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TUTORIAL: GAS TURBINE PERFORMANCE

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ABSTRACT

The power and efficiency characteristics of a gas turbine are the result of a complex interaction of different turbo machines and a combustion system.

In this tutorial, we will address the basic characteristics of each of the components in a gas turbine (compressor, gas generator turbine, power turbine) and the impact of typical control limits and control concepts. The goal is to provide explanations for the operational characteristics of typical industrial gas turbines, emphasizing the interaction between the gas turbine components. The concept of component matching is explained. 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

In particular, the following topics will be discussed:

- The gas turbine as a system
- Thermodynamics and aerodynamics
- 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
- Thermo dynamical 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.

The concepts developed will be used to derive basic principles for successful condition monitoring and performance testing of gas turbines.

INTRODUCTION

Gas turbines have been used for many aerospace and industrial applications for many years (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 turbo machines and a combustion system.

Figure 1. Gas turbine applications

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.

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 turbine (Brayton cycle, Figure 3). The turbine provides the necessary power to operate the compressor.

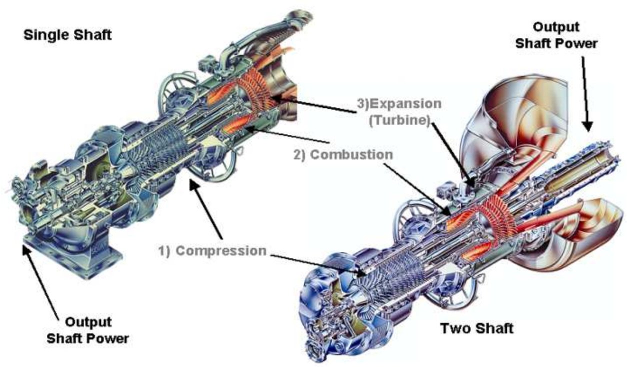

Figure 2. Single-shaft (cold end drive) and two-shaft (hot end drive) gas turbines

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 to 2 and 3 to 7 describes the work output of the turbine, i.e., most of the work generated by the expansion 3 to 7 is used to provide the work 1 to 2 to drive the compressor.

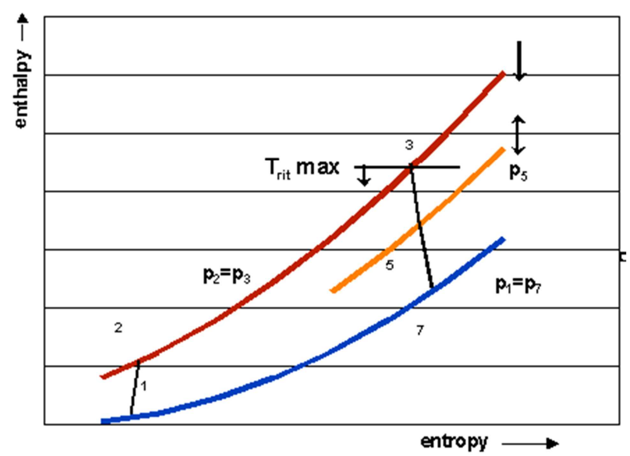

Figure 3. Enthalpy-entropy diagram for the Brayton cycle

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. Lines 5 to 7 describe the work output of the power turbine.

For a perfect gas, enthalpy and temperature are related by

$$\Delta h = c_p \Delta T$$

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.

We can thus describe the entire process, 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.

$$-c_{p_a}(T_2 - T_1) + c_{pe}(T_3 - T_7) = P/W$$
$$c_{pe}(T_3 - T_2) = E_f W_f / W$$

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

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, we can re-arrange the equation above to find:

$$-c_{p_a}(T_2 - T_1) = c_{pe}(T_3 - T_5)$$
$$c_{pe}(T_5 - T_7) = P/W$$

This relationship neglects mechanical losses(which are in the order of 1%) 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 don't 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

$$P = \frac{\Delta h}{W} = \frac{c_p}{W}(T_2 - T_1) = \frac{c_p}{W}\frac{T_1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

and the turbine

$$P = \frac{\Delta h}{W} = \frac{c_p}{W}(T_3 - T_7) = \frac{c_p}{W}\cdot\eta_t\cdot T_3\left[1 - \left(\frac{p_7}{p_3}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

In the two equations, ideal gas laws are assumed.

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

$$\eta_{th} = \frac{P}{W_f E_f}$$

and the heat rate is

$$HR = \frac{1}{\eta_{th}} = \frac{W_f E_f}{P}$$

In this paper, T_3, TIT, and TRIT will be (loosely) referenced as firing temperatures. The differences which lie simply in 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 GT cycle.

A QUICK EXCURSION TO AERODYNAMICS

Any gas turbine consists of several turbo machines. 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 the conversion of kinetic flow energy (i.e., velocity) into pressure or vice versa.

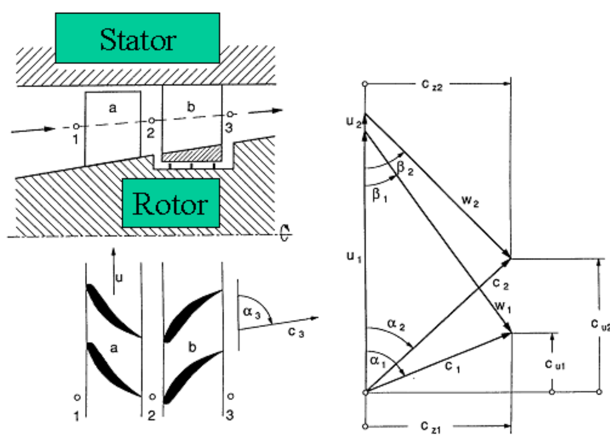

Figure 4. Velocities in a typical compressor stage

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

$$\Delta h = u_2 c_{u2} - u_1 c_{u1}$$

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

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

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.,

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

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

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

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 the Mach number of a turbo machine, 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:

$$M_n = \frac{u}{\sqrt{\gamma RT}} = \frac{2\pi DN}{\sqrt{\gamma RT}}$$

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:

- 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, we 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:

$$N_{corr} = \frac{N}{\sqrt{T / T_{ref}}}$$

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, we will also use this simplified expression N/√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).

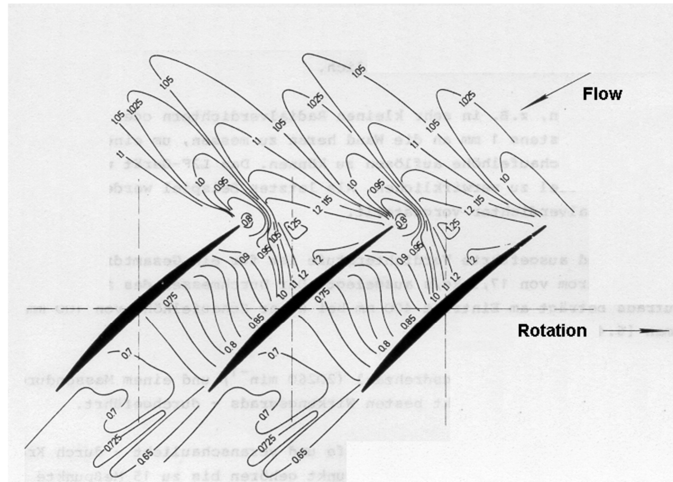

Figure 5. Mach number distribution for typical transonic compressor blades. The flow enters at supersonic speeds and is decelerated to subsonic speeds at the exit (Schodl, 1977)

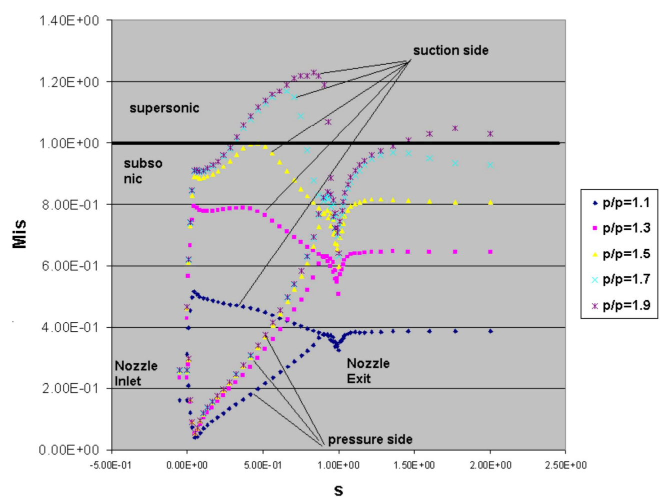

Figure 6. Velocity distribution in a turbine nozzle at different pressure ratios. As soon as the maximum local flow velocity exceeds Mach 1 (at a pressure ratio of 1.5 in this example), the inlet flow can no longer be increased (Kurz, 1991).

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. Figure 7 and Figure 8 with the compressor 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).

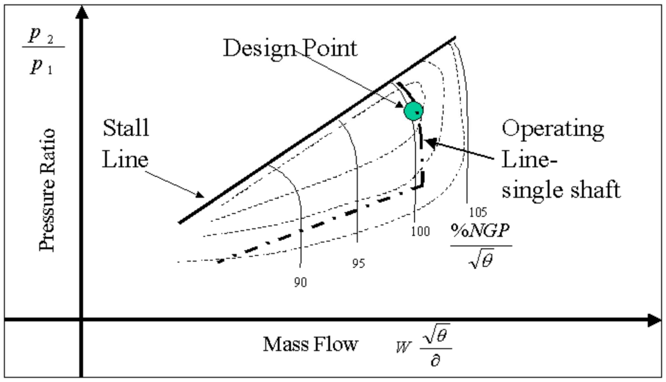

Figure 7. Typical compressor performance map with operating lines for a single-shaft engine

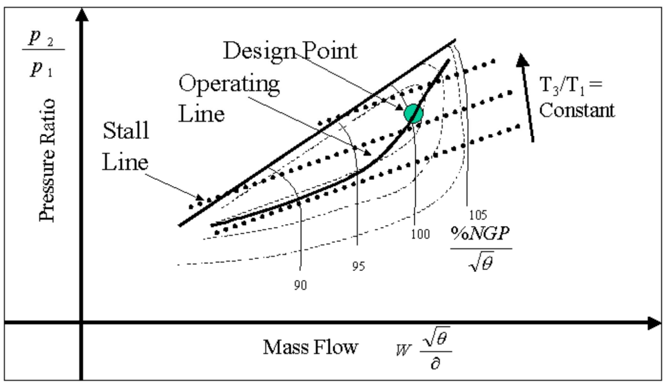

Figure 8. Typical compressor performance map with operating lines for a two-shaft engine

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

$$\frac{N_{GP,op1}}{\sqrt{T_{1,op1}}} = \frac{N_{GP,op2}}{\sqrt{T_{1,op2}}}$$

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, we require for the firing temperature:

$$\frac{T_{3,op1}}{T_{1,op1}} = \frac{T_{3,op2}}{T_{1,op2}}$$

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 by 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 engine, 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%. 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 to accomplish the cooling (Figure 9): The air is pushed through the inside of the blade with goal to remove 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 though 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, 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.

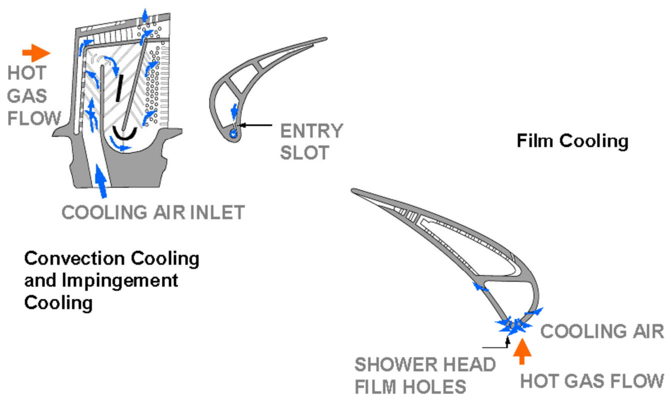

Figure 9. Typical cooling arrangements: Convection and impingement cooling (left) and film cooling (right).

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:

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

The heat capacity $\tilde{c}_{pe}$ in the equation above is a suitable average heat capacity. Modern combustors convert the energy stored in the fuel almost completely into heat (Typical combustion efficiencies for natural gas burning engines range above 99.9%). This is also evident from the fact that the results of incomplete combustion, namely CO and unburned hydrocarbons, are emitted only in the parts-per-million level.

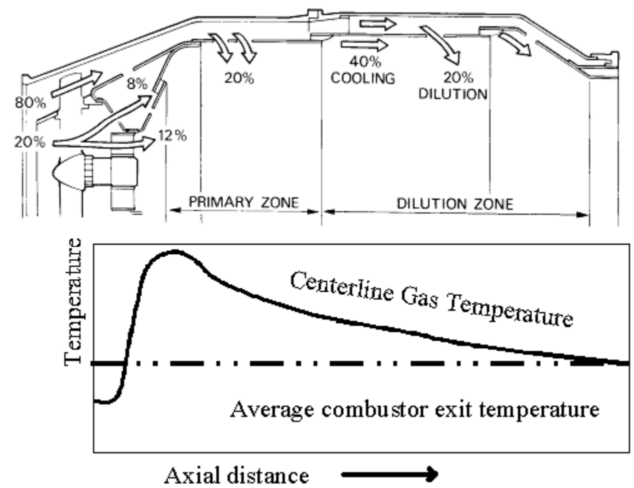

Figure 10. Axial temperature distribution in a combustor

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 to reduce the gas temperature lead to an acceptable combustor exit temperature. The flow will also incur a pressure loss due to friction and mixing.

Lean Premix Combustion (LPM) Systems

To further reduce the NOx emissions of gas turbines, Lean Premix combustion systems were developed.

The general idea behind any Lean Premix 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 NOx production (Figure 11).

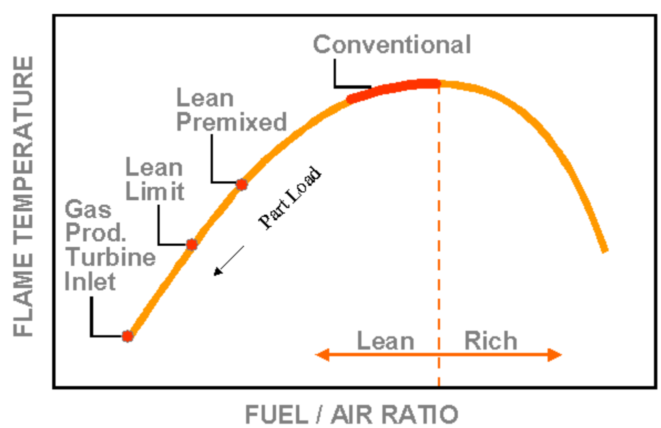

Figure 11. Flame temperature as a function to fuel-to-air ratio

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 unburned hydrocarbons (UHC). The necessity to control the fuel-to-air ratio closely yields different part-load behavior when comparing gas turbines with conventional combustors and LPM engines.[1] At certain levels of part load, LPM 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 LPM engine. This sounds paradoxical because the amount of air available at the combustor in part-load operation has to be less for a LPM engine (to maintain the fuel-to-air ratio) than for an engine with conventional combustion. However, due to the bleeding of air in a LPM 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 LPM engines. Once the bleed valve opens, the part-load efficiency of a LPM 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 LPM 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 LPM engines are always lower than for engines with conventional combustion.

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 ppm basis) in LPM 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. But 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.

[1]Regarding the requirements for Lean Premix engines, multi-spool engines show no fundamental differences from single-spool engines.

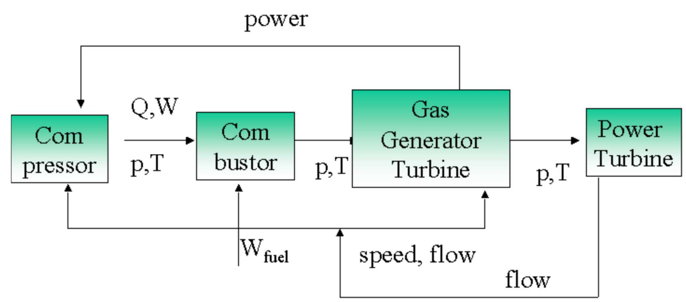

Figure 12. Interaction between the gas turbine components

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

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 both 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, 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

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 to 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. We 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 Figure 7 and Figure 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 (Figure 7 and Figure 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.[2] 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 GP turbine has to be high enough to force the flow through the power turbine.

Typical compressor and turbine maps are shown in Figure 8 and Figure 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.

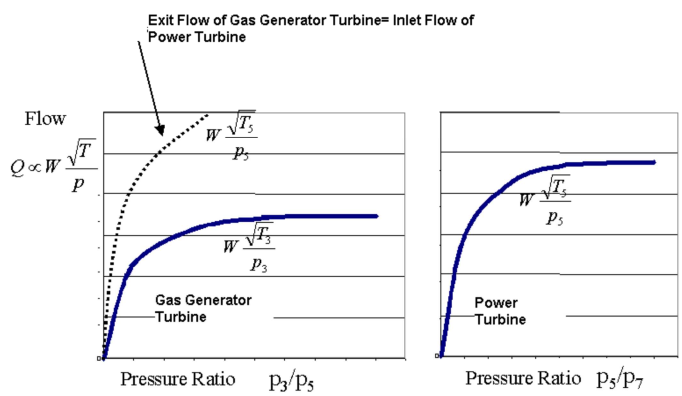

Figure 13. Schematic turbine performance map for two turbines (gas generator and power turbine) in series

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:

$$\frac{p_5}{p_a} = \frac{p_2}{p_a}\frac{p_3}{p_2}\frac{p_5}{p_3}$$

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 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

[2]Some engines are configured as multi-spool 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 LP compressor and turbine rotates inside the shaft connecting the HP compressor and turbine. In general, all the operating characteristics described above also apply to these engines.

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:

1. Increasing the firing temperature (without changing speed or geometry) will lead to a lower mass flow.
2. Increasing the gas generator speed, thus increasing p_2 and p_3, will allow for a larger mass flow.
3. 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 IGV's 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 IGV's 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 to trim the engine such that the rated T_3 is always reached at full corrected NGP. Therefore, at high ambient temperatures, when the gas generator would normally slow down, IGV's 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) we see that the compressor in a two-shaft engine operates for most of the load points close to its best efficiency.

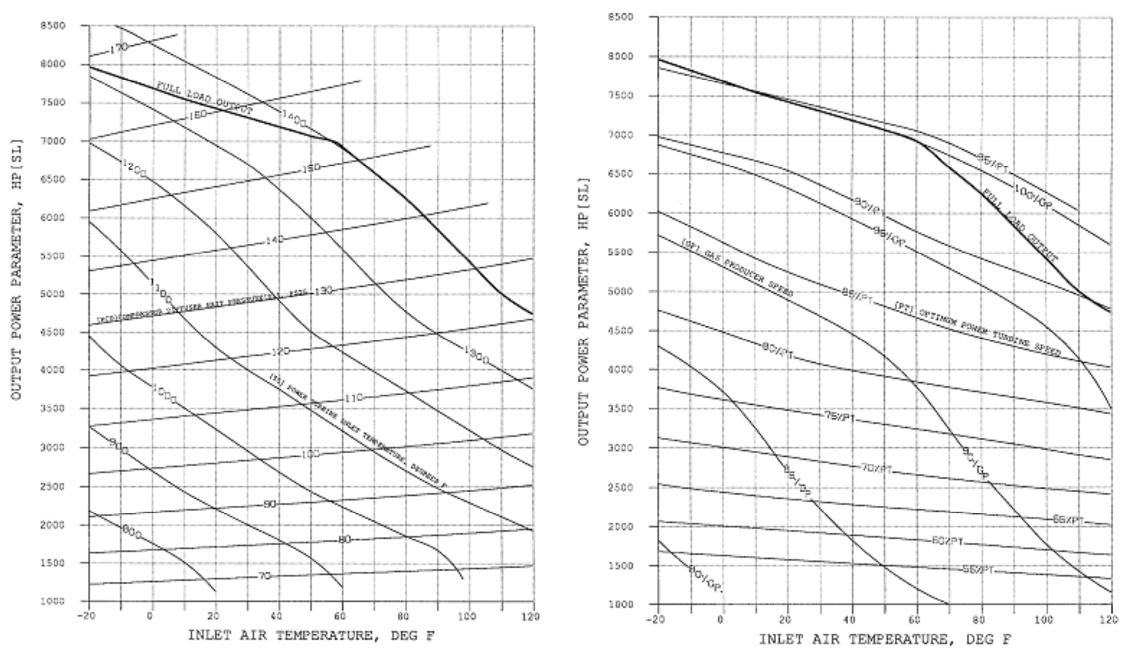

Figure 14. Two-shaft gas turbine performance map

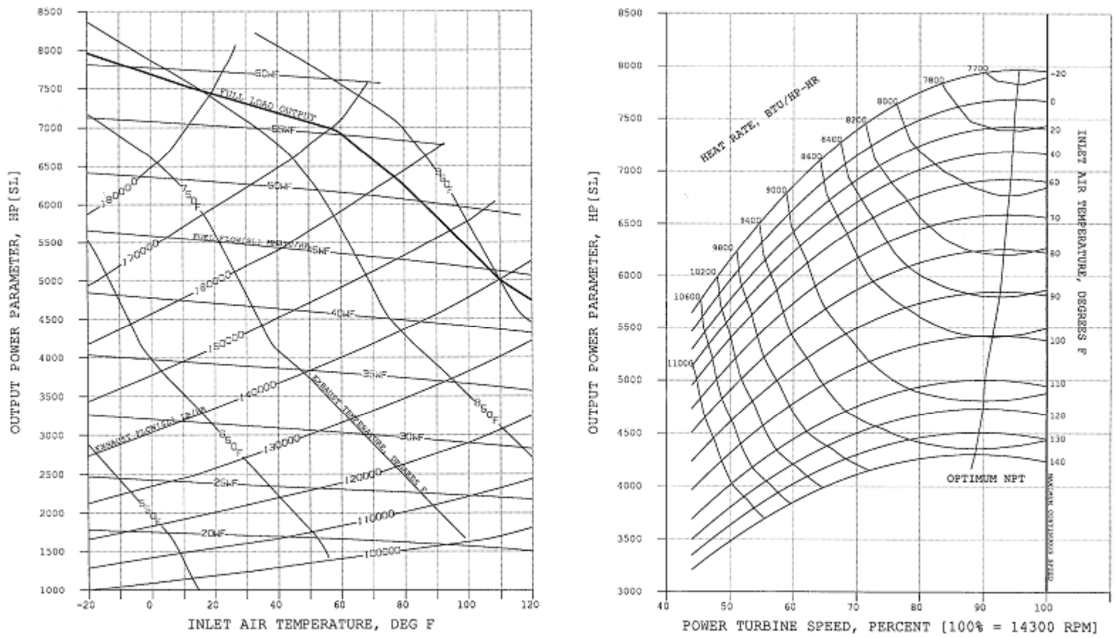

Figure 15. Two-shaft gas turbine performance map

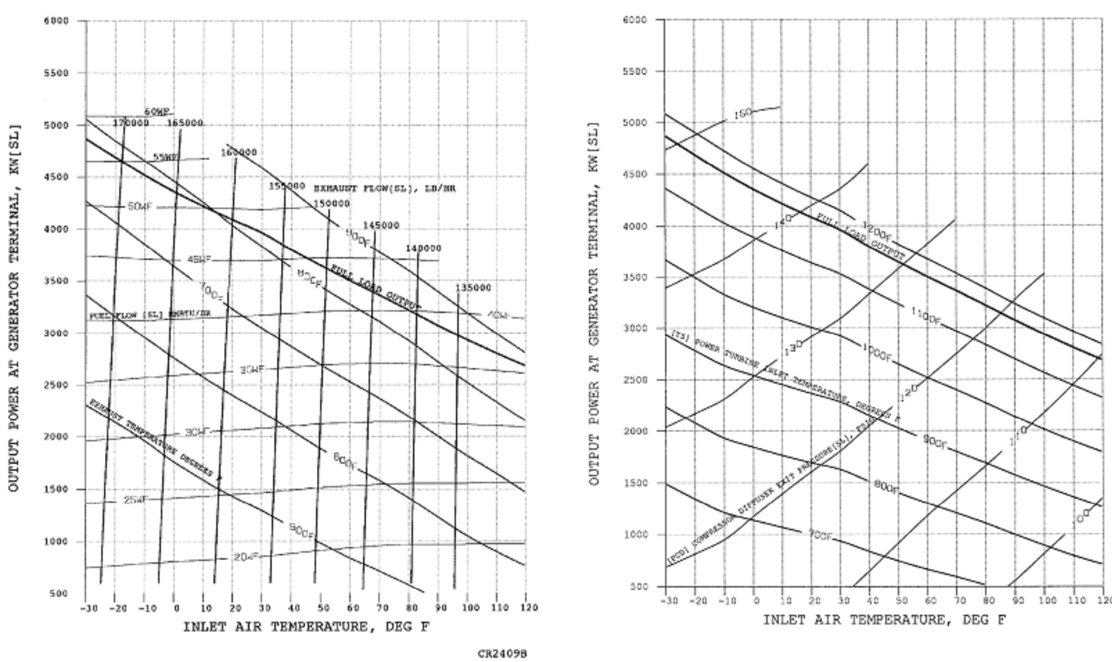

Figure 16. Single-shaft gas turbine performance map

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
- 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 and Figure 15)

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 the 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 turbine to exhibit an altered match temperature:

Gas fuel with a low heating value or water injection increases 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 power and torque. This type of turbine is only of limited use for mechanical drive purposes. However,the two-shaft engine, having a free power turbine, 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 regardless 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 five or six percent over a speed range from half to full speed.

Load

Any gas turbine will experience a reduced efficiency at part load (Figure 17). 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 is 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.

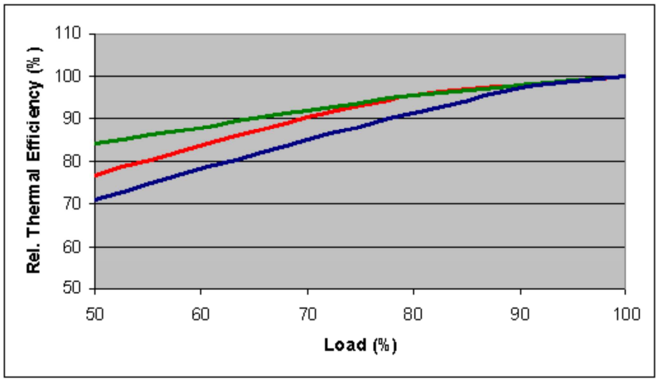

Figure 17. Thermal efficiency of typical industrial gas turbines as a function of load

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 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 Lean Preix (LPM) combustion and conventional combustion are very similar. The need to bleed combustion air for two-shaft engines typically leads to lower part load efficiency for engines equipped with LPM 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 equilibrium where the following requirements have to be met:

1. The compressor power equals gas generator turbine power. This determines the available pressure upstream of PT.
2. 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 DryLowNox 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 Q5 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 this the optimum power turbine speed, the power output and the efficiency of the power turbine will be lower (Figure 15). The impact of changing the power turbine speed is easily described by:

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

This equation can be derived from basic relationships (Brun and Kurz, 2000) 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 15). The heat rate becomes for a constant gas generator operating point:

$$\frac{HR}{HR_{opt}} = \frac{P_{opt}}{P}$$

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) do 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:

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

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 the 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 flame-out (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 flame-out 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 flame-out at part load or transient situations, for example

- bleeding air overboard
- using variable inlet guide vanes
- 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 the stage characteristics of compressor stages (see explanation on Euler Equations) because they change the head making capability of the stage by increasing or reducing the pre-swirl 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 T3). 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 (T5) 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 non-uniformly distributed in the reference planes (ie 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 (T7). 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 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:

1. 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.
2. 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_{Ggcorr} (ie 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 to 2 and 3 to 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 to 3, with a higher compressor discharge temperature and considering that the firing temperature T3 is limited, we see that less heat input is possible, ie., less fuel will be consumed .The expansion process has, due to the lower p2 = p3, 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, equilibrium will be found at a lower flow, and a lower pressure ratio, thus a reduced power output.

1. 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.
2. The relevant Reynolds number changes

At full load, single-shaft engines will run a 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 de-icing, fogging, evaporative cooling and other systems. The exhaust system may include a silencer, ducting, and waste heat recovery systems. All these system 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 21 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.

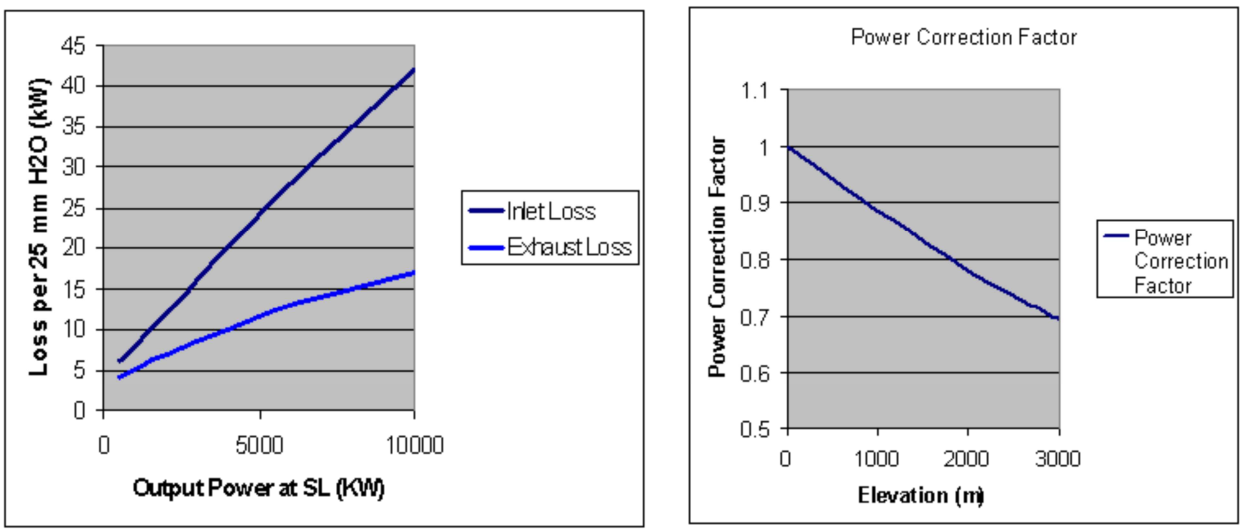

Figure 18. Correction factors for inlet losses, exhaust losses, and site elevation

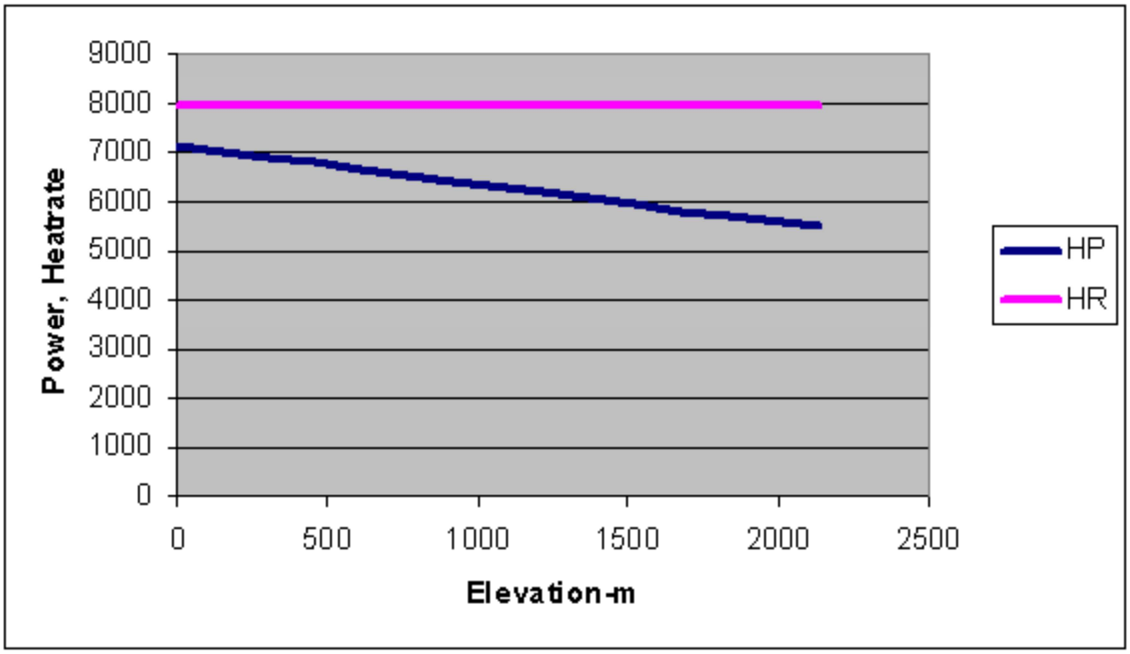

Figure 19. Power and heat rate as a function of site elevation (typical)

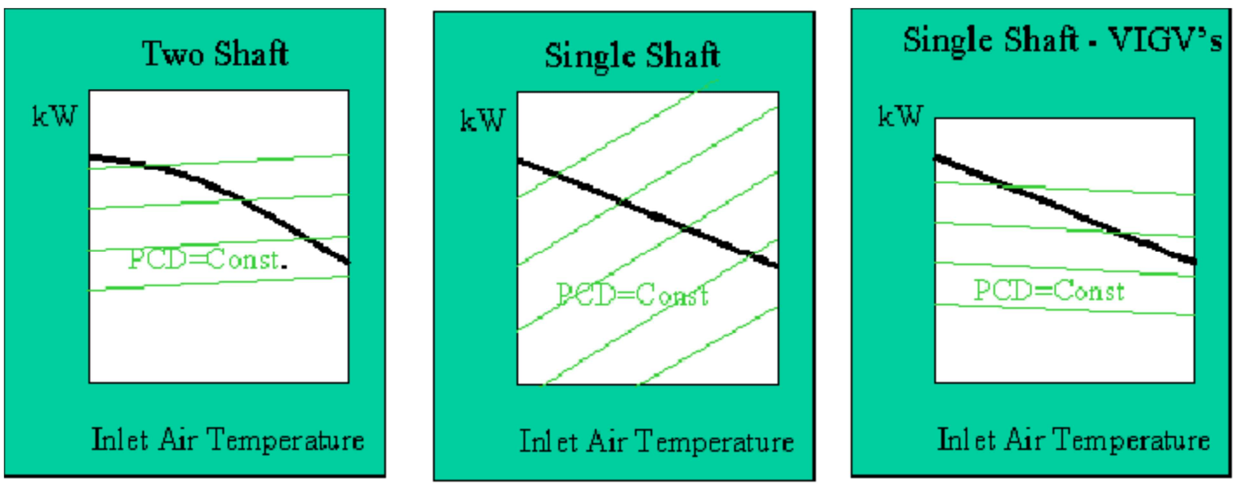

Figure 20. Effect of low fuel gas pressure on different engine designs. PCD is the pressure in the combustor.

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 (Figure 21 and Figure 19). 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.

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:

$$\delta = \frac{p_{ambient}(in\_'' Hg)}{29.929'' Hg}$$

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

$$p_a = p_{sealevel} \cdot e^{-\frac{elevation(ft)}{27200}}$$

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 CO_2 or N_2) has 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 1220.

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

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

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 though 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 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 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.

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 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.

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, 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 et al., 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 co-exist 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 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 start-up 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 towards 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 et al., 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%) or in terms of the specific humidity ($kg_{H20}/kg_{dry\ air}$). Figure 21 illustrates, this, relating relative humidity for a range of temperatures with the specific humidity.

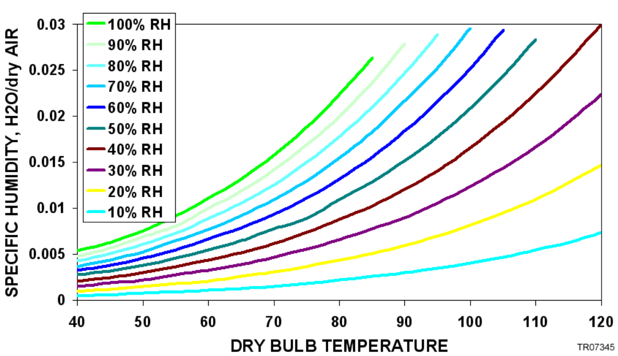

Figure 21. Graphic explanation of specific and relative humidity as a function of temperature

The main properties of concern that are affected by humidity changes are density, specific heat, and enthalpy. Because the molecular weight of water (18 g/mol) is less than dry air (28 g/mol), density of ambient air actually decreases with increasing humidity. When the density of the ambient air decreases the total mass flow will decrease, which then will decrease thermal efficiency and output power.

Performance of the combustor and turbines as a function of humidity is dominated by the changes in specific heat and enthalpy. Increases in water content will decrease temperatures during and after combustion (the same reason water is injected into the fuel to reduce NO_x levels).

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%) even at high temperatures of 38°C (100°F). The water content changes the thermodynamic properties of air (such as density and heat capacity) and thus causes a variety of changes in the engine.

For single-shaft engines, increasing humidity will decrease temperatures at the compressor exit. Humidity also causes decreased flame temperatures at a given fuel air ratio. As a result T2, combustor exit temperature, TRIT and T5 all decrease with an increase in humidity. Since the speed is set in single-shaft engines, the controls system will increase fuel flow in order to get T5 temperature up to the topping set point. Despite the increase in fuel flow, the total exhaust flow still decreases due to the decrease in airflow. Output power increases throughout the range of temperatures and humidity experienced by the engines, which shows that the increased fuel energy input has a greater influence on output power than does the decreased total flow.

In two-shaft engines, we have to distinguish whether the engine runs at maximum speed (NGP topped), or at maximum firing temperature (T5 topped). Increasing humidity will decrease air density and mass flow when running *NGP topped*, which will *decrease output power*. This is the general trend in output power noticed in all two-shaft engines when running NGP topped. As previously discussed, increased humidity causes lower T2, Flame temperature, TRIT, and T5 temperatures. When running *T5 topped*, the trend in output power reverses due to the engine increasing fuel flow to increase temperatures, and results in *increased output power*. So for two-shaft engines, output power will be seen to increase when running T5 topped, and to decrease when running NGP topped.

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. Today, these maps are usually embedded in software programs that allow the calculation of performance parameters of the engine.

Typical engine performance maps are shown in Figure 16 for single-shaft engines and in Figure 14 and Figure 15 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 to determine 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, we want to determine what the reason is for an engine to behave the way it does[3].

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 fine-tuned 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 fine-tuned 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 have on 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 build-up of material that causes increased surface roughness and to some degree changes the shape of the airfoil (if the material build up forms thicker layers of deposits). Many of the contaminants are smaller than 2 µm. Fouling can normally 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 aero components as scale. High temperature oxidation, on the other hand, is the chemical reaction between the components 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 aero engine 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 build up 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 non-recoverable degradation, the latter effect can at least be partially reversed by washing the compressor.

[3] 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 a 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 bin 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 15.

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 that 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 NON-RECOVERABLE DEGRADATION

The distinction between recoverable and non-recoverable 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 on-line 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 manufacturer’s recommendations regarding shut-down 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, or industrial exhausts will primarily effect 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 a steady state conditions. We should briefly discuss the engine operation during load transients, 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.

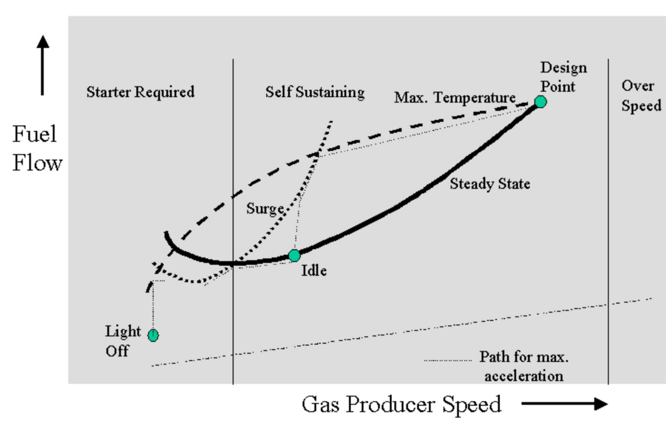

Figure 22. Start and acceleration map for a gas turbine

If the load suddenly drops, the maximum rate deceleration is limited by the flame out limits of the engine.

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	throughflow area
c	velocity vector in stationary frame
cp	specific heat
E_f	lower heating value
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
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 = 2?N/60

ACRONYMS

GG	-Gas generator
IGV	-Inlet guide vane
NGP	-Gas Generator Speed
PCD	-Gas Generator Compressor discharge pressure
PT	-Power turbine
TRIT	-Turbine rotor inlet temperature
VIGV	-Variable inlet guide vane

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

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APPENDIX A: GAS TURBINE CYCLE CALCULATION

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

p_a=14.73psia (1.013bara) p_2=147.3 psia (10.13bara) W= 100lbs/s (45 kg/s) η_c =85% T_a=100F(37.8°C) = 560R (311K)

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

pa=14.73psia (1.013bara) p3=147.3 psia (10.13bara) W= 100lbs/s (45 kg/s) η_c =85% T3=1600F(870°C) = 2060R (1144K)

We use the relationships for work H and power P:

$$H = c_p \Delta T$$

$$P = W \cdot H$$

and the gas properties for air: c_p=0.24BTU/lbR (1.007 kJ/kgK) ; γ=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

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

and the compressor discharge temperature is therefore

560R + 613R = 713°F (379°C).

This indicates, that the compressor consumed the work

H = .24 x 613 = 147 BTU/lb (= 1.007 x 341 = 344kJ/kg),

and the power

P= 100 lbs/s x 147BTU/lb=14700BTU/s= 20800hp (= 45 kg/s x 344kJ/kg =15480 kJ/s = 15480 kW)

The power extraction of the turbine causes a temperature drop

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

and thus an exhaust temperature of

T_7=1600-844= 756°F (=1144-469= 402°C)

Thereby extracting the work

H =.24 x 844 = 202 BTU/lb (=1.007 x 469 = 472 kJ/kg),

producing a power of

P=100 lbs/s x202BTU/lb=20200BTU/s = 28583hp (=45 kg/s x 472kJ/kg =21240 kJ/s = 21240 kW)

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

P_{net}= 28583hp-20800hp = 7783hp (= 21240kW-15480kW = 5760kW)

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

Q = W c_p ΔT = 100 x 0.24 x (1600-713) = 21300BTU/s = 76.7MMBTU/hr (= 45 x 1.007 x (870-379) = 22250 kJ/s = 80.1 GJ/hr)

With this, the engine heat rate can be calculated from

HR = 76.7/ 7783 = 9850 BTU/hphr (= 80.1/5760 = 13900 kJ/kWh),

and the thermal efficiency is

η_{th} =5760kW/22250kJ/s=25.9%.

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 (cp) and gas constant(R) of all these constituents are known, so it is easy to calculate the overall cp and γ, once the mole fractions of the constituents y_i are known:

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

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

The following table gives γ,cp and R of the above substances at 10 bar, 800°C:

	c_p (kJ/kgK)	R(kJ/kgK)	γ= 1/(1-R/ c_p)
Air	1.156	287	1.33
CO2	1.255	189	1.18
H2O	2.352	461	1.24

Table B-1: Specific heat (c_p), ratio of specific heats γ and gas constants (R) for some components of the exhaust gas

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

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

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 1st 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.