DLN engines.

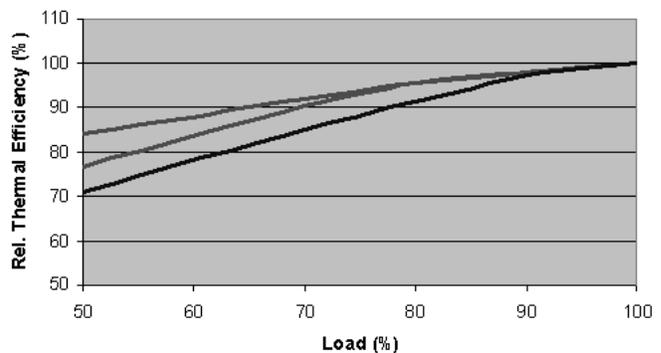


Figure 15. Thermal Efficiency of Typical Industrial Gas Turbines as a Function of Load.

To operate the engine at part load, the fuel flow is adjusted, until some control parameter (for example the flow through the driven compressor, or a certain kW value for a generator) is satisfied. Other adjustments, such as guide vane settings or bypassing combustion air may be necessary.

For a single shaft engine, which has to operate at constant gas generator speed (to keep the generator frequency constant), this means that the firing temperature will be changed with load. A governor will keep the speed constant and will increase the fuel flow with increasing load, thus increasing the firing temperature, until the control limit is reached. Due to the constant speed, the airflow through the engine will not vary greatly between full load and part load. This means that the fuel to air ratio drops significantly at part load and the combustor exit temperature drops significantly from full load to part load. Therefore, most single shaft dry low NOx engines use variable stator vanes on the engine

compressor to vary the airflow, and thus keep the fuel to air ratio relatively constant.

For a given mass flow, any increase in firing temperature would increase the volume flow through the turbine section. Therefore the pressure supplied from the compressor has to be increased, because the turbine nozzle is at or near choked conditions. Since the compressor also operates at constant speed, the result is a reduction of mass flow until equilibrium is reached. A typical performance map for a single shaft engine shows this increase in compressor discharge pressure at increased load. If the engine is equipped with variable inlet guide vanes (VIGVs) to keep the fuel to air ratio in the combustor constant, a reduction in load will require a closing of the VIGVs to reduce the airflow. Closing the VIGVs also reduces the pressure ratio of the compressor at constant speed.

For single shaft engines as described above, the part load efficiencies of gas turbines with DLN combustion and conventional combustion are very similar. The need to bleed combustion air for two shaft engines typically leads to a lower part load efficiency for engines equipped with DLN combustors.

For a two shaft engine, both gas generator speed and firing temperature change with load. An increase in load at the power turbine will cause the fuel flow to increase. Because the gas generator is not mechanically coupled with the power turbine, it will accelerate, thus increasing airflow, compressor discharge pressure, and mass flow. The increase in gas generator speed means that the compressor now operates at a higher Mach number. At the same time the increased fuel flow will also increase the firing temperature. The relative increase is governed by the fact that the power turbine requires a certain pressure ratio to allow a given amount of airflow pass. This forces an equilibrium where the following requirements have to be met:

- The compressor power equals gas generator turbine power. This determines the available pressure upstream of the power turbine (PT).
- The available pressure ratio at the power turbine is sufficient to allow the airflow to be forced through the power turbine.

Depending on the ambient temperature relative to the engine match temperature, the fuel flow into the engine will either be limited by reaching the maximum firing temperature or the maximum gas generator speed. The ambient temperature, where both control limits are reached at the same time, is called engine match temperature.

Variable stator vanes at the engine compressor are frequently used, however, not for the purpose of controlling the airflow. This is due to the fact that in two shaft engines, the airflow is controlled by the flow capacities of the gas generator turbine and power turbine nozzles. To control the fuel to air ratio in the combustor, another control feature has to be added for two shaft engines with dry low Nox combustors. Usually, a certain amount of air is bled from the compressor exit directly into the exhaust duct. This leads to the fact that while the airflow for two shaft engines is reduced at part load, the reduction in airflow is larger for an engine with a standard combustion system. Like in single shaft engines, the combustor exit temperature at part load drops significantly for engines with standard combustion, while it stays relatively high for DLN engines. The drop in combustor temperature in engines with standard combustion, which indicates the leaner fuel to air ratio, automatically leads to NOx emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction, because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel to air ratio.

Power Turbine Speed

For any operating condition of the gas generator, there is an optimum power turbine speed at which the power turbine operates at its highest efficiency, and thus produces the highest amount of

power for a given gas generator operating point. Aerodynamically, this optimum point is characterized by a certain ratio of actual flow Q_5 over rotating speed N_{PT} . The volumetric flow depends on the ambient temperature and the load. This explains why the optimum power turbine speed is a function of ambient temperature and load.

If the power turbine does not operate at the optimum power turbine speed, the power output and the efficiency of the power turbine will be lower (Figure 14b). The impact of changing the power turbine speed is easily described by:

$$\frac{P}{P_{opt}} = 2 \cdot \frac{N_{PT}}{N_{PT,opt}} - \left(\frac{N_{PT}}{N_{PT,opt}} \right)^2 \quad (18)$$

This equation can be derived from basic relationships (Brun and Kurz, 2001) and is pretty accurate for any arbitrary power turbine.

When using this relationship, it must be considered that the optimum power turbine speed ($N_{pt,opt}$) depends on the gas generator load and the ambient temperature (Kurz and Brun, 2001). In general, the optimum power turbine speed is reduced for increasing ambient temperatures and lower load (Figure 14b). The heat rate becomes for a constant gas generator operating point:

$$\frac{HR}{HR_{opt}} = \frac{P_{opt}}{P} \quad (19)$$

Off-optimum speed of the power turbine reduces the efficiency and the ability to extract head from the flow. Even if N_{GG} (and the fuel flow) does not change, the amount of power that is produced by the PT is reduced. Also, because of the unchanged fuel flow, the engine heat rate increases and the exhaust temperature increases accordingly. Theoretically, any engine would reach its maximum exhaust temperature at high ambient, full load, and locked PT.

Another interesting result of the above is the torque behavior of the power turbine, considering that torque is power divided by speed:

$$\frac{\tau}{\tau_{opt}} = 2 - \left(\frac{N_{PT}}{N_{PT,opt}} \right) \quad (20)$$

The torque is thus a linear function of the speed, with the maximum torque at the lowest speed. This explains one of the great attractions of a free power turbine: to provide the necessary torque to start the driven equipment is usually not difficult (compared to electric motor drives or reciprocating engines) because the highest torque is already available at low speeds of the power turbine.

Influence of Emission Control Technologies

All emission control technologies that use lean-premix combustion require a precise management of the fuel to air ratio in the primary zone of the combustor (i.e. where the initial combustion takes place) as well as a precise distribution of combustor liner cooling and dilution flows. Deviations in these areas can lead to increased NOx production, higher CO or UHC levels, or flameout (Greenwood, 2000).

Lean-premix combustion achieves reduction in NOx emissions by lowering the flame temperature. The flame temperature is determined by the fuel to air ratio in the combustion zone. A stoichiometric fuel to air ratio (such as in conventional combustors) leads to high flame temperatures, while a lean fuel to air ratio can lower the flame temperature significantly. However, a lean fuel to air mixture also means that the combustor is operating close to the lean flameout limit.

Any part load operation will cause reduction of fuel to air ratio, because the reduction in air flow is smaller than the reduction in fuel flow. Several different approaches to control the fuel to air ratio are possible to avoid flameout at part load or transient situations, for example:

- Bleeding air overboard,
- Using variable inlet guide vanes, and
- Managing the ratio between fuel burned in lean premix mode and in a diffusion flame,

to name a few. Obviously, all these approaches can have an effect on the part load performance characteristics of the gas turbine.

Variable Inlet and Stator Vanes

Many modern gas turbines use variable inlet guide vanes and variable stator vanes in the engine compressor. Adjustable vanes allow altering of the stage characteristics of compressor stages (refer to the explanation on Euler equations in the previous section, "A QUICK EXCURSION TO AERODYNAMICS") because they change the head making capability of the stage by increasing or reducing the preswirl contribution. This means that for a prescribed pressure ratio they also alter the flow through the compressor. It is therefore possible to change the flow through the compressor without altering its speed. There are three important applications:

1. During startup of the engine it is possible to keep the compressor from operating in surge.
2. The airflow can be controlled to maintain a constant fuel to air ratio in the combustor for dry low NOx applications on single shaft machines.
3. Two shaft engines can be kept from dropping in gas generator speed at ambient temperatures higher than the match temperature, i.e., the gas generator turbine will continue to operate at its highest efficiency.

Accessory Loads

Accessory loads are due to mechanically driven lube oil or hydraulic pumps. While the accessory load can be treated fairly easily in a single shaft engine—its power requirement is subtracted from the gross engine output—this is somewhat more complicated in a two shaft machine.

In a two shaft gas turbine, the accessory load is typically taken from the gas generator. In order to satisfy the equilibrium conditions the gas generator will have to run hotter than without the load. This could lead to more power output at conditions that are not temperature limited. When the firing temperature is limited (i.e., for ambient temperatures above the match point), the power output will fall off more rapidly than without the load. That means that an accessory load of 50 hp may lead to power losses at the power turbine of 100 or more hp at higher ambient temperatures. The heat rate will increase due to accessory loads at all ambient temperatures. The net effect of accessory loads can also be described as a move of the match point to lower ambient temperatures.

Control Temperature

One of the two operating limits of a gas turbine is the turbine rotor inlet temperature (TRIT or T_3). Unfortunately, it is not possible to measure this temperature directly—a temperature probe would only last for a few hours at temperatures that high. Therefore, the inlet temperature into the power turbine (T_5) is measured instead. The ratio between T_3 and T_5 is determined during the factory test, where T_5 is measured and T_3 is determined from a thermodynamic energy balance. This energy balance requires the accurate determination of output power and air flow, and can therefore be performed best during the factory test.

It must be noted that both T_3 and T_5 are circumferentially and radially very nonuniformly distributed in the reference planes (i.e., at the combustor exit, at the rotor inlet, at the power turbine inlet). Performance calculations use a thermodynamic average temperature. This is not exactly the temperature one would measure as the average of a number of circumferentially distributed temperature probes.

Rather than controlling T_3 the control system limits engine operations to the T_5 that corresponds to the rated T_3 . However, the ratio between T_3 and T_5 is not always constant, but varies with the ambient temperature; the ratio T_3/T_5 is reduced at higher ambient temperatures. Modern control algorithms can take this into account.

Engines can also be controlled by their exhaust temperature (T_7). For single shaft engines, measuring T_7 or T_5 are equivalent choices. For two shaft engines, measuring T_7 instead of T_5 adds the complication that the T_7 control temperature additionally depends on the power turbine speed, while the relationship between T_3 and T_5 does not depend on the power turbine speed.

INFLUENCE OF AMBIENT CONDITIONS

Ambient Temperature

Changes in ambient temperature have an impact on full-load power and heat rate, but also on part-load performance and optimum power turbine speed (Figure 14). Manufacturers typically provide performance maps that describe these relationships for International Organization for Standardization (ISO) conditions. These curves are the result of the interaction between the various rotating components and the control system. This is particularly true for DLN engines.

If the ambient temperature changes, the engine is subject to the following effects:

- The air density changes. Increased ambient temperature lowers the density of the inlet air, thus reducing the mass flow through the turbine, and therefore reduces the power output (which is proportional to the mass flow) even further. At constant speed, where the volume flow remains approximately constant, the mass flow will increase with decreasing temperature and will decrease with increasing temperature.
- The pressure ratio of the compressor at constant speed gets smaller with increasing temperature. This can be determined from a Mollier diagram, showing that the higher the inlet temperature is, the more work (or head) is required to achieve a certain pressure rise. The increased work has to be provided by the gas generator turbine, and is thus lost for the power turbine, as can be seen in the enthalpy-entropy diagram.

At the same time $N_{G\text{corr}}$ (i.e., the machine Mach number) at constant speed is reduced at higher ambient temperature. As explained previously, the inlet Mach number of the engine compressor will increase for a given speed, if the ambient temperature is reduced. The gas generator Mach number will increase for reduced firing temperature at constant gas generator speed.

The enthalpy-entropy diagram (Figure 3) describes the Brayton cycle for a two shaft gas turbine. Lines 1-2 and 3-4 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 4-5 describes the work output of the power turbine. At higher ambient temperatures, the starting point 1 moves to a higher temperature. Because the head produced by the compressor is proportional to the speed squared, it will not change if the speed remains the same. However, the pressure ratio produced, and thus the discharge pressure, will be lower than before. Looking at the combustion process 2-3, with a higher compressor discharge temperature and considering that the firing temperature T_3 is limited, one sees that less heat input is possible, i.e., less fuel will be consumed. The expansion process has, due to the lower $p_2 = p_3$, less pressure ratio available, or a larger part of the available expansion work is being used up in the gas generator turbine, leaving less work available for the power turbine.

On two shaft engines, a reduction in gas generator speed occurs at high ambient temperatures. This is due to the fact that the equilibrium condition between the power requirement of the compressor (which increases at high ambient temperatures if the pressure ratio must be maintained) and the power production by

the gas generator turbine (which is not directly influenced by the ambient temperature as long as compressor discharge pressure and firing temperature remain) will be satisfied at a lower speed.

The lower speed often leads to a reduction of turbine efficiency. The inlet volumetric flow into the gas generator turbine is determined by the first stage turbine nozzle, and the Q_3/N_{GG} ratio (i.e., the operating point of the gas generator turbine) therefore moves away from the optimum. Variable compressor guide vanes allow keeping the gas generator speed constant at higher ambient temperatures, thus avoiding efficiency penalties.

In a single shaft, constant speed gas turbine one would see a constant head (because the head stays roughly constant for a constant compressor speed), and thus a reduced pressure ratio. Because the flow capacity of the turbine section determines the pressure-flow-firing temperature relationship, an equilibrium will be found at a lower flow and a lower pressure ratio, thus a reduced power output.

- The compressor discharge temperature at constant speed increases with increasing temperature. Thus, the amount of heat that can be added to the gas at a given maximum firing temperature is reduced.
- The relevant Reynolds number changes.

At full load, single shaft engines will run at temperature topping at all ambient temperatures, while two shaft engines will run either at temperature topping (at ambient temperatures higher than the match temperature) or at speed topping (at ambient temperatures lower than the match temperature). At speed topping, the engine will not reach its full firing temperature, while at temperature topping, the engine will not reach its maximum speed.

The net effect of higher ambient temperatures is an increase in heat rate and a reduction in power. The impact of ambient temperature is usually less pronounced for the heat rate than for the power output, because changes in the ambient temperature impact less the component efficiencies than the overall cycle output.

Inlet and Exhaust Pressure Losses

Any gas turbine needs an inlet and exhaust system to operate. The inlet system consists of one or several filtration systems, a silencer, ducting, and possibly deicing, fogging, evaporative cooling, and other systems. The exhaust system may include a silencer, ducting, and waste heat recovery systems. All these systems will cause pressure drops, i.e., the engine will actually see an inlet pressure that is lower than ambient pressure and will exhaust against a pressure that is higher than the ambient pressure. These inevitable pressure losses in the inlet and exhaust system cause a reduction in power and cycle efficiency of the engine. The reduction in power, compared to an engine at ISO conditions, can be described by simple correction curves, which are usually supplied by the manufacturer. The ones shown in Figure 16 describe the power reduction for every inch (or millimeter) of water pressure loss. These curves can be easily approximated by second order polynomials. The impact on heat rate is easily calculated by taking the fuel flow from ISO conditions and dividing it by the reduced power.

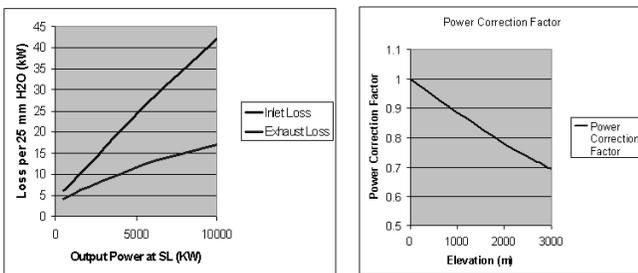


Figure 16. Correction Factors for Inlet Losses, Exhaust Losses, and Site Elevation.

Ambient Pressure

The impact of operating the engine at lower ambient pressures (for example, due to site elevation or simply due to changing atmospheric conditions) is that of a reduced air density (Figures 16 and 17). The engine, thus, sees a lower mass flow (while the volumetric flow is unchanged). The changed density only impacts the power output, but not the efficiency of the engine. However, if the engine drives accessory equipment through the gas generator, this is no longer true because the ratio between gas generator work and required accessory power (which is independent of changes in the ambient conditions) is affected.

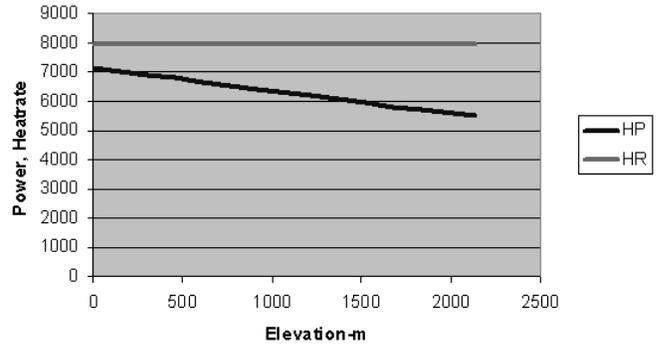


Figure 17. Power and Heat Rate as a Function of Site Elevation (Typical).

The impact is universal for any engine, except for the result of some secondary effects such as accessory loads. If the ambient pressure is known, the performance correction can be easily accomplished by:

$$\delta = \frac{P_{ambient} (in - " Hg)}{29.929" Hg} \tag{21}$$

If only the site elevation is known, the ambient pressure at normal conditions is:

$$P_{ambient} = P_{sealevel} \cdot e^{-\frac{elevation(ft)}{27200}} \tag{22}$$

Fuel

While the influence of the fuel composition on performance is rather complex, fortunately the effect on performance is rather small if the fuel is natural gas. Fuel gas with a large amount of inert components [such as carbon dioxide (CO₂) or nitrogen (N₂)] have a low Wobbe index, while substances with a large amount of heavier hydrocarbons have a high Wobbe index. Pure methane has a Wobbe index of about 1220.

In general, engines will provide slightly more power if the Wobbe index:

$$WI = \frac{LHV}{\sqrt{SG}} \tag{23}$$

is reduced. This is due to the fact that the amount of fuel mass flow increases for a given amount of fuel energy when the Wobbe index is reduced. This increases the mass flow through the turbine section, which increases the output of the turbine. This effect is to some degree counteracted by the fact that the compressor pressure ratio increases to push the additional flow through the flow restricted turbine. In order to do this, the compressor will absorb somewhat more power. The compressor will also operate closer to its stall margin. The above is valid irrespective of whether the engine is a two shaft or single shaft engine.

The fuel gas pressure at skid edge has to be high enough to overcome all pressure losses in the fuel system and the combustor

pressure, which is roughly equal to the compressor discharge pressure p_2 . The compressor discharge pressure at full load changes with the ambient temperature, and, therefore, a fuel gas pressure that is too low for the engine to reach full load at low ambient temperature may be sufficient if the ambient temperature increases.

If the fuel supply pressure is not sufficient, single and two shaft engines show distinctly different behavior, namely:

- A two shaft engine will run slower, such that the pressure in the combustor can be overcome by the fuel pressure (Figure 18). If the driven equipment is a gas compressor (and the process gas can be used as fuel gas), “bootstrapping” is often possible. The fuel gas is supplied from the gas compressor discharge side. If the initial fuel pressure is sufficient to start the engine and to operate the gas compressor, the gas compressor will increase the fuel gas pressure. Thus the engine can produce more power, which in turn will allow the gas compressor to increase the fuel pressure even more, until the fuel gas pressure necessary for full load is available.

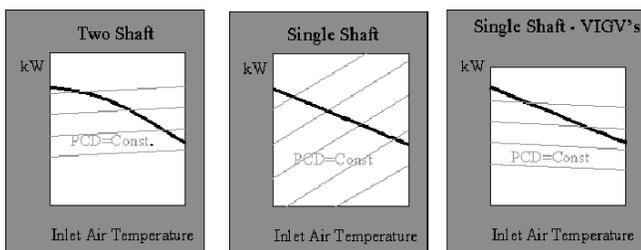


Figure 18. Effect of Low Fuel Gas Pressure on Different Engine Designs. PCD Is the Pressure in the Combustor.

A single shaft engine, which has to run at constant speed, will experience a severe reduction in possible firing temperature and significant loss in power output, unless it uses VIGVs. With VIGVs, the compressor exit pressure, and thus the combustor pressure can also be influenced by the position of the VIGVs, thus leading to less power loss (Figure 18).

Without VIGVs, the only way to reduce gas generator compressor discharge pressure (PCD) is by moving the operating point of the compressor on its map. This can be done by reducing the back pressure from the turbine, which requires a reduction in volume flow. Since the speed is fixed, only a reduction in firing temperature—which reduces the volume flow through the gas generator if everything else remains unchanged—can achieve this. A reduced volume flow will reduce the pressure drop required for the gas generator turbine.

Industrial gas turbines allow operation with a wide variety of gaseous and liquid fuels. To determine the suitability for operation with a gas fuel system, various physical parameters of the proposed fuel need to be determined: heating value, dew point, Joule-Thompson coefficient, Wobbe index, and others (Elliott, et al., 2004). However, fuel borne contaminants can also cause engine degradation. Special attention should be given to the problem of determining the dew point of the potential fuel gas at various pressure levels. In particular the treatment of heavier hydrocarbons and water must be addressed. Since any fuel gas system causes pressure drops in the fuel gas, the temperature reduction due to the Joule-Thompson effect has to be considered and quantified (Kurz, et al., 2004).

Gas fuels for gas turbines are combustible gases or mixtures of combustible and inert gases with a variety of compositions covering a wide range of heating values and densities. The combustible components can consist of methane and other low molecular weight hydrocarbons, hydrogen, and carbon monoxide. The major inert components are nitrogen, carbon dioxide, and water vapor. It is generally accepted that this type of fuel has to be completely gaseous at the entry to the fuel gas system and at all points downstream to the fuel nozzle (ASME B133, 1992).

Gaseous fuels can vary from poor quality wellhead gas to high quality consumer or “pipeline” gas. In many systems, the gas composition and quality may be subject to variations (Newbound and Wagiealla, 2003). Typically, the major sources of contaminants within these fuels are:

- Solids.
- Water.
- Heavy gases present as liquids.
- Oils typical of compressor oils.
- Hydrogen sulfide (H_2S).
- Hydrogen (H_2).
- Carbon monoxide (CO).
- Carbon dioxide (CO_2).
- Siloxanes.

Other factors that will affect turbine or combustion system life and performance include lower heating value (LHV), specific gravity (SG), fuel temperature, and ambient temperature.

Some of these issues may coexist and be interrelated. For instance, water, heavy gases present as liquids, and leakage of machinery lubricating oils may be a problem for turbine operators at the end of a distribution or branch line, or at a low point in a fuel supply line.

Water in the gas may combine with other small molecules to produce a hydrate—a solid with an ice-like appearance. Hydrate production is influenced, in turn, by gas composition, gas temperature, gas pressure, and pressure drops in the gas fuel system. Liquid water in the presence of hydrogen sulfide (H_2S) or CO_2 will form acids that can attack fuel supply lines and components. Free water can also cause turbine flameouts or operating instability if ingested in the combustor or fuel control components.

Heavy hydrocarbon gases present as liquids provide many times the heating value per unit volume than they would as a gas. Since turbine fuel systems meter the fuel based on the fuel being a gas, this creates a safety problem, especially during the engine startup sequence when the supply line to the turbine still may be cold. Hydrocarbon liquids can cause:

- Turbine overfueling, which can cause an explosion or severe turbine damage.
- Fuel control stability problems, because the system gain will vary as liquid slugs or droplets move through the control system.
- Combustor hot streaks and subsequent engine hot section damage.
- Overfueling the bottom section of the combustor when liquids gravitate toward the bottom of the manifold.
- Internal injector blockage over time, when trapped liquids pyrolyze in the hot gas passages.

Liquid carryover is a known cause for rapid degradation of the hot gas path components in a turbine (Meher-Homji and Gabriles, 1998).

The condition of the combustor components also has a strong influence, and fuel nozzles that have accumulated pipeline contaminants that block internal passageways will probably be more likely to miss desired performance or emission targets. Thus, it follows that more maintenance attention may be necessary to assure that combustion components are in premium condition. This may require that fuel nozzles be inspected and cleaned at more regular intervals or that improved fuel filtration components be installed.

Relative Humidity

The impact on engine performance would be better described by the water content of the air (say, in mole percent). Since the water

concentration in the air for the same relative humidity increases with increasing temperature, the effects on engine performance are negligible for low ambient temperatures and fairly small (in the range of 1 or 2 percent) even at high temperatures of 38°C (100°F). Since the water content changes the thermodynamic properties of air (such as density and heat capacity), it causes a variety of changes in the engine, such that on some engines performance is increased with increased humidity, while other engines show reduced performance at increased humidity.

WHAT DO TYPICAL MAPS SHOW?

Because the gas turbine performance varies significantly from one design to the other, the procedure to determine the performance of the engine for a specified operating point is to use the manufacturer's performance maps.

Typical engine performance maps are shown in Figure 14c for single shaft engines and in Figure 14a and b for two shaft engines. In general, these maps can be used to determine the engine full load output at a given ambient temperature and a given power turbine speed. They also show the fuel flow at any load, as well as exhaust flow and temperature. Additional maps allow correction for inlet and exhaust losses as well as for the site elevation. For diagnostic purposes, the maps also allow determination of the expected compressor discharge pressure, control temperature (typically power turbine inlet temperature or exhaust temperature), and gas generator speed at any operating point. Discrepancies between the expected and the actual values may be indicative of engine problems.

In order to fully understand the information displayed on engine performance maps, one wants to determine what the reason is for an engine to behave the way it does.

API 616 (1998) prescribes another form of representing engine performance than described above. The map for single shaft engines in API 616 (1998) is not particularly useful for single shaft engines driving generators, because it shows the performance as a function of gas generator speed. For these generator set applications, however, the gas generator speed is always constant. The API 616 maps used to represent two shaft engines do not allow description of the engine performance at varying ambient temperatures. Also, the control temperature for two shaft engines is usually not the exhaust temperature (as postulated in one of the API 616 curves), but the power turbine inlet temperature. The most useful curve in API 616 is essentially a subset of the power turbine curve in Figure 14b.

It should be noted that, particularly in the field, the measurement of power output, heat rate, exhaust flow, and exhaust temperature are usually rather difficult (Brun and Kurz, 2001). Understanding the operating principles of the engine is therefore a useful tool of interpreting data.

PERFORMANCE DEGRADATION

Any prime mover exhibits the effects of wear and tear over time. The problem of predicting the effects of wear and tear on the performance of any engine is still a matter of discussion. Because the function of a gas turbine is the result of the finetuned cooperation of many different components, the gas turbine has to be treated as a system, rather than as isolated components (Kurz and Brun, 2001).

Treating the gas turbine package as a system reveals the effects of degradation on the match of the components as well as on the match with the driven equipment. The mechanisms that cause engine degradation, are:

- Changes in blade surfaces due to erosion or fouling, and the effect on the blade aerodynamics.
- Changes in seal geometries and clearances, and the effect on parasitic flows.
- Changes in the combustion system (e.g., which result in different pattern factors).

The function of a gas turbine is the result of the finetuned cooperation of many different components. Any of these parts can show wear and tear over the lifetime of the package, and thus can adversely affect the operation of the system. In particular the aerodynamic components, such as the engine compressor, the turbines, the driven pump or compressor have to operate in an environment that will invariably degrade their performance. The understanding of the mechanisms that cause degradation, as well as the effects that the degradation of certain components can cause for the overall system, are a matter of interest.

Several mechanisms cause the degradation of engines:

- Fouling is caused by the adherence of particles to airfoils and annulus surfaces. The adherence is caused by oil or water mists. The result is a buildup of material that causes increased surface roughness and to some degree changes the shape of the airfoil (if the material buildup forms thicker layers of deposits). Many of the contaminants are smaller than 2 μm . Fouling can normally be eliminated by cleaning.
- Hot corrosion is the loss of material from flow path components caused by chemical reactions between the component and certain contaminants, such as salts, mineral acids, or reactive gases. The products of these chemical reactions may adhere to the aerocomponents as scale. High temperature oxidation, on the other hand, is the chemical reaction between the component's metal atoms and oxygen from the surrounding hot gaseous environment. The protection through an oxide scale will in turn be reduced by any mechanical damage such as cracking or spalling, for example, during thermal cycles.
- Erosion is the abrasive removal of material from the flow path by hard particles impinging on flow surfaces. These particles typically have to be larger than 20 μm in diameter to cause erosion by impact. Erosion is probably more a problem for aeroengine applications, because state-of-the-art filtration systems used for industrial applications will typically eliminate the bulk of the larger particles. Erosion can also become a problem for driven compressors or pumps where the process gas or fluid carries solid materials.
- Damage is often caused by large foreign objects striking the flow path components. These objects may enter the engine with the inlet air, or the gas compressor with the gas stream, or are the result of broken off pieces of the engine itself. Pieces of ice breaking off the inlet, or carbon buildup breaking off from fuel nozzles can also cause damage.
- Abrasion is caused when a rotating surface rubs on a stationary surface. Many engines use abradable surfaces, where a certain amount of rubbing is allowed during the run-in of the engine, in order to establish proper clearances. The material removal will typically increase seal or tip gaps.

While some of these effects can be reversed by cleaning or washing the engine, others require the adjustment, repair, or replacement of components.

It should be noted that the determination of the exact amount of performance degradation in the field is rather difficult. Test uncertainties are typically significant, especially if package instrumentation as opposed to a calibrated test facility is used. Even trending involves significant uncertainties, because in all cases the engine performance has to be corrected from datum conditions to a reference condition.

Three major effects determine the performance deterioration of the compressor:

- Increased tip clearances
- Changes in airfoil geometry
- Changes in airfoil surface quality

While the first two effects typically lead to nonrecoverable degradation, the latter effect can at least partially be reversed by washing the compressor.

The overall effect of degradation on an engine compressor yields added losses and lower capability of generating head. Typically, a degraded compressor also will have a reduced surge or stall margin (Spakovszki, et al., 1999). This will not have any significant effect on the steady-state operation, as long as other effects that lower the stall margin (such as water or steam injection) are avoided (Brun, et al., 2005). For a given speed of a degraded compressor, each subsequent stage will see lower Mach numbers (because of the higher temperature) and an increased axial velocity component (because $\rho = p/RT$, where p is reduced, T is increased, thus the density gets reduced).

The net effect will be that, while in the new machine all stages were working at their optimum efficiency point at design surge margins, the degradation will force all stages after the first one to work at off-optimum surge margins and lower than design efficiency. This will not only lower the overall efficiency and the pressure ratio than can be achieved, but also the operating range.

Calculations for a typical axial compressor (Kurz and Brun, 2001) reveal that the combined effects of airfoil fouling and increased clearances lead to loss of pressure ratio, loss of efficiency, and loss of range or stall margin. In particular the increased clearances cause choke at lower flow.

Recoverable and Nonrecoverable Degradation

The distinction between recoverable and nonrecoverable degradation is somewhat misleading. The majority of degradation is recoverable; however the effort is very different depending on the type of degradation. The recovery effort may be as small as water or detergent online washing, or detergent on-crank washing. The degradation recovery by any means of washing is usually referred to as recoverable degradation. However, a significant amount of degradation can be recovered by engine adjustments (such as resetting variable geometry). Last, but not least, various degrees of component replacement in overhaul can bring the system performance back to as-new conditions.

Protection Against Degradation

While engine degradation cannot entirely be avoided, certain precautions can clearly slow the effects down. These precautions include the careful selection and maintenance of the air filtration equipment, and the careful treatment of fuel, steam, or water that are injected into the combustion process. It also includes obeying manufacturers' recommendations regarding shutdown and restarting procedures. For the driven equipment surge avoidance, process gas free of solids and liquids and operation within the design limits need to be mentioned. With regards to steam injection, it must be noted that the requirements for contaminant limits for a gas turbine are, due to the higher process temperatures, more stringent than for a steam turbine.

The site location and environment conditions, which dictate airborne contaminants, their size, concentration, and composition, need to be considered in the selection of air filtration. Atmospheric conditions, such as humidity, smog, precipitation, mist, fog, dust, oil fumes, industrial exhausts, will primarily affect the engine compressor. Fuel quality will impact the hot section. The cleanliness of the process gas, entrained particles, or liquids will affect the driven equipment performance. Given all these variables, the rate of degradation is impossible to predict with reasonable accuracy.

Thorough on-crank washing can remove deposits from the engine compressor blades and is an effective means for recovering degradation of the engine compressor. The engine has to be shut down and allowed to cool down prior to applying detergent to the engine compressor while it rotates at slow speed. Online cleaning, where detergent is sprayed into the engine running at load, can extend the periods between on-crank washing, but it cannot replace it. If the compressor blades can be accessed with moderate effort (for example, when the compressor casing is horizontally split), hand cleaning of the blades can be very effective.

TRANSIENT BEHAVIOR

All the above considerations were made with the assumption that the engine operates at steady-state conditions. The author should briefly discuss the engine operation during load transients, i.e., when load is added or removed.

Figure 19 shows the engine limits (for a two shaft engine) from start to the full load design point. The engine initially is accelerated by a starter. At a certain gas power (GP) speed, fuel is injected and light-off occurs. The fuel flow is increased until the first limit, maximum firing temperature, is encountered. The engine continues to accelerate, while the fuel flow is further increased. Soon, the surge limit of the engine compressor limits the fuel flow. While the starter continues to accelerate the engine, a point is reached where the steady-state operating line can be reached without violating surge or temperature limits. At this point, the engine can operate self sustaining, i.e., the starter can disengage. The maximum acceleration (i.e., the maximum load addition) can now be achieved by increasing to the maximum possible fuel flow. However, the maximum possible fuel flow is limited by either the surge limit of the engine compressor or the maximum firing temperature. If the load suddenly drops, the maximum rate deceleration is limited by the flameout limits of the engine.

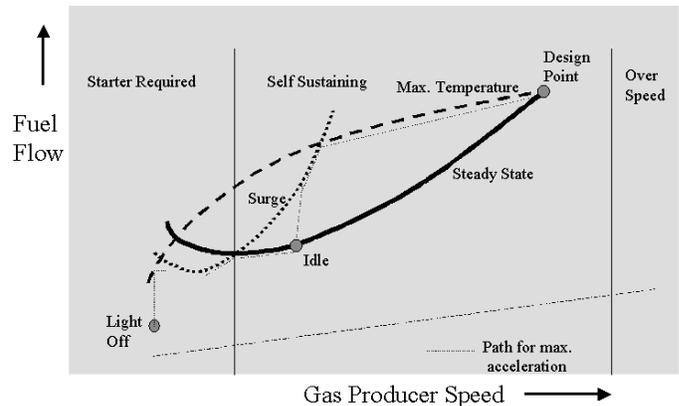


Figure 19. Start and Acceleration Map for a Gas Turbine.

CONCLUSION

The previous pages have given some insight into the working principles of a gas turbine, and what the effect of these working principles on the operating characteristics of gas turbines is. Based on this foundation, it was explained what the effects of changes in ambient temperature, barometric pressure, inlet and exhaust losses, relative humidity, accessory loads, different fuel gases, or changes in power turbine speed are. The topics presented aim at enhancing the understanding of the operation principles of a gas turbine in industrial applications.

NOMENCLATURE

A	= Through-flow area
c	= Velocity vector in stationary frame
c_p	= Specific heat
E_f	= Lower heating value
h	= Enthalpy
L	= Length
M	= Mach number
M_n	= Machine Mach number
N	= Rotational speed in rpm
N_{GP}	= Gas generator speed
p	= Stagnation pressure
P	= Power
Q	= Volumetric flow rate
q	= Heat flow
R	= Specific gas constant

- s = Entropy
- T = Temperature
- U = Blade velocity
- V = Velocity vector
- W = Mass flow rates
- w = Velocity vector in rotating frame
- Δ = Difference
- η = Efficiency
- τ = Torque
- γ = Specific heat ratio
- ω = Rotational speed in rad/sec = 2πN/60

Subscripts

- 1 = At engine inlet
- 2 = At engine compressor exit
- 3 = At turbine inlet
- 5 = At power turbine inlet
- 7 = At engine exit
- t = Turbine
- c = Compressor
- I = Tangential direction

APPENDIX A—
GAS TURBINE CYCLE CALCULATION

A gas turbine may be designed for the following parameters for the compressor:

- $p_a = 14.73$ psia (1.013 bara) $p_2 = 147.3$ psia (10.13 bara) $W = 100$ lb/s (45 kg/s) = 85 percent
- $T_a = 100^\circ\text{F}$ (37.8°C) = 560 R (311 K)

and the turbine (neglecting the fuel mass flow and the combustor pressure drop):

- $p_a = 14.73$ psia (1.013 bara) $p_3 = 147.3$ psia (10.13 bara) $W = 100$ lb/s (45 kg/s) = 85 percent
- $T_3 = 1600^\circ\text{F}$ (870°C) = 2060 R (1144 K)

One uses the relationships for work H and power P:

$$H = c_p \Delta T \tag{A-1}$$

$$P = W \cdot H$$

and the gas properties for air: $c_p = 0.24$ Btu/lbR (1.007 kJ/kgK); $\gamma = 1.4$ (this is a simplified assumption, because the gas properties of the exhaust gas are somewhat different from air).

The compressor temperature rise is:

$$T_2 - T_1 = \frac{T_1}{\eta_c} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] = 613R \quad (341K) \tag{A-2}$$

and the compressor discharge temperature is therefore:

$$560R + 613R = 713^\circ F (379^\circ C) \tag{A-3}$$

This indicates that the compressor consumed the work:

$$H = 24 \times 613 = 147 \text{ Btu/lb} (= 1.007 \times 341 = 344 \text{ kJ/kg}), \tag{A-4}$$

and the power:

$$P = 100 \text{ lb/s} \times 147 \text{ Btu/lb} = 14700 \text{ Btu/s} = 20800 \text{ hp} \tag{A-5}$$

$$(= 45 \text{ kg/s} \times 344 \text{ kJ/kg} = 15480 \text{ kJ/s} = 15480 \text{ kW})$$

The power extraction of the turbine causes a temperature drop:

$$T_3 - T_7 = \eta_t \cdot T_3 \left[1 - \left(\frac{p_7}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] = 844R \quad (469K) \tag{A-6}$$

and thus an exhaust temperature of:

$$T_7 = 1600 - 844 = 756^\circ F (= 1144 - 469 = 402^\circ C) \tag{A-7}$$

Thereby extracting the work:

$$H = 24 \times 844 = 202 \text{ Btu/lb} (= 1.007 \times 469 = 472 \text{ kJ/kg}), \tag{A-8}$$

producing a power of:

$$P = 100 \text{ lb/s} \times 202 \text{ Btu/lb} = 20200 \text{ Btu/s} = 28583 \text{ hp} \tag{A-9}$$

$$(= 45 \text{ kg/s} \times 472 \text{ kJ/kg} = 21240 \text{ kJ/s} = 21240 \text{ kW})$$

The net engine output is the difference between the power produced by the turbine and absorbed by the compressor:

$$P_{net} = 28583 \text{ hp} - 20800 \text{ hp} = 7783 \text{ hp} \tag{A-10}$$

$$(= 21240 \text{ kW} - 15480 \text{ kW} = 5760 \text{ kW})$$

With a compressor exit temperature of 713°F (379°C) and a turbine inlet temperature of 1600°F (870°C), one needs to add heat to bring the gas from 713°F (379°C) to 1600°F (870°C):

$$Q = W c_p \Delta T = 100 \times 0.24 \times (1600 - 713) = 21300 \text{ Btu/s} = 76.7 \text{ MM Btu/hr} \tag{A-11}$$

$$(= 45 \times 1.007 \times (870 - 379) = 22250 \text{ kJ/s} = 80.1 \text{ GJ/hr})$$

With this, the engine heat rate can be calculated from:

$$HR = 76.7 / 7783 = 9850 \text{ Btu/hphr} \tag{A-12}$$

$$(= 80.1 / 5760 = 13900 \text{ kJ/kWh}),$$

and the thermal efficiency is:

$$\eta_{th} = 5760 \text{ kW} / 22250 \text{ kJ/s} = 25.9\% \tag{A-13}$$

APPENDIX B—
THERMODYNAMICAL PARAMETERS
FOR EXHAUST GASES

The specific heat and ratios of specific heats, which determine the performance of the turbine section, depend on the fuel to air ratio, the fuel composition, and the relative humidity of the air. The main constituents of the exhaust gas are nitrogen, oxygen, carbon dioxide, and water. The specific heat (c_p) and gas constant (R) of all these constituents are known, so it is easy to calculate the overall c_p and γ , once the mole fractions of the constituents y_i are known:

$$c_p = \sum_{i=1}^N y_i c_{p,i} \tag{B-1}$$

$$\gamma = \frac{c_p}{c_p - R_m}$$

Table B-1 gives γ , c_p , and R of the above substances at 10 bar, 800°C (145.04 psi, 1472°F).

Table B-1. Specific Heat (C_p), Ratio of Specific Heats and Gas Constants (R) for Some Components of the Exhaust Gas.

	cp (kJ/kgK)	R (kJ/kgK)	$\gamma = 1/(1-R/cp)$
Air	1.156	287	1.33
CO ₂	1.255	189	1.18
H ₂ O	2.352	461	1.24

Assuming an expansion over a constant pressure ratio and constant efficiency, then:

$$\Delta T = \frac{\Delta h}{c_{p,ex}} = \frac{\eta_t \cdot \Delta h_s}{c_{p,ex}} = \eta_t T_3 \left[1 - \left(\frac{p_5}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (\text{B-2})$$

Thus, the temperature differential over the turbine depends on the fuel gas composition and the water content of the inlet air. The practical consequence of the above is the fact that most engines measure T_5 rather than T_3 for control purposes. The ratio T_5/T_3 is often assumed constant. However, in reality it is dependent on the fuel to air ratio, the fuel gas, and the water content of the air (i.e., the relative humidity and the ambient temperature).

Another effect that must be considered is the fact that the relationship between the pressure ratio and the Mach number depend on γ . That means that the maximum volumetric flow through the first stage GP nozzle and the first stage PT nozzle also depend on the exhaust gas composition, which means that different fuel compositions (if the differences are very large) can influence the engine match.

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