

GAS TURBINE PERFORMANCE—WHAT MAKES THE MAP?



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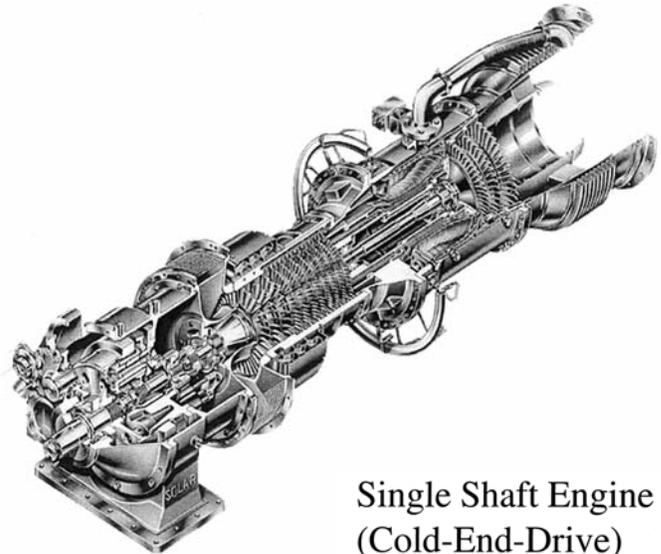
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ABSTRACT

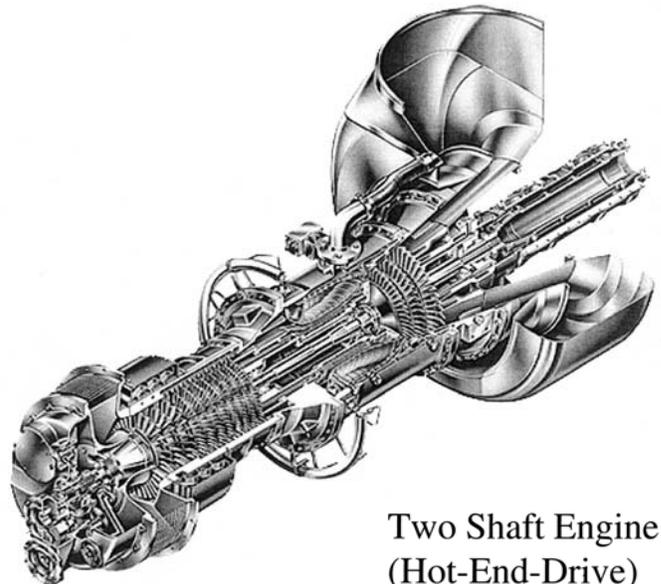
The power and efficiency characteristics of a gas turbine are the result of a complex interaction of different turbomachines and a combustion system. This tutorial addresses the basic characteristics of each of the components, and defines the rules under which they interact, as well as typical control limits and control concepts. Important nondimensional characteristics such as Mach number and Reynolds number are introduced. The concept of component matching is explained.

INTRODUCTION

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) 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 1). The power and efficiency characteristics of a gas turbine are therefore the result of a complex interaction of different turbomachines and a combustion system. This tutorial addresses the basic characteristics of each of the components, and defines the rules under which they interact, as well as typical control limits and control concepts. Important nondimensional characteristics such as Mach number and Reynolds number are introduced. The concept of component matching is explained.



Single Shaft Engine
(Cold-End-Drive)



Two Shaft Engine
(Hot-End-Drive)

Figure 1. Single-Shaft (Cold End Drive) and Two-Shaft (Hot End Drive) Gas Turbines.

Based on this foundation, 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 explained.

The following topics are discussed:

- The gas turbine as a system
- Component matching
- Off-design behavior of gas turbines
- Low fuel gas pressure
- Accessory loads
- Single-shaft versus two-shaft engines
- Variable inlet and stator vanes
- Control temperature
- Transient behavior
- Thermodynamical parameters of exhaust gases

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.

WHAT DO TYPICAL PERFORMANCE 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 2 for single-shaft engines and in Figures 3, 4, and 5 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 (Figure 5) 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 (1998) is essentially a subset of the power turbine curve in Figure 4.

It should be noted that, particularly in the field, the measurement of power output, heat rate, exhaust flow, and exhaust temperature is usually rather difficult (Brun and Kurz, 1998). Understanding the operating principles of the engine is therefore a useful tool of interpreting data.

GENERAL REMARKS ON GAS TURBINE THERMODYNAMICS

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 approximately constant pressure, and then expanding the hot gas in turbine (Brayton cycle). The turbine provides the necessary power to operate the compressor. Whatever power is left is used as the mechanical output

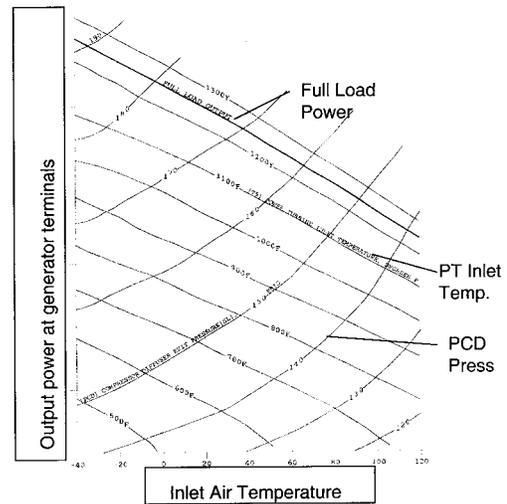
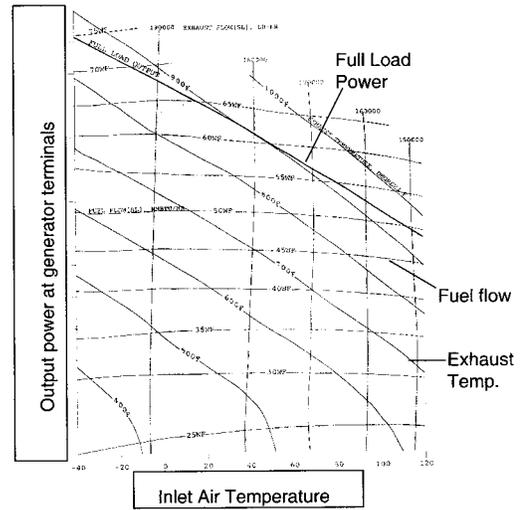


Figure 2. Typical Performance Maps for a Single-Shaft Gas Turbine, Driving a Generator.

of the engine. This thermodynamic cycle can be displayed in an enthalpy-entropy (h - s) diagram (Figure 6). 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 afterward 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 4-5 describes the work output of the power turbine.

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

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

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 $c_{p,a}$, $c_{p,e}$, and $c_{p,a}$ being suitable averages):

$$c_{p,a}(T_1 - T_2) + c_{p,e}(T_3 - T_7) = \frac{P}{W} \quad (2)$$

$$c_p(T_3 - T_2)W = E_f W_f \quad (3)$$

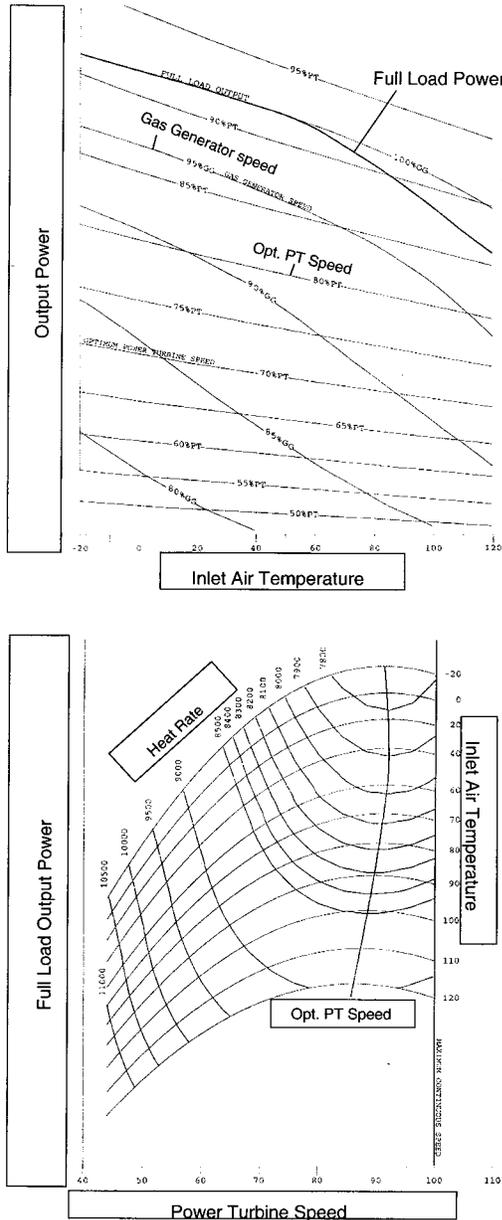


Figure 3. Typical Performance Maps for a Two-Shaft Gas Turbine (First Part).

In Equation (2), the first term is the work input by the compressor, the second term describes the work extracted by the turbine section. Equation (3) yields the temperature increase from burning the fuel in the combustor.

For two-shaft engines, two additional relationships can be derived, taking into account that the gas generator turbine has to balance the power requirements of the compressor, and that the useful power output is generated by the power turbine:

$$c_{p,a}(T_2 - T_1) = c_{p,e}(T_3 - T_5) \quad (4)$$

$$c_{p,e}(T_5 - T_7) = \frac{P}{W} \quad (5)$$

GENERAL REMARKS ON TURBOMACHINERY AERODYNAMICS

Any gas turbine consists of several turbomachines. First, there is an air compressor, and after the combustion has taken place, there

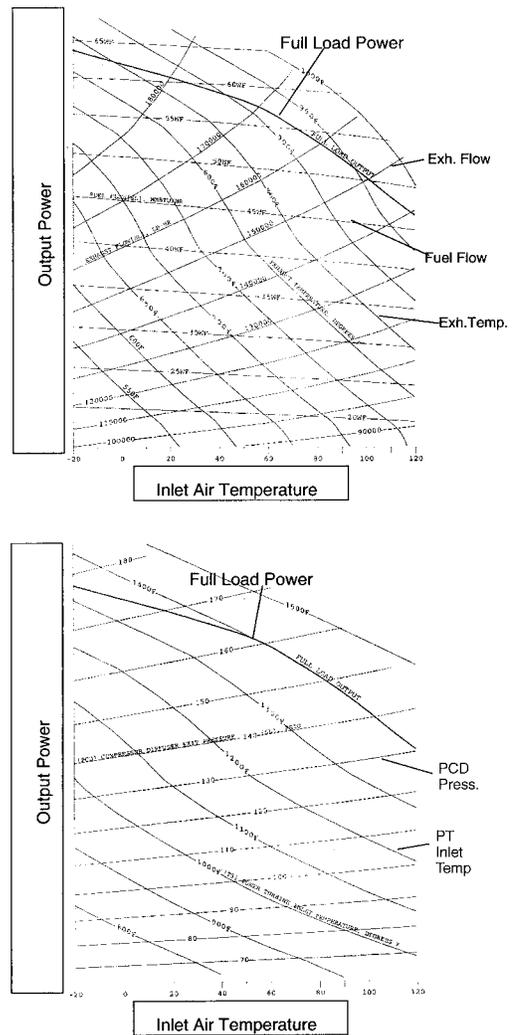


Figure 4. Typical Performance Maps for a Two-Shaft Gas Turbine (Second Part).

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 (two-shaft design), or only of a gas generator turbine (single-shaft, Figure 1).

Compressor

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 on the air. The fundamental law describing the conversion of mechanical energy into pressure in a turbomachine is Euler’s law:

- A compressor stage imparts energy on the fluid (air) by increasing the fluid’s angular momentum (torque). The increase of the tangential velocity component of the air is simply (Figure 7):

$$\Delta c_u = c_{u2} - c_{u1} \quad (6)$$

The mathematical expression for torque is:

$$\tau = W \cdot r \cdot c_u \quad (7)$$

and thus the change of angular momentum of the fluid is:

$$\Delta \tau = W \cdot \Delta(r \cdot c_u) = W \cdot (r_2 c_{u2} - r_1 c_{u1}) \quad (8)$$

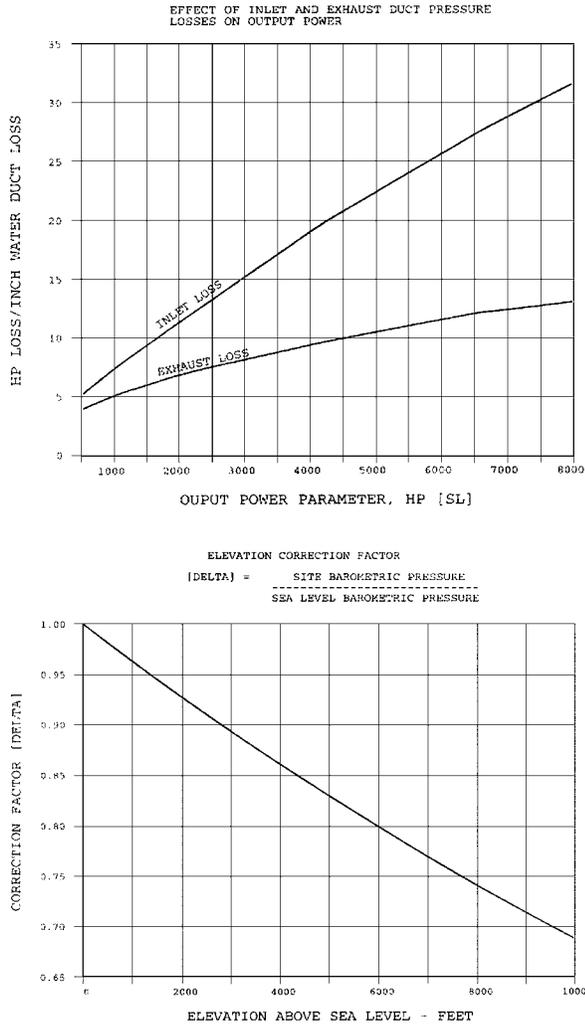


Figure 5. Typical Correction Curves for Elevation and Inlet and Exhaust Losses.

Power (work/time) is defined as torque multiplied by the angular speed, i.e.,:

$$P = \omega \cdot \Delta \tau = \omega \cdot W \cdot \Delta (r \cdot c_u) = \omega \cdot W \cdot (r_2 c_{u2} - r_1 c_{u1}) \quad (9)$$

This equation is called Euler's angular momentum equation. From Euler's equation one can also determine the head (enthalpy difference) per stage as follows:

$$H = h_2 - h_1 = \frac{P}{W} = \omega \cdot (r_2 c_{u2} - r_1 c_{u1}) \quad (10)$$

And since it is known that $h = c_p T$ one can estimate the temperature increase across the compressor stage:

$$T_2 = \frac{\omega}{c_p} \cdot (r_2 c_{u2} - r_1 c_{u1}) + T_1 \quad (11)$$

Since this is a compressor we are really more interested in the pressure increase than the temperature increase per stage. The isentropic relationships give us a function between pressure and temperature ratios. However, when employing the isentropic relationships in this context one has to remember that they apply for ideal (no losses) thermodynamic processes. In this case the process is not ideal and thus a compressor efficiency has to be introduced:

$$\eta_c = \frac{\text{Ideal (Isentropic) Head}}{\text{Actual Head}} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \quad (12)$$

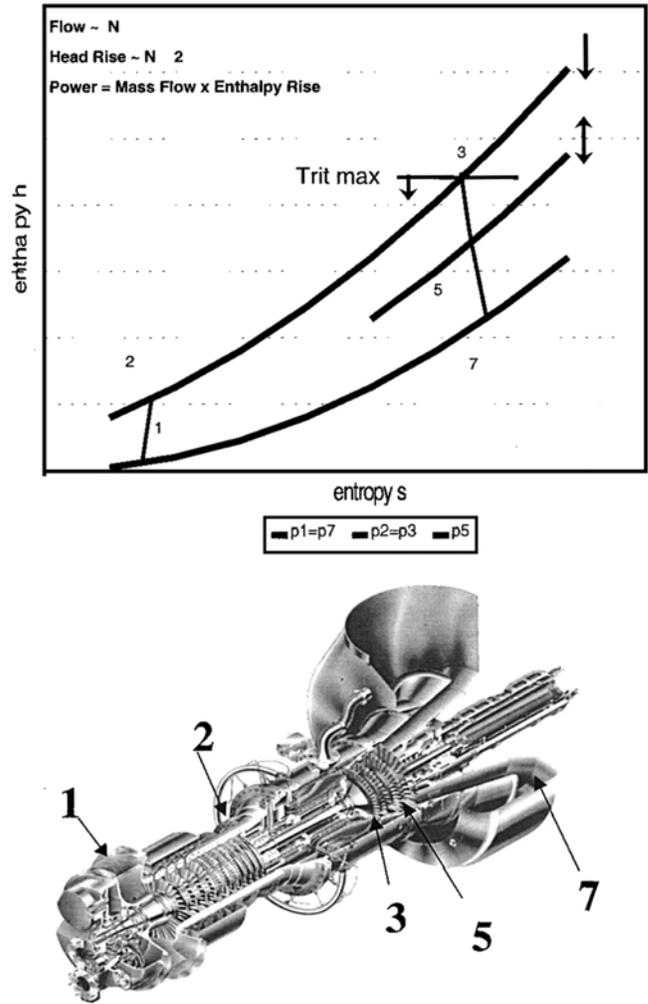


Figure 6. Enthalpy-Entropy Diagram for the Brayton Cycle.

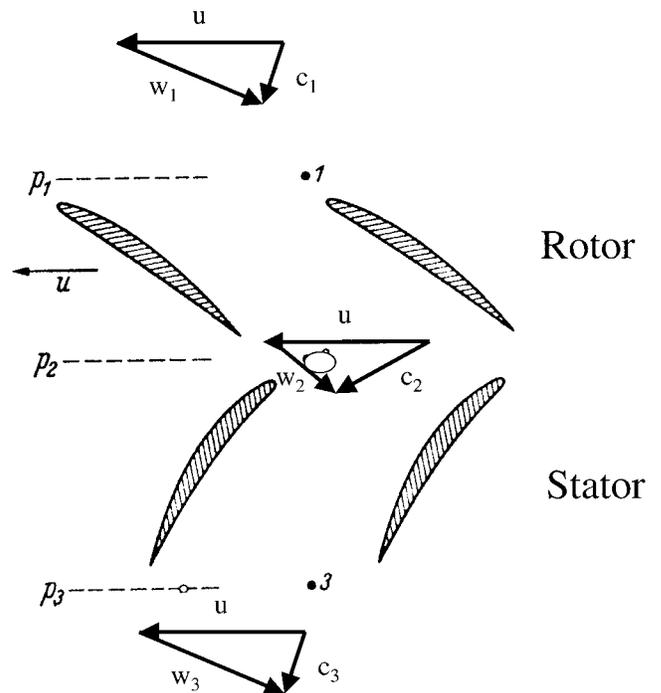


Figure 7. Energy Conversion in a Compressor Stage.

When combining the above efficiency with the expression for temperature and the isentropic relationship for temperature and pressure ($T_2/T_1 = (P_2/P_1)^{\gamma/(\gamma-1)}$) we obtain after some algebra:

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta \omega}{c_p T_1} \cdot (r_2 c_{u2} - r_1 c_{u1})\right)^{\frac{\gamma}{\gamma-1}} \quad (13)$$

This expression gives us the pressure ratio per stage as a function of the tangential air velocity. Using the expression for head one can also rewrite this as follows (to relate head to pressure ratio):

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta}{c_p T_1} \cdot H\right)^{\frac{\gamma}{\gamma-1}} \quad (14)$$

The above set of equations is an extremely powerful engineering tool. From basic knowledge of the tangential flow velocities inside the compressor rotating blade passage one can determine the head, required power, temperature ratio, and pressure ratio across a compressor stage.

Turbine

The same relationships that apply to the compressor can also be applied to the turbine.

$$P = \omega \cdot \Delta \tau = \omega \cdot W \cdot \Delta(r \cdot c_u) = \omega \cdot W \cdot (r_2 c_{u2} - r_1 c_{u1}) \quad (15)$$

All that has to be considered is that while the compressor imparts energy to the fluid, thus $c_{u2} > c_{u1}$, the turbine extracts energy from the fluid, and thus $c_{u2} < c_{u1}$. The visible expression of this fact is the change in the sign for the power. The pressure ratio and temperature ratio over the turbine are thus:

$$T_2 = \frac{\omega}{c_p} \cdot (r_2 c_{u2} - r_1 c_{u1}) + T_1 \quad (16)$$

and

$$\frac{P_2}{P_1} = \left(\frac{\eta \omega}{c_p T_1} \cdot (r_2 c_{u2} - r_1 c_{u1}) + 1\right)^{\frac{\gamma}{\gamma-1}} \quad (17)$$

Mach Number

The Mach number is one of the most important parameters influencing the performance of turbomachinery components. An increasing Mach number can completely change the aerodynamic behavior of a component (shown later in Figure 12). Mach numbers are used to compare a velocity with the speed of sound:

$$M = \frac{w}{\sqrt{\gamma \cdot R \cdot T}} \quad (18)$$

The Mach number increases with increasing velocity w , and decreasing temperature T . It also depends on the gas composition, which determines the ratio of specific heats γ and the gas constant R .

Frequently, the “machine” Mach number M_n is 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 :

$$M_n = \frac{u}{\sqrt{\gamma \cdot R \cdot T}} = \frac{2\pi D N}{\sqrt{\gamma \cdot R \cdot T}} \quad (19)$$

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.

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

$$NGP_{corr} = \frac{NGP}{\sqrt{\frac{\gamma \cdot R \cdot T}{(\gamma \cdot R \cdot T)_{ref}}}} \quad (20)$$

Even though NGP_{corr} (corrected gas generator speed) is not dimensionless, it is a convenient way of writing the machine Mach number of the component. In the following text, the simplified expression N/\sqrt{T} , which is based on the above explanations, will also be used.

Any aerodynamic component performance will change if the characteristic Mach numbers are changed. The dramatic change of the flow velocities in a turbine nozzle is shown in Figure 8. For turbine nozzles, one of the effects connected with the Mach number is that it limits the maximum flow that can pass through the nozzle. Beyond a certain pressure ratio, the amount of actual flow that can pass through the nozzle can no longer be increased by increasing the pressure ratio. With increasing pressure ratio, 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 throughflow (which is proportional to the velocity at the inlet into the nozzle) can no longer be increased (Kurz, 1991).

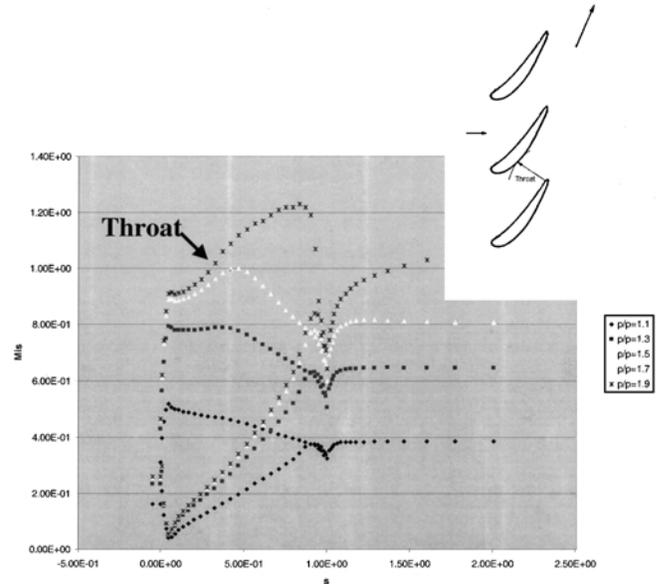


Figure 8. Velocity Distribution in a Turbine Nozzle at Different Pressure Ratios.

Another effect of changes in the Mach number is the strong dependency of losses and enthalpy rise or decrease for a given blade row on the characteristic Mach number. Figure 9 shows how the losses increase for an axial compressor stage, while the range is reduced with a rising Mach number (Cohen, et al., 1996). The practical conclusion is that the performance of any aerodynamic component will change if the characteristic Mach number changes.

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

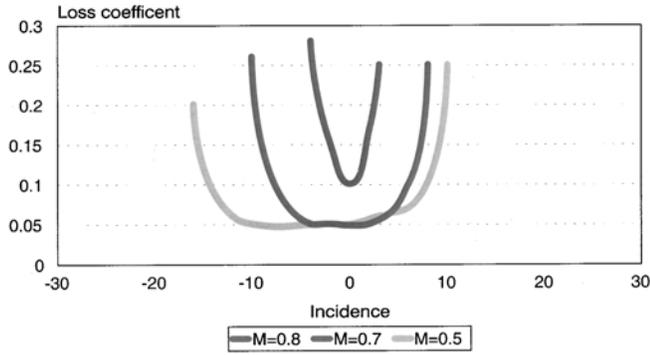


Figure 9. Losses and Range of an Axial Compressor Stage at Different Mach Numbers.

temperature for the engine compressor is the ambient temperature, the characteristic temperature for the gas generator (GP) turbine and the power turbine is the firing temperature T_3 and the power turbine inlet temperature T_5 , respectively. The Mach number is indeed the most important nondimensional characteristic for the gas turbine behavior.

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 compressor and gas generator turbine can be expressed by $NGP_{corr} = constant$ (which leads to identical Mach numbers for the compressor):

$$NGP_{corr} = constant = \frac{NGP_{op1}}{\sqrt{\gamma \cdot R \cdot T_{1,op1}}} = \frac{NGP_{op2}}{\sqrt{\gamma \cdot R \cdot T_{1,op2}}} \quad (21)$$

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:

$$\frac{T_{3,op1}}{T_{1,op1}} = \frac{T_{3,op2}}{T_{1,op2}} \quad (22)$$

The engine heat rate will remain roughly constant, while the engine power will be changed proportionally to the change in inlet density. The emphasis lies on the word roughly, because this approach does not take effects like Reynolds number changes, changes in clearances with temperature, changes in gas characteristics (γ, c_p), 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. The Reynolds number is defined by:

$$Re = \frac{w \cdot L}{\nu} \quad (23)$$

The kinematic viscosity ν depends on the gas, the temperature, and the pressure. 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

(Kurz, 1991). 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.

THE GAS TURBINE AS A SYSTEM

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. The components are designed to work together at their highest efficiencies at a single operating 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, we 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

Consider the single-shaft gas turbine performance first. 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 10 and 11, respectively.

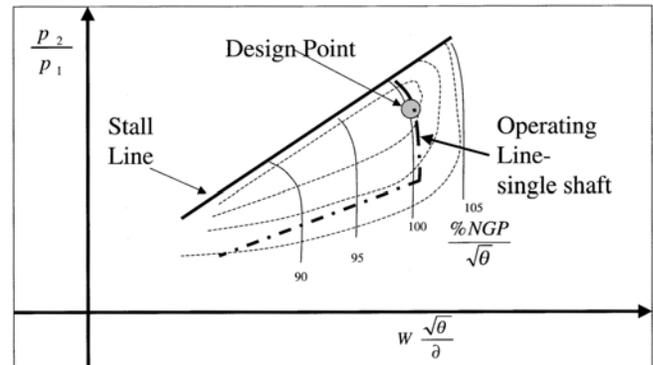


Figure 10. Typical Compressor Performance Map with Operating Lines for a Single-Shaft Engine.

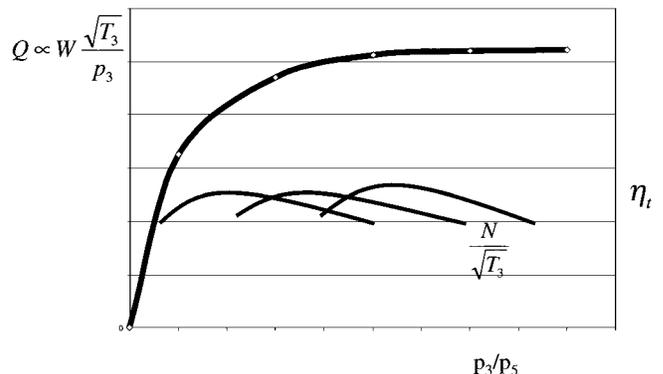


Figure 11. Typical Turbine Performance Map.

The relationships between the work input and the pressure ratio of turbine and compressor (assume $p_7 = p_1$ and $W_c = W_t = W$ to make the example simpler) are given by:

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (24)$$

$$\frac{\Delta T_t}{T_3} = \eta_t \left[1 - \left(\frac{p_7}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (25)$$

These relationships are a consequence of the relationships between temperature rise and pressure ratio in isentropic systems. The introduction of the component efficiencies then bridges the gap between the isentropic and the real process.

The gas generator turbine and the compressor operate on the same shaft, thus:

$$\frac{N}{\sqrt{T_3}} = \frac{N}{\sqrt{T_1}} \cdot \sqrt{\frac{T_1}{T_3}} \quad (26)$$

A possible algorithm for determining an arbitrary operating point is described in APPENDIX B.

Two-Shaft Engines

A similar example for the performance of a two-shaft gas turbine is now described. 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 GP turbine has to be high enough to force the flow through the power turbine.

Typical turbine and compressor maps are shown in Figures 11 and 12, respectively. Note that the compressor operating points are very different between a single-shaft and a two-shaft engine.

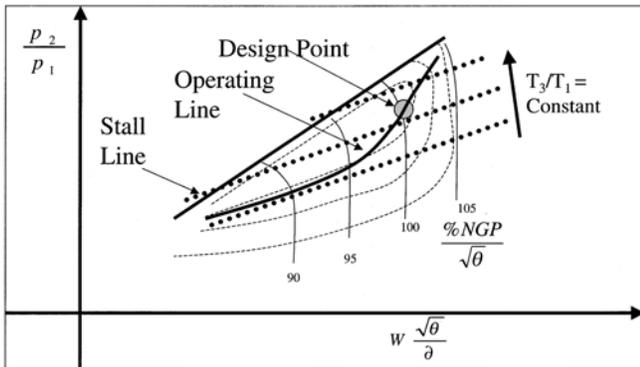


Figure 12. Typical Compressor Performance Map with Operating Lines for a Two-Shaft Engine.

One can again write for the relationships between the work input and the pressure ratio of components. To make the example easier to follow, again assume $p_7 = p_1$ and $W_c = W_t = W$. Unlike for the single-shaft engine, the pressure ratio for the turbine, p_3/p_5 , is not known beforehand.

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (27)$$

$$\frac{\Delta T_t}{T_3} = \eta_t \left[1 - \left(\frac{p_5}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (28)$$

However, an additional condition is found, because the absorbed compressor power must be provided by the gas generator turbine:

$$c_{p,c} \frac{\Delta T_c}{T_1} \cdot \frac{1}{\eta_m} = c_{p,t} \frac{\Delta T_t}{T_3} \frac{T_3}{T_1} \quad (29)$$

Also, the gas generator turbine and the compressor operate on the same shaft, thus:

$$\frac{N}{\sqrt{T_3}} = \frac{N}{\sqrt{T_1}} \cdot \sqrt{\frac{T_1}{T_3}} \quad (30)$$

In APPENDIX C a possible algorithm for determining an arbitrary operating point is described.

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:

$$\frac{p_5}{p_a} = \frac{p_2}{p_a} \frac{p_3}{p_2} \frac{p_5}{p_3} \quad (31)$$

This has some interesting implications: If the power turbine operates at a nonchoked condition, the gas generator will be restrained to operate at a fixed pressure ratio for each power turbine pressure ratio.

The pressure ratio is controlled by the swallowing capacity of the power turbine. Moreover, if the power turbine is choked, it will cause the gas generator turbine to operate at one fixed nondimensional point (Figure 13). In reality, the problem may be even less complex. The gas generator turbine first-stage nozzle often also operates at or near choked flow conditions. In this case, the actual flow through this nozzle is constant. This means that the mass flow through the turbine can be calculated from:

$$W = \frac{p_3}{A^*} \cdot \sqrt{\frac{\gamma}{RT_3}} \left(\frac{\gamma+1}{2} \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} \quad (32)$$

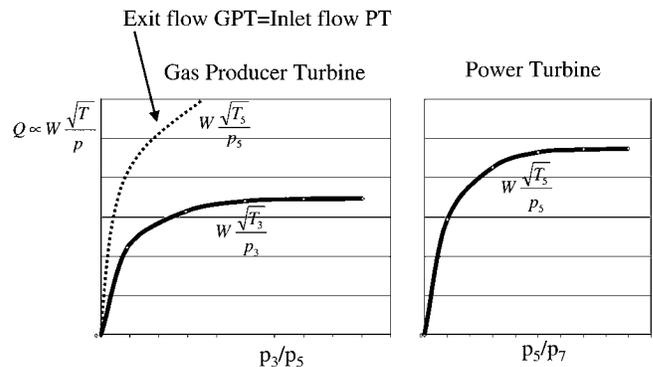


Figure 13. Typical Performance Maps for a Gas Generator Turbine and a Power Turbine in Series.

The mass flow is only dependent on the combustor exit pressure p_3 (which can be substituted by the compressor exit pressure p_2), the firing temperature T_3 , the gas composition (which determines γ), and the geometry of the nozzle, which determines A^* (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 the geometry, will lead to a lower mass flow.
- Increasing the GP speed, thus increasing p_2 , will allow for a larger mass flow.
- With variable inlet guide vanes (IGVs) the airflow can be altered, 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). 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.

The setting of the flow function above obviously has a great influence on the possible operating points of the gas generator. Analyzing the equations above shows that for a high resistance of the power turbine (i.e., low mass flow W for a given p_2/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 swallowing capability). This effect is used to match the power turbine with the gas generator for different ambient temperatures. This concept will be discussed more in the next section.

COMPONENT MATCHING

Each gas turbine consists of three partially independent aerodynamic components (compressor, gas generator turbine, and power turbine). These components need to be "matched," such that the overall performance of the gas turbine is optimized for a defined operating ambient temperature.

Two-shaft engines have a power turbine that is not mechanically coupled with the gas generator. The speed of the gas generator is therefore not controlled by the speed of the driven equipment (as it is 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.

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 14).

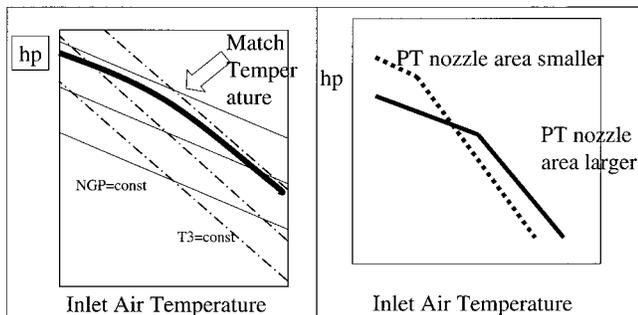


Figure 14. Matching of the Power Turbine.

As discussed previously, the first power turbine nozzle determines the amount of pressure ratio needed by the power turbine to allow a certain gas flow. In turn, this 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 (see 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 generator 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 temperature topping.
- Accessory loads also have the effect of leading to premature temperature topping.
- When IGVs open too far, increased airflow W is the result. Again, the matching equations indicate that early topping is the result.

The match of any engine can be altered by altering the flow characteristic of the first power turbine nozzle. Opening this nozzle, which means that the flow out of this nozzle is more in the axial direction than before, will lower the flow resistance of the power turbine. Therefore, the match point will move to higher ambient temperatures (Figure 14).

Matching of a Gas Generator and Free Power Turbine

Simplified mass and energy conservation retain the essential physical relationships and allow derivation of the two fundamental matching conditions, using the compressor pressure ratio $R_c = p_2/p_1$. Mass conservation,

$$W = const = \frac{Q_3 p_3}{RT_3} \quad (33)$$

in a choked nozzle, where

$$W \frac{\sqrt{T_3}}{A^* p_3} = \left[\frac{\gamma}{R} \left(\frac{2}{\gamma+1} \right)^{(\gamma+1)/(\gamma-1)} \right]^{1/2} \quad (34)$$

yields

$$\sqrt{\frac{T_3}{\theta}} \propto \frac{Q_3}{W_c \sqrt{\theta}} R_c \quad (35)$$

Energy conservation yields

$$\frac{T_3}{\theta} \propto T_{std} \left(1 + K \frac{Q_3}{Q_5} \right) \frac{R_c^{\gamma-1} - 1}{\eta_c \eta_{gp}} \quad (36)$$

Note that the matching is not affected by elevation or inlet pressure loss. However, flow capacities of the gas generator turbine nozzle Q_3 and the power turbine nozzle Q_5 , as well as component efficiencies directly affect the matching. K is considered a constant.

The turbine geometry determines both actual flows Q_3 and Q_5 , as well as the gas generator turbine efficiency. The compressor geometry and speed set the airflow. With variable IGVs the airflow can be altered, thus also setting a new T_3 and a different compressor pressure ratio R_c . The relationship between R_c and T_3 remains, however, unchanged: The turbine flow capacities alone set 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 Q_3 remains constant).

If, however, η_{gpt} increases due to the change in speed, T_3/θ has to drop, leading to an increase in W_c : 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 NGP.

Matching of Single-Shaft Engines

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.

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 operated at varying speeds, the two-shaft configuration is the most advantageous. The running lines for single-shaft and two-shaft units are shown in the compressor maps (Figures 10 and 12).

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. There is little change in compressor temperature rise because the efficiency is also reduced. This means that the compressor power remains essentially fixed. With a two-shaft engine, however, reducing net power output involves a reduction in compressor speed and hence in air flow, pressure ratio, and temperature rise. The compressor power needed is therefore appreciably lower than for the single-shaft engine. It should also be evident from a comparison of the maps that the compressor operates over a smaller range of efficiency in a two-shaft engine.

Comparing the output characteristics of single- and two-shaft engines, indicates that at low ambient temperatures, the power and heat rate of the single-shaft machine tend to become better than for the two-shaft machine. This is due to the fact that the two-shaft engine operates at speed topping, i.e., with a declining firing temperature, while a single-shaft engine is able to operate at the maximum firing temperature over the full range of ambient temperatures. It should be noted that a two-shaft engine with variable geometry can overcome this disadvantage. Both types are penalized in the same way by the increase in compressor Mach number (both run at speed topping, but with falling temperature the Mach number rises). At high ambient temperatures, the performance is almost identical (because both engines reach their maximum firing temperature).

The two types also have different characteristics regarding the supply of waste 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 differences in efficiency between single-shaft and two-shaft machines at part load depend largely on the particular design of the components, and not on the number of shafts.

The torque characteristics, however, 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 as shown in Figure 15. This type of turbine is not very suitable for mechanical drive purposes. The normal flat torque curve of an internal combustion engine is shown dotted for comparison.

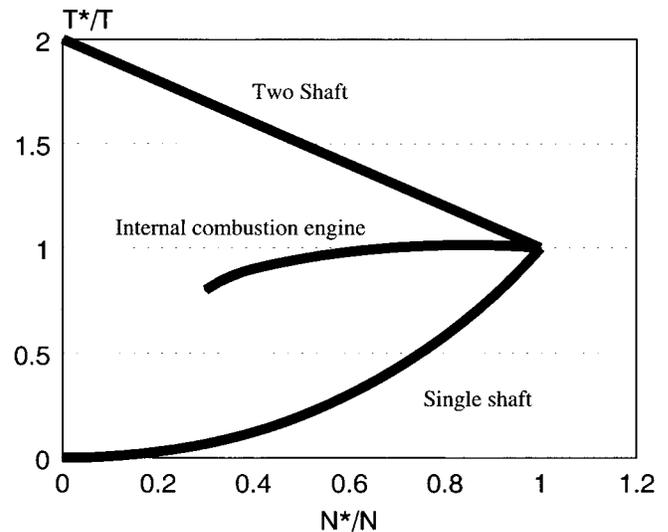


Figure 15. Torque Characteristics of a Single-Shaft and a Two-Shaft Engine, Compared with an Internal Combustion Engine.

The two-shaft unit, having a free power turbine, however, has a more 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. 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. Thus at fixed gas generator operating conditions, reduction in output speed results in an increase in torque as shown in the speed-torque map. It is quite possible to obtain a stall torque of twice the torque delivered at full speed. The torque characteristic is also clearly superior to that of a reciprocating engine (Figure 13).

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 3 suggests that the efficiency penalty will not be greater than about 5 or 6 percent over a speed range from half to full speed. Thus quite a large increase in torque can be obtained efficiently when the output speed is reduced to 50 percent of its maximum value.

OFF-DESIGN BEHAVIOR OF GAS TURBINES OR —WHAT CREATES THE PERFORMANCE MAPS

This section will explain the behavior of single and two-shaft turbines at the following off-design conditions:

- Varying ambient temperature
- Varying ambient pressure
- Varying humidity

- Varying load
- Varying power turbine (PT) speed

Also touched on will be the effect of accessory loads (like lube oil pumps or other pumps driven through a power takeoff directly from the gas generator), and the effect of insufficient fuel gas pressure. The means of controlling the behavior of the engine, such as fuel control valves, bleed valves, and variable geometry, will also be included.

The term firing temperature will be used in a generic way, since for these considerations it is inconsequential whether this term is defined as T_3 , TIT, or TRIT. Usually, T_3 and TIT are defined as the combustor exit or gas generator inlet temperatures, while TRIT designates the inlet temperature into the first gas generator turbine rotor. Due to the introduction of cooling flows in the first stage nozzle, TRIT is slightly lower than TIT. However, TRIT is more meaningful in judging the heat exposure of the first stage rotor, which experiences higher mechanical stresses than the first stage nozzle.

Several concepts need to be introduced to help in the understanding of the behavior. These are:

- The h-s diagram of the gas turbine cycle
- The concept of equal mach numbers (corrected NGP)
- The influence of variable geometry
- The concept of matching
- Typical control concepts

The h-s diagram has been introduced previously (Figure 6). From the h-s diagram can be derived some general ideas of the gas turbine operation. Since the mass flow through the entire engine is almost constant (neglecting the additional fuel mass flow), the enthalpy difference created in the compressor has to be balanced by the enthalpy extracted by the GP turbine. Whatever isentropic enthalpy (i.e., pressure ratio) is left, can be used in the power turbine.

In this discussion, the possibility of variable geometry, especially variable inlet guide vanes (VIGV), will be initially disregarded. At this point, note that VIGVs are able to vary the compressor map while the speed remains constant. Counter swirl will increase flow and head (normally at the expense of efficiency), swirl will reduce head and flow. Variable geometry introduces an additional control variable that can affect changes in the engine operation independent of gas generator speed and firing temperature, which are both controlled by the fuel flow. For example it can be used to keep NGP at optimum level even at high ambient temperatures, when NGP would normally drop.

Important considerations about off-design behavior of a gas turbine include two important dimensionless figures, the Mach and the Reynolds number. Changes in either one of them can lead to significant changes in the performance characteristics of turbomachinery components. Another point worth mentioning is that the performance of any gas turbine depends to some extent on the size of running clearances. These clearances can depend more or less on the prevalent operating temperature. No general relationships are available to describe the opening and closing of clearances with operating temperature.

It also should be noted that some engines are limited in power output (especially at low ambient temperatures) not for aerodynamic reasons, but for mechanical reasons: The engine shaft may be designed for a certain maximum torque, which would be exceeded at low ambient temperatures if the engine were to be run at maximum firing temperature or maximum gas generator speed.

Varying Ambient Temperatures

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

1. The density changes. At constant speed, where the volume flow remains approximately constant, the mass flow (and with it the

power output) will increase with decreasing temperature and will decrease with increasing temperature.

2. The pressure ratio of the compressor at constant speed gets smaller with increasing temperature. This is due to:

$$\Delta h_s = c_p T_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{\kappa}{\kappa-1}} - 1 \right) \quad (37)$$

and

$$\Delta h_s \propto N^2 \quad (38)$$

In other words, to maintain a given pressure ratio (in a two-shaft engine), more speed, more head, and thus more power is needed to drive the compressor (Figure 16). This power 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. Typically, 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/NGP ratio (i.e., the operating point of the gas generator turbine) therefore moves away from the optimum. Variable compressor guide vanes allow keeping up the gas generator speed constant at higher ambient temperatures, thus avoiding efficiency penalties.

A single-shaft, constant speed gas turbine will see a constant head (because the head stays roughly constant for a constant compressor speed), and thus a reduced pressure ratio. Again, because the compressor requires a larger portion of the turbine power, the power output is reduced.

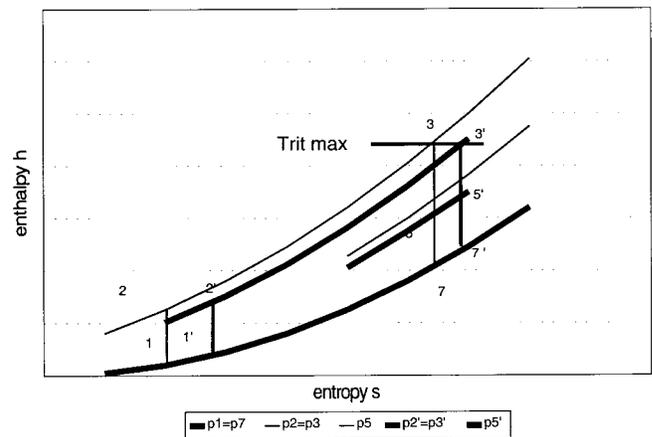


Figure 16. Enthalpy-Entropy Diagram for a Gas Turbine, Showing the Effect of Increasing Ambient Temperature.

3. 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.

4. NGP_{corr} (alas the 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 a reduced firing temperature at a constant gas generator speed.

5. The relevant Reynolds number changes.

The net effect of higher ambient temperatures is an increase in heat rate and a reduction in power.

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 enthalpy-entropy diagram (Figure 16) 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 at 1'. 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 and 2'-3', and considering that the firing temperature T_3 is limited, less heat input is possible (because T_3-T_2 is smaller than T_3-T_2'). The expansion process 3'-4'-5' has, due to the lower $p_2 = p_3$, less pressure ratio available. Furthermore, 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. The effects of increased ambient temperature are therefore a lower power turbine output and less fuel flow (however, also a lower efficiency).

The 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. Engine performance maps typically show a reduction of engine mass flow with increasing ambient temperature. On the engine performance map for a two-shaft engine, one can see that the decline in full load power is much steeper at temperatures above the match point. This distinguishes a two-shaft engine (without adjustable geometry) from a single-shaft engine. The effect of the gas generator speed reduction is more significant than the reduction in firing temperature at low ambient temperatures.

Change in Ambient Pressure

If the ambient pressure changes, only the density of the gas changes, while the volumetric flow and all component efficiencies remain the same. This means that the heat rate is unaffected and the power is reduced proportional to the density reduction. 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 5).

Inlet and Exhaust 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 de-icing, 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 pressure losses reduce the engine performance (Figure 5).

Effects of Humidity

The effects of humidity become clearer if we look at the water content of the inlet air. Relative humidity is the ratio between the amount of water in the air compared to the maximum amount of water the air can carry under the prevailing conditions of pressure

and temperature. At low temperatures, even a high relative humidity does not mean there is a large amount of water in the air, whereas at high temperatures, the mole or weight percentage of water can be rather high.

The engine performance at low ambient temperatures will thus hardly show any effects of changing relative humidity, while the engine performance at high ambient temperatures will be effected somewhat by changes in relative humidity (Figure 17). What are the effects of increasing the amount of water vapor in the air?

- Because water vapor is lighter than air, the mass flow of the engine will be reduced at constant speed.
- For the same reason, the pressure ratio for a given head and speed will go down.
- The specific heat of the exhaust gas will increase with increased amounts of water vapor (refer to thermodynamic parameters for exhaust gases). Therefore the turbine can create more power under otherwise the same conditions.
- The measurement of the control temperature, i.e., the ratio between TRIT and T_5 (refer to thermodynamic parameters for exhaust gases) may be affected.

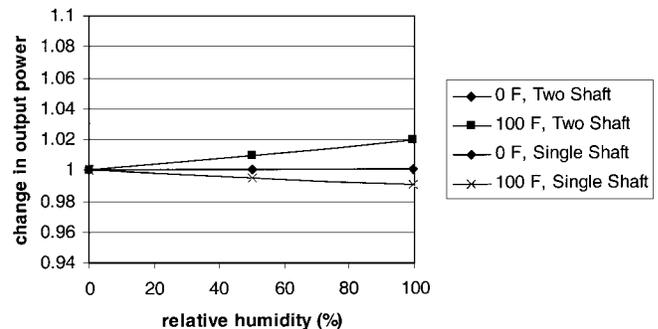


Figure 17. Effect of Relative Humidity on Gas Turbine Performance (Example).

For a two-shaft engine, the following is expected: When the engine operates on NGP topping (the engine speed cannot be increased) higher humidity rather reduces power (items 1 and 2), and when the engine operates on T_5 topping higher humidity rather increases power (items 3 and 4). However, engines with “active guide vane control” (AGVC), i.e., engines where the variable guide vanes of the compressor can be modulated according to a control schedule, are on NGP topping and T_5 topping on warm and hot days. The intent of the AGVC is to modulate the IGVs to maintain maximum NGP while also maintaining maximum T_5 . But, when the humidity increases, T_5 drops (item 4), and the engine becomes topped on NGP at cooler T_5 , thus the power is reduced rather than increased.

Another issue to consider is that the engine is usually controlled at a constant T_5 , not a constant TRIT. For example, at a constant T_5 , the TRIT of a given engine design may drop more than that for another engine design. For the relationship between T_5 and TRIT refer to the section on control temperature. The result is a net loss of power for the former, while leaving the latter design with a net gain in power.

For a single-shaft, constant speed gas turbine, at full load and constant TRIT, two effects counteract each other: Increasing relative humidity decreases air mass flow at the constant NGP (which would decrease power, item 1), but the increasing humidity increases the specific heat (item 3) to offset the loss of air flow. Whether the engine gains or loses power with relative humidity depends on the particular engine design.

While the above sounds rather cumbersome, it must be noted that the overall effect of changes in relative humidity is rather small (Figure 17).

Part Load

To operate the engine at part load, the fuel flow is adjusted, until some control parameter (for example the flow through the driven gas compressor, or a certain kW value for a generator) is satisfied.

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 reduced with load. The airflow will remain almost constant, and the compressor discharge pressure will reduce only slightly. If variable guide vanes are used (for example, to keep the fuel to air ratio within a small window for emissions control), the airflow will be reduced, while the firing temperature drops less than without variable guide vanes.

For a two-shaft engine, both gas generator speed and firing temperature drop at part load. The drop in gas generator speed means that the compressor now operates at a lower Mach number, and thus behaves the same way as for a full load point at a higher ambient temperature. The airflow and usually the compressor discharge pressure are reduced.

For both single- and two-shaft engines, part load operation typically causes an increase in heat rate. How fast the heat rate increases with part load depends largely on the engine component characteristics, but hardly on the configuration, i.e., whether it is a single-shaft or two-shaft engine.

Changing Power Turbine Speed

For any given set of hot gas inlet conditions, the power turbine will have one optimum speed, which is the speed where the power turbine works at its highest efficiency, thus producing the highest output power. The optimum working point of a turbine is characterized by a given ratio of tip speed u (which depends on the rotor speed NPT) and axial velocity c_{ax} . The axial component of the velocity c_{ax} is proportional to the volume flow at the power turbine inlet. The volume 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.

Off-optimum speed of the power turbine reduces the efficiency and the ability to extract head from the flow. Even if NGP (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.

To understand the effects of changing the power turbine speed, one has to consider the velocity polygons. Figure 18 shows velocity polygons for a reaction turbine:

- Leaving the flow area and the flow constant (because the flow condition from the gas generator is not changed), the axial portion of the velocity remains unchanged. Also, the exit flow angles of both rotor (in the absolute frame) and stator (in the rotating frame) will not change, because they are set by the geometry of the blades. The tip speed will change from u to u^* due to the change in speed.

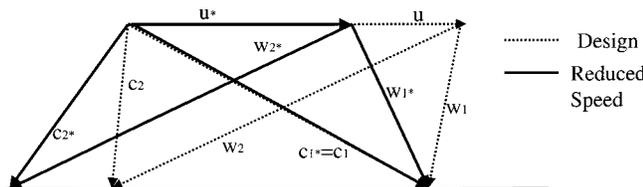


Figure 18. Velocity Polygon for a Turbine Stage at Design and Off-Design Speed.

For the design point:

$$\begin{aligned} c_{u2} &\cong 0 \\ c_{u1} &\cong u \\ h &\cong u^2 \end{aligned} \quad (39)$$

The change in speed from N to N^* yields

$$\begin{aligned} c_{u2} &\cong u^* - u \\ c_{u1} &\cong u \\ h &\cong u^*(c_{u1}^* - c_{u2}^*) \cong u^*(2u - u^*) \end{aligned} \quad (40)$$

and, with $P = W \times h$, we get

$$\frac{P^*}{P} = \frac{h^*}{h} = 2 \frac{N^*}{N} - \left[\frac{N^*}{N} \right]^2 \quad (41)$$

This correlation yields the speed-power relationship for the turbine stage, and is plotted in Figure 19, which is remarkably similar to a real power turbine characteristic as in Figure 3. The drop in power for off-optimum speeds is in reality somewhat larger, because we neglected the reduction of the turbine efficiency due to higher incidence angles and higher swirl in the exit flow. The pressure ratio over the turbine therefore remains almost constant for a wide flow range.

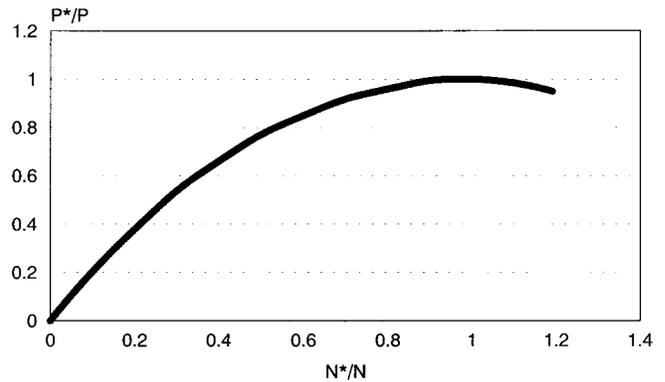


Figure 19. Theoretical Turbine Characteristic Derived from the Velocity Polygon.

Another interesting result of the above is the torque behavior of the power turbine:

$$\frac{\tau^*}{\tau} = \frac{P^*}{P} \frac{N}{N^*} = 2 - \frac{N^*}{N} \quad (42)$$

The torque is thus a linear function of the speed, with the maximum torque at the lowest speed (Figure 15). 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.

Different Fuel Gas

Gas turbines can operate on a wide variety of gas and liquid fuels. A way of characterizing gas fuels is through the Wobbe index, which describes the amount of energy contained within a given volume of fuel:

$$WI = \frac{LHV}{\sqrt{SG}} \quad (43)$$

Fuel gas with a large amount of inert components (such as CO_2 or N_2) have a low Wobbe index, while substances with a large amount of heavier hydrocarbons have a high Wobbe index. Pipeline quality natural gas has a Wobbe index of about 1200.

The Wobbe index by itself does not describe the flame characteristics of the fuel. For example: hydrogen has a low Wobbe index, but a very hot flame, while gas with a high CO_2 content also has a low Wobbe index, but a relatively cool flame. Nonetheless, the

Wobbe index clearly describes the amount of fuel volume necessary to provide the prescribed fuel energy flow.

The amount of fuel gas that is added and burned in the combustor influences the amount of exhaust gas, which, now at high pressure and high temperature, is expanded through the turbine section. The power generated in the turbine section is obviously proportional to the mass flow through the turbine section. Similarly, the power needed to drive the air compressor is proportional to the mass flow through the compressor.

If the Wobbe index of the fuel gas is low, the fuel mass flow must be increased to operate the gas turbine at full load. Even though the fuel mass flow is a very small part of the overall mass flow (about 1 to 3 percent), the mass flow through the turbine section is increased, while the compressor air mass flow remains roughly the same. Thus, the added mass flow causes the overall turbine output to increase proportionally to the added mass flow. The air compressor is forced to operate at a slightly higher discharge pressure due to the added mass flow that has to be pushed through the turbine section. The net effect is an increase in output power.

The above is valid irrespective of whether the engine is a two-shaft or single-shaft engine.

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 NO_x production, higher CO or UHC levels, or flame-out.

Lean-premix combustion achieves reduction in NO_x 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 closer to the lean flame-out limit.

Any part load operation, and even more so during load transients, 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 flame-out at part load or transient situations, for example:

- Bleeding air overboard,
- Using variable inlet guide vanes, and
- Managing the air distribution between primary zone and dilution zone through a valve system,

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

Low Fuel Gas Pressure

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 gas generator compressor discharge pressure (PCD). Note that PCD usually decreases with increasing ambient temperature (refer to discussion of h-s diagram). Insufficient fuel supply pressure at low ambient may well be sufficient for full load operation at higher ambient temperatures. 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 PCD pressure can be overcome by the fuel pressure (Figure 20). 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.

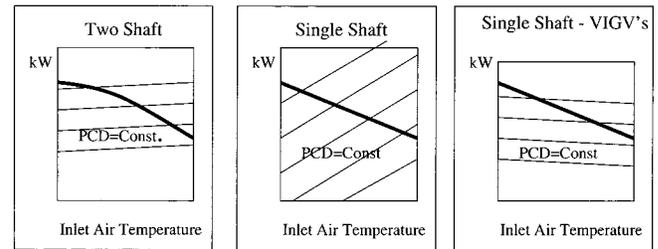


Figure 20. Effect of Low Fuel Gas Pressure on Different Engine Designs.

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, PCD can also be influenced by the position of the VIGVs, thus leading to less power loss (Figure 20).

Without VIGVs, the only way to reduce PCD pressure 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.

Accessory Loads

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.

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 explanation on Euler equations) because they change the head making capability of the stage by increasing or reducing the preswirl contribution (Figure 21). 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:

- During startup of the engine it is possible to keep the compressor from operating in surge.
- The airflow can be controlled to maintain a constant fuel to air ratio in the combustor for dry low NO_x applications on single-shaft machines.

- 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.

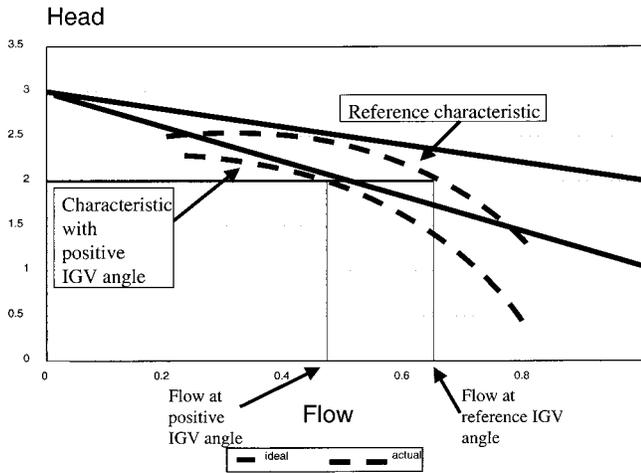


Figure 21. Head-Flow Characteristic for a Compressor with Variable Inlet Guide Vanes.

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 thermodynamic relationship between T_3 and T_5 is described in the section “THE GAS TURBINE AS A SYSTEM.” 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. On single-shaft engines, the exhaust temperature T_7 can be used for purposes of monitoring TRIT. This is not practical for two-shaft engines, because T_7 also depends on the power turbine speed.

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 and with the relative humidity: The ratio T_3/T_5 is reduced at higher ambient temperatures. Maintaining a constant T_5 means that the engine will not reach its maximum allowed T_3 or TRIT on hot days, thus penalizing its power output.

The reason that the ratio T_3/T_5 depends on the ambient conditions is quite interesting: The water content of air depends on the ambient temperature and the relative humidity. At constant relative humidity, the content of water in the air increases with increasing ambient temperature. As outlined in the discussion on combustion products, the amount of water in the combustion gas has a significant impact on properties like γ and c_p in the combustion gas. With the pressure ratio over the gas generator turbine essentially constant, T_3/T_5 is a function of γ , and therefore a function of the ambient temperature and the relative humidity.

Engines with T_3 control use an algorithm in the control system that corrects the T_3/T_5 ratio according to the ambient temperature. This has no impact on the life of the engine, because both T_3 and T_5 controls obey the same upper limit for TRIT. There are also no hardware differences. The capability to reach its full TRIT on hot days improves the performance of the T_3 controlled version of the two-shaft engines significantly.

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.

TRANSIENT CONDITIONS

Change of Load

In a single-shaft, constant speed engine, the only controlled operating parameter is the firing temperature. 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. For a given mass flow, any increase in firing temperature will increase the volume flow through the turbine section. Therefore, the pressure drop over the turbine increases. This means that the pressure supplied from the compressor has to be increased. 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 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 a two-shaft engine the situation is somewhat different: 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, and compressor discharge pressure. At the same time the increasing 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 GP turbine power. This determines the available pressure upstream of 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 and 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.

Transient Behavior

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

Figure 22 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 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 flame-out limits of the engine.

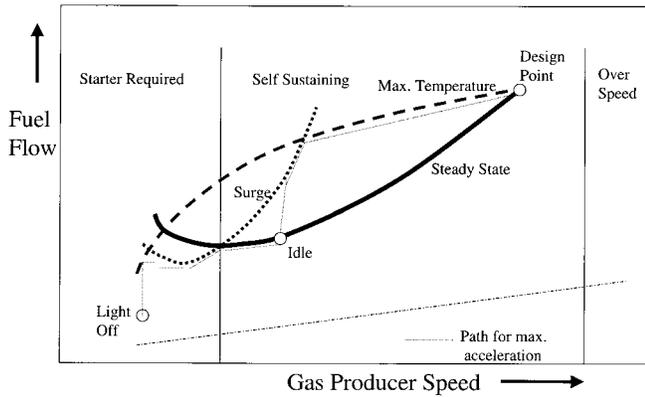


Figure 22. Start and Acceleration Map for a Gas Turbine.

CONCLUSION

The previous sections 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 = Throughflow area
- c = Velocity vector in stationary frame
- c_p = Specific heat
- H = Head
- h = Enthalpy
- L = Length
- M = Mach number
- M_n = Machine Mach number
- N = Rotational speed in rpm
- p = Stagnation pressure
- P = Power
- Q = Volumetric flow rate
- q = Heat flow
- q_R = Lower heating value
- R = Specific gas constant
- s = Entropy
- T = Temperature
- U = Blade velocity
- W = Mass flow rates
- w = Velocity vector in rotating frame
- ν = Kinematic viscosity
- Δ = Difference
- η = Efficiency
- τ = Torque
- γ = Specific heat ratio
- ω = Rotational speed in rad/sec = $2\pi N/60$

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
- u = Tangential direction
- s = Isentropic

APPENDIX A—
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. For mixtures of gases the following equations can be applied to determine c_p , R, and γ .

$$c_{p,ex} = \frac{1}{100} \sum c_{p,i} m_i \tag{A-1}$$

$$R_{ex} = \frac{1}{100} \sum R_i m_i \tag{A-2}$$

$$\gamma_{ex} = \frac{c_{p,ex}}{c_{p,ex} - R_{ex}} \tag{A-3}$$

m_i is the mole fraction of the individual component.

The main constituents of the exhaust gas are nitrogen, oxygen, carbon dioxide, and water. The specific heat (c_p) and gas constants (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 are known. Table A-1 gives γ , c_p , and R of the above substances at 10 bar, 800°C.

Table A-1. Specific Heat (c_p), Ratio of Specific Heats (γ) and Gas Constants (R) for Some Components of Exhaust Gas.

	c_p (kJ/kgK)	R(kJ/kgK)	$\gamma = 1/(1-R/c_p)$
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:

$$\Delta T = \frac{\Delta h}{c_{p,ex}} = \frac{\eta_t \cdot \Delta h_s}{c_{p,ex}} = \eta_t T_3 \left(1 - \left(\frac{p_4}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right) \tag{A-4}$$

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 depends on the exhaust gas composition, which means that different fuel compositions (if the differences are very large) can influence the engine match.

APPENDIX B—
ALGORITHM FOR SINGLE-SHAFT ENGINES

The equations:

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \tag{B-1}$$

$$\frac{\Delta T_t}{T_3} = \eta_t \left[1 - \left(\frac{p_7}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (\text{B-2})$$

$$\frac{N}{\sqrt{T_3}} = \frac{N}{\sqrt{T_1}} \cdot \sqrt{\frac{T_1}{T_3}} \quad (\text{B-3})$$

are all coupled by the firing temperature ratio T_3/T_1 (Cohen, et al., 1996). This ratio has to be determined by trial and error for operation at any arbitrary point of the compressor characteristic using the following method:

1. On the compressor map, chose any point on the $N/\sqrt{T_1}$ line. The characteristics then yield p_2/p_1 , $W\sqrt{T_1}/p_1$, $\Delta T_c/T_1$.
2. Guess a value of p_3/p_5 . The turbine characteristic then gives $W\sqrt{T_3}/p_3$, and T_3/T_1 can be obtained from the second equation, and furthermore allows the calculation of $N/\sqrt{T_3}$.
3. With the turbine efficiency from the map, $\Delta T_t/T_3$ can be calculated and, using the power balance equation, another ratio T_3/T_1 results.
4. Because this ratio T_3/T_1 normally is not the same as the initial one, a new ratio p_3/p_5 has to be guessed, until the same ratio T_3/T_1 is obtained. At this point, a compatible operating point is found, if the resulting T_3 is within the allowed operating limits of the engine.
5. The power output finally becomes the difference between turbine power and compressor absorbed power:

$$P_{pt} = (c_{p,t} \Delta T_t - c_{p,c} \Delta T_c) \cdot \frac{1}{\eta_m} \cdot W = (c_{p,t} T_3 \eta_t \left[1 - \left(\frac{p_3}{p_a} \right)^{\frac{\gamma-1}{\gamma}} \right] - c_{p,c} T_1 \frac{1}{\eta_c \eta_m} \left[\left(\frac{p_2}{p_a} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]) \cdot W \quad (\text{B-4})$$

Using the above method, one can determine the performance of a single-shaft gas turbine at any operating point. Note that the above method works for engines with or without variable geometry. The difference would be introduced by the component maps.

APPENDIX C—

ALGORITHM FOR TWO-SHAFT ENGINES

The equations (assume $p_7 = p_1$ and $W_c = W_t = W$)

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (\text{C-1})$$

$$\frac{\Delta T_t}{T_3} = \eta_t \left[1 - \left(\frac{p_5}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (\text{C-2})$$

$$c_{p,c} \cdot \frac{\Delta T_c}{T_1} \cdot \frac{1}{\eta_m} = c_{p,t} \cdot \frac{\Delta T_t}{T_3} \cdot \frac{T_3}{T_1} \quad (\text{C-3})$$

$$\frac{N}{\sqrt{T_3}} = \frac{N}{\sqrt{T_1}} \cdot \sqrt{\frac{T_1}{T_3}} \quad (\text{C-4})$$

are all coupled by the firing temperature ratio T_3/T_1 (Cohen, et al., 1996). This ratio can be determined by trial and error for operation at any arbitrary point of the compressor characteristic, using the following method:

1. In the compressor map, chose any point on the $N/\sqrt{(T_1)}$ line. The characteristics then yield p_2/p_1 , $W\sqrt{(T_1)}/p_1$, and $\Delta T_c/T_1$ from above.

2. Guess a value for p_3/p_5 . The turbine characteristic gives $W\sqrt{(T_3)}/p_3$, and T_3/T_1 can be obtained from the second equation. Furthermore, this characteristic allows calculation of $N/\sqrt{(T_3)}$, using the same speed N as previously for the compressor.

3. With the turbine efficiency from the map, $\Delta T_t/T_3$ can be calculated and, using the power balance equation, another ratio T_3/T_1 results. Because this ratio T_3/T_1 normally is not the same as the initial one, a new ratio p_3/p_5 has to be guessed, until the same ratio T_3/T_1 is obtained. At this point, a compatible operating point is found, if the resulting T_3 is within the allowed operating limits of the engine.

4. For the power turbine, the following relations apply:

$$\frac{W\sqrt{T_5}}{p_5} = \frac{W\sqrt{T_1}}{p_1} \cdot \frac{p_1}{p_5} \cdot \sqrt{\frac{T_5}{T_1}} \quad (\text{C-5})$$

$$\frac{\Delta T_{pt}}{T_5} = \eta_{pt} \left[1 - \left(\frac{p_a}{p_5} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (\text{C-6})$$

This condition adds another constraint, because the available pressure ratio p_5/p_1 has to be sufficient to move the mass flow W through the power turbine section. If the available pressure ratio is too low, another operating point at a lower mass flow (i.e., lower gas generator speed) has to be selected.

5. Combining the relationships for the gas generator and the power turbine yields:

$$\frac{W\sqrt{T_5}}{p_5} = \frac{W\sqrt{T_3}}{p_3} \cdot \frac{p_3}{p_5} \cdot \sqrt{\frac{T_5}{T_3}} \quad (\text{C-7})$$

$$\sqrt{\frac{T_5}{T_3}} = \sqrt{1 - \eta_t \left[1 - \left(\frac{p_5}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (\text{C-8})$$

$$\frac{p_5}{p_a} = \frac{p_2}{p_a} \cdot \frac{p_3}{p_2} \cdot \frac{p_5}{p_3} \quad (\text{C-9})$$

6. Finally, the engine output is the power that the power turbine generates at the available gas generator exit temperature and the pressure ratio over the power turbine:

$$P_{pt} = c_p \cdot \Delta T_{pt} \cdot W = c_p T_5 \eta_{pt} \left[1 - \left(\frac{p_7}{p_5} \right)^{\frac{\gamma-1}{\gamma}} \right] W \quad (\text{C-10})$$

Note that the power turbine efficiency depends strongly on the actual power turbine speed deviation from its optimum speed.

REFERENCES

- API Standard 616, 1998, "Gas Turbines for Refinery Service," Fourth Edition, American Petroleum Institute, Washington, D.C.
- Brun, K. and Kurz, R., 1998, "Measurement Uncertainties Encountered During Gas Turbine Driven Compressor Field Testing," ASME Paper 98-GT-1.
- Cohen, H., Rogers, G. F. C., and Saravanamuttoo, H. I. H., 1996, *Gas Turbine Theory*, Longman, Harlow.
- Kurz, R., 1991, "Transonic Flow through Turbine Cascades with 3 Different Pitch-to-Chord Ratios," Proc. 10. ISABE, Nottingham, United Kingdom.
- Kurz, R., Brun, K., and Legrand, D. D., 1999, "Field Performance Testing of Gas Turbine Driven Compressor Sets," *Proceedings of the Twenty-Eighth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 213-230.