GAS TURBINES FOR ENERGY NETWORK SYMPOSIUM
BANFF, ALBERTA, CANADA
OCTOBER 2019

19-GTEN-201

GAS TURBINE PERFORMANCE FOR MECHANICAL DRIVE APPLICATIONS

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Keywords: *Gas Turbine, Performance, Aerodynamics, Thermodynamics*

Abstract
Industrial gas turbines show performance characteristics that distinctly depend on ambient and operating conditions. Application of these gas turbines, as well as the control and condition monitoring, require to consider the influence of site elevation, ambient temperature and relative humidity, but also by the speed of the driven equipment, the fuel, and the load conditions. It is further necessary to understand the performance characteristics of the gas turbine components and their interaction. The paper explains the performance characteristics based on the performance of the engine compressor, the combustor and the turbine section, and certain control strategies. It introduces fundamental concepts that help to understand the flow of energy between the components. Further discussed are control concepts.

Introduction

The industrial gas turbines discussed in this paper are two shaft designs, because the offer the capability to vary the speed of the driven equipment over a wide range, thus providing a effective and efficient method to adapt the driven equipment to changing operating conditions.

How Does a Gas Turbine Work?

Explanations of the working principles of a gas turbine have to start with the thermodynamic principles of the Brayton cycle, which essentially defines the requirements for the gas turbine components. Since the major components of a gas turbine perform based on aerodynamic principles, we will explain these, too. (Kurz et al [1], Brun et al.[2])

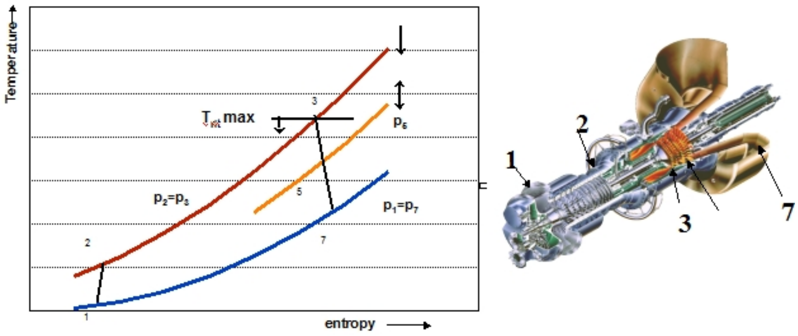

Figure 1: Brayton Cycle

The Brayton or gas turbine cycle (Figure 1) involves compression of air (or another working gas), the subsequent heating of this gas (either by injecting and burning a fuel or by indirectly heating the gas) without a change in pressure, followed by the expansion of the hot, pressurized gas. The compression process consumes power, while the expansion process extracts power from the gas. Some of the power from the expansion process can be used to drive the compression process. If the compression and expansion process are performed efficiently enough, the process will produce useable power output. The process is thus substantially different from a steam turbine (Rankine) cycle that does not require the compression process, but derives the pressure increase from external heating. The process is similar to processes used in Diesel or Otto reciprocating engines that also involve compression, combustion, and expansion. However, in a reciprocating engine, compression, combustion, and expansion occur at the same place (the cylinder), but sequentially, in a gas turbine, they occur in dedicated components, but all at the same time.

The major components of a gas turbine include the compressor, the combustor, and the turbine.

The compressor (usually an axial flow compressor, but some smaller gas turbines also use centrifugal compressors) compresses the air to several times atmospheric pressure. In the combustor, fuel is injected into the pressurized air from the compressor and burned, thus increasing the temperature. In the turbine section, energy is extracted from the hot pressurized gas, thus reducing pressure and temperature. A significant part of the turbine's energy (from 50 – 70 percent) is used to power the compressor, and the remaining power can be used to drive mechanical equipment (gas compressors and pumps). Industrial gas turbines for mechanical drives are built with a number of different arrangements for the major components:

- Two-shaft gas turbines consist of two sections: the gas producer (or gas generator) with the gas turbine compressor, the combustor, and the high pressure portion of the turbine on one shaft and a power turbine on a second shaft (Figure 2). In this configuration, the high pressure or gas producer turbine only drives the compressor, while the low pressure or power turbine, working on a separate shaft at speeds independent of the gas producer, can drive mechanical equipment.
- Multiple spool engines: Industrial gas turbines derived from aircraft engines sometimes have two compressor sections (the HP and the LP compressor), each driven by a separate turbine section (the LP compressor is driven by an LP

turbine by a shaft that rotates concentric within the shaft that is used for the HP turbine to drive the HP compressor), and running at different speeds. The energy left in the gas after this process is used to drive a power turbine (on a third, separate shaft), or the LP shaft is used as output shaft.

The energy conversion from mechanical work into the gas (in the compressor) and from energy in the gas back to mechanical energy (in the turbine) is performed by appropriately manipulating gas flows with stationary and rotating airfoils. Leonard Euler (in 1754) equated the torque produced by a turbine wheel to the change of circumferential momentum of a working fluid passing through the wheel. Somewhat earlier (in 1738), Daniel Bernoulli stated the principle that (in inviscid, subsonic flow) an increase in flow velocity is always accompanied by a reduction in static pressure and vice versa, as long as no external energy is introduced.

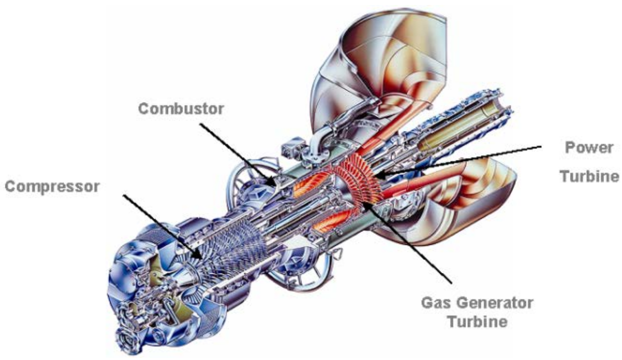

Figure 2: Typical Industrial Gas Turbine

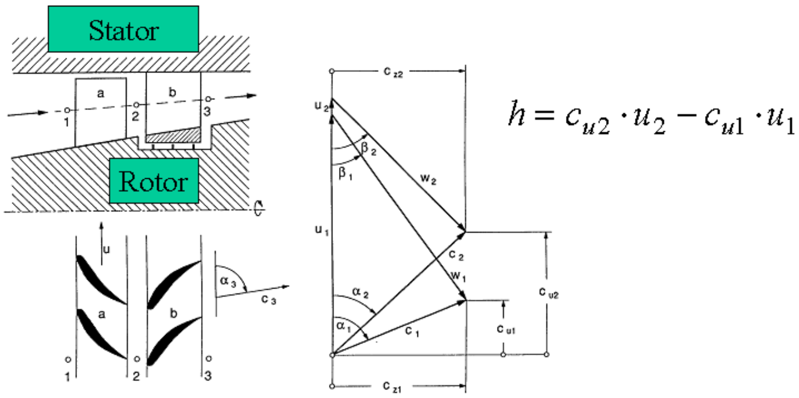

Figure 3: Velocities in a Typical Compressor Stage. Mechanical Work h Transferred to the Air is Determined by the Change in Circumferential Momentum of the Air.

While Euler's equation applies Newton's principles of action and reaction, Bernoulli's law is an application of the conservation of energy. These two principles explain the energy transfer in a turbomachinery stage (Figure 3).

The compressed air from the compressor enters the gas turbine **combustor**. Here, the fuel (natural gas, natural gas mixtures, hydrogen mixtures, diesel, kerosene, and many others) is injected into the pressurized air and burns in a continuous flame. The flame temperature is usually so high that any direct contact between the combustor material and the flame has to be avoided, and the combustor has to be cooled using air from the engine compressor. Additional air from the engine compressor is mixed into the combustion products for further cooling. Since the 1990s, combustion technology has focused on systems often referred to as dry low NO_x combustion, or lean-premix combustion. The idea behind these systems is to make sure that the mixture in the flame zone has a surplus of air, rather than allowing the flame to burn under stoichiometric conditions. This lean mixture, assuming the mixing has been done thoroughly, will burn at a lower flame temperature and thus, produce less NO_x.

The exhaust gases from the combustor, that is the combustion products and the applied cooling air, are now expanded through a gas producer turbine and a power turbine, again applying the aerodynamic principles described above.

The components of the gas turbine work together as follows, with initial focus on the gas generator only: The compressor and the turbine that drives the compressor run at the same speed, and the (speed dependent) power produced by the turbine has to match the (speed dependent) power absorbed by the compressor. The compressor operating point also has to be such that the compressor produces enough discharge pressure to push the mass flow through the turbine section. In most cases, the turbine nozzle is choked, which means the volumetric flow through the turbine nozzle is fixed. Therefore, if the firing temperature in the combustor is increased, the compressor discharge pressure has to increase also, to compensate for the gas density reduction as a result of the higher temperature. The energy left in the exhaust gas, after the gas producer turbine has satisfied the power needs of the compressor, is converted to mechanical energy by further expanding the gas through the power turbine (Figure 4).

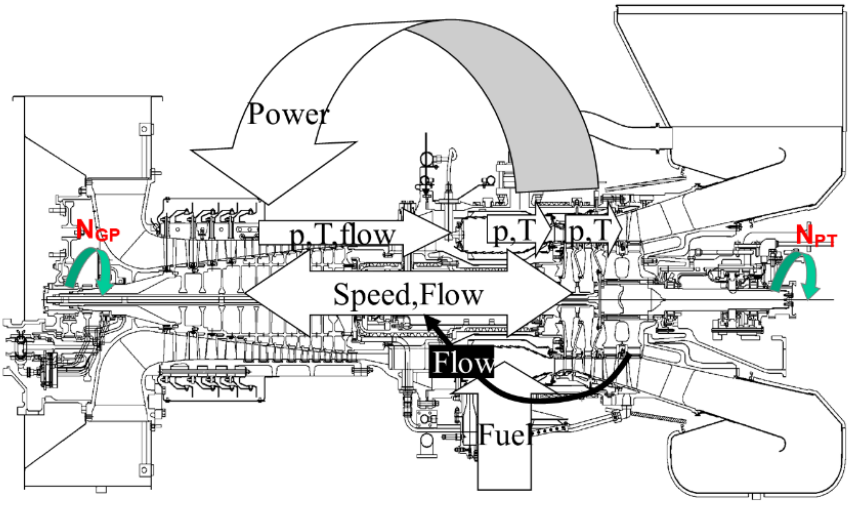

Figure 4: Gas turbine component interaction

The conversion of heat released by burning fuel into mechanical energy 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 1**). The turbine provides the necessary power to operate the compressor. Whatever power is left is used as the mechanical output of the engine. This thermodynamic cycle can be displayed in an enthalpy-entropy (h-s) diagram (**Figure 1**). The air is compressed in the engine compressor from state 1 to state 2. The heat

added in the combustor brings the cycle from 2 to 3. The hot gas is then expanded. In a single shaft turbine, the expansion is from 3 to 7, while in a two shaft engine, the gas is expanded from 3 to 5 in the gas generator turbine and afterwards from 5 to 7 in the power turbine. The difference between Lines 1-2 and 3-7 describes the work output of the turbine, i.e. most of the work generated by the expansion 3-7 is used to provide the work 1-2 to drive the compressor.
In a two shaft engine , the distances from 1 to 2 and from 3 to 5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 5-7 describes the work output of the power turbine.

In this paper, T3, 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.

The gas generator section is controlled by the amount of fuel that is supplied to the combustor. Its two operating constraints are the firing temperature (T.I.T., TRIT) and the maximum gas generator speed for two shaft engines. If the fuel flow is increased, both firing temperature and gas generator speed increase, until one of the two operating limits is reached.[1] The power turbine speed and load has no impact on this balance. 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.

Factors influencing the available power at the power turbine output shaft include:

- Ambient Temperature
- Ambient Pressure
- Power Turbine Speed
- Inlet / Exhaust Pressure Losses
- Fuel
- Accessory Loads
- Relative Humidity

[1] At the match temperature of the engine, both limits are reached at the same time. At ambient temperatures below the match temperature, the speed limit is reached first. At ambient temperatures above the match temperature, the firing temperature becomes the limiting factor.

Factors influencing the heat rate or efficiency of the engine include:

- Load
- Ambient Temperature
- Power Turbine Speed
- Inlet / Exhaust Pressure Losses
- Fuel
- Accessory Loads
- Ambient Pressure (indirectly)
- Relative Humidity

Dry Low NOx (DLN) engines employ additional means of control. The general idea behind any DLN combustor currently in service is to generate a thoroughly mixed lean fuel and air mixture prior to entering the combustor of the gas turbine. The lean mixture is responsible for a low flame temperature, which in turn yields lower rates of NOx production (Figure 5). 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). Therefore it is desirable to keep the combustor temperatures at part load at the a similar level as at full load.

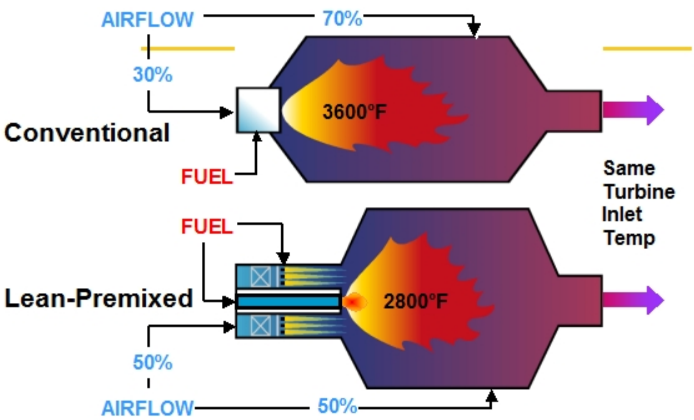

Figure 5: Conventional and Lean-Premix (DLN) Combustion Systems

The necessity to control the fuel-to-air ratio closely yields different part-load behavior when comparing gas turbines with conventional combustors and DLN engines (Stansel, [3]).[2] At certain levels of part load, DLN engines usually bleed a certain amount of air from the compressor exit directly into the exhaust duct. Therefore, while the airflow for any two-shaft engines is reduced at part load, the reduction in airflow is greater for a conventional combustion engine than for a DLN engine. This sounds paradoxical because the amount of air available at the combustor in part-load

[2]Regarding the requirements for DLN engines, multi-spool engines show no fundamental differences from single-spool engines.

operation has to be less for a DLN engine (to maintain the fuel-to-air ratio) than for an engine with conventional combustion. However, due to the bleeding of air in a DLN engine, the flow capacity of the turbine section is artificially increased by the bleeding duct.

The combustor exit temperature at part load drops significantly for engines with conventional combustion, while it stays high for DLN engines. Once the bleed valve opens, the part-load efficiency of a DLN engine drops faster than for an engine with conventional combustion. Since the opening of the bleed valve is driven by emissions considerations, it is not directly influenced by the load. Regarding emissions, the drop in combustor temperature in engines with conventional combustion, leading to a leaner fuel-to-air ratio, automatically leads to NOx emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel-to-air ratio. However, the NOx emissions levels of DLN engines are always lower than for engines with conventional combustion.

The Control of Gas Turbines

The primary control system for a gas turbine has as its main task to avoid unsafe, or damaging operating conditions for the gas turbine. This means it will prevent the gas turbine rotors to run too fast or too slow, it will limit the firing temperature, and will create alarm or shutdowns if vibration exceed acceptable limits. It may also prevent component pressures to increase beyond safe limits, are prevent situations where torque limits are exceeded. The safe operating range of a gas turbine creates a window. Within this window, the control systems sets the gas turbine operation according to the needs of the process.

Basic Process Control with a Gas turbine driver

The Control system for a gas turbine driver process control is set up to run the engine to maximum gas generator speed (i.e full load), unless it runs into another limit first. Limits can be for the driven compressor suction pressure, compressor discharge pressure or compressor flow. If, for example, suction pressure is controlled, the engine will run at full load unless the suction pressure drops below its set point. In that case, the gas producer speed is reduced. In the case of discharge pressure control or flow control, the engine will run at full load unless the discharge pressure or the compressor flow exceeds its set point.

For generator drives, the control is relatively simple: The goal of the control effort is to maintain a constant generator speed. The control system will increase the fuel flow to increase the power output if the generator speed drops, and it will reduce the fuel flow and power if the generator speed increases.

The interaction between compressor characteristic and system characteristic then becomes a basic ingredient for the control approach. Figure 6 shows how the power input provided by the driver can be used to control the compressor operating point within the constraint of the system behavior (Brun and Kurz,[4]).

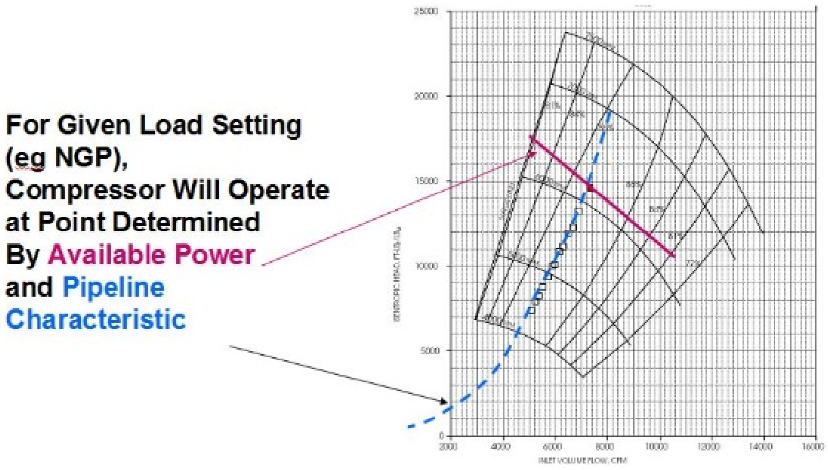

Figure 6: Available Power, Compressor Map and Pipeline Characteristic

Compressor power P is a function of mass flow W and actual head H, and thus related to the coordinates in the compressor map (inlet density ρ, inlet flow Q, isentropic head H_s and efficiency η):

$$P = W \cdot H = \rho Q \cdot \frac{H_s}{\eta}$$

This defines the line of constant power in Figure 6.

Off Design Performance

Change in ambient conditions and load conditions cause the gas turbine to operate at changing component conditions.

We will find that the ambient temperature (or, more precisely, the engine compressor inlet temperature) has a significant impact on both power and heat rate of the gas turbine. So does the site elevation (or, actually, the barometric pressure at site). We also find, that gas turbines are most efficient at full load, with a drop in efficiency at part load. In two shaft engines, the power turbine speed impact engine output and efficiency (Brun and Kurz [2]).

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. The off-design performance behavior is the result of the interaction between the various rotating components and the control system.

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, which in turn reduces the power output. The power output is proportional to the mass flow. At constant speed, where the volumetric flow remains approximately constant, the mass flow will increase with decreasing temperature and it 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. It also shows that the higher the inlet temperature, the more work (or head) is required to achieve a certain pressure rise in the compressor (Figure 6). The increased work has to be provided by the gas generator turbine. Therefore, less power is available from the power turbine, as can be seen in the enthalpy-entropy diagram (Figure 6). At the same time the machine Mach number at constant speed is reduced at higher ambient temperature, and vice versa, the Mach number of the engine compressor will increase for a given speed, if the ambient temperature is reduced. The gas generator turbine Mach number will increase for reduced firing temperature at constant gas generator speed.

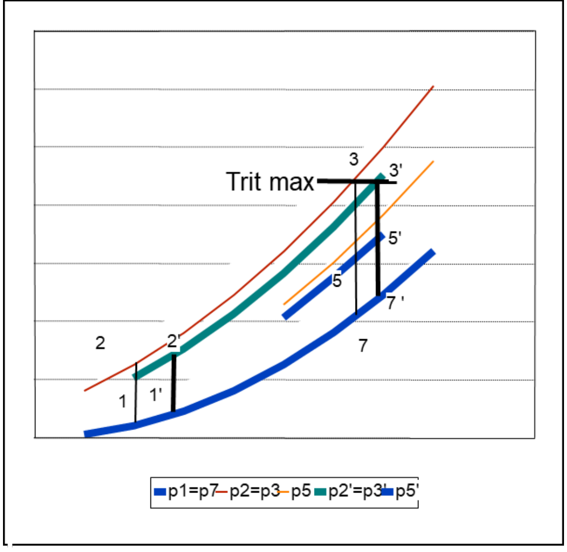

Figure 7: Brayton Cycle (Temperature over Entropy, with lines of constant pressure) at different inlet temperatures (inlet temperature 1' is higher than inlet temperature 1)

The Enthalpy-Entropy Diagram **(Figure 7)** describes the Brayton cycle for a two-shaft gas turbine. Lines 1-2 and 3-5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 5-7 describes the work output of the power turbine. 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 compressor discharge pressure, will be lower than before. Looking at the combustion process 2-3, with a higher compressor discharge temperature and considering that the firing temperature T_3 is limited, we see that less heat input is possible, ie. less fuel will be consumed. The expansion process has, due to the lower starting pressure p_3, less pressure ratio available. Thus, a larger portion 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 would increase at high ambient temperatures if the pressure ratio had to 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) can only be satisfied at a lower speed.
The lower gas generator speed N_{GG} often leads to a reduction of turbine efficiency: The inlet volumetric flow Q_3 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 to keep he gas generator speed constant at higher ambient temperatures, thus avoiding efficiency penalties.

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

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

4-The relevant Reynolds number changes. This usually does not cause any significant performance 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 (Figure 7).

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.

The full load power of a gas turbine is determined by one of two operational limits: Maximum firing temperature and maximum gas generator speed.

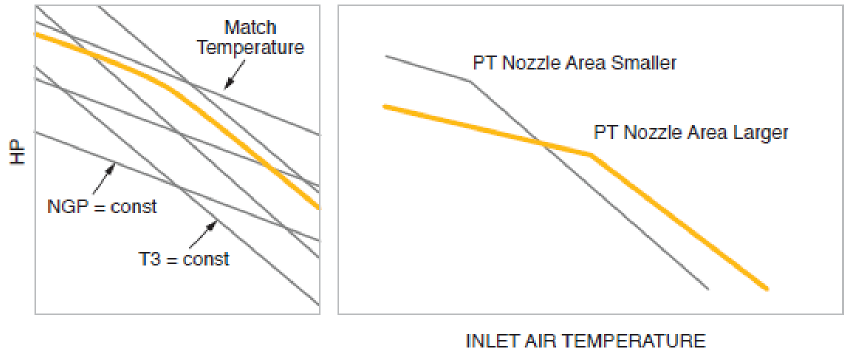

Figure 8: Full load power for a two shaft engine is limited by either maximum gas producer speed or maximum firing temperature.

While a single shaft engine runs at a defined speed, its maximum load is determined by the maximum firing temperature. A two shaft engine is either limited by maximum speed or maximum temperature, depending on ambient temperature (Figure 8). The match temperature is the ambient temperature where the engine will reach both limit simultaneously. This match temperature can be affected by the flow capacity of the power turbine nozzle (Figure 8). A nozzle with a larger flow area will move the match point to a higher ambient temperature. The statements above assume engines that have no adjustable geometry. As explained later, adjustable compressor vanes allow to maintain the maximum gas producer speed even at temperatures higher than the match temperature.

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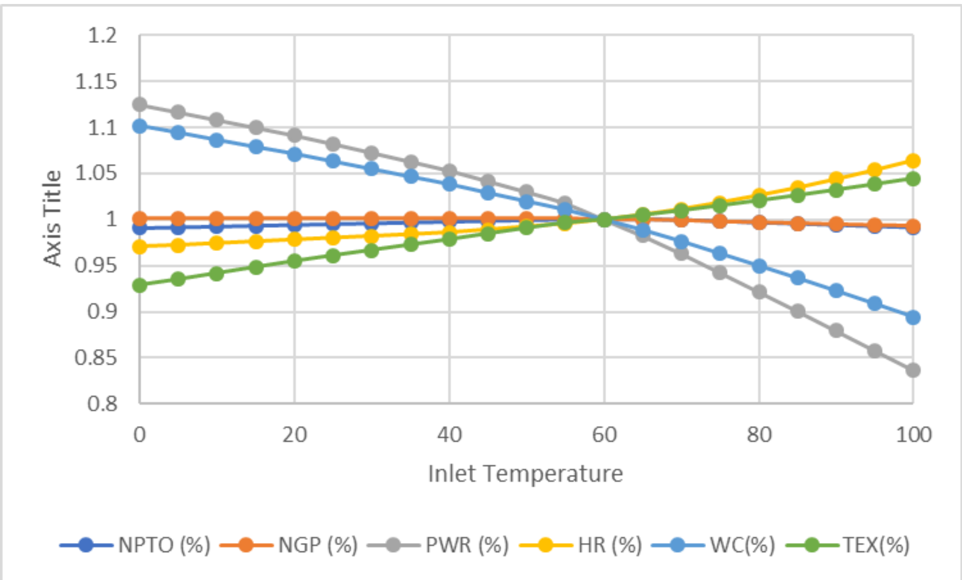

Figure 9: Full Load performance parameters, 2shaft engine, for ambient temperatures from 0°F to 100°F (-18°C to 38°C).

Figure 9 shows some of these details: The power changes more with the ambient temperature than the heat rate (see above). The optimum power turbine speed changes with ambient temperature. With increasing ambient temperature, inlet air mass flow is reduced (due to the reduced air density), and exhaust temperature increases. Not shown is the engine compressor discharge pressure. It will drop at higher ambient temperatures, because less mass flow has to pass the gas generator turbine nozzle

Part Load Operation

Running a two shaft engine in part load means that the gas generator speed is reduced. Depending on the control mode, the firing temperature and/or the airflow through the engine are reduced. This in turn will also lead to a reduction in compressor exit pressure ratio. Notably, the optimum power turbine speed is also reduced. This is advantageous in many applications where the gas turbine drives a gas compressor, which often will also run slower when the operating condition consumes less power (Figure 10).

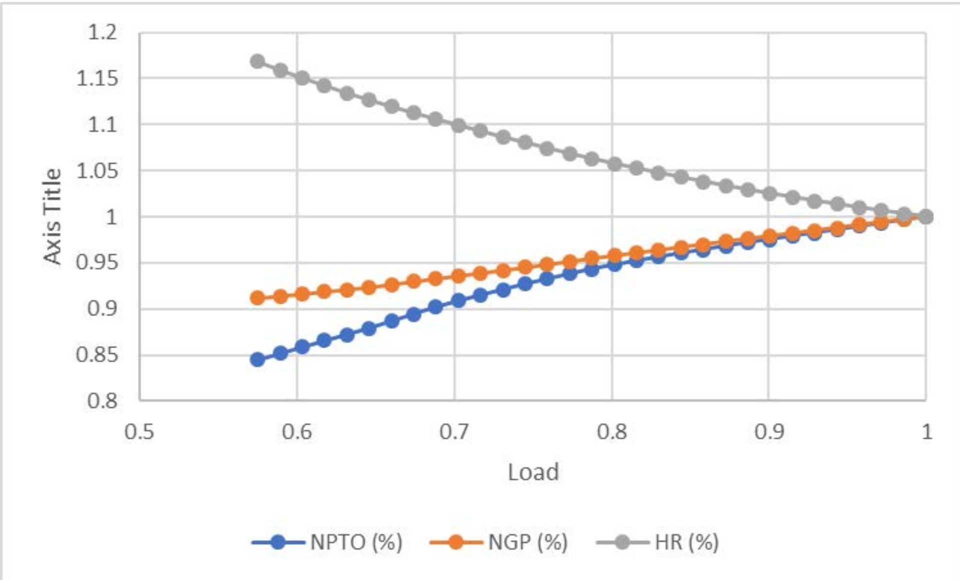

Figure 10: Two shaft gas turbine parameters at part load (50% to 100%)

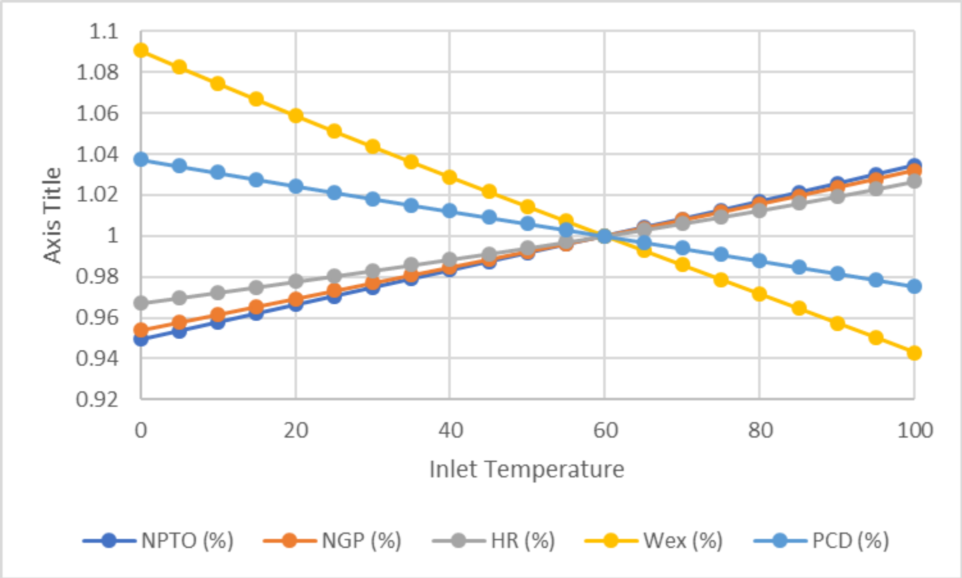

Figure 11: 2 shaft Engine at constant power output and varying ambient temperature from 0°F to 100°F (-18°C to 38°C), running at optimum power turbine speed.

For a two shaft engine running at constant power over a range of ambient temperatures (Figure 11), the gas producer speed increases with higher ambient temperatures, because the engine runs at higher relative load. Despite the lower gas producer speed at lower temperatures, the compressor discharge pressure rises, because the higher mass flow through the engine requires this rise to be able to push the flow through the choked gas producer nozzle. Notably, the optimum power turbine speed also changes.

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 (Wilcox et al.,[5]; Orhon et al.,[6]). The exhaust system may include a silencer, ducting, and waste heat recovery systems. Appropriate air filtration has been found to be the key to avoid engine performance degradation (Kurz et al.[7])
All these systems will cause pressure drops, i.e. the engine will actually see an inlet pressure that is lower than ambient pressure, and will exhaust against a pressure that is higher than the ambient pressure. These inevitable pressure losses in the inlet and exhaust system cause a reduction in power and cycle efficiency of the engine. The reduction in power, compared to an engine at ISO conditions, can be described by simple correction curves, which are usually supplied by the manufacturer. The ones shown in Figure 12 describe the power reduction for the respective inlet and exhaust pressure loss.

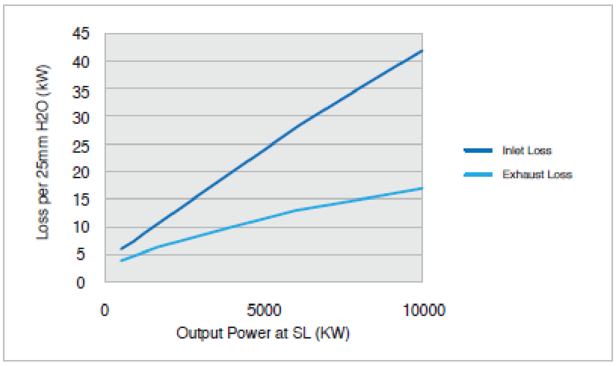

Figure 12: Impact of Inlet and Exhaust System pressure loss.

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. The engine, thus, experiences 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 of site elevation 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 at normal conditions is:

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

Fuel

Industrial gas turbines can use a wide variety of liquid and gas fuels [7]. 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 CO2 or N2) have a low Wobbe index, while substances with a large amount of heavier hydrocarbons have a high Wobbe index. Pure methane has a Wobbe index of about 1220.

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

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

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. The effect is to some degree counteracted by the fact that the compressor pressure ratio has to increase to push the additional flow through the choked turbine nozzle. 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 for both two shaft or single shaft engines.

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. If the available fuel gas pressure is too low for the engine to reach full load at a low ambient temperature it may be sufficient when 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 13) . 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 driven 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 firing temperature and a significant loss in power output, unless it uses VIGV's. With VIGV's, the compressor exit pressure, and thus the combustor pressure can also be influenced by the position of the VIGV's, thus leading to less power loss (Figure 13).
Without VIGV's, the only way to reduce the compressor discharge (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.

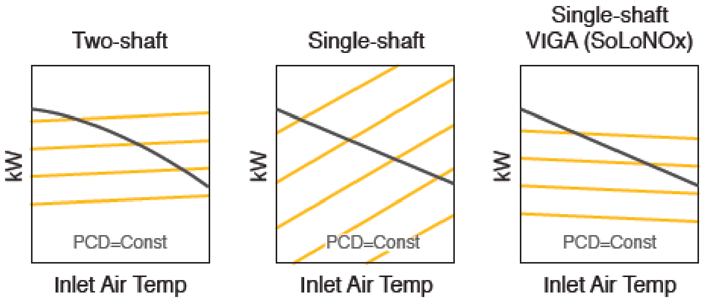

Figure 13: Fuel gas pressure and engine output.

Relative Humidity

The impact of humidity 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 14 illustrates this, relating relative humidity for a range of temperatures with the specific humidity.

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). Since the water content changes the thermodynamic properties of air (such as density and heat capacity), it causes a variety of changes in the engine, such that on some engines the output power is increased with increased humidity, while other engines show reduced performance at increased humidity.

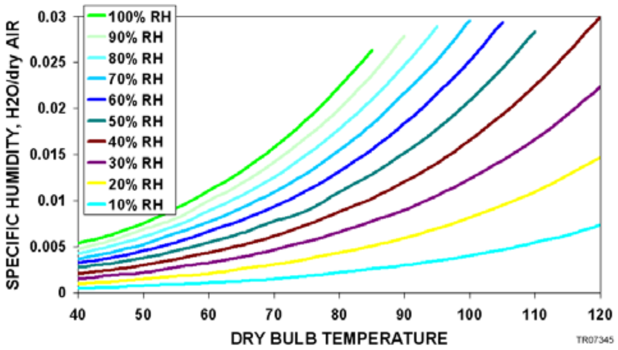

Figure 14: Specific and relative humidity as a function of temperature

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), the density of ambient air decreases with increasing humidity. Since the water concentration in the air for the same relative humidity increases with increasing temperature (Figure 14), 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).

When the density of the ambient air decreases the total mass flow will decrease, which then will decrease output power. The performance of the combustor and the 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. For the same reason water is injected into the fuel to reduce NO_x levels.

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, the combustor exit temperature, TRIT and T5 all decrease with an increase in humidity. Since the speed is constant 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.

Power Turbine Speed

The power turbine receives hot pressurized gas from the gas generator. The flow through the power turbine is then set by the flow capacity of the power turbine nozzle. The power turbine output, for a given operating point of the gas producer, depends on the speed of the power turbine (Figure 15). For any power turbine inlet pressure and flow, there is an optimum speed. At the optimum speed, the flow leaves the power turbine with little or no swirl, while at off-optimum speeds, the flow will have swirl. Therefore, the power turbine will extract less energy from the gas, which also leads to an increased exhaust temperature [2].

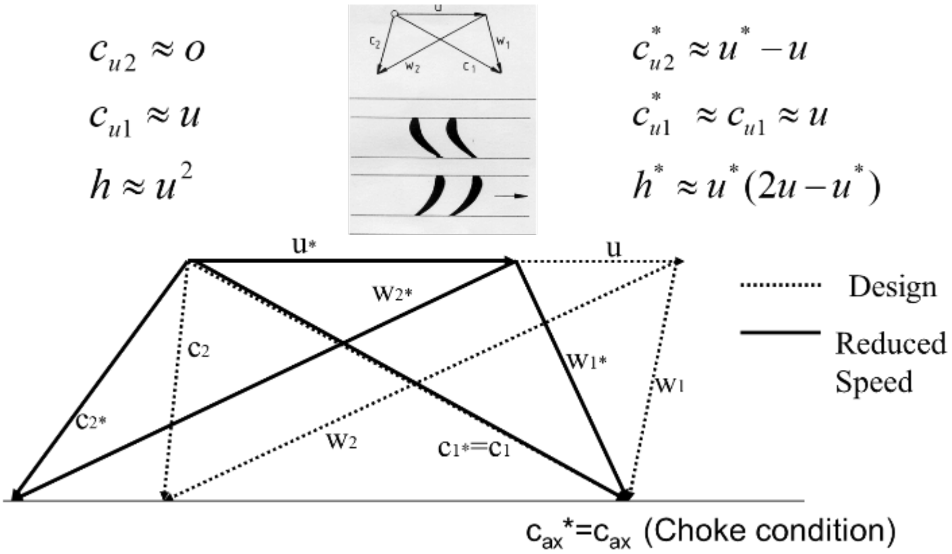

Figure 15: Off optimum power turbine speed.

From Figure 15 we see that the ratio work from the power turbine at the design point h_{opt} relative to some off-design speed h, for constant flow is:

$$\frac{h}{h_{opt}} = \frac{u(2u_{opt} - u\ )}{u_{opt}^2}$$

Constant flow is a valid assumption for a choked turbine nozzle. Thus, mass flow stays the same, thus the impact of changing the power turbine speed is easily described by:

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

Figure 16 shows this relationship.

The power turbine speed is then the result of the equilibrium between the speed dependent power of the power turbine, and the likewise speed dependent power consumption of the driven equipment (Figure 16).

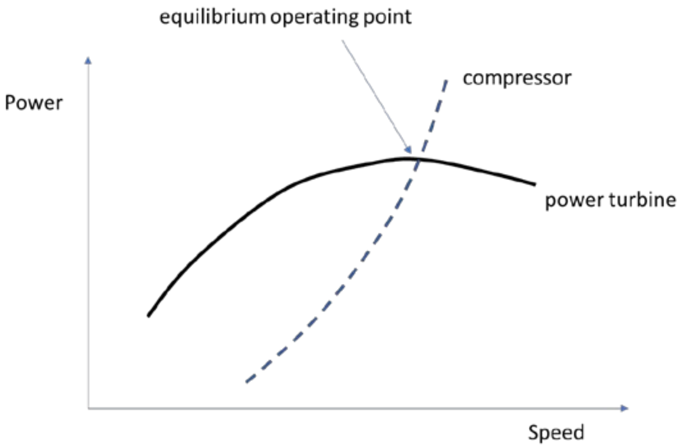

Figure 16: Speed-Power relationship for a driven centrifugal compressor and the power turbine. The power turbine curve assumes a constant gas generator operating condition.

Engine Compressor and Guide vanes

Most modern industrial gas turbines use adjustable guide vanes (IGVs) for their air compressor. The impact of adjusting guide vanes on the compressor performance map is shown in Figure 17. If the suction and discharge pressure are kept constant, the flow will be reduced if the guide vanes are closed (positive IGV angle), thus inducing a pre-swirl into the flow.

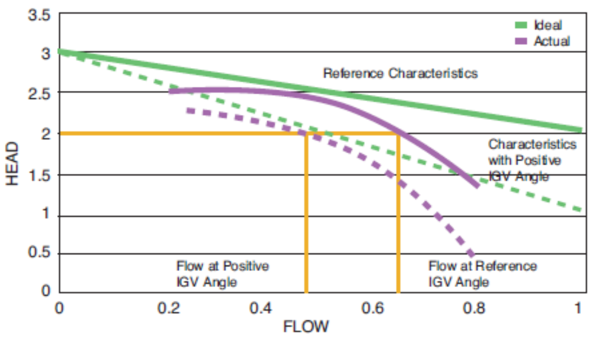

Figure 17: Effect of adjustable guide vanes for constant discharge pressure

The actual impact of modulating guide vanes in a gas turbine depends on the way the engine is operated:

In a single shaft engine, especially for a generator drive, where the control system keeps the gas turbine speed constant, a positive IGV angle will indeed reduce the flow through the compressor. This can be useful in cases where the firing temperature must be kept about constant to manage emission control in part load.

In a two shaft engine, where the speed of the gas generator is set by the equilibrium between compressor absorbed power and gas generator turbine produced power, modulating the guide vanes will only change the speed of the gas generator, because the flow and discharge pressure are determined by the choked flow through the gas generator turbine nozzle. Since normally the gas producer speed drops at high ambient temperatures and part load (while the actual flow through the nozzle stays constant), this feature allows to maintain the operating point of the gas producer turbine at its optimum.

Summary

Figure 18 shows the influence of ambient pressure and ambient temperature on gas turbine power and heat rate. The influence of ambient temperature on gas turbine performance is very distinct. Any industrial gas turbine in production will produce more power when the inlet temperature is lower, and less power when the ambient temperature gets higher. The rate of change cannot be generalized, and is different for different gas turbine models ([8],[9]). Full-load gas turbine power output is typically limited by the constraints of maximum firing temperature and maximum gas producer speed (or, in twin spool engines, by one of the gas producer speeds). Gas turbine efficiency is less impacted by the ambient temperature than the power.

The air humidity does impact power output, but to a small degree, (generally, not more than 1 to 3%, even on hot days). The impact of humidity tends to increase at higher ambient conditions.

Lower ambient pressure (for example, due to higher site elevation) will lead to lower power output, but has practically no impact on efficiency. It must be noted that the pressure drop due to the inlet and exhaust systems impact power and efficiency negatively with the inlet pressure drop having a more severe impact.

Gas turbines operated in part load will generally loose some efficiency. Again, the reduction in efficiency with part load is very model specific. Most gas turbines show a very small drop in efficiency for at least the first 10% of drop in load. In two-shaft engines, the power turbine speed impacts available power and efficiency. For any load and ambient temperature, there is an optimum power turbine speed. Usually, lowering the load (or increasing the ambient temperature) will lower the optimum power turbine speed. Small deviations from the optimum (by say +/- 10%) have very little impact on power and efficiency.

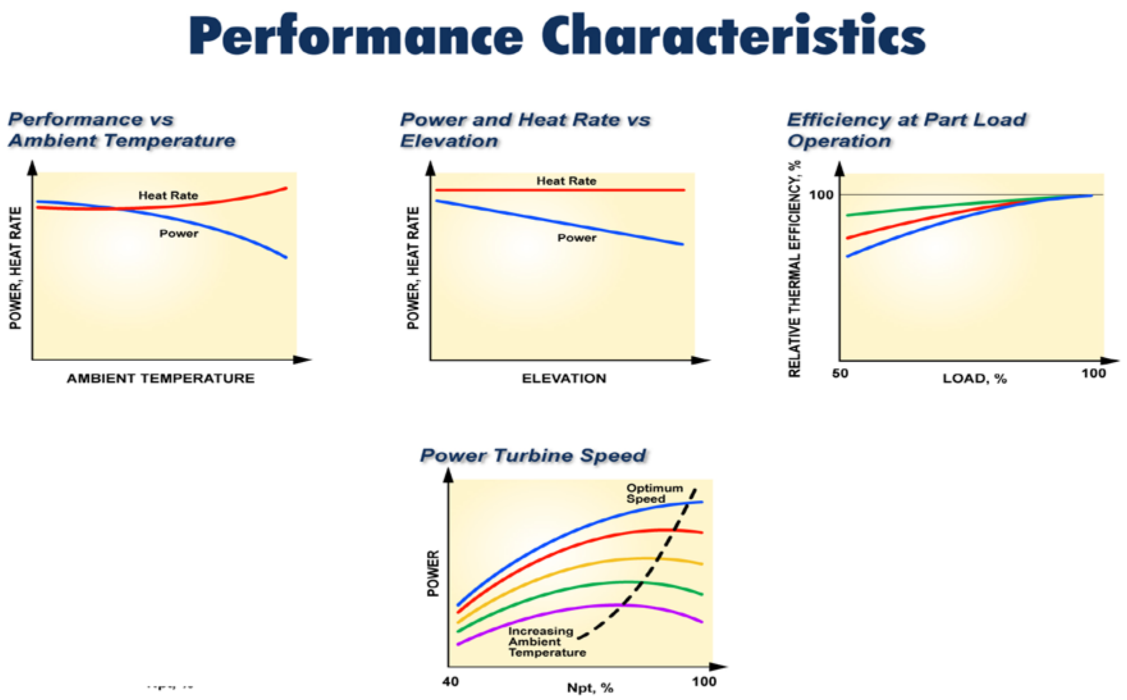

Figure 18: Performance Characteristics

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