PERFORMANCE OF INDUSTRIAL GAS TURBINES

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Dr. Brun is the Director of Research & Development at Elliott Group where he leads a group of over 60 professionals in the development of turbomachinery and related systems for the energy industry. His past experience includes positions in product development, engineering, project management, and executive management at Southwest Research Institute, Solar Turbines, General Electric, and Alstom. He holds nine patents, authored over 350 papers, and published three textbooks on energy systems and turbomachinery. Dr. Brun is a Fellow of the ASME and won an R&D 100 award in 2007 for his Semi-Active Valve invention. He also won the ASME Industrial Gas Turbine Award in 2016 and 11 individuals ASME Turbo Expo Best Paper awards. Dr. Brun is the chair if the 2020 Supercritical CO2 Power Cycles Symposium, past chair of the ASME-IGTI Board of Directors, the ASME Oil & Gas Applications Committee, and ASME sCO2 Power Cycle Committee. He is also a member of the API 616 Task Force, the ASME PTC-10 task force, the Asia Turbomachinery Symposiums Committee, and the Supercritical CO2 Symposium Advisory Committee. Dr. Brun is currently the Executive Correspondent of Turbomachinery International Magazine and Associate Editor of the ASME Journal of Gas Turbines for Power.

ABSTRACT

Industrial gas turbines have 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, the speed of the driven equipment, the fuel, and the load conditions. The reasons for these performance characteristics can be explained by the behavior the gas turbine components and their interaction.

The tutorial 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, both for single shaft and two shaft machines, driving generators, compressors, or pumps.

Methods are introduced that allow to use performance 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.

INDUSTRIAL GAS TURBINES

Industrial gas turbines produce mechanical power, and serve to drive generators, pumps or gas compressors (Figure 1). They share the working principles with gas turbines that are used for aircraft propulsion (Figure 2). The first useful gas turbines, both for industrial uses as well as for aircraft propulsion appeared in the 1930's. Some fundamental improvements in aerodynamics, heat transfer and material sciences allowed their successful design (Kurz and Brun, 2019). Industrial gas turbines have been applied very successfully in many oil and gas related applications, providing electric power for offshore installations and remote oil and gas fields. They are used to power centrifugal compressors for gas transmission, gas storage, gas lift, gas export and gas injection applications, or as drivers for pumps (Sheard, 2014, Brun and Kurz, 2018).

Figure 1: Industrial Gas Turbine powers a gas compressor

Figure 2: Gas Turbines as aircraft engines

COMPONENTS

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 to 60 percent) is used to power the compressor, and the remaining power can be used to drive generators or mechanical equipment (gas compressors and pumps). Industrial gas turbines are built with a number of different arrangements for the major components:

- Single-shaft gas turbines have all compressor and turbine stages running on the same shaft
- 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 3). 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.

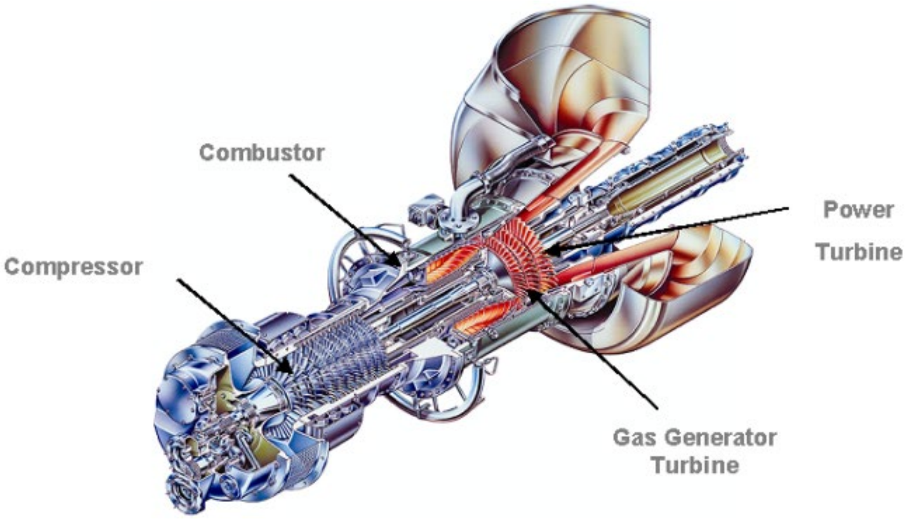

Figure 3: Components of a typical industrial gas turbine

- Multiple spool engines: Industrial gas turbines derived from aircraft engines sometimes have two compressor sections (the HP and the LP compressor), each is driven by a separate turbine section (the LP compressor is driven by an LP turbine connected to a shaft that rotates concentric within another 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.

WORKING PRINCIPLES

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., 2013).

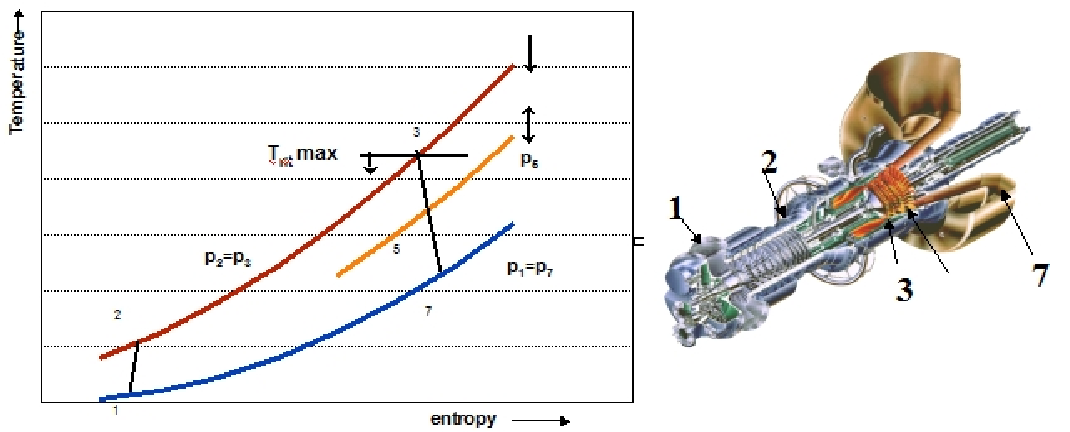

Figure 4: Brayton cycle

The Brayton or gas turbine cycle (Figure 4) 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, which is 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. This principle is used for any gas turbine, from early concepts by C. G. Curtiss (in 1895), F. J. Stolze (in 1899), S. Moss (in 1900), Lemale and Armengaud (in 1901), to today's jet engines and industrial gas turbines (Meher-Homji, 2000). 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 Brayton cycle 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, while in a gas turbine, they occur in dedicated components, but continuously and at the same time.

In a gas turbine, the conversion of heat released by burning fuel into mechanical energy is achieved by first compressing air in the air compressor, then injecting and burning fuel at (ideally) constant pressure, and then expanding the hot gas in turbine (Brayton Cycle, **Figure 4**). The turbine provides the necessary power to operate the compressor. Whatever power is left is available as the mechanical output of the engine. This thermodynamic cycle can be displayed in a temperature (or enthalpy)-entropy (h-s) diagram (**Figure 4**). The air is compressed in the engine compressor from state 1 to state 2. The heat added in the combustor brings the cycle from state 2 to 3. The hot gas is then expanded: In a single shaft turbine, the expansion is from state 3 to 7, while in a two-shaft engine, the gas is expanded from state 3 to 5 in the gas generator turbine and afterwards from state 5 to 7 in the power turbine. The difference between state 1 to 2 and state 3 to 7 describes the work output of the gas turbine. Some of the work generated by the expansion from state 3 to 7 is used to provide the work (state 1 to 2) to drive the compressor.

In a two-shaft engine, the distance from state 1 to 2 and from state 3 to 5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. The path from state 5 to 7 describes 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 cp , cp,e and cp,a are suitable averages)

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

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

For two shaft engines, where the gas generator turbine has to balance the power requirements of the compressor, and the useful power output is generated by the power turbine, 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 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

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

and the turbine

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

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

$$\eta_{th}=\frac{P}{W_fE_f}$$

and the heat rate is

$$HR=\frac{1}{\eta_{th}}=\frac{W_fE_f}{P}$$

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. Appendix A shows an example for a typical GT cycle.

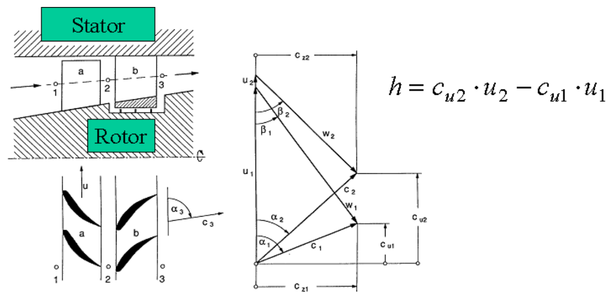

Figure 5: Velocities in a typical compressor stage. Mechanical work h transferred to the air is determined by the change in circumferential momentum of the air**.**

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 the means of aerodynamics, by appropriately manipulating gas flows. In 1754, Leonard Euler 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. 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 5).

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 (Stansel, 2018; Glassman, 1996)

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 (Figure 6). 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 (Figure 7). In most cases, the turbine nozzle is choked, which means the volumetric flow through the turbine nozzle is fixed (Figure 8). 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. In a single shaft machine, with a constant speed gas generator, all compressor operating points are on a constant speed line, and a load increase requires an increase in firing temperature, and thus an increase of compressor discharge pressure at constant speed (Figure 7a). For a two shaft engine, the gas generator speed and the firing temperature increase with load (because the speed is not fixed, but the result of a power equilibrium). Still, the compressor discharge pressure will also increase with firing temperature (Figure 7b).

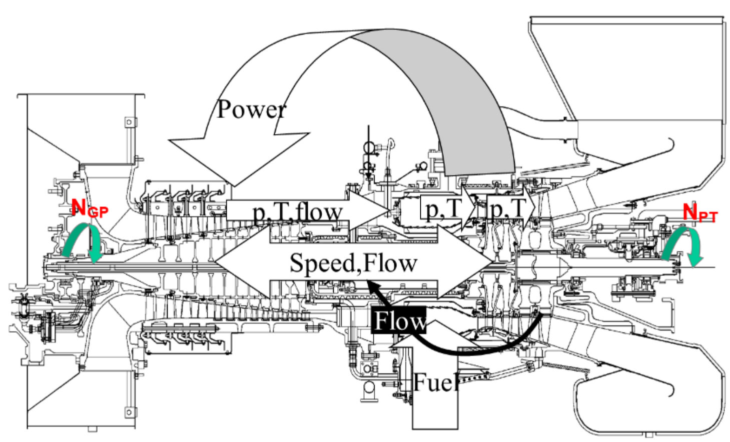

Figure 6: Gas turbine component interaction

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 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 (Kurz 2005; Kurz and Brun, 2000).

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.

In single shaft engines for generator drives, the controller adjusts the fuel flow to keep the gas turbine speed constant.

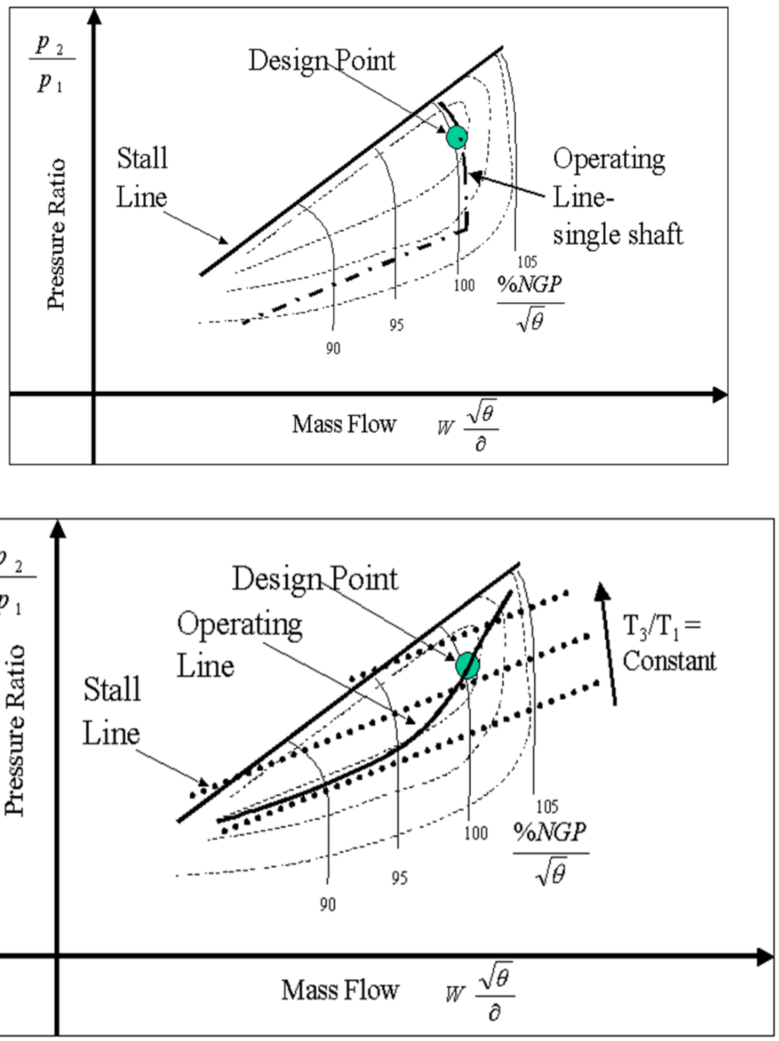

Figure 7: (top) compressor map, single shaft engine, (bottom) compressor map, two-shaft engine

1At 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.

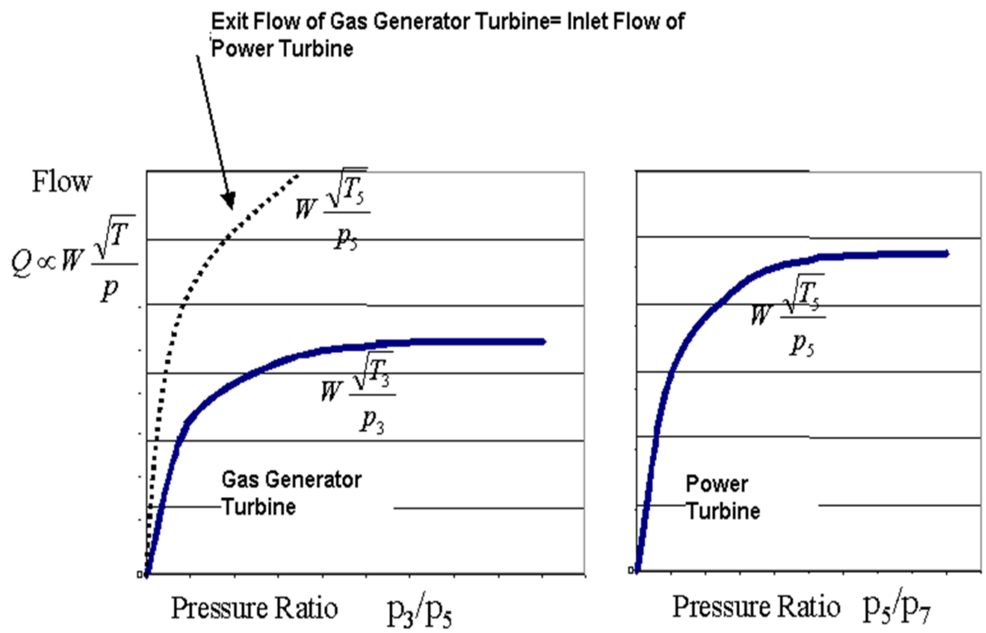

Figure 8: Gas generator turbine and power turbine.

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

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. 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 level of products related to incomplete combustion, such as CO and unburned hydrocarbons (UHC). Therefore, it is desirable to keep combustor temperatures at part load at a similar level as at full load.

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.2 At certain levels of part load, DLN engines usually bleed a certain amount of air from

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

the compressor exit directly into the exhaust duct. Therefore, while the airflow for any two-shaft engines is reduced at part load, the reduction in airflow is greater for a conventional combustion engine than for a DLN engine. This sounds paradoxical because the amount of air available at the combustor in part-load operation has to be less for a DLN engine (to maintain the fuel-to-air ratio) than for an engine with conventional combustion. However, due to the bleeding of air in a DLN engine, the flow capacity of the turbine section is artificially increased by the bleeding duct.

The combustor exit temperature at part load drops significantly for engines with conventional combustion, while it stays high for DLN engines. Once the bleed valve opens, the part-load efficiency of a DLN engine drops faster than for an engine with conventional combustion. Since the opening of the bleed valve is driven by emissions considerations, it is not directly influenced by the load. Regarding emissions, the drop in combustor temperature in engines with conventional combustion, leading to a leaner fuel-to-air ratio, automatically leads to NOx emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel-to-air ratio. However, the NOx emissions levels of DLN engines are always lower than for engines with conventional combustion (Stansel, 2018).

Single shaft, constant speed engines allow to control the combustor exit temperature using variable compressor guide vanes. This allows to reduce the airflow through the compressor at part load, thus maintaining a reasonably constant firing temperature at part load.
Gas Turbines for industrial use can be purpose built industrial gas turbines or derived from aircraft engines ('aero-derivatives') For the latter, the distinction is often made between 'light' or 'hybrid' industrial gas turbines and 'heavy-duty' industrial gas turbines. As far as engine efficiency is concerned, there is no fundamental difference between aero-derivative gas turbines and modern light or hybrid gas turbines. A key difference exists between the aforementioned types and heavy duty machines: Maintenance and overhaul for heavy duty machines has often to be performed on site, while aero-derivatives and light/hybrid industrial engines are light enough to allow for engine exchanges, with significant advantages regarding availability. There is also no fundamental efficiency difference between single shaft, two shaft, or multi spool engines, assuming they are of comparable size.

OFF DESIGN PERFORMANCE
Change in ambient conditions and load conditions cause the gas turbine components to operate at changing operating conditions. The result are performance curves as outlined in Figure 9.

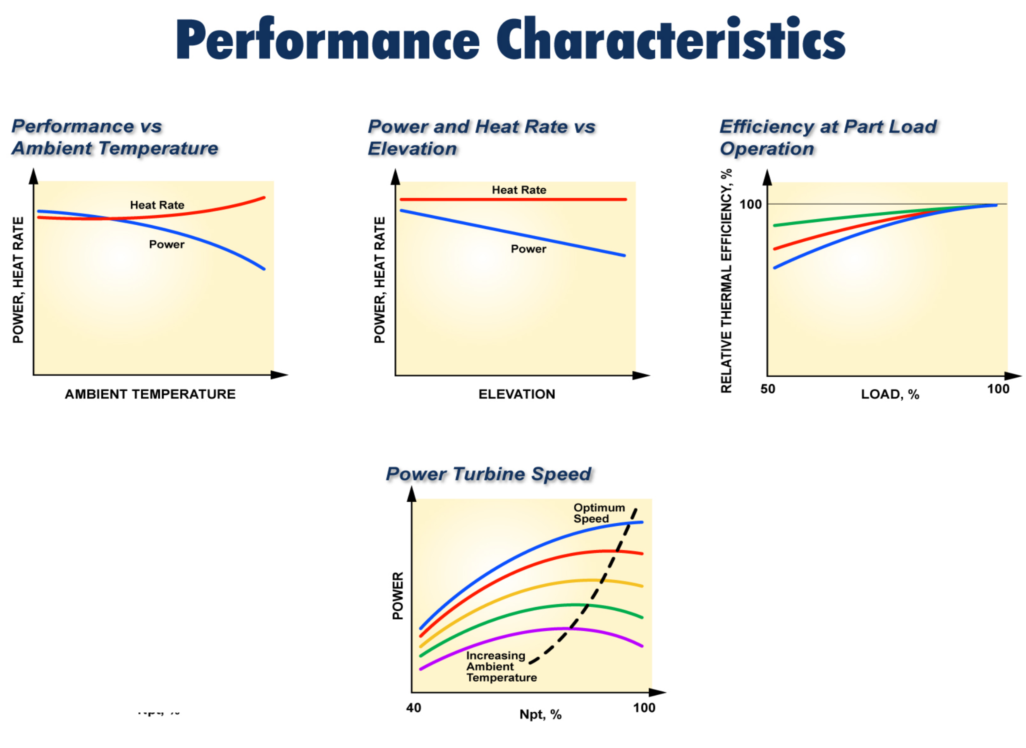

Figure 9: Performance Characteristics

We 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. The site elevation (or, actually, the barometric pressure at site) mostly impacts the output power of the gas turbine. 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 impacts engine output and efficiency.

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 9). The off-design performance curves are 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 10). 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 10).
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.

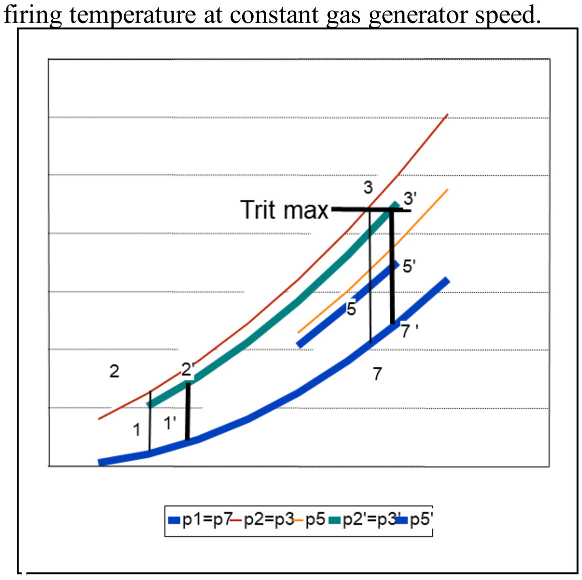

Figure 10: Brayton Cycle (Temperature over Entropy, with lines of constant pressure) at different inlet temperatures (inlet temperature 1' is higher than inlet temperature 1)

The Enthalpy-Entropy Diagram **(Figures 4 and 10)** 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 11).

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.

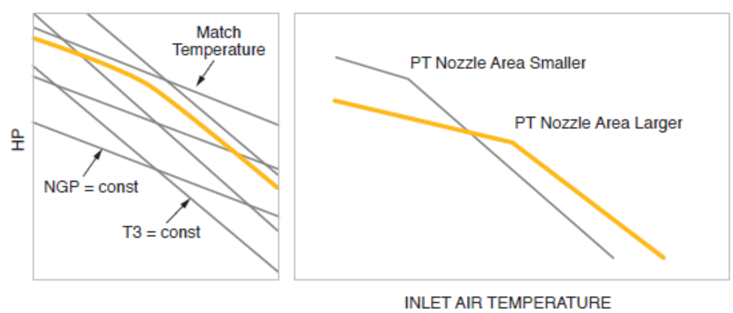

Figure 11: Full load power for a two shaft engine is limited by either maximum gas producer speed or maximum firing temperature.

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 11). 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 11). 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.

Figure 12 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.

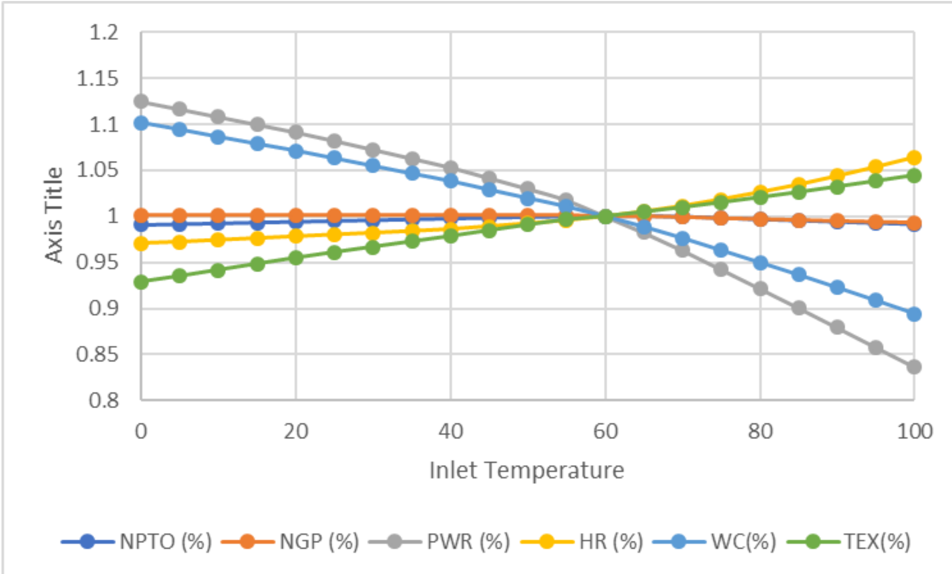

Figure 12: Full Load performance parameters, 2shaft engine, for ambient temperatures from 0°F to 100°F (-18°C to 38°C).

Part Load Operation

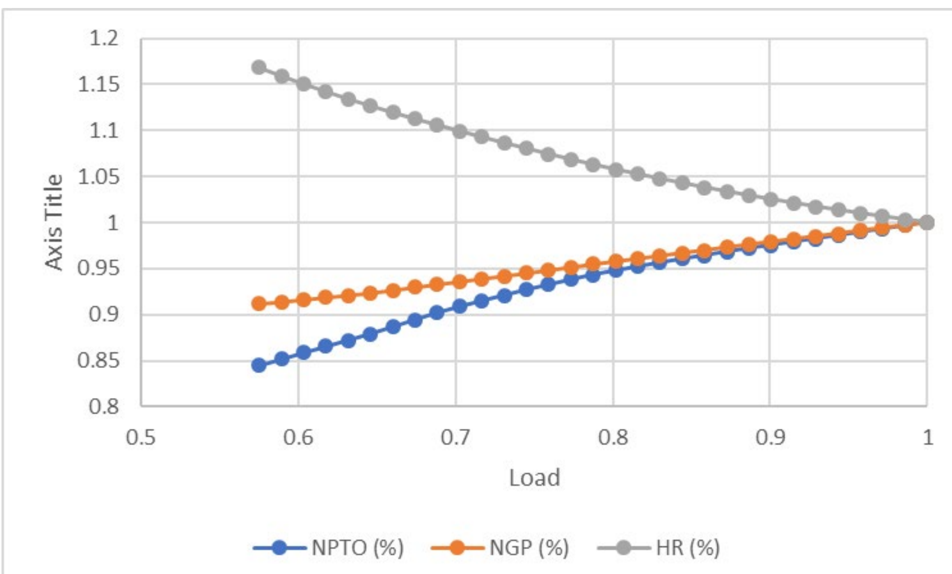

Figure 13: Two shaft gas turbine parameters at part load (50% to 100%)

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 13).

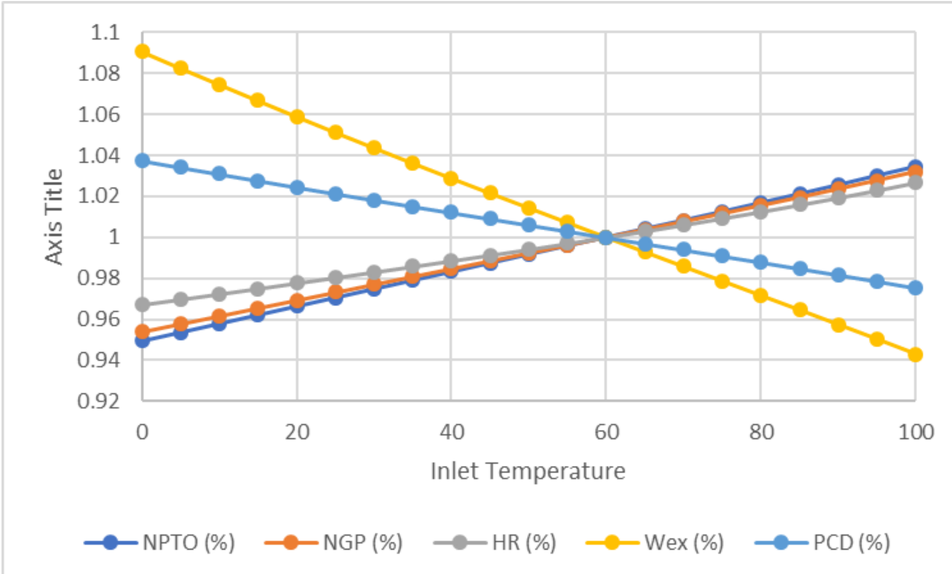

Figure 14: 2 shaft Engine at constant power output and varying ambient temperature from 0°F to 100°F (-18°C to 38°C), running at optimum power turbine speed.

For a two shaft engine running at constant power over a range of ambient temperatures (Figure 14), 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., 2011; Orhon et al., 2015). 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., 2009, Kurz et al., 2012)

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 15 describe the power reduction for the respective inlet and exhaust pressure loss.

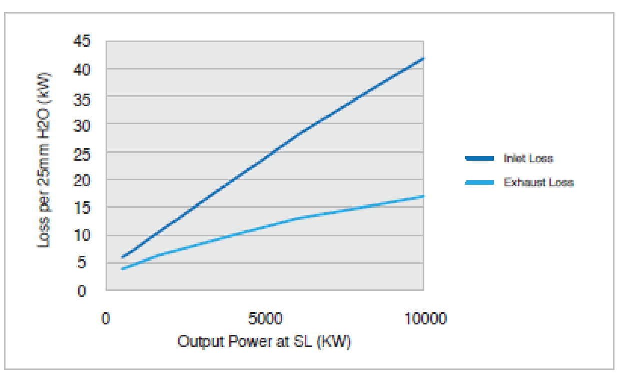

Figure 15: Impact of Inlet and Exhaust System pressure loss.

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 (Figure 9). 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 (Elliott et al., 2004; Kurz et al., 2006)

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 16) . 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 16).

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.

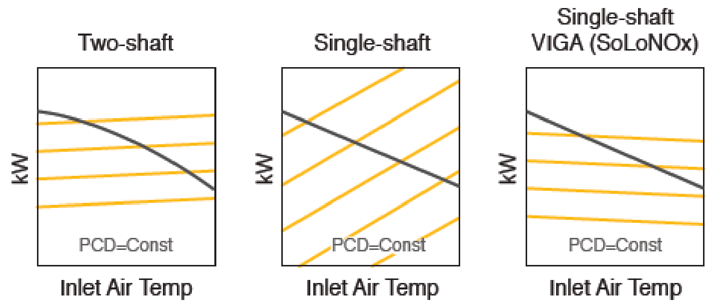

Figure 16: Fuel gas pressure and engine output.

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 17 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.

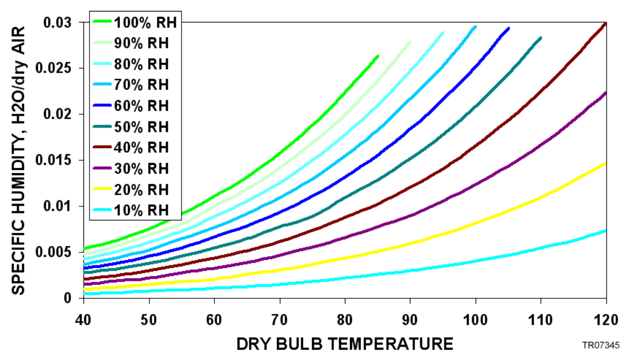

Figure 17: Specific and relative humidity as a function of temperature

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 17), 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

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 18). For any power turbine inlet pressure and flow, here 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.

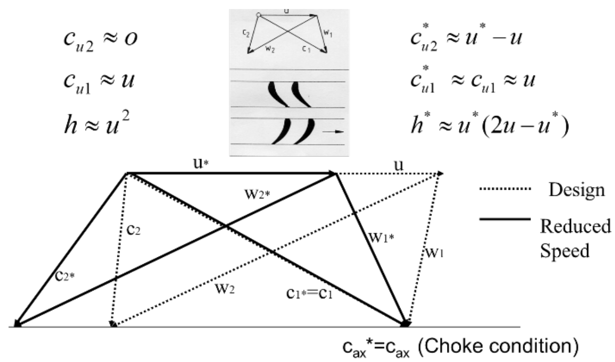

Figure 18 : Off optimum power turbine speed.

From Figure 18 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 9 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 19).

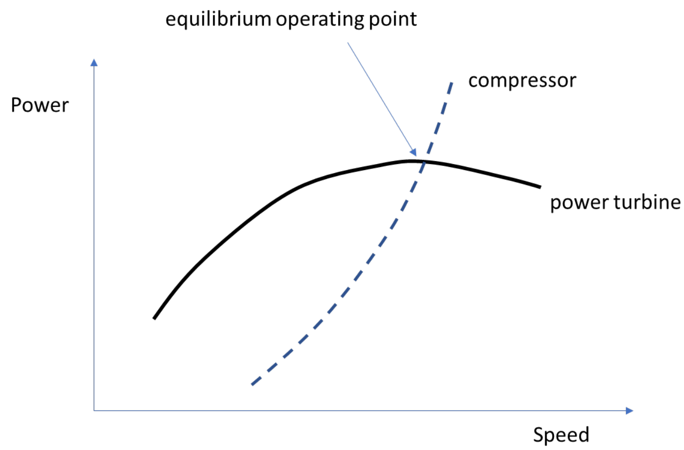

Figure 19: Speed-Power relationship for a driven centrifugal compressor and the power turbine. The power turbine curve assumes a constant gas generator operating condition.

Engine Compressor and guide vanes

Most modern industrial gas turbines use adjustable guide vanes for their air compressor. The impact of adjusting guide vanes on the compressor performance map is shown in Figure 20. 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.

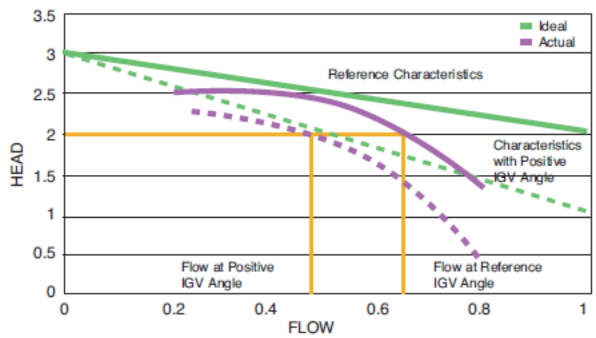

Figure 20: Effect of adjustable guide vanes for constant discharge pressure

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 the gas producer speed would drop at high ambient temperatures and at part load (while the actual flow through the nozzle stays constant), guide vane adjustments allow the gas producer speed to stay constant, and thus maintain the operating point of the gas producer turbine at its optimum.

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 21 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.

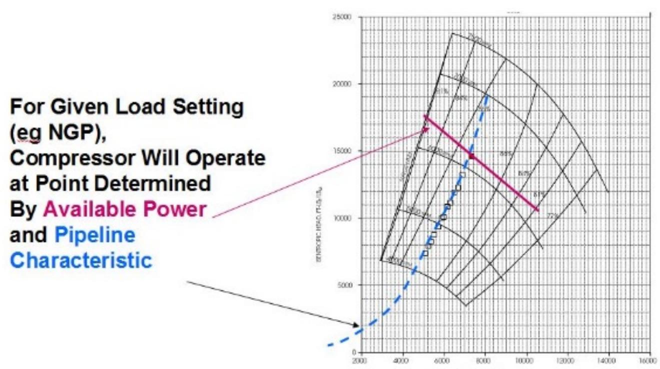

Figure 21: Available Power, Compressor Map and Pipeline Characteristic

Compressor power P is a function of mass flow W and actual head H, and thus related to the coordinates in the compressor map (of 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 21.

Further, the transient system behavior must be considered (Figure 22). A pipeline for example can be operated in a transient condition by feeding more gas into the pipeline than what is taken off on the other end. This is usually referred to as line packing. In general, pipelines are operated under slowly changing operating conditions. While a pipeline under steady state conditions requires a unique station pressure ratio for a given flow (Figure 9), this is no longer true under transient conditions: If the pipeline operates under transient conditions, for example during line pack after a fast increase in driver power, or, if one of the compressors has to be shut down, the steady state relationships are no longer valid. Dynamic studies of pipeline behavior reveal a distinctly different reaction of a pipeline to changes in station operating conditions than a steady state calculation. In steady state (or, for slow changes), pipeline hydraulics dictate an increase in station pressure ratio with increased flow, due to the fact that the pipeline pressure losses increase with increased flow through the pipeline. However, if a centrifugal compressor receives more driver power, and increases its speed and throughput rapidly, the station pressure ratio will react very slowly to this change. This is due to fact that initially the additional flow has to pack the pipeline (with its considerable volume) until changes in pressure become apparent. Thus, the dynamic change in operating conditions would lead (in the limit case of a very fast change in compressor power) to a change in flow without a change in head. If the power setting is maintained, the compressor operating point would then start to approach the steady state line again, albeit at a higher speed, pressure ratio, flow, and power.

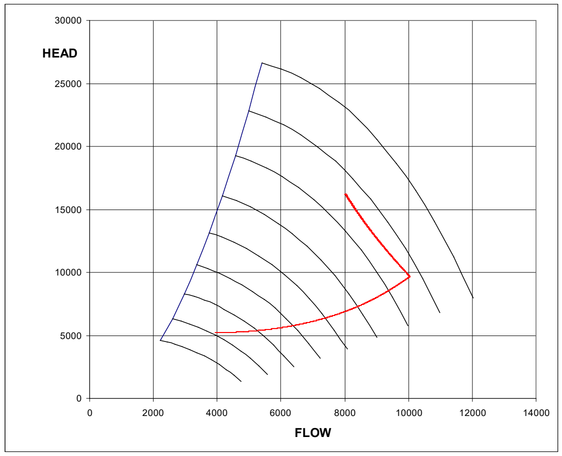

Figure 22: Typical Operating points if transient conditions are considered, in this case due to a fast engine acceleration from 50% to 100% load (Kurz et al [13]).

Interaction between the system and the compressor

The gas compressor, driven by a two shaft gas turbine can meet the process needs by varying its speed. For any situation, the process determines the suction and discharge pressure the driven gas compressor 'experiences'. Based on some control setting (available power, speed, guide vane setting) the compressor will react to the situation by providing a certain amount of flow to the system. Thus, the flow into the system is a result of the compressor characteristic (its map) and some external control setting (Figures 19 and 21). Different controls elicit different scenarios in these control situations: If we control the compressor by the level of power that's supplied, then the speed at which the compressor runs is an outcome of the interaction between compressor and process. If we control the speed of the compressor, the required power is an outcome. The same is true for a constant speed machine (which in that sense is just a special case of a compressor that's forced to operate at a set speed).

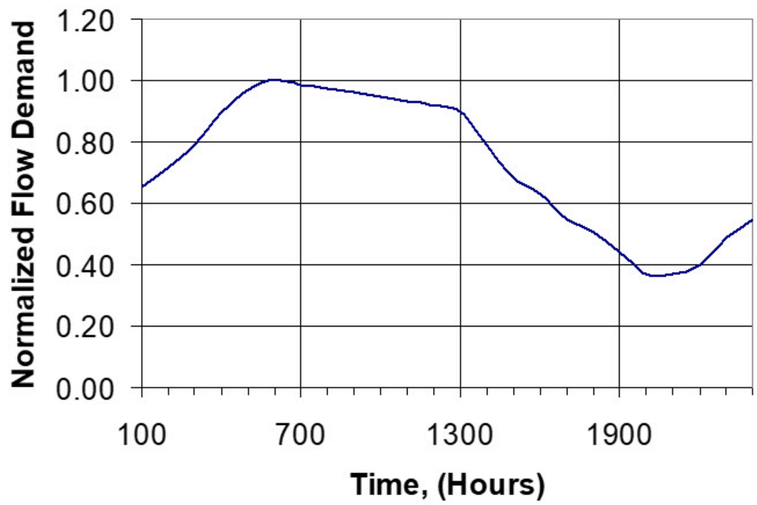

Figure 23: Diurnal variation in flow demand for a pipeline station.

APPLICATION AND SIZING

Applying gas turbines in industrial applications requires the knowledge of the principles and dependencies outlined above. The dependency of gas turbine performance on ambient temperature. In most places of the world, the ambient temperature varies over a wide range, and the available power output of the gas turbine with it (Figure 23). In addition, the required load may also change with the time of the year, or even the time of the day. The sizing of machinery, both regarding the driver and the driven equipment, must take these effects into account. Additionally, driver and driven equipment must be matched regarding their speed (Taher and Meher-Homji, 2012).

Further, the driven compressor operating point is also a function of power from the gas turbine, and the characteristic of the installation (Figures 21, 24)

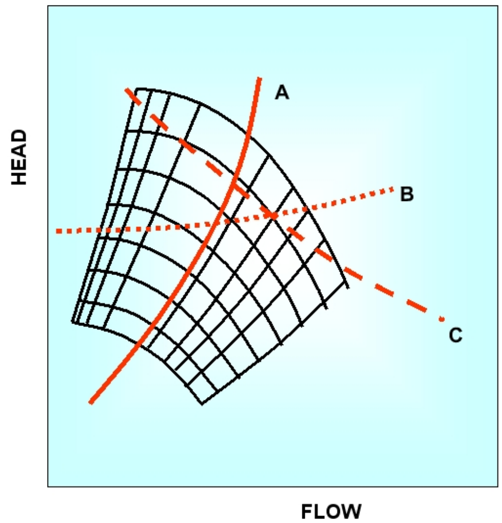

Figure 24: System Characteristics and Compressor Map

CONDITION MONITORING

The performance aspect of conditions monitoring involves generally the comparison of a measured performance parameter with an expected value (Kurz et al., 2013) . Trending then becomes the logging of deviations between measured and expected parameters. This identifies the two key ingredients of condition monitoring:

- A (digital) performance model for the gas turbine, including its components.
- Sensors and algorithms that allow the measurement and processing of relevant performance data.

In addition, one must have ways to exclude invalid data, be it from sensor failures, sensor inaccuracy, and non-steady state behavior, and to get a sense of the accuracy of the data. Also, the digital performance models can be improved by the use of test data on individual engines. These improvements can come from tests on an individual engine, to customize the digital model for this engine, as well as the use of data from a number of the same engines.

The intersection of machine learning methods and gas turbine sensor data has expanded rapidly in the last decade to include numerous applications of regression, clustering, and even neural network algorithms. Learning algorithms have pushed traditional engine health management into the realm of prognostic health management. Focus is generally placed on industrial gas turbines with an industry standard monitoring system. Allen et al (2017) for example explore beyond gas path analysis with a novel use of machine learning algorithms to engine component classification. Other applications can involve the optimization of maintenance schedules (Allen et al., 2018), by estimating degradation rates, and the economic impact of delaying maintenance. One of the key challenges is the capability to distinguish data that is valid and suitable for an analysis, versus data that does not lend itself to be used, for example because it was taken during transient conditions, or because it comes from a faulty sensor (Venturini et al., 2013). Examples may include situations where the driven compressor is used to measure the power output of the gas turbine and the gas composition for said driven compression is unknown, or has changed.

The requirements for sensors and algorithms that allow the measurement and processing of relevant performance data are essentially the same as for a performance test of a gas turbine. In general, the following parameters, or a subset thereof, should be measured:

-Speed of all shafts
-pressures at the inlet and exit of all components
-temperatures at the inlet and exit of all components
-output power
-fuel flow, composition and temperature
-relative humidity of the inlet air
-bleed flows
-emissions can also provide input about the health state of the engine.

The temperature of the gas entering the turbine section is difficult to measure due to the high temperature levels. Usually it is derived from other measurements, such as the temperature measurement into the power turbine, or the exhaust gas temperature.

A particular difficulty arises from determining the output power. If the gas turbine drives a generator, the power output can easily determined by the electrical output at the generator terminal, and then corrected by known generator and gearbox efficiencies. If the gas turbine drives one compressor or multiple compressors, the power measurement requires to fully instrument the driven compressors, and to calculate the absorbed power by thermodynamic measurements and flow measurements. This can be avoided, if the torque at the power turbine is measured with a torque sensing coupling. If the gas turbine drives a pump, thermodynamic measurements are not possible, and either the pump efficiency is assumed, or a torque measuring coupling is installed.
Another approach is to determine the power turbine output from a thermodynamic measurement, using pressures and temperatures upstream and downstream of the power turbine, and the measured or estimated flow. In all cases, the accuracy of the data has to be determined, and an uncertainty analysis has to be conducted. Of importance is the fact that the uncertainty is not just determined by the instrumentation, but also by the operating condition of the equipment (Kurz and Brun, 2001).

Further, the parameters in a given measurement plane are not necessarily uniform. The temperature field of the gas leaving the combustor or, entering a power turbine, for example, is radially and circumferentially non-uniform. Also, cooling flows usually cannot be directly measured.

Gas turbine performance deteriorates over time. Several mechanisms cause this degradation (Kurz et al., 2009):

Fouling is caused by the adherence of particles to airfoils and annulus surfaces. Oil or water mists on the surface significantly increase the amount of particles that are captured, and typically only small particles (up to 10 μm) stick to the surface. Smoke, oil mists, carbon, and sea salt particles are typical examples. The build-up of dirt particles increases surface roughness and may change the shape of the airfoil. Fouling can be controlled by appropriate air filtration systems, and can often be reversed to some degree by detergent washing of components (Suman et al., 2014, Orhon et al,,2015).

Hot corrosion is the loss or deterioration of material of flow path components caused by chemical reactions between the component and certain contaminants, such as salts (for example sodium and potassium), mineral acids or reactive gases (such as hydrogen sulfide or sulfur oxides). Since many industrial gas turbines are located near the sea, sea salt (sodium chloride) is often a potential offender. Sodium sulfate is often the result of the combination of sulfur in the fuel and sodium chloride in the air.

Corrosion is caused both by inlet air contaminants and by fuel and combustion derived contaminants. Fuel side corrosion is typically more noted and severe with heavy fuel oils and distillates than with natural gas because of impurities and additives in the liquid fuels that leave aggressive deposits after combustion. Corrosion is often produced by salts such as sodium and potassium, but lead and vanadium are also common contributors.

Erosion is the abrasive removal of material from the flow path by hard or incompressible particles or droplets impinging on flow surfaces. These particles typically have to be larger than 10μm in diameter to cause erosion by impact. State of the art filtration systems used for industrial applications can keep these larger particles from entering the engine. Erosion can become a problem for engines using water droplets for inlet cooling or water washing.

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. Bearings tend to become softer (reduction in stiffness) due to an increase in clearance over time that causes an increase in journal orbital amplitude. The larger orbit can result in material removal at blade tips and seals , which will increase seal or tip gaps.

Damage may also be caused by *foreign objects* striking the flow path components. These objects may enter the engine with the inlet air, 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.

Fouling, Corrosion, Hot Corrosion and Erosion can to some degree controlled by appropriate inlet air filtration.

While fouling effects can be reversed by cleaning or washing the engine, other effects require the adjustment, repair or replacement of components. It is thus common to distinguish between recoverable and non-recoverable degradation. Any degradation mechanisms that can be reversed by on-line and off-line water washing are considered recoverable degradation. Degradation mechanisms that require the replacement of parts are considered non-recoverable, because they usually require an engine overhaul.

The determination of the exact amount of performance degradation in the field is rather difficult, due to ubiquitous test uncertainties (Kurz and Brun, 2001). Even trending involves some uncertainties, in particular when data from transient operating conditions has to be identified (Ceschini,et al., 2017)

One of the open questions in a number of publications was the capability of very fine dust to alter the air foil shape and surface quality sufficiently to affect a performance deterioration. The data shows the performance of an engine with an air filtration system that filters out all but the smallest particles. It thus seems that the data indicates that very fine particles indeed will not cause any fouling at all. This also proves a statement in Kurz and Brun (2012): The effect of the type of air filtration on fouling far outweighs any effect of engine design, or engine susceptibility (Meher-Homji et al., 2009).

An example of engine degradation, and the impact of air filtration is shown in Figure 25 with the power margin versus new engine level performance for a 4.6 MW gas turbine installation (Burnes et al. 2018). The power loss and gains are coincident with the water washes over the observation period. The conventional air filters were replaced with HEPA filters around 2/3/2012. The one noticeable exception is the general overall power loss demonstrated over the entire duration despite the full elimination of fouling with the new filter media resulting in a non-degrading compressor. Figure 26 shows the compressor and turbine efficiency diagnostic scalars again at the same time interval. While the compressor parameters show no degradation after the HEPA filters were installed, the turbine efficiency clearly deteriorates, leading to a loss of power over time. Hardware evaluation during overhaul identified the turbine to operate with higher running clearances which in turn results in a lower turbine efficiency. This particular issue explains this set of

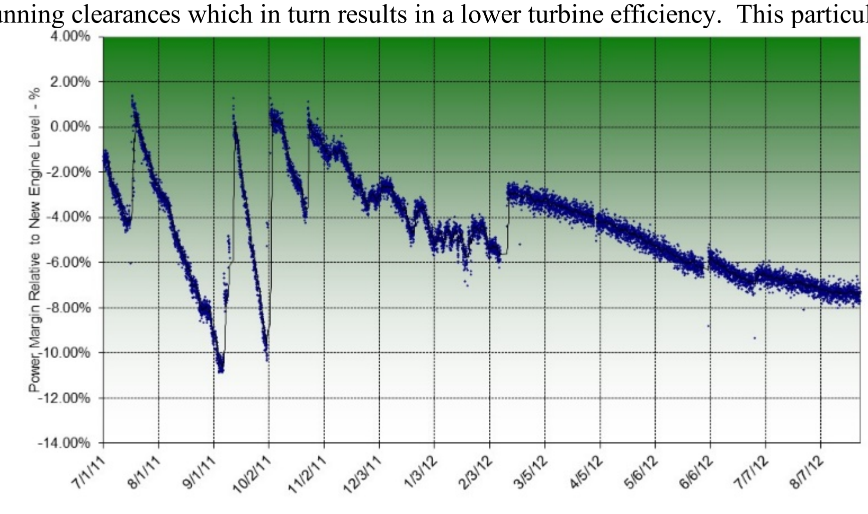

Figure 25: Field Data Power Margin

data, and shows how an analysis of this type can give deep insight into the state of the engine. Initially, the effectiveness of the newer filter media was questioned observing only the power, but looking deeper into the diagnostic results, it could be seen that the compressor was performing in an as new state from the point in which the new filter was installed.

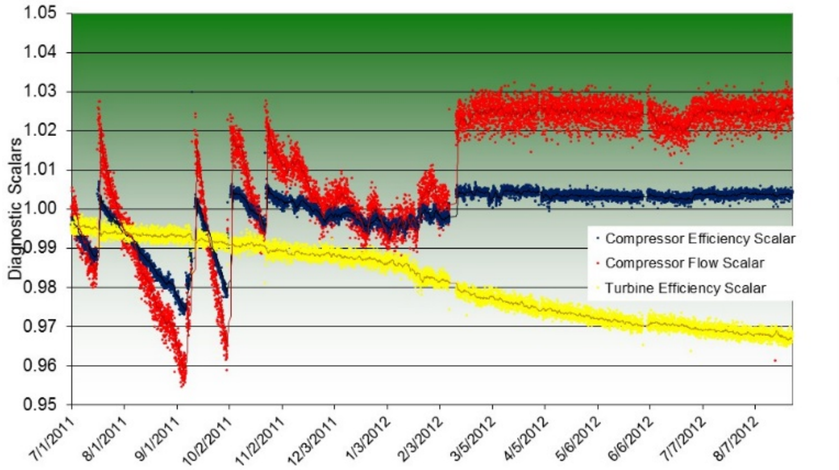

Figure 26: Field Data Component Diagnostic Scalars, Compressor and Turbine.

The final case study is from a thorough evaluation made on a 11MW engine with a conventional combustor after an overhaul return reaching the typical service interval. This unit was well maintained and operated at part load while in service at a site in South Africa. Without any significant rubs from the rotating components, there was negligible non-recoverable performance degradation from the new to the as received condition. Figure 27 shows gas producer speed (NGP) at or below 90%, which is well below full load (~100%) speed throughout the nearly 5 years of operation in the field. Since this is an engine with conventional combustion system, the firing temperature drops significantly at part load operation based on the control methodology. The cooler operating temperature was beneficial to the hot section components. Surface finish data was compiled for all the hot section flow path components. It was found that each component would be compliant to new engine surface finish requirements even after reaching TBO. Figure 28 illustrates where the airfoil surface finish measurements were obtained. Tables were created measuring a representative sample of air foils to determine values relative to the drawing requirement. Tables 1 shows normalized surface finish data for this engines' 1st stage nozzle taken after tear down at overhaul. Average values at or below 1 indicate a surface finish that would be acceptable for a new engine.

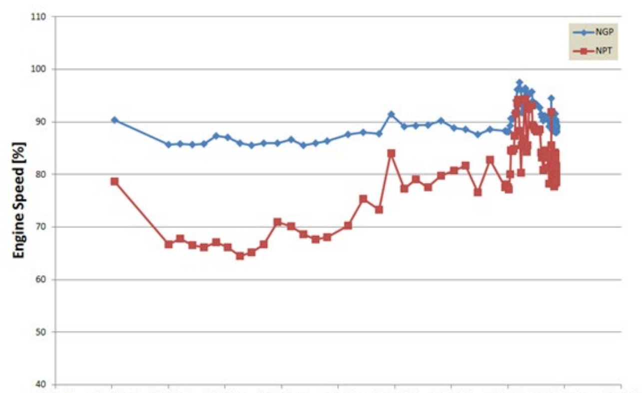

Figure 27: Two shaft engine, 11MW, Field Data Engine Speeds.

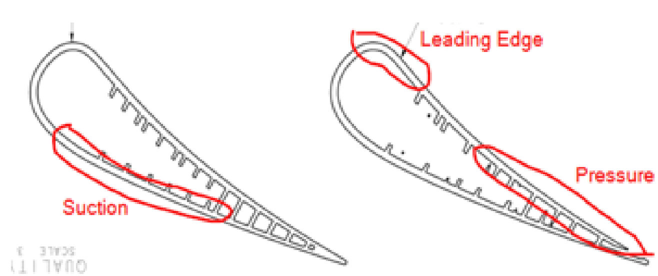

Figure 28 : Airfoil Locations of Surface Finish Measurements

Stage 1 Nozzles Surface Finish (Viewed aft looking forward)				
Serial Number	Left Vane Suction Side Airfoil Trailing Edge	Right Vane Pressure Side Airfoil Trailing Edge	Right Vane Leading Edge	Inner Shroud
1	0.889	0.932	1.143	0.843
2	1.029	0.537	1.190	0.803
3	0.937	0.598	1.370	0.878
4	0.683	0.771	1.144	0.598
5	0.719	0.657	1.324	0.776
6	0.633	0.710	1.389	0.813
7	0.984	0.675	0.990	0.556
8	0.860	0.624	0.651	0.562
9	0.671	0.570	1.116	0.595
10	0.644	0.524	1.210	0.643
AVERAGE	0.805	0.660	1.153	0.707

Table 1: Stage 1 Nozzle Surface Finish

This information, along with the general good condition all flow path components, seals, cases/shrouds, and bearings helped explain the as received finding of negligible performance loss compared to the as new condition

CONCLUSIONS

This tutorial explained the reasons behind the performance characteristics of industrial gas turbines, driving generators, compressors, or pumps. The performance behavior depends on ambient and operating conditions. The function of the components such as the engine compressor, the combustor and the turbine section, and certain control strategies of a gas turbine and their interaction is outlined. Fundamental concepts that help to understand the flow of energy between the components are explained. Further discussed are control concepts, both for single shaft and two shaft machines, Performance characteristics are also determined by decisions made during the design process of an engine, so the tutorial will give an explain of that process.

Methods are introduced that allow to use 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.

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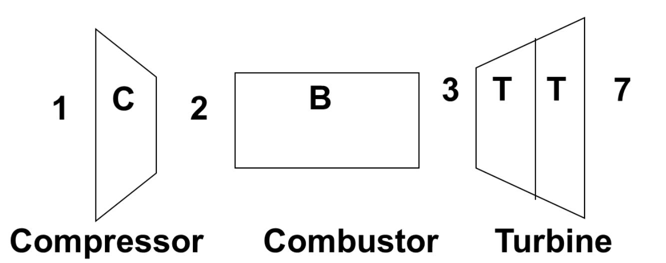
APPENDIX A: GAS TURBINE CYCLE CALCULATION



Work = $h_2 - h_1$ Work = $h_3 - h_2$Work= $h_7 - h_3$
= $C_p (T_2 - T_1)$ = $C_p (T_3 - T_2)$ = $C_p (T_7 - T_3)$

Figure A-1: Simplified Gas Turbine Cycle

The following example is not intended to represent the cycle calculation for a specific gas turbine. It is rather the simplest possible demonstration of a Brayton cycle, strictly assuming ideal, yet compressible, gas, with a constant heat capacity, and the same gas throughout the cycle.
In the actual gas turbine, one would see a different heat capacity c_p and ratio of specific heat γ for the compressor, the combustor, and the turbine section. However, this does not change the general principle for a Brayton cycle as outlined below. Further, we also did not account for the added mass flow from the fuel.

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)

p_a=14.73psia (1.013bara) p_3=147.3 psia (10.13bara) W= 100lbs/s (45 kg/s) η_c=85% T_3=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 (1.007kJ/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 1600F (870°C), we need to add heat to bring the gas from 713°F (379°C) to 1600F (870°C):

$Q = W\, c_p\, \Delta T$ = 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%.