43rd Turbomachinery & 30th Pump Users Symposia (Pump & Turbo 2014)
September 23-25, 2014 | Houston, TX | pumpturbo.tamu.edu

GAS TURBINE DEGRADATION

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ABSTRACT

There is a strong incentive for gas turbine operators to minimize and control performance degradation, as this directly affects profitability. The area of gas turbine recoverable and non-recoverable performance degradation will be comprehensively treated in this tutorial. Deterioration mechanisms including compressor and turbine fouling, erosion, increased clearances, and seal distress will be covered along with their manifestations, rules of thumb, and mitigation approaches. The treatment will deal with simple cycle gas turbines in power generation and mechanical drive applications and will also address

the impact of performance deterioration on combined and cogeneration cycles.

The goal of this tutorial is to provide a non-mathematical treatment of performance deterioration to help plant operators grasp the underlying causes, effects, and measurement of gas turbine performance degradation. Topics covered will include fouling, effect of blade surface roughness, erosion, corrosion, losses of ancillary systems (inlet/ outlet), clearances effects, and the impact of fuels on GT combustion and degradation.

To understand the impact of degradation, basic concepts of gas turbine matching and off design operation will be addressed first. The three major sources of performance deterioration will be discussed:

- Recoverable Deterioration- can be removed by actions during operation of the gas turbine.
- Unrecoverable Deterioration – can be recovered by an overhaul but not during operation.
- Permanent Deterioration- residual deterioration present even after a major overhaul.

Control aspects and their interaction with performance deterioration mechanisms will also be covered. Lastly, condition monitoring approaches focused on the detection of deterioration will be reviewed.

HOW DOES A GAS TURBINE WORK?

To discuss the degradation of gas turbines, we briefly have to explain the working principles of the gas turbine and its components.

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).

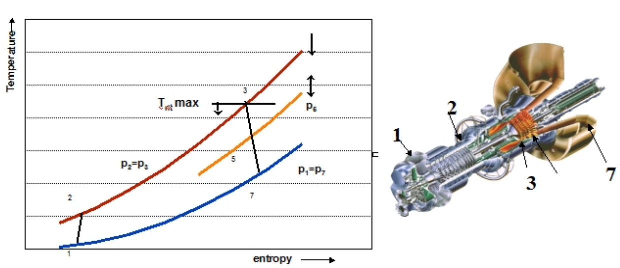

Figure 1: Brayton cycle

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. 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 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 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 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.

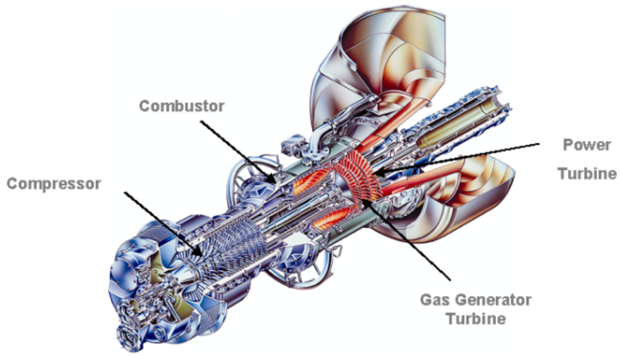

Figure 2: Typical industrial gas turbine

- 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.

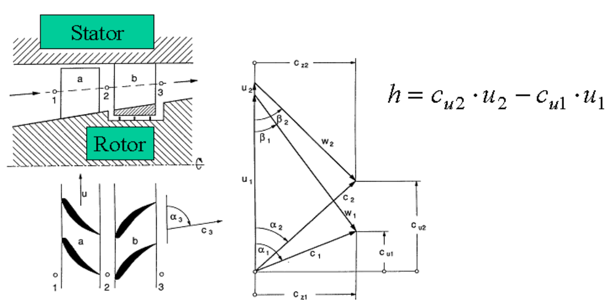

Figure 3: Velocities in a typical compressor stage. Mechanical work h transferred to the air is determined by the change in circumferential momentum of the air.

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 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.

Some component characteristics are particularly relevant for the degradation behavior and are mentioned here:

- The airflow through the engine is typically controlled by the fact that the turbine nozzle is choked. Thus, any geometry change or change in boundary layer will directly impact the airflow.
- When the engine compressor operates at design aerodynamic speed (i.e., Machine Mach Number), the stages in the axial compressor are usually fairly equally loaded. At low aerodynamic speed (i.e., low mechanical speed, or high ambient temperature) the front stages become higher loaded than the rear stages and vice versa for high aerodynamic speed (i.e., high mechanical speed or low ambient temperature) (Williams, 2008, Kappis and Guidati, 2012).
- The impact of fouling in the compressor can also change the stage loading across the compressor.

COMPONENT INTERACTION

When the compressor, the gas generator turbine, and the power turbine (if applicable) are combined in a gas turbine, the operation of each component experiences operating constraints that are caused by the interaction (matching) between the components**.**

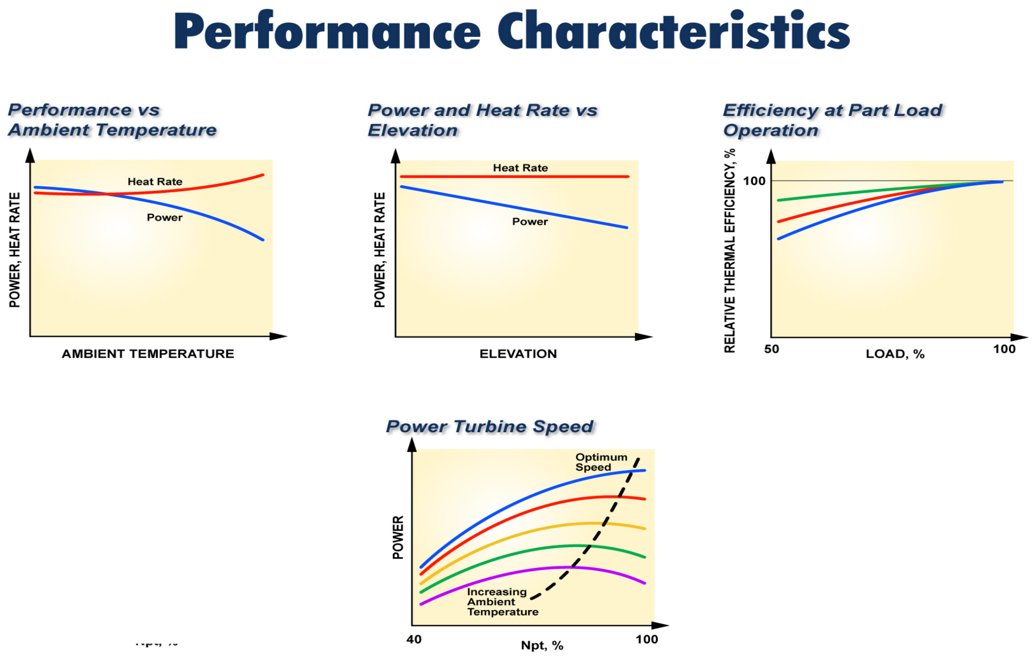

Figure 4: Performance characteristics of gas turbines

For example, the engine compressor will compress a certain mass flow, which in turn dictates the compressor discharge pressure necessary to force the mass flow through the turbine section.

On a two-shaft engine, the gas generator turbine provides the power for the compressor. This determines the gas generator speed, where the compressor-absorbed power and the gas generator turbine-produced power are at equilibrium.

On a single-shaft engine, where the gas generator speed is held constant, the firing temperature has to be at a level where the power from the turbine, after the necessary power for the compressor is subtracted, satisfies the load demand of the driven electric generator. The firing temperature influences both the power that the turbines can produce, but it also impacts the discharge pressure necessary from the compressor (Kurz, 2005).

The components are designed to work together at their highest efficiencies at a design point, but the operation of the components at any other than the design point must also be considered (Kurz 2005).

In a single shaft machine, the gas turbine operates at constant speed, meaning any operating point of the engine compressor (for given ambient conditions) lies on a single speed line. Load increases are initiated by increasing the fuel flow, which in turn increases the firing temperature. Due to the fact that the first turbine nozzle is usually choked, the compressor operating point moves to a higher pressure ratio to compensate for the reduced density (from the higher firing temperature).

A two-shaft gas turbine consists of an air compressor, a combustor, a gas generator turbine on one shaft, and a power turbine on a second shaft.[1]

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

[1]Some engines are configured as multi-spool engines. In this case, the gas generator has a low-pressure compressor driven by a low-pressure turbine and a high-pressure compressor driven by a high-pressure turbine. For this configuration, the shaft connecting the LP compressor and turbine rotates inside the shaft connecting the HP compressor and turbine. In general, all the operating characteristics described above also apply to these engines.

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. A detailed discussion can be found in Kurz and Brun (2000) and Kurz (2005).

GAS TURBINE PERFORMANCE CHARACTERISTICS

The gas turbine power output is a function of the speed, the firing temperature, as well as the position of certain secondary control elements, like adjustable compressor vanes, bleed valves, and in rare cases, adjustable power turbine vanes. The output is primarily controlled by the amount of fuel injected into the combustor. Most single-shaft gas turbines run at constant speed when they drive generators. In this case, the control system modifies fuel flow (and secondary controls) to keep the speed constant, independent of generator load. Higher load will, in general, lead to higher firing temperatures. Two-shaft machines are preferably used to drive mechanical equipment, because being able to vary the power turbine speed allows for a very elegant way to adjust the driven equipment to process conditions. Again, the power output is controlled by fuel flow (and secondary controls), and higher load will lead to higher gas producer speeds and higher firing temperatures.

Figure 4 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. 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 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 output and efficiency.

DEGRADATION IN GAS TURBINE SYSTEMS

Any prime mover exhibits the effects of wear and tear over time. Because the function of a gas turbine is the result of the fine-tuned cooperation of many different components, the emphasis of this tutorial is on the entire gas turbine as a system, rather than on isolated components. Treating the gas turbine as a system, reveals the effects of degradation on the match of the components. This contributes insights into the problem of gas turbine system degradation. (Kurz et al., 2008). Based on some detailed studies on the mechanisms that cause engine degradation, namely: (1) changes in blade surfaces due to erosion or fouling and the effect on the blade aerodynamics; (2) changes in seal geometries and clearances and the effect on parasitic flows; and (3) changes in the combustion system (e.g., which results in different pattern factors), the relative effects of degradation can be determined

To evaluate the actual state of gas turbine-driven equipment, the performance degradation of the equipment must be determined. The most common degradation mechanisms are:

- Airfoil (blade) fouling and pitting (loss of flow profile caused by corrosion and surface deposits). Other flow path components (such as casing walls, etc.) are affected in similar ways.
- Opening of blade clearances (typically caused by softening bearings and increased rotor unbalance).

- Leading and trailing edge erosion (caused by mechanical solid and liquid impacts on the rotor blades).

The different degradation mechanisms and their relative importance are discussed in detail below.

As outlined in some detail above, the function of a gas turbine and of a gas turbine package is the result of the fine-tuned cooperation of many different components. Any of these parts can show wear and tear over the lifetime of the package and, thus, can adversely affect the operation of the system. In particular, the aerodynamic components, such as the engine compressor, the turbines, and the driven compressor have to operate in an environment that will invariably degrade their performance. Degradation of individual engine components has a compounded effect on engine performance, because the change in component performance characteristics leads to a mismatch of these components on the engine level, as well as on the component level. The impact of individual component degradation is also influenced by the control system and the control modes of the engine. Single-shaft engines, operating at constant speed, will show different degradation behavior than two-shaft engines. The impact of degradation on two-shaft engines depends on the control mode they are in, i.e., whether the gas generator speed or the firing temperature is the limiting factors. Additional, the method and location of measuring the control temperature will determine the behavior of the engine in degraded conditions

DEGRADATION MECHANISMS

Several mechanisms cause the degradation of engines (Kurz et al., 2008):

- Fouling
- Corrosion, hot corrosion, and oxidation
- Erosion and Abrasion
- Particle Fusing
- Mechanical Degradation

FOULING

Fouling is caused by the adherence of particles to airfoils and annulus surfaces. The adherence is increased by oil or water mists. The result is a build-up of material that causes increased surface roughness and, to some degree, changes the shape of the airfoil. Particles that cause fouling are typically smaller than 2 to 10 µm. Smoke, oil mists, carbon, and sea salt are common examples. Fouling can be controlled by an appropriate air filtration system. Fouling can often be reversed to some degree by detergent washing of components. Fouling in the colder sections of the engine can normally be eliminated by cleaning. Components operating at elevated temperatures are also subject to fouling, but the dirt will often be baked to the surfaces, and can usually not be removed by cleaning processes available in the field.

There are generally four different types of fouling deposits that can be found inside an axial compressor: salts, heavy hydrocarbons (oils and waxes), carbon dirt, and other dirt.. A gas turbine compressor acts like a very effective air filter in that it collects and deposits a significant percentage of any solids that are carried by the ambient air and are ingested into the gas turbine inlet. Also, due to the heat of compression, the temperature of the air passing through the compressor will increase to above 600°F such that any solids or chemicals dissolved in the water will drop out and deposit on the compressor blades.

Salts: Salts (and other chlorides) in combination with moisture are primarily responsible for metal surface pitting in gas turbine compressors. As one of the most common deposits, salts lead to the formation of the deposit of chlorides in microscopic surface cracks and subsequent subsurface corrosion (evidenced as characteristics pits (small holes) on the surface of the blades. Salts do not enter the gas turbine from the ambient air (as is typical in offshore applications) but would be introduced by the plant water (i.e., fogging and water wash).

Oils and Waxes: Oils and waxes are usually residues from compressor washing or ambient air contamination. Generally, the components do not cause significant damage but can act as binding agents for dirt or sand in the compressor and, thus, can contribute to fouling.

Carbon: Carbon (or coke) deposits on compressor blades indicate that exhaust gases from the gas turbine or other internal combustion engines are entering the axial compressor. Coking inside the rotor can also be caused by local overheating or lube oil leakage. Deposits will bind with oils to form a surface deposit on the blades, which is very hard to remove with online and offline cleaning. Most will deposit in the first couple of stages of the compressor and will not be carried downstream.

Dirt: Sand and other types of dust (for example from farming, industrial processes, drilling etc.) are primarily introduced into the gas turbine through the inlet filter and is an indication of inadequate inlet filtration or filter dirt saturation. The likely source of any sand or dirt particles entering the gas turbine at subject installation is from the inlet air as any water carrying sand or dirt would have been captured by the high efficiency fogging water filter.

Engine fouling is caused by the buildup of contaminants on internal surfaces. Adhesion of particles on blade surfaces alters the profiles and results in a loss of engine power and an increase in fuel consumption. Fouling in intercoolers reduces compression heat removal. Air contamination can be significantly reduced by the use of more efficient air filtration systems, especially those that reduce the quantity of smaller particles ingested. Therefore, it is essential that the air filtration system be optimized to remove small particles, while also being capable of protecting against larger contaminants. Many improved media filter elements can be used as replacement elements during filter change-out. Filter companies have developed media to capture more dust particles in the 0.1 to 2 µm size range, while still offering satisfactory filtration of larger particles and acceptable service life. However, generally speaking, the more efficient the filtration efficiency of small particles, the higher the pressure loss across the inlet system, or the larger the filtration system becomes.

Composition of the particles is important in determining the rate of fouling. Because of very high mass flows, even very low levels of ingested foreign material can produce a substantial input of contaminants through gas turbine engines (see discussion above). For example, 1 ppmw of impurities entering the gas turbine results in over 10,000 lbs of ingested material for a Frame 5 type gas turbine over an 8,000-hour operating period. Even after several stages of cleanup (including barrier filters) of ambient inlet air, deposits commonly form in gas turbine compressors due to ingested particulate so that compressor washing is periodically needed to restore efficiency by removing deposits. Details on compressor fouling may be found in Meher-Homji et al (2013).

CORROSION, HOT CORROSION, AND OXIDATION

Corrosion is the wearing away of surface metals due to a chemical reaction of the metal with the environment (unlike erosion or fretting which are mechanisms of the wearing away of surface metal due to mechanical surface action). Usually the metal reacts with oxygen in the air but there are many other chemical reactions that can participate in the different corrosion mechanisms. Certain types of corrosion (such as oxidation, sulfidation, and hot corrosion) are mainly attacking the hot section of the gas turbine. Other types, such as crevice corrosion and pitting, are rather found in the compressor section of the gas turbine. For gas turbine applications the most important corrosion mechanisms are (Brun and Kurz, 2010): Oxidation, type 1 hot corrosion, type 2 hot corrosion, corrosion pitting, and crevice corrosion. These mechanisms are relevant for gas turbines and can impact their life and performance.

Corrosion can be caused both by inlet air contaminants and by fuel, water, or 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. Salt-laden air can also cause corrosion on unprotected engine parts.

Oxidation of a metal is the loss of one or more electrons, causing the metal to go from a neutral state to become a positively charged ion. This results in the formation of a surface metal oxide. Rusting is a typical oxidation process. The oxide layer on the surface can either be beneficial, as a protective barrier (also called a passivation film) over the metal, or detrimental, when the corrosion continues at a rate that leads to a rapid reduction of the mechanical properties of the metal. Oxidation is the chemical reaction between a component and the oxygen in its surrounding gaseous environment. Oxidation of turbine section components is relatively easy to predict and measures can be taken to control it since it primarily involves relatively simple metal/oxygen reactions. The oxidation rate increases with temperature. Metal loss due to oxidation can be reduced by the formation of protective oxide scales. Due to the high temperatures significant oxidation is often observed in the combustor or on the hot section turbine blades; this should be monitored carefully as it can weaken these highly stressed parts. It is important to not confuse this oxidation with type 1 and 2 hot corrosion, which will be discussed below, or with basic metal melting caused by a local hot spot.

Hot corrosion, also called high temperature corrosion requires the interaction of the metal surface with another chemical substance at elevated temperatures. Hot corrosion is a form of accelerated oxidation that is produced by the chemical reaction between a component and molten salts deposited on its surface. Hot corrosion comprises a complex series of chemical reactions, making corrosion rates very difficult to predict. It is the accelerated oxidation of alloys caused by the deposit of salts (e.g. Na_2SO_4). Type I or high temperature hot corrosion, occurs at a temperature range of 730 to 950°C. Type II or low temperature hot corrosion occurs at a temperature range of 550 to 730°C. Inside gas turbines, sulfidation, and vanadium assisted hot corrosion are the most potent causes for premature metal weakening and failure. For example hot stress corrosion cracking of turbine blades, which is effectively the continuous weakening of the blade metal surface from hot corrosion and the subsequent growth of subsurface cracks under mechanical stresses, has been the recognized root cause for many aircraft and ground based gas turbine engine failures. Hot Corrosion requires the interaction of the metal surface with sodium sulfate or potassium sulfate, salts that can form in gas turbines from the reaction of sulfur oxides, water, and sodium chloride (salt) or potassium chloride, respectively. Hot corrosion is caused by the diffusion of sulfur from the molten sodium sulfate into the metal substrate which prevents the formation of the protective oxidation film and results in rapid removal of surface metal. For hot corrosion to occur both sulfur and salt (e.g., sodium chloride or potassium chloride) have to be present in the very hot gas stream in and downstream of the combustor. Sulfur and salt can come from the inlet air, from the fuel, or water (if water is injected).

Vanadium and lead are found in certain types of liquid fuels. Vanadium and lead assisted hot corrosion mechanisms are different in that molten vanadates (vanadium oxides such as vanadium-pentoxide) or lead slag dissolve (flux) the protective metal surface oxide layer and allow the diffusion of new oxygen into the metal substrates. I.e., the protective oxide layer is destroyed, leaving the underneath metal unprotected against oxidation. As this process continues as long as either vanadates or lead is present, the surface corrosion can be very rapid. Most liquid fuels contain small traces of vanadium, potassium, and sometimes lead (from accidental mixing with leaded gasoline during transport). It is critically important to maintain the gas turbine's fuel quality within the OEM specifications for these trace elements, treat/wash the fuel, or use vanadium inhibitors if necessary.

Sulfidation is the reaction between a metal and a sulfur and oxygen-containing atmosphere to form sulfides and/or oxides. This means, in particular that sulfidation only requires the presence of sulfur in the combustion air or the fuel, but it does not require the presence of sodium or potassium. In essence, sulfidation attack is a form of accelerated oxidation resulting in rapid degradation of the substrate material due to loss of corrosion protection. Whereas during oxidation protective oxide scales can form, the metallic sulfides formed are not protective. This accounts for the rapid rate of degradation produced by sulfidation attack.

Corrosion of compressor components, which operates at much lower temperatures, is unlikely during engine operation because the compressor is dry. However, during shutdowns where cold surfaces can condense water, chemical species such as salts or sulfur compounds can be absorbed in the water producing an acidic, corrosive liquid. This liquid phase can result in *aqueous corrosion* of compressor components through a variety of mechanisms, e.g., generalized, pitting, and crevice corrosion, and stress corrosion cracking. Compressor Coatings, where used, are very effective in preventing these types of corrosion.

Corrosion Pitting, also simply called "pitting," this is a localized corrosion mechanism that leads to the formation of small but deep holes in the metal surface. As these holes are often not detected because the remainder of the metal part may appear completely clean, shiny, and polished, they represent a significant danger for unexpected failures. Pitting is often found on gas turbine compressor blades and it is caused by the intrusion of conductive impurities, such as salt water, into small surface cracks in the metal surface. As the water evaporates, the concentration of the impurity (usually sodium, sulfate, or chloride) increases, which results in highly localized corrosion and consequent deepening of the cracks. This process continues until a deep crack forms, metal pieces break off, and a pit (or hole) is formed. The pit can severely weaken the blade and also cause stress concentrations. Pitting can be initiated by very small microscopic surface defects or scratches, which are often not visually detectable. Most often, pitting on compressor blades occurs in gas turbine application where salt is ingested into the compressor (from the inlet air) and if the unit's operation is highly cyclic with many starts and stops. During any prolonged shutdown, water condensate forms on the blades, dissolves the blade salt deposits, and then enters into microscopic surface cracks to initiate the pitting process. To limit compressor blade pitting, anti-corrosive blade coatings can be utilized, and the axial compressor should be thoroughly water washed before any extended shutdown period (to remove salt deposits), and the inlet filtration system should be designed to minimize salt intrusion.

The physical process of *crevice corrosion* is similar to pitting, but rather than inside metal surface cracks, crevice corrosion occurs in pre-existing tight gaps, such as contact areas between parts, underneath grout, seals, and gaskets, or below hardened dirt or blade foulant. Concentration factors of impurities in these crevices can reach several millions. Because these areas cannot easily be inspected without disassembly, crevice corrosion presents a substantial risk of catastrophic failure. In gas turbines, crevice corrosion at the highly stressed mating surfaces between rotor blade base and disk slots can go undetected for years until either the blades are disassembled (which is usually not done unless the gas turbine is repaired/overhauled) or the blade structural support fails and they are liberated into the gas path (resulting in domestic object damage).

Other corrosion mechanisms, such as weld corrosion, microbial corrosion, galvanic corrosion, and green rot are also possible mechanisms in gas turbines and their ancillary systems, but these are less common. Hydrogen embrittlement is also sometimes observed in gas turbine fuel system if the fuel composition contains elemental hydrogen but this is not usually considered a classical corrosion mechanism.

Since all types of corrosion can be either enabled or accelerated by the presence of certain contaminants in fuel, air or water supplied to the gas turbine, inspection, maintenance and proper fuel, air and water treatment are important factors in the prevention of corrosion. Ingested contaminants can result in corrosion to the compressor, combustion, and turbine sections of gas turbine engines.

Corrosion can be controlled by proper maintenance procedures, good air filtration, attention to fuel and water. It is important to note that corrosion processes are often self-propagating, and will continue even after the source is removed or abated.

To avoid hot section corrosion, it is very important to minimize the amount of Na+K entering the machine. Maximum allowed limits for sodium and potassium (Na+K) depend on the type of gas turbine involved, and there is some variance between the different OEMs. However, typical guidelines are as follows:

- Lower-firing temperature heavy-duty models:
  0.5 to 1.0 ppm max.
- Higher-firing temperature heavy-duty models:
  0.2 to 0.5 ppm max.
- Most aeroderivative models:
  0.1 to 0.5 ppm max.

By definition these limits apply to “total Na+K” entering the hot gas path from all sources; i.e. from fuel, air and any water or steam that may be injected for NOx control or power augmentation. As can be seen, these limits are very severe.

HOT SECTION COATINGS

There are a large number of commercially available coatings for gas turbine hot sections. These coatings can be designed for increased corrosion resistance of higher oxidation resistance. Hot gas path coatings can be broken into three main groups:

- diffusion aluminide,
- overlay MCrAlY,
- thermal barrier.

Coatings are an effective approach to diminish the impact of hot section distress and deterioration along with other lines of defense such as high quality air filtration.

EROSION

Erosion is the abrasive removal of material from the flow path by hard or incompressible particles impinging on flow surfaces. These particles typically have to be larger than 10 μm in diameter to cause erosion by impact. Erosion is more a problem for aircraft engines, because state of the art filtration systems used for industrial applications will typically eliminate the bulk of the larger particles. Erosion can become a problem for engines using water droplets for inlet cooling or water washing.

Abrasive solid particles attack rotating parts. Collisions between high-speed rotating blades and airborne particles result in metal fragments being ejected from blade surfaces. Particles as small as ten microns in diameter can cause erosion. Particle composition and shape can significantly affect erosion rates. Blade profiles are so carefully designed that even minor abrasions can alter the profiles to an extent that engine performance is affected. Erosion is an expensive problem, since it causes permanent damage, eventually requiring parts refurbishment or replacement. Erosion is proportional to particle concentration and, in severe service with poor filtration, can significantly reduce engine life.

Damage may also be caused by *foreign objects* striking the flow path components. These objects may enter the engine with the inlet air or the gas compressor with the gas stream or are the result of broken off pieces of the engine itself. Pieces of certain types of ice breaking off the inlet or carbon build up breaking off from fuel nozzles can also cause damage.

ABRASION

Abrasion is caused when a rotating surface rubs on a stationary surface. Many engines use abradable surfaces, where a certain amount of rubbing is allowed during the run-in of the engine, in order to establish proper clearances. The material removal will typically increase seal or tip gaps. Part of this is also age related, as bearings tend to become softer (reduction in stiffness) due to an increase in clearance over time that causes an increase in journal orbital amplitude.

While some of the effects of fouling can be reversed by cleaning or washing the engine, most other types of damage 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 online and offline 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. There are some grey areas, because some degradation effects can be recovered by control system adjustments (that are, however, difficult to perform in the field due to limited capabilities to measure mass flow and performance). It should be noted, that the determination of the exact amount of performance degradation in the field is rather difficult. Test uncertainties are typically significant, especially if package instrumentation as opposed to a calibrated test facility is used. Even trending involves some uncertainties, because in all cases, the engine performance has to be corrected from datum conditions to a reference condition.

PARTICLE FUSING

The fusion of particles on hot surfaces leads to another source of problems. While dry, 2 to 10 μm size particles could pass through older engines, causing little or no damage. However, these particles can cause problems in new generation, hotter running engines. If the fusion temperature of the particles is lower than the turbine operating temperature, the particles will melt and stick to hot metal surfaces. This can cause severe problems since the resultant molten mass can block cooling passages, alter surface shape, and severely interfere with heat transfer, often leading to thermal fatigue. Affected surfaces are usually permanently damaged and will eventually need replacement.

MECHANICAL DEGRADATION

Causes of mechanical degradation include wear in bearings and seals, coupling problems, excessive vibration and noise, or problems in the lube oil system. Mechanical degradation includes:

- Gas turbine component creep or thermal ratcheting. The creep deformation of a nozzle can cause aerodynamic problems and performance changes.
- Bearing wear and increased losses
- Gearbox losses
- Coupling problems
- Excessive misalignment causing higher bearing loads and losses
- Combustor nozzle coking or mechanical damage
- Excessive rotor unbalance
- Excessive parasitic loads in auxiliary systems due to malfunctions
- Excessive mechanical losses in the driven equipment (generator or compressor)

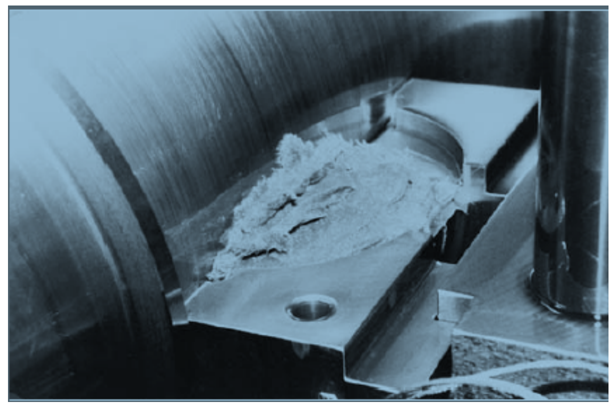

Figure 5: Uncleanliness in gear box

It is interesting to note that mechanical deterioration can also be caused by improper maintenance activities or the lack of proper inspection procedures prior to reassembly of machinery. Rags that were found in the oil galley of a journal bearing on a high-speed gearbox of a 20 MW gas turbine generator train can be seen in Figure 5. These were found in both the drive and non-drive end bearings when the gearbox was opened up after an extended run.

Probably the most common indicator of mechanical degradation has been vibration. It is most important to note that several problems that manifest themselves as vibration may in fact have underlying causes that are aerodynamic (or performance) related in nature.

Bearing problems are often caused by low oil pressure (malfunction in pump or leaks), line blockage, or excessive loads due to factors such as misalignment. The lube oil supply pressure and scavenge temperatures can be measured and correlated to a parameter such as rotor speed. The expected pattern during speed changes can be noted and subsequent checks made during transients.

Combustor fuel nozzles can, at times, plug up. There can be several causes for this such as coking, erosion, and misassembly. Temperature distortions can create a host of problems in the hot section. Severe temperature distortions can create serious dynamic loads on blading possibly inducing fatigue problems. The pattern of the EGT spreads can be monitored during transient conditions to indicate nozzle problems. Blade failures that can be induced by performance and mechanical factors have been detailed by Meher-Homji and Gabriles (1998), and specific issues and deterioration problems related to gas and liquid fuels are detailed in Meher-Homji and Bromley (2010)

DEGRADATION OF COMPONENTS

Blades

Compressors experience fouling, erosion, deposits, corrosion, and other damage on the individual airfoil (blades). Fouling and, to some extent, erosion generate a blade surface with increased roughness. Any increased roughness can increase the friction losses. It also may cause early transition from laminar to turbulent boundary layers, which increases loss production. Erosion, deposits, or damages to the airfoil change the geometric shape of the airfoil. On a well-designed airfoil, optimized for the application, this will always reduce the performance of the airfoil. The deterioration of the turbine blades is accompanied by changes in exit angles and increased losses. Profile loss increases of 100% for 1% added or reduced thickness on well-designed subsonic turbine blades are not unusual. Increased surface roughness also causes performance degradation. The main influence of degradation appears around the optimum incidence angles, while the far off-optimum performance is less influenced. Added roughness on the pressure side of the blades has a smaller effect compared to added roughness on the suction side.

Thicker boundary layers on the blades and sidewalls reduce the flow capacity, especially near choking conditions. On the other hand, if the trailing edge erodes, the throat width of the blade is increased, thus allowing more flow but with lower work capability. In general, all these influences will create higher losses and less turning (in other words, less work is exchanged between blade and working fluid). This means that the following row of airfoils will see different incidence angles, higher temperatures, lower (for compressors) or higher (for turbines) pressures, different densities and flow velocities.

Erosion typically has the most significant effects on the blade leading edges (Figure 5). This can significantly affect the location and extent of the transition from laminar to turbulent boundary layers. Because the heat transfer characteristics of a boundary layer depend in addition to its thickness on its state (i.e., laminar, turbulent, transitional, separated), leading edge erosion can influence the heat balance of the blade.

Giebmanns et al., (2012) evaluated the effect of a contributor to non-recoverable degradation, the modification of leading edge contours on transonic compressor airfoils: Again, both efficiency and flow capacity suffer, but in this case, efficiency drops more severely than flow capacity and pressure ratio. Detailed evaluation of the loss behavior in a cascade showed that most of the efficiency reduction can be attributed to increased shock losses.

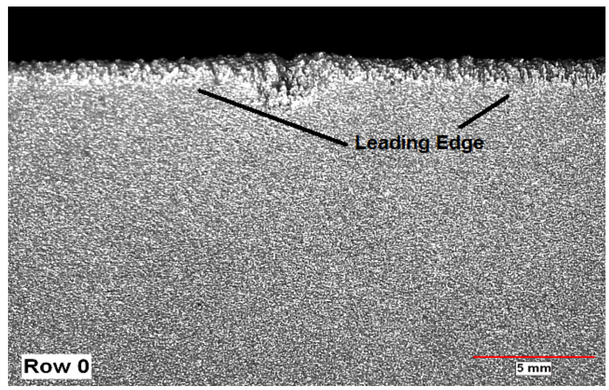

Figure 6: Eroded leading edge of a rotor blade of the first stage of a compressor.

Matz et al., (2008) report results form an evaluation about a gas turbine with evaporative cooling after 4,000 hours of operation. They found erosion of the compressor leading edges at the first ten stages of the compressor. A possible explanation was that the droplets were large enough to survive past the theoretical prediction of their complete evaporation. The large droplets were suspected to be remnants of surface films being shed from upstream blade rows.

Turbine blades and nozzles, when subjected to increased surface roughness or local erosion damage show increased profile losses. Additionally, the resulting increased boundary layer thickness reduces the flow capacity, especially for blades or nozzles operating near their choke limit. On the other hand, erosion or corrosion may affect trailing edges, which typically leads to an increase in flow capacity, usually accompanied by a reduction in work produced.

Clearances

Clearances between stationary and rotating parts (i.e., between stationary blades and the rotating hub or between rotating blades and the stationary casing) have a tendency to open up during the aging process of the equipment. This results in higher and unwelcome leakage flows. These leakage flows reduce the possible head capability and the efficiency of the components. In compressors, an increase of the rotor tip clearance from 1% of blade chord to 3.5% of blade chord reduces the pressure ratio of the stage by up to 15%.

The loss production is due to intensive mixing of leakage flows with the main flow, thus producing losses and reducing the effective through flow area (blockage). The effects of tip clearance and added roughness after ingestion of sand lead to about the same magnitude of performance deterioration. However, the degradation mechanism by sand ingestion is more relevant for aero applications than for industrial applications.

Giebmanns et al., (2013) present data on the effect of increased clearance in transonic fan blades. For small increases in clearance, they detect a near linear influence on mass flow reduction and efficiency reduction. In their data, for a given increase in clearance, efficiency degraded slower than the flow capacity. Similar results were found by other researchers (ref Kurz, GT2000-0345).

Compressor

Three major effects determine the performance deterioration of the compressor: (1) increased tip clearances; (2) changes in airfoil geometry, and (3) changes in airfoil surface quality. While the first two effects typically lead to non-recoverable degradation, the latter effect can be at least partially reversed by washing the compressor.

The overall effect of degradation on an engine compressor can be described through added losses, less flow capacity, and lower capability of generating head (i.e., pressure ratio). Typically, a degraded compressor will also have a reduced surge or stall margin. This will not have any significant effect on the steady state operation unless other factors aggravate the situation.

The degradation of individual stages has a cumulative effect: While in a new machine all stages are working at their optimum efficiency point at design surge margins, stage degradation will force all stages after the first one to work at off optimum inlet flow. Therefore, they operate at lower than design efficiency. This will not only lower the overall efficiency and the pressure ratio that can be achieved, but also the operating range.

Rodriguez et al., (2013), used a stage stacking technique where they assign efficiency and flow deterioration factors to individual stages of the gas turbine compressor. Their results indicate that efficiency and, to some lesser degree, mass flow decline particularly steep at high compressor Mach numbers (i.e., at full load, maximum gas generator speed, and low ambient temperatures). They also find that fouling of the front stages tends to have its biggest impact on throughput, while fouling in the rear stages has a bigger impact on compressor efficiency.

Damage Caused by Hydrocarbon Liquids in Fuel Gas

Gas turbine hot section damage can occur due to the presence of liquids in the fuel gas, plugging of fuel nozzles, or due to internal gas turbine fires and explosions. The presence of hydrocarbon liquid can cause over-firing, either in all or a few combustion chambers. The presence of liquid hydrocarbons in natural gas depends on the temperature and pressure of the gas. Despite precautions, it is not uncommon for natural gas to entrain liquids. Liquid carryover from pipeline gas scrubbers can occur due to:

- Foaming
- Gas flow and pressure that exceeds scrubber capacity
- Improper operation of a centrifugal separator
- Surges in liquid due to rapid fluctuations in gas pressure

Liquids have different volumetric heating values and have different flow characteristics in piping and fuel nozzles. Thus, when liquids are present in natural gas, gas turbine performance is affected depending on the concentration of the liquid compared to the gas. At low concentration levels, the liquid is typically in aerosol form, and the turbine control system will react by reducing fuel flow due to the apparent higher heating value of the fuel. The flame will change color from transparent blue to a luminous flame with a color ranging from yellow to red. This can also occur if iron sulfide or sodium compounds are present. With liquids in the fuel, the combustion may be rough, resulting in gas supply pipe vibration causing wear of transition pieces or manifolds. At moderate liquid concentration levels, the liquid gets segregated due to inertia effects and is unevenly distributed to the fuel distribution system. The control system may then behave in an unstable manner because of exhaust gas temperature variations.

Combustion imbalance from one combustion chamber to another can cause high exhaust gas spreads in excess of 45°C (80°F) which have a damaging effect on the blades. If a significant level of liquid is present in the gas, (causing the heating value to be greater than 110 percent of the nominal value) then rapid destruction of hot gas components can occur. Exhaust gas temperature spreads can become exceedingly high and in extreme cases, rotating blades can melt. If the melting is uniform, the vibration levels may not increase. Complete hot gas path destruction can, at times, occur in a period of five to ten minutes.

Damage due to liquids in the natural gas fuel can include:

- Damage to the premixer hardware and combustor due to autoignition, flashback, or pulsations.
- Transition piece failures: This can be due to several factors. Greater cyclic stresses due to liquids being introduced intermittently cause damaging pressure pulsations. At times, seals rupture allowing compressor discharge air to enter at the first stage nozzle which causes severe temperature distortion and is damaging to the blades. Unburned liquid droplets can ignite creating a flame near the stator nozzle.
- Thermal distress of blading, caused by a distorted temperature profile.
- Compressor surge, due to rapid increase in back pressure.

It is therefore extremely important for natural gas fuel to be properly treated *and have a superheat temperature of at least 28℃ (50℉) above the hydrocarbon or moisture dew point*. As was mentioned in an earlier section, gas analysis for determination of dew point should consider all components in the gas. It is also important to note that in a case where several gas turbines are being fed by a fuel line, depending on the location of the gas turbine, a situation may exist where some experience problems and other do not.

Low Fuel Nozzle Pressure Ratio and Excessive Pressure Fluctuations

The combustor fuel nozzle tip acts as a metering orifice. The fuel nozzle pressure ratio (FNPR) is defined as

FNPR = (Fuel gas pressure)/(Combustor liner pressure)

Excessive pressure fluctuations in the combustor liner would propagate upstream into the fuel nozzle. These pressure pulsations coupled with low fuel nozzle pressure ratios cause a high variation in the fuel flow and heat release. Pulsations can damage hot gas components. Liquids in the fuel can contribute to a lower fuel nozzle pressure ratio. The liquids cause an increase in the heating value per unit volume. Consequently, a smaller volumetric flow is required. The reduction in flow results in lower nozzle pressure ratios which further compound the pulsation problems.

Problems in the fuel system that can cause blockage of the fuel nozzles such as coking will also cause severe vibratory stresses on the blades. Excessive coking on the fuel nozzles will cause:

- Higher pressure to occur on the other fuel nozzles causing the flame to move downstream in the combustor
- Flow and temperature distortion which can damage the turbine

Balancing of nozzle flow should be done as precisely as possible. A target of 1 percent or less is recommended and will have a long-term benefit to the operation of the turbine. It is important to note that the blockage of the fuel nozzles may also be due to external factors such as debris derived from the fuel filters. It is most important to have a means for detecting such problems by an EGT monitoring system which examines qualitative pattern changes and not just absolute limits.

Turbine Section

Three major effects determine the performance deterioration of the turbine section: (1) increased tip clearances; (2) changes in airfoil geometry, and (3) changes in airfoil surface quality. Unlike in the compressor, high operating temperatures in turbine sections usually lead to non-recoverable degradation.

The overall effect of degradation on the turbine (both gas generator and power turbine) can be described through added losses, different flow capacity, and lower capability to perform work. It should be noted that degradation in the turbine section can impact the flow capacity in different ways: Erosion, especially of the trailing edges, may increase the flow capacity, while changes in airfoil quality may reduce the flow capacity because a thicker boundary layer will increase blockage.

The degradation of individual stages has a cumulative effect: While in a new machine all stages are working at their optimum efficiency point at design surge margins, stage degradation will force all stages after the first one to work at off optimum inlet flow. Therefore, they operate at lower than design efficiency. This will lower the overall efficiency. It also will change the flow capacity of the turbine, which subsequently impacts the operating point of the engine compressor.

Recoverable and Non-Recoverable Degradation

The distinction between recoverable and non-recoverable degradation is somewhat misleading. The majority of degradation is recoverable; however, the effort is very different depending on the type of degradation. The recovery effort may be as small as water or detergent online washing or detergent on-crank washing. The degradation recovery by any means of washing is usually referred to as recoverable degradation. Figure 7 shows how a combination of frequent online washing and occasional offline washing can slow the engine performance deterioration by restoring the engine compressor performance. However, a certain amount of degradation can be recovered by engine adjustments (such as resetting variable geometry). Last but not least, various degrees of component replacement in overhaul can bring the system performance back to as-new conditions.

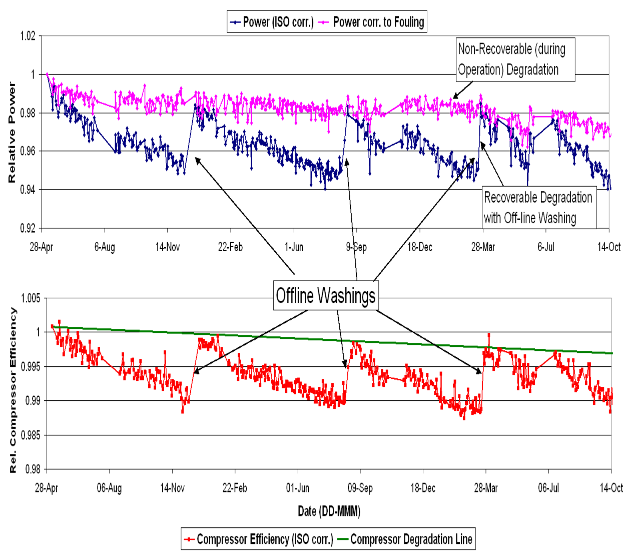

Figure 7: Impact of fouling, online and offline washing on compressor performance and engine power output (Hepperle et al., 2011)

PERFORMANCE IMPACT OF DEGRADATION

Degradation of engine components has a compounded effect on engine performance, because the change in component performance characteristics leads to a mismatch of these components on the engine level, as well as on the component level. The impact of individual component degradation is also influenced by the control system and the control modes of the engine. Single-shaft engines, operating at constant speed, will show different degradation behavior than two-shaft engines. The impact of degradation on two-shaft engines depends on the control mode they are in, i.e., whether the gas generator speed or the firing temperature is the limiting factor. Additionally, the method and location of measuring the control temperature will determine the behavior of the engine in degraded conditions (Kurz and Brun, 2001).

It should be noted that separate effects may occur together and affect several components: Compressor degradation will impact pressure ratio, efficiency, and flow capacity, albeit to various degrees depending on the type of degradation. The individual impact of reduced compressor efficiency (predominantly due to fouling), reduced compressor flow capacity (opening of clearances and fouling), altered gas producer turbine flow capacity (corrosion, erosion, fouling), and reduced gas producer turbine efficiency (fouling, erosion) cannot be treated as isolated events, but each of them impacts the operating conditions of other components. Since in many instances, the operating point on the map of a component is different (but not very different) between the new and the degraded engine, the actual efficiency in the cycle calculation is the result of component degradation *and* the move of the operating point.

An observed reduction in engine flow is a result of a higher power consumption of the compressor (generally related to a deteriorated compressor), or the reduced flow capacity of the turbine section. Frith cropped the blades of the compressor of a two-shaft turboprop engine, thus simulating the effect of tip rubs. This crop reduced both airflow and compressor efficiency, but also compressor pressure ratio. This indicates that the increased power consumption of the compressor leads to a reduced pressure ratio. This in turn (since the flow capacity of the turbine was not changed), causes a reduction in air flow.

As will be seen in the examples below as well as in studies by Kappis (2013) and Kappis and Guidati (2012), the impact of component degradation on the overall engine performance depends on the engine load and the ambient temperature. The engine compressor, as stated earlier, will become more front loaded as the ambient temperature is increased, and thus becomes more sensitive to fouling of the front stages with increased ambient temperature.

Reduced Compressor Flow Capacity

Let us consider the impact of reduced compressor flow capacity, which can be the result of fouling or increased clearances (Figure 8). Spakovszky et al., as well as Khalid et al., relate increased rotor clearances to an increase in flow blockage, thus reduced compressor flow capacity. We find the same level of compressor flow blockage leading to more power degradation in a two-shaft engine than in a single-shaft engine. It is interesting that both for single-shaft or for two-shaft engines, there is a very small increase in heat rate due to the compressor blockage at higher ambient temperatures (Figure 9 and Figure 10). Only for temperatures below 50°F, the heat rate starts to increase slightly at about the same rate in single-shaft and two-shaft machines.

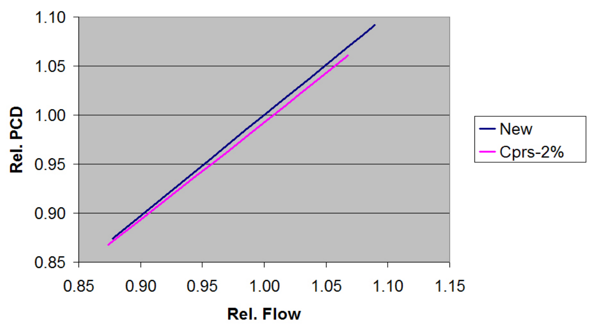

Figure 8: Impact of reduced compressor flow capacity (reduction of 2% from new compressor) on the operating lines of the compressor in a two-shaft gas turbine

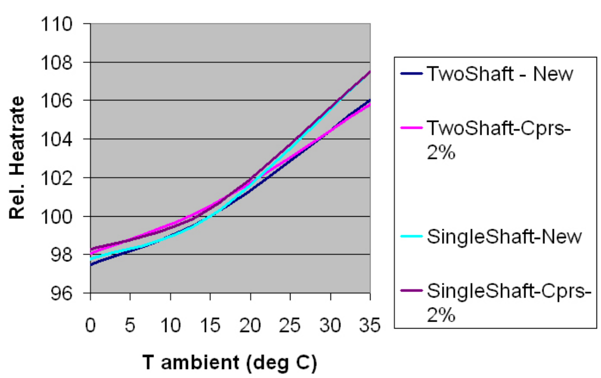

Figure 9: Impact of reduced compressor flow capacity (reduction by 2%) on full load heat rate for a single-shaft and a two-shaft gas turbine at varying ambient temperatures

It should be noted, that the flow blockage and efficiency deterioration of the engine compressor will likely be higher for higher ambient temperatures (Kappis and Guidati, 2012). The study below assumes constant compressor deterioration.

Also, the loss in available power is more pronounced at low ambient temperatures. This is probably due to higher Mach numbers at low ambient temperatures, which make the compressor more sensitive to changes in flow capacity.

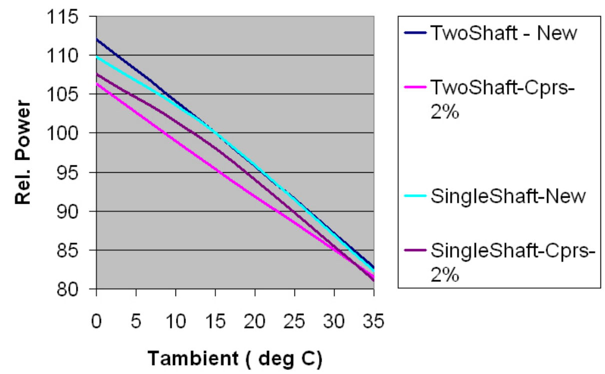

Figure 10: Impact of reduced compressor flow capacity (reduction by 2%) on full load power for a single-shaft and a two-shaft a gas turbine at varying ambient temperatures

An operating point at part load (i.e., below maximum firing temperature and below maximum gas generator speed) can still be maintained with a degraded engine, albeit at a higher firing temperature and a higher gas generator speed than for the new condition (Figure 11).

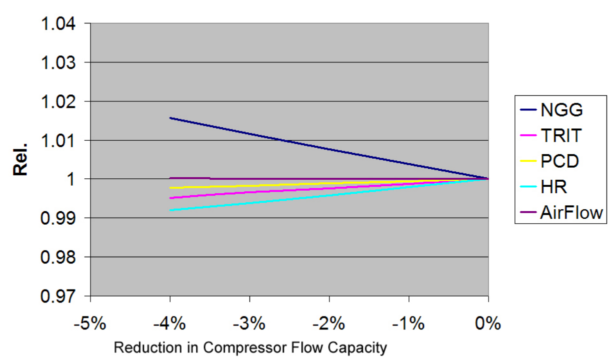

Figure 11: Impact of reduced compressor flow capacity on gas generator speed, firing temperature, compressor discharge pressure, heat rate, and air flow in a gas turbine at constant load

The relative loss in efficiency is significantly lower than for an engine at full load for the same amount of degradation. Airflow and pressure change very little. Compressor deterioration by itself will usually cause higher power losses than losses in heat rate, because a higher compressor exit temperature (due to lower efficiency) at a fixed firing temperature will reduce the possible fuel flow.

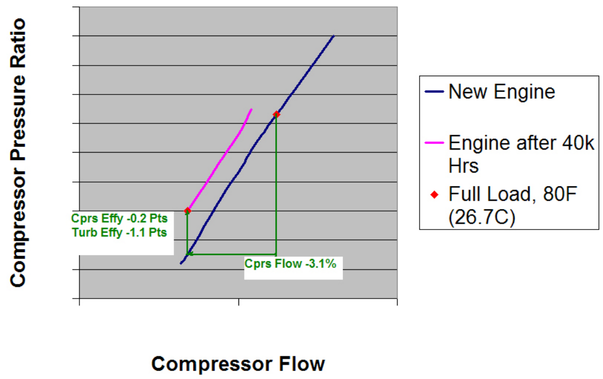

Figure 12: Compressor operating line for a gas turbine after 40,000 actual operating hours (Test data)

Due to component interactions we also find, that the level of degradation on any component causes different levels of power and efficiency degradation for the engine depending on load and ambient conditions.

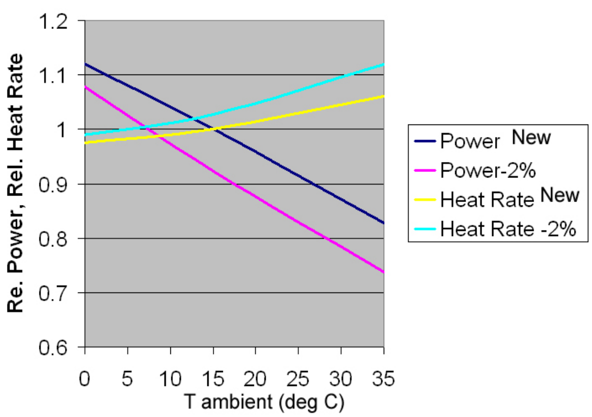

Figure 13: Impact of reduced gas generator turbine efficiency (by 2%) on full load power and heat rate for a two-shaft gas turbine

Reduced Gas Generator Efficiency

As another example, we discuss the impact of reduced gas generator efficiency. Degradation in the hot section is mostly not recoverable, unless the engine is overhauled. Deposits from sulfur containing fuels or damage to coatings, as well as the effects of corrosion and erosion, will reduce the efficiency of the gas generator turbine. As shown in Figure 13, the impact on power and heat rate increases with increasing ambient temperature.

Freire et al., provide operational data for a 23 MW aeroderivative gas turbine in offshore service. The gas turbine was running on liquid fuel (Diesel) for the first 3,300 hours and thereafter on natural gas. Data was taken for a total of 22,000 hours. The engine was water washed regularly. The data shows significant scatter (which is not surprising due to water washing interventions), but indicates a linear loss of efficiency for both compressors, as well as the overall engine. The gas generator turbine deteriorates very fast during the operation on Diesel, but afterwards also shows a linear deterioration. The power turbine shows almost no (less than 0.2%) deterioration in efficiency. The overall engine efficiency was reduced by 2.5% after 22,000 hours. Noteworthy is in particular the rapid performance degradation of the gas generator turbine during the 3,300 hours of operation on Diesel (over 2.5%), which points to the impact of fuel on degradation.

Compressor Contamination

Kappis and Guidati (2012) simulated different degradation scenarios and their impact on the compressor performance. They evaluated a situation, where all compressor stages are equally fouled, with situations where the compressor front stages are heavier fouled than the rear stages, as well as situations where the rear stages are heavier fouled than the front stages. Heavier fouling of the front stages is the normal result of foulant ingestion, as reported, for example, by Tarabrin et al., (1998). Heavier fouling of the rear stages is often the result of online water washing, where part of the dirt is moved from the front stages to the rear stages. They show that both uniform fouling as well as heavier fouling on the front stages or rear stages reduces compressor mass flow and efficiency more for low aerodynamic speeds (i.e., higher ambient temperatures). However, the impact on compressor performance of more heavily fouled rear stages was significantly lower than for the more heavily fouled front stages for all ambient temperatures.

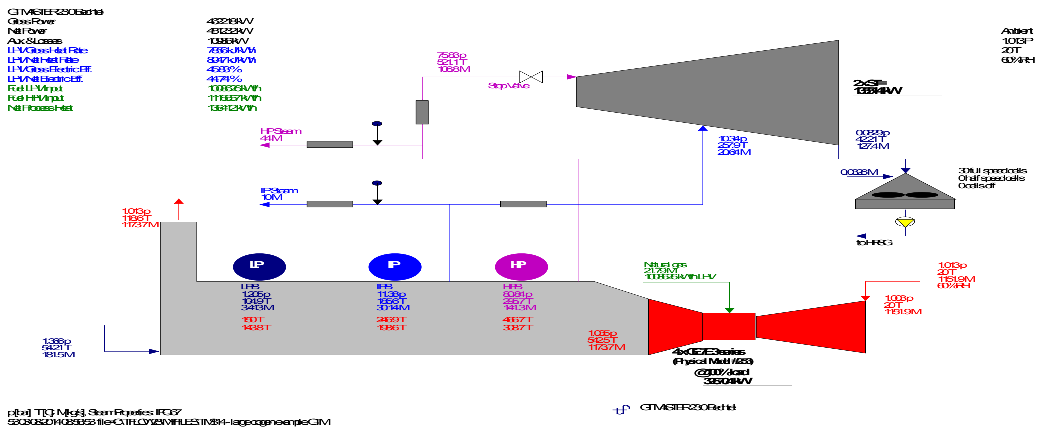

Figure 14: Large Cogen plant – new and clean condition

Combined Cycles

With the introduction of high performance gas turbines and increasing fuel costs, maintaining of the highest efficiency in gas turbine combined cycle/ Cogen systems becomes an imperative. To show the importance of deterioration effects on the overall cycle, let us consider a hypothetical large scale combined cycle facility made up of four Frame 7EA units making process steam and power that would be typical in a large refinery like a Cogen facility as shown in Figure 13. This is a representative design using thermal heat balance software. In this diagram, all pressures (P) are in bara, temperatures in C, and mass flow in kg/sec.

Salient features of the system in a new and clean state:

- Configuration- 4 GTs + 2 condensing ST, air cooled condenser
- GT power (4 gas turbines) = 326.7 MWe
- ST power (2 condensing steam turbines) = 135.5 MWe
- Net process heat = 136,412 kW th (HP steam and IP steam)
- Net electric efficiency (LHV) = 44.34%
- Net plant power = 451,232 kWe
- CHP efficiency = 58.26%
- ST throttle conditions 76 bara/521C ; LPT exit pressure = 0.0829 bara
- Fuel = natural gas , supplied @ 25 C LHV @ 25 C = 46280.22 kJ/kg
- G.T. @ 100 % rating, inferred TIT control model, CC limit
- Site ambient conditions: 1.013 bar, 20 C, 60% RH
- Total inlet loss = 10 millibar, exhaust loss = 21.98 millibar
- Inlet filter = 10 millibar; duct & stack = 5.00, HRSG = 16.98 millibar

The fuel cost for 20 years of operation at 5.5 $/MMBTU for this facility without assuming any fuel escalation would be $2.8 Billion. Assuming moderate escalation of 2%, the cost would be closer to $3.5 billion. The revenue from the electricity considering 7 c/KWhr over twenty years would be $5.5 Billion and with escalation around $6.68 Billion. Clearly, the importance of understanding and controlling performance deterioration within the full cycle is of paramount importance.

The situation with a deteriorated gas turbines is shown in Figure 15. The deterioration imposed is a 5% reduction in GT airflow and a decrease of 3% in the GT compressor efficiency. The process steam flows have been kept constant, as these are of critical importance to most Cogen plant steam recipients.

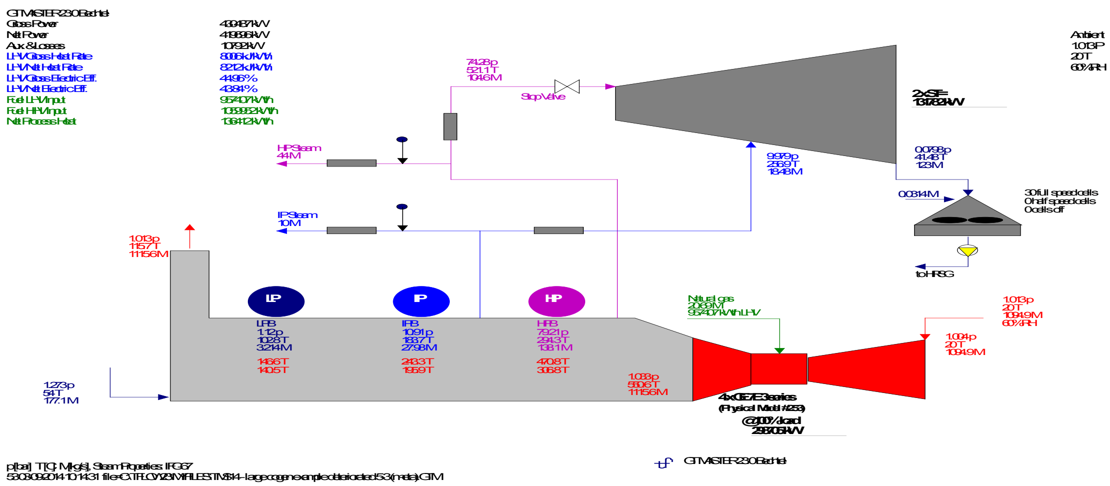

Figure 15: Large Cogen plant – deteriorated conditions

Table 1. Cogen plant salient parameters, under new and clean and deteriorated conditions.

Parameter	New and Clean	Deteriorated
Net Plant Power, kW	451,232	419,616
GT Power, kW	326,704	298,705
ST Power, kW	135,514	131,782
LHV Heat Rate, KJ/kwHr	8,047	8,212
ST Throttle P(bara)/T DegC/M (kg/sec)	75.83 bara/521C/ 106.8 kg/sec	74.28 bara/521C/ 104.6 kg/sec

A comparison of the new and clean and deteriorated situation is shown in Table 1. The revenue and fuel consumption are shown in Figure 16. In this figure, an annual escalation of 2% has been considered for both fuel and electricity. As is typically the case, the loss in revenue from electrical sales is a major factor. Under GT deteriorated conditions, it is often a source of surprise to see that at base load conditions, the fuel flow actually decreases by a small amount, hence the fuel costs decrease. While it is true that the GT efficiency worsens, the absolute value of the fuel (kg/sec) consumed drops. The gas turbine cycle parameters are shown in Figure 14 and Figure 15.

With the deregulation of the electric utility market and highly competitive cogeneration market and an environment in which fuel costs are high, plant operators have to understand and control all forms of gas turbine performance deterioration. The sections ahead will cover the sensitivity of combined cycles to various deterioration factors.

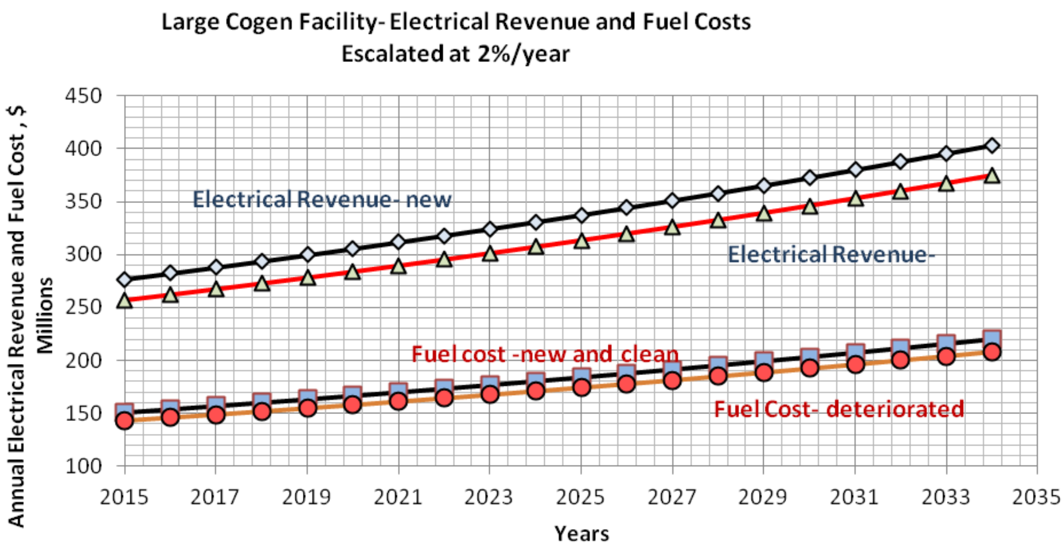

Figure 16: Cogen plant electrical revenues and fuel costs over time (escalation assumed 2%)

Degradation can occur due to the deterioration of the individual components of the combined cycle which include:

The Gas Turbine- this is by far the most sensitive component and can be impacted by compressor fouling of the compressor and turbine section, erosion, corrosion/ oxidation, and impacts of increases in inlet filter differential pressure. Problems can include both recoverable and unrecoverable deterioration. As the GT contributes the largest contribution to CCPP degradation, operators should consider the best air filters, and condition monitoring to monitor degradation. Careful attention should also be paid to fuel gas treatment and scrubbing and, in the case of liquid fuels, careful monitoring of contamination should be done. A careful program of online and crank washing should be instituted to minimize the impact of compressor fouling.

HRSG – Degradation in the HRSG can be caused due to fouling and scaling or sludge buildup on the water side tube surfaces in the evaporator section. Some oxidation can also occur in the superheater tubes where scale can develop. Scaling can affect the heat transfer in the tubes, causing a loss in heat transfer. Some degradation can be recovered by gas side and water side cleaning. Replacing gas side baffles and seals can also help in performance retention. Degradation of the HRSG can be expressed and modeled in terms of water and gas side fouling factors. Details may be found in Pasha et al., (2005). In terms of control, operators should ensure good water chemistry to minimize fouling and operational problems. Trending of key HRSG performance parameter by using DCS information is also helpful, especially if it is coupled to advanced heat balance software that can help discriminate between actual degradation and off design operation effects.

Steam Turbine- similar to the GT and HRSG. Degradation here is both recoverable and non-recoverable. Degradation can be caused by a host of factors including turbine rubs and increased clearances, causing increased steam leakage at the stages, diaphragm, and end seals. Solid particle erosion can also be a factor when iron oxide particles are entrained in the steam causing erosion of the steam path - typically in the nozzles, control valves, first HP, IP, and reheat stages. Moisture in the LP blades (Wilson line) can cause erosion in the LP section.

Condenser- depending on the type of condenser, deterioration here or air in leakage can increase the back pressure on the ST, causing an increase in the heat rate of the cycle. In the case of water cooled condensers, fouling and growth of slime films can impair the heat transfer, increasing the water side fouling factor. Some recovery can be attained by tube cleaning and by replacing seals and gaskets.

Parametric Analysis of Combined Cycle Degradation

In order to evaluate degradation effects of a gas turbine in a combined cycle, a thermodynamic model was created of a CCPP plant as shown in Figure 17. The combined cycle consists of one gas turbine rated at 81.6 MWe at site conditions of 20°C and 60% RH.

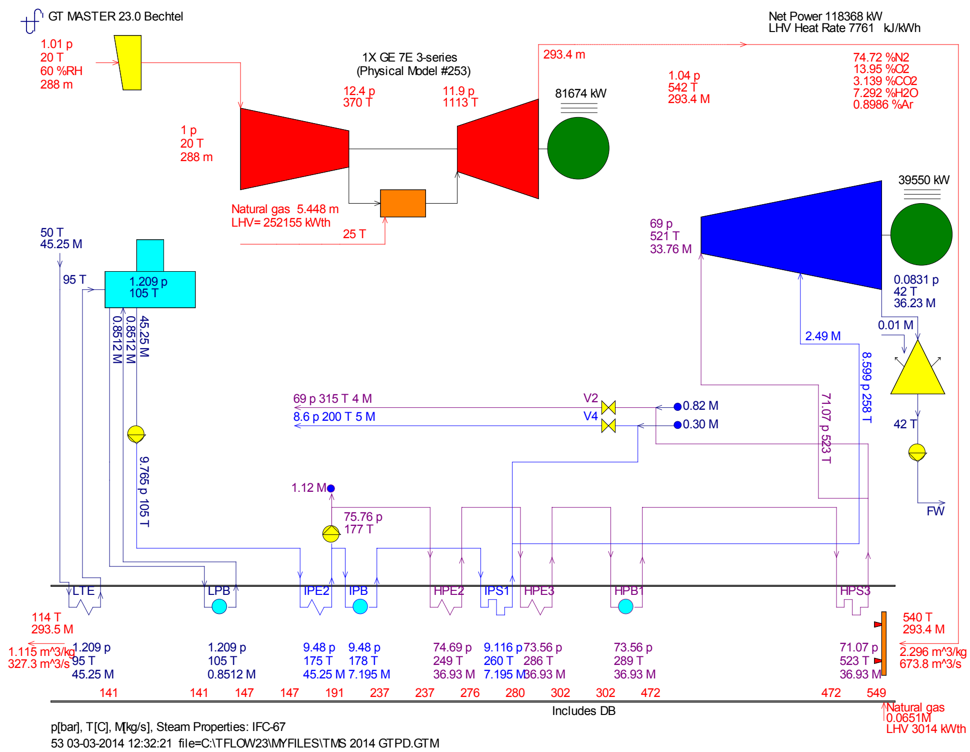

Figure 17: Cycle schematic of Cogen system for parametric analysis

The cycle consists of a 2-pressure level HRSG (Note that the LP drum shown is just for the de-aerator). Salient features of this cycle include:

- Gross Power = 121,224 kW
- Net Power = 118,368 kW
- Aux and Losses = 2,857 kW
- LHV Net Heat Rate = 7,761 kJ/kWhr
- LHV Net Electric Eff = 46.7 %
- Net Process Heat = 22,689 kWth

To evaluate the degradation effects, the following items were examined:

- Inlet filter pressure drop on the CCPP performance
- GTG compressor fouling effects (flow and efficiency reduction of axial flow compressor)
- GTG turbine section reduction in efficiency effects
- Diverter valve leakage

IMPACT OF INLET FILTER PRESSURE DROP

The impact on the combined cycle of an increase in the inlet filter pressure loss is shown in Figure 18. The inlet filter differential pressure is increased from the normal value of 10 mbar through 20 mb, and the impact on key CCPP parameter are depicted in the figure normalized to the value at the base case. Impact of Gas Turbine Compressor Fouling

To examine the impact of compressor fouling a number of fouling steps on the GT axial flow compressor were imposed- as axial compressor fouling includes both a drop in mass flow and compressor efficiency. The steps are shown in Table 2

Table 2: Fouling imposed

Fouling Step	Drop In Mass Flow % Points	Drop In Compressor Eff % Points
0 (No Deterioration)	0	0
1	1	0.5
2	2	1
3	3	1.5
4	4	2
5	5	2.5
6	6	3
7	7	3.5

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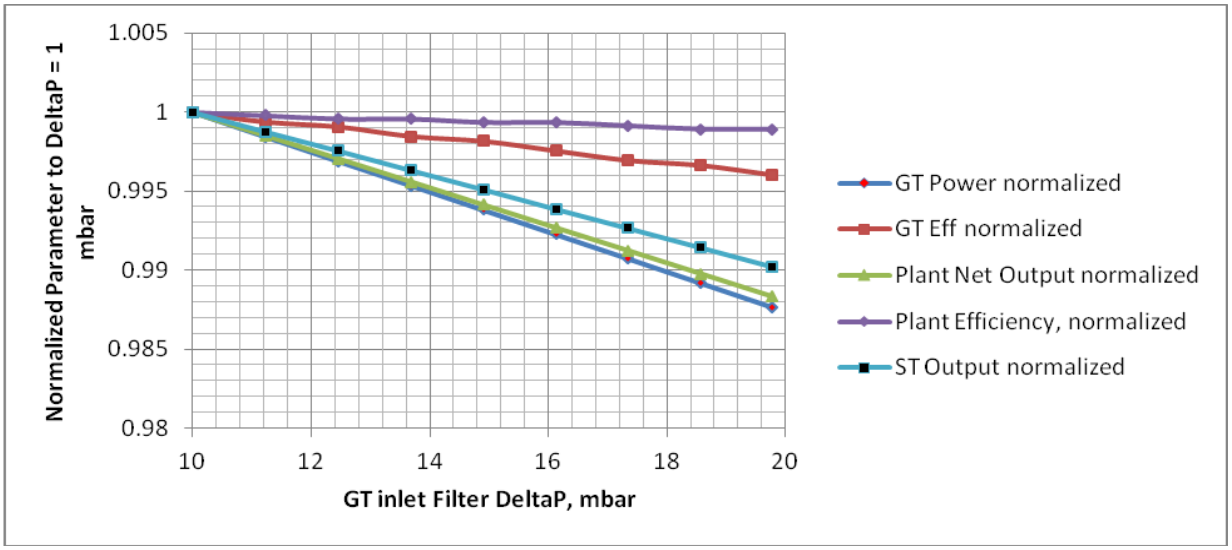

Figure 18: Impact of variation in GT inlet differential pressure increase on CCPP parameters

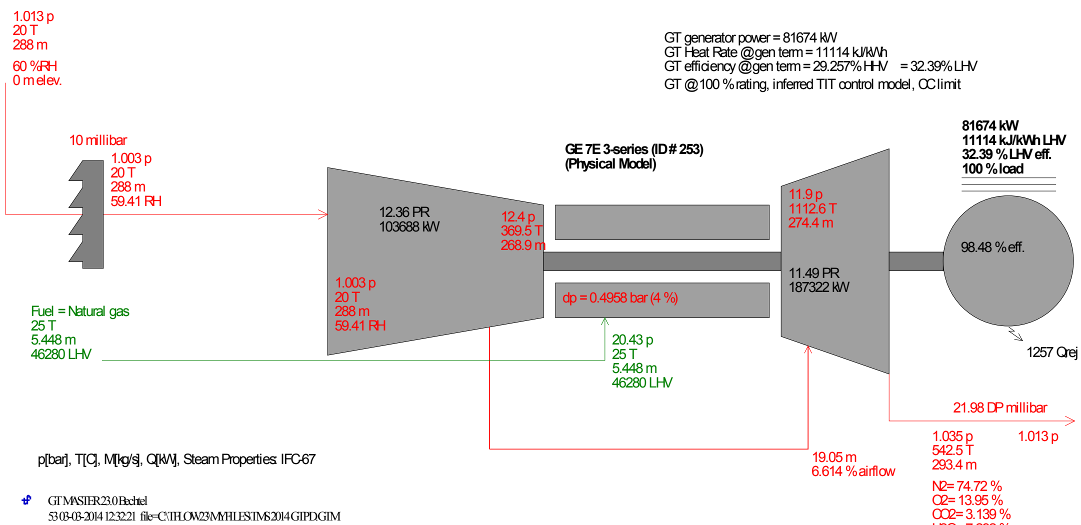

Figure 19: GT cycle (Undeteriorated)

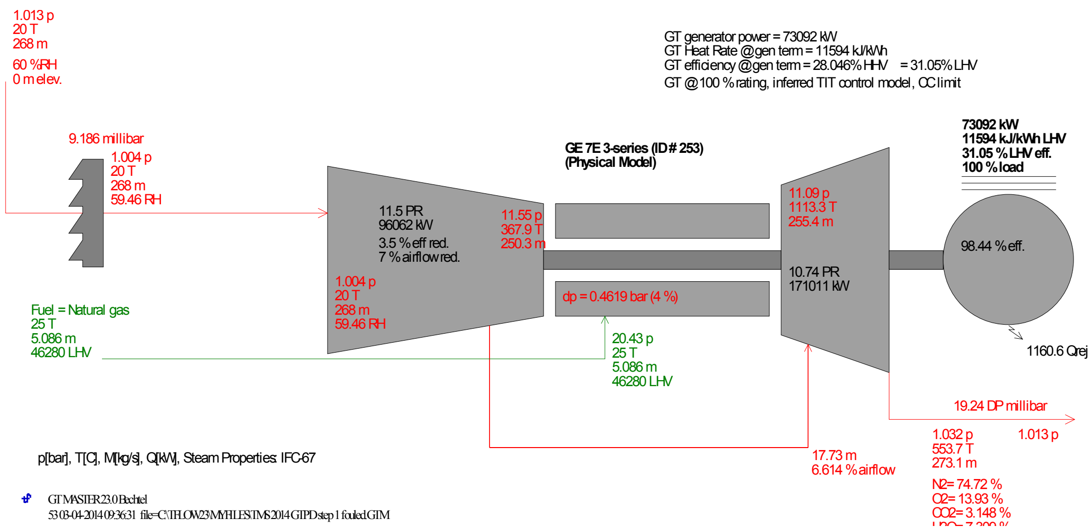

Figure 20: GT Cycle (Deteriorated, 7% airflow drop and 3.5% efficiency reduction). Cycle parameters shown on figure

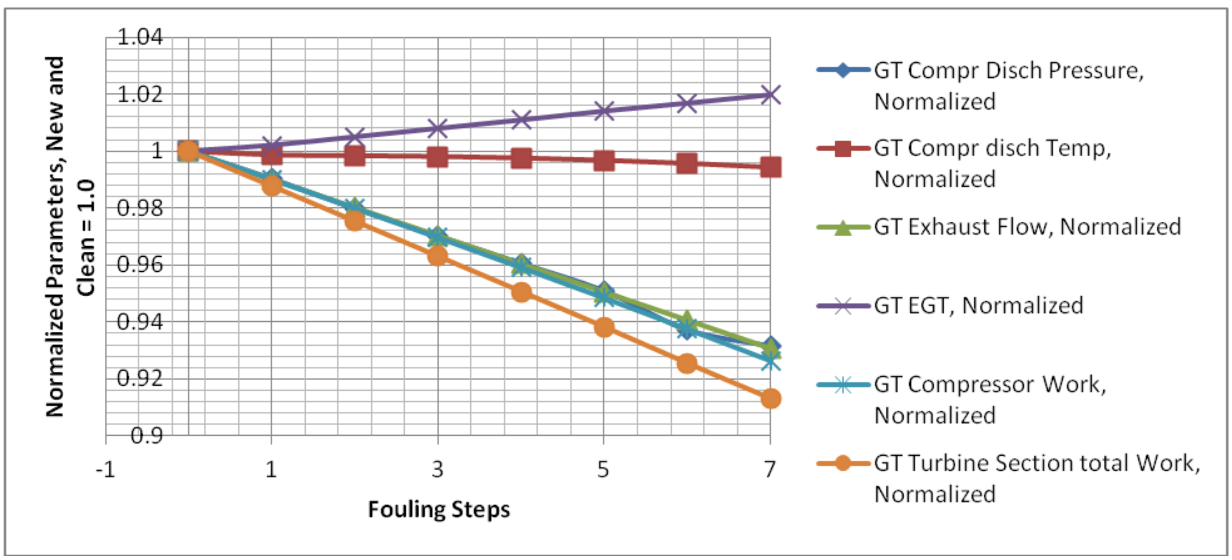

Figure 21: Impact of GT fouling on GT parameters

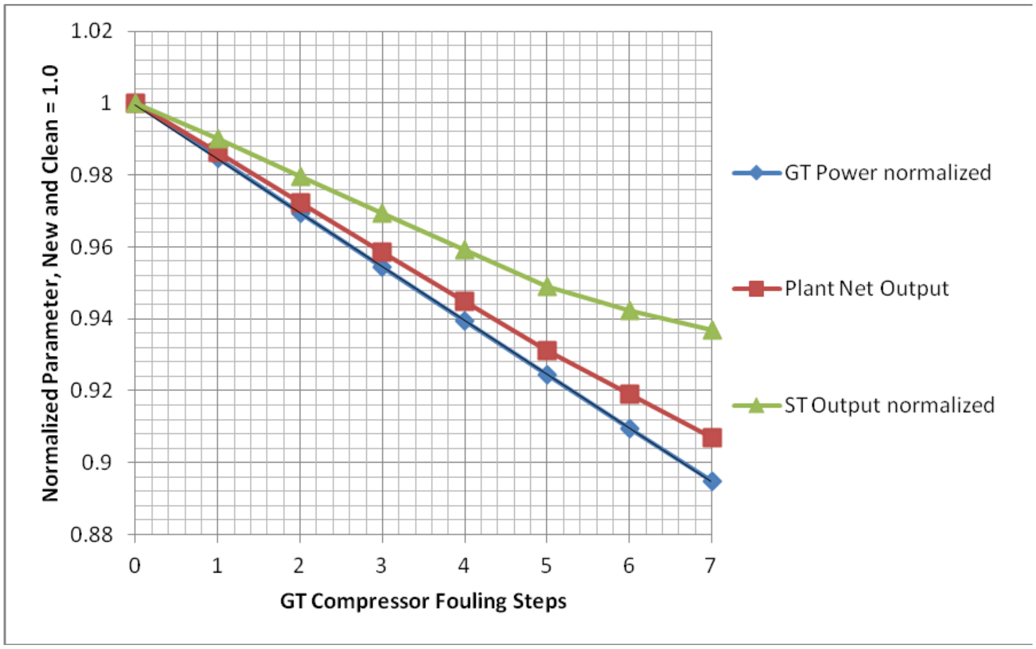

Figure 22: Impact of GT compressor fouling on combined cycle plant output, GT output and ST output

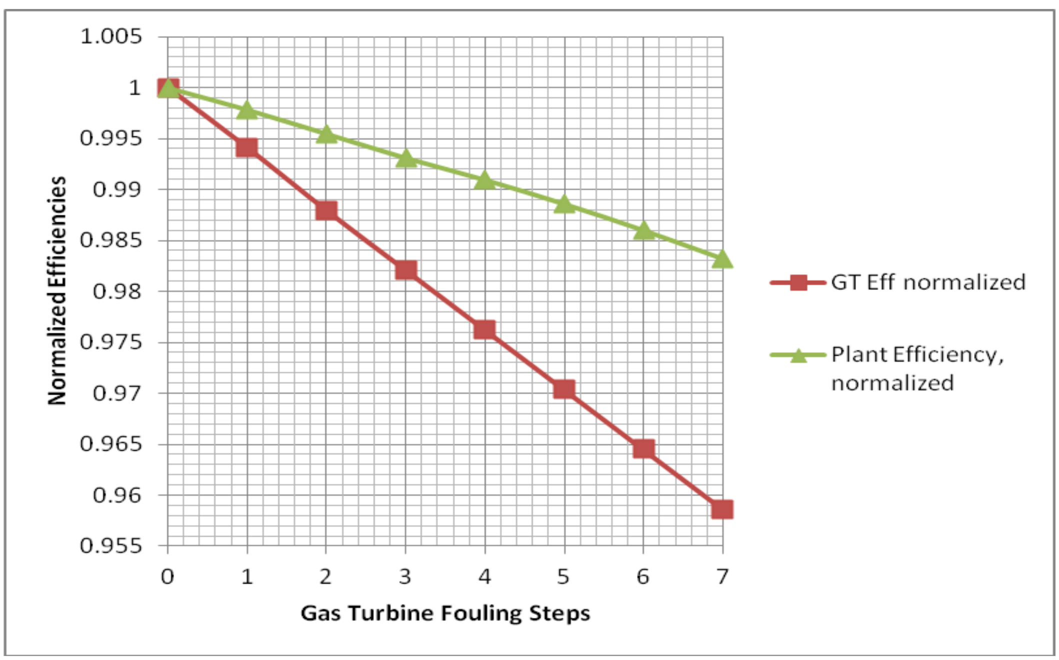

Figure 23: Impact of GT compressor fouling on GT efficiency, and combined cycle plant deficiency

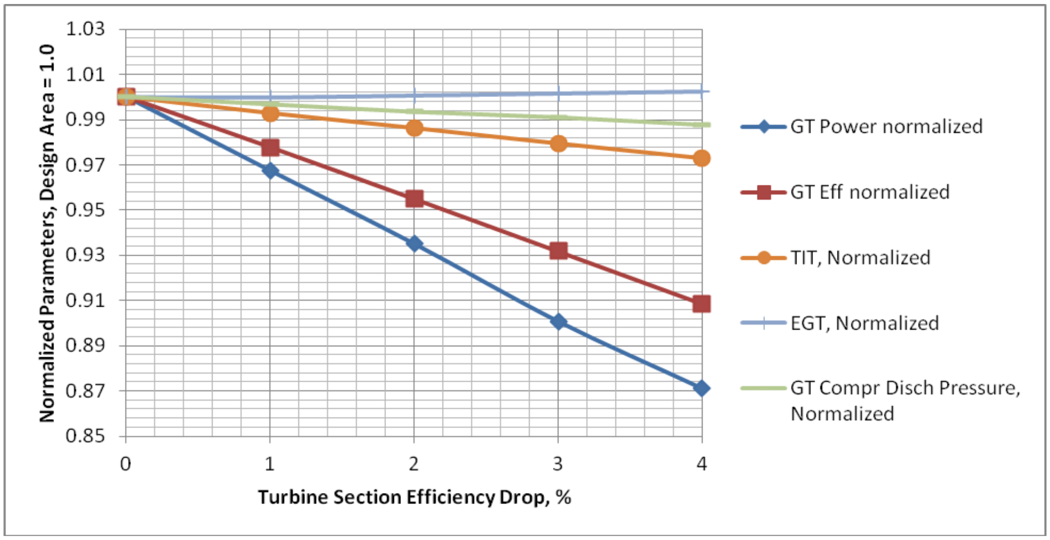

Figure 24: Impact of GT turbine section efficiency drop on GT parameters

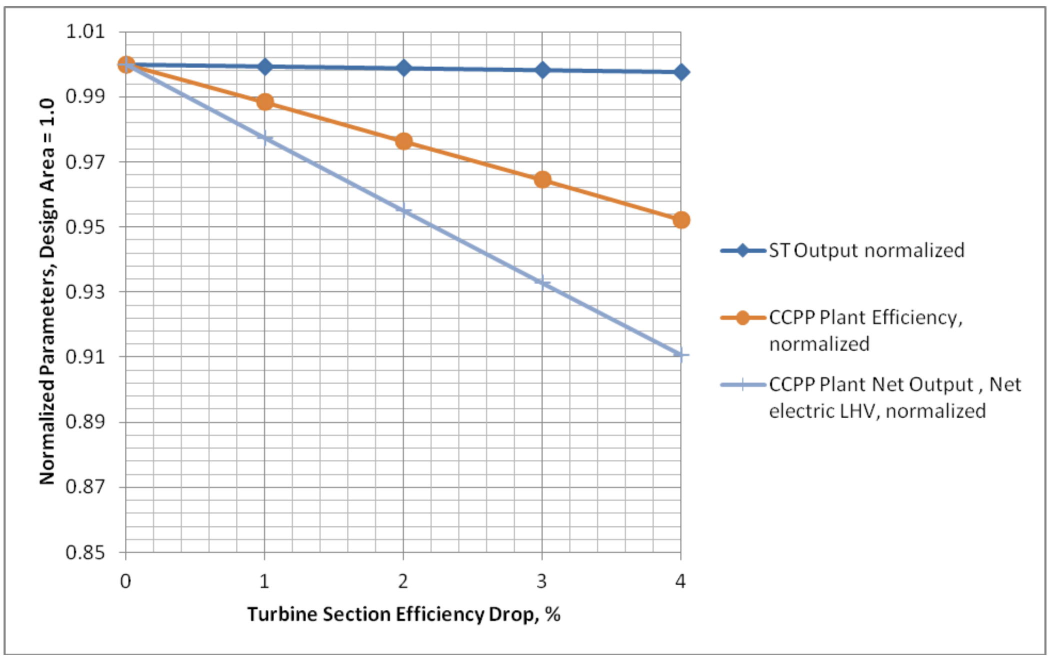

Figure 25: Impact of GT turbine section efficiency drop on combined cycle parameters

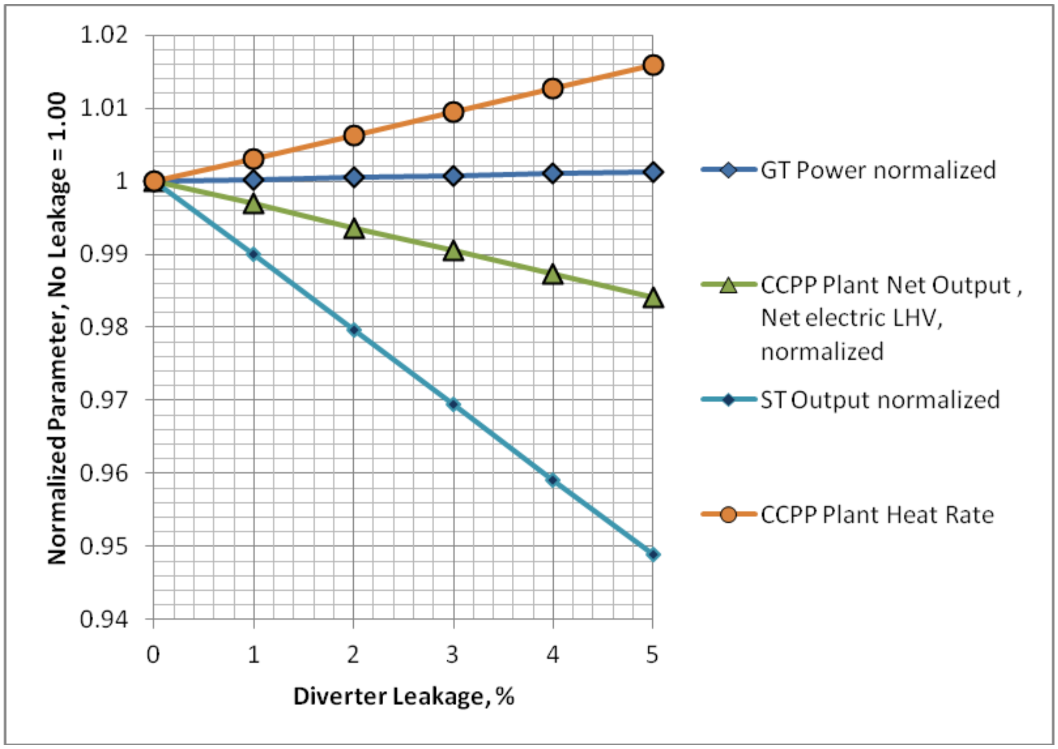

Figure 26: Plant output versus diverter leakage

The impact on the gas turbine cycle alone is shown in Figure 19 which shows Step 0 (no deterioration) and Step 7 is shown in Figure 20

TURBINE SECTION EFFICIENCY DROP

The impact of a drop in turbine section efficiency on gas turbine parameters is shown in Figure 24, and the impact on combined cycle plant parameters can be seen in Figure 25.

DIVERTER VALVE LEAKAGE

Diverter valves can leak causing a drop in cycle efficiency and output. Figure 26 provides a representation of the impact of diverter valve leakage on combined cycle parameters.

MONITORING OF HRSGS

The approach temperature can be calculated by taking the saturation temperature at the steam drum operating pressure and subtracting the temperature of the water leaving the economizer section. This can typically be calculated using the instrumentation present in the control room. As the ambient temperature decreases, the approach temperature will also decrease, indicating that the economizer outlet temperature is getting closer to saturation temperature and that the risk of steaming within the economizer is increasing. As the ambient temperature increases, approach and economizer effectiveness decrease. This assumes a gas turbine without any form of inlet cooling.

- *Low approach temperature* - Very low approach temperature could indicate that the economizer is steaming, and this can cause water hammer, vapor lock, and performance deterioration in the entire economizer section.
- *High approach temperature* – If the calculated approach is higher than the design value, this would indicate economizer underperformance which could be caused by gas bypass, tube outer surface problems (such as damaged fins), tube inner surface damage, air/vapor pockets or low water velocity.

By definition, the pinch temperature is the difference between the exhaust gas temperature leaving the evaporator section and the saturated temperature within the evaporator tubes. A typical pinch temperature may be 30°F. It is not a parameter that can be easily measured without special instrumentation. It can be derived by computational means.

In general, the pinch temperature will behave inversely with ambient temperature. If duct burners are used, the pinch temperature will increase. A high pinch temperature, measured or calculated, might indicate an underperforming evaporator section caused by gas bypass, damaged fins, or fouled fins, or tube inner surface fouling. A low pinch temperature indicates that the evaporator is working better than designed.

Stack temperature is a good indicator of overall HRSG health. An excessively high stack temperature indicates external fouling of the tubes or gas bypass. The feedwater temperature is an important parameter and should be checked. It is important to note that supplemental firing will reduce the stack temperature for a fixed pinch.

PROTECTION AGAINST DEGRADATION

While engine degradation cannot entirely be avoided, certain precautions can clearly slow the effects down. These precautions include the careful selection and maintenance of the air filtration equipment and the careful treatment of fuel, steam, or water that are injected into the combustion process.

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

While the rate of deterioration is slowed by frequent online washing, thorough on-crank washing can yield a more significant recovery. However, if the wrong detergents are used, some of the solvent may stick to the blades, or the washing process simply transports contaminant from the front stages of the compressor to rear stages or the turbine section. No matter how good the washing, the rear stages of the compressor will not get cleaned effectively. If the compressor blades can be accessed with moderate effort, hand cleaning of the blades can be very effective.

The rate of degradation of components depends to a large degree on the amount and size of degrading substances entering the engine. The quality of air, the type and quality of fuel, the quality of water or steam determine the rate at which the engine losses power and efficiency.

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

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

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

Air Filtration System

Fouling of inlet filters occurs progressively over time. This leads to increased pressure drop in the inlet system, and as a result, reduced engine power and efficiency. Figure 27 shows the relative impact of the pressure loss in the inlet system on power and efficiency. Self-cleaning filters, where appropriate, or changing of filter pads or cartridges can reverse this pressure loss. It must be noted that air filtration systems are always a compromise between filtration effectiveness, pressure loss, and size or cost of the system.

The filtration system has to be appropriate for the type of contamination that is expected. Some types of filters are very effective for small particle sizes, some are specifically designed for high dust loads; others are effective in keeping droplets (with potentially dissolved contaminants) out of the engine. A complete discussion of air filtration options was given by Wilcox et al., (2011).

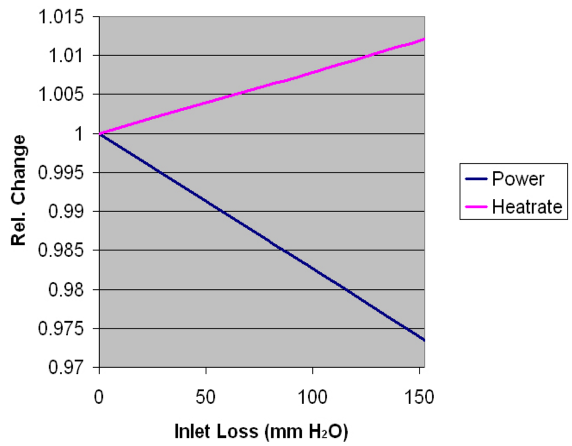

Figure 27: Impact of inlet pressure loss on engine power and heat rate

Schroth and Cagna (2008) reported, that the use of three stage high efficiency filters (H11 end stage) allowed keeping the performance deterioration below 2% for a year of operation without having to water wash the engine. The results were gathered for machines in the size range of 4.5 to 232 MW, operating in facilities in Southeast Asia and Europe. The useful filter life time for all stages was reported to exceed one yea, and reached up to three years, although the ultimate useful lifetime had not been determined, since the final pressure drop of the filters had not been reached at the time the report was written. In this study, the authors also evaluated the trade-off between a higher inlet pressure drop versus lower fouling rates of a three-stage high efficiency filter (F6-F9-H11) system and a two-stage system (F6-F8). They found that the higher pressure loss of the three-stage system is more than compensated by the reduced fouling and the lower frequency of shutdowns for water washing.

Similar results are also reported by Hepperle et al., (2011), where upgrades in filtration efficiency significantly reduced the power loss from fouling. The biggest effect, a reduction in power losses due to fouling of 3.5% per year, was reported when an H10 rear stage was added to an existing G4-F8 air filtration system.

Water washing

Several different methods of gas turbine compressor cleaning have been applied over the years, but ***"wet cleaning"*** has been found to be by far the most effective and economic technique. However, today's sophisticated large industrial engines and blade coatings require appropriately designed cleaning systems to ensure operational safety, reliability and optimum efficiency. Two different wet cleaning techniques, known as offline (crank wash) and online cleaning, are generally applied. Under extreme fouling conditions, hand washing of the IGVs may have to be conducted if time permits. During overhauls, hand cleaning of the full axial compressor is most effective. There has also been some recent interest in foam cleaning of gas turbines (Kurz et al., 2013).

Fouling is best controlled by a combination of two methods. The first line of defense is to employ a high quality air filtration system. However, as fouling will inevitably occur, compressor washing should be used to control its impact. This is an area in which strong and divergent opinions exist. Washing efficacy is so site-specific that approaches which work for one site may not be appropriate for another. Controversy is often caused by polarized opinions relating to wash procedures, wash media, and techniques.

Offline wet cleaning (also known as crank washing) is a typical "soak and rinse" procedure for which the gas turbine must be shut down and cooled. The compressor is rotated at crank-speed while a cleaning fluid is injected via nozzles or jet lances. Offline washing is almost always carried out with the aid of a detergent, and extremely effective power recovery can usually be achieved. Typically, wheel space temperatures must be below 200°F to avoid thermal shock, and the offline water wash is done with the machine on crank. The downtime for a crank wash depends mainly on the time it takes for cooling the engine. Larger, heavy-duty engines can take eight to ten hours to cool, whereas, on light aeroderivative engines, only one-and-a half to three hours may be needed because of the low metal mass. Offline cleaning is most effective when carried out in several steps that involve the application of a soap and water solution, followed by several rinse cycles using water alone.

Online washing is now very popular as a means to control fouling by avoiding the problem from developing. The primary objective of online washing is to extend the operating period between offline washes by minimizing the build-up of deposits in the compressor, and thereby reducing the on-going incremental power losses. Online washing is performed with the unit in full operation, and techniques and wash systems have now evolved to a point where this can be done effectively and safely. Outages or shutdown periods are not required. Optimal compressor cleaning can normally be achieved by adopting a combined program of regular and routine online washing (for example, every few days or weekly), plus periodic offline washing during planned outages.

Kurz and Brun (2012) show the result of a variety of measures to reverse the effects of degradation. In the example, the engine was returned to the factory after several thousand hours of operation. The initial run of the engine, without any adjustments whatsoever and at the IGV and T5 topping temperature from the original factory test, showed the TIT was 40°F (22°C) below design T_3, and the gas generator speed was 3% below design speed. After adjusting the guide vanes to get back to the desired design T_3 and gas generator speed, the engine improved power and reduced heat rate. Then the engine was detergent-washed and continued to improve performance. Next the individual stages of compressor variable vanes were adjusted to the factory settings, improving power. After all of the adjustments, the engine, compared to the factory testing when the engine was new, had lost 2.5% in power and 1.2% in heat rate. These results show that a significant amount of apparent degradation can be reversed by washing and adjustments.

In another example, Kurz et al., (2013), an industrial gas turbine was operated 3,500 hours without a detergent wash. The inlet air contained jet engine exhausts and was salt laden. Borescope inspections had shown deposits on the compressor and turbine. An offline detergent wash was performed. Control system data was recorded at full load before and after detergent washing. The data was taken with the ambient temperature below the design match temperature.

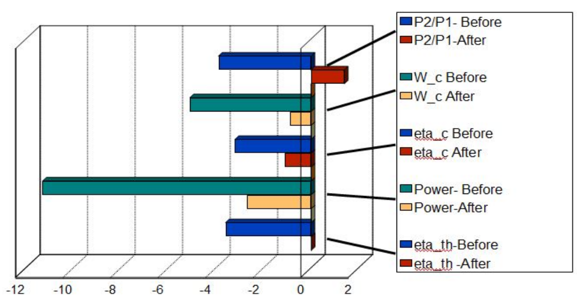

Figure 28: Impact of On-crank/ Offline water wash

The improvements in compressor pressure ratio, compressor efficiency, power, and heat rate are as expected, because the washing was reported to be very black and dirty. The engine was also underfired, because one of the effects of degradation is the reduction in the T_3/T_5 ratio. These improvements explain the very substantial 9.7% increase in output power. After washing and given the test uncertainties, this engine appeared to be performing essentially the same as when new Figure 28.

Roupa et al., evaluated the effectiveness of different washing fluids for different types of foulants on compressor blades, when used for online washing. The paper also includes wind tunnel measurements of the losses in clean and contaminated compressor cascades. The fouling evaluated was for dirt sticking on blades contaminated with a layer of crude oil, while for another series of tests, the sticking of particles on the blade was enhanced by a water soluble agent. Two wash fluids were investigated, aqueous surfactant and demineralized water. Droplet size range was from 50 to 250 µm. For the crude oil contaminated blades, the aqueous surfactant showed better cleaning results than demineralized water, while the blades contaminated with water soluble sticking agent showed about the same results for both cleaning agents. The wind tunnel measurements also provide clear evidence that fouling not only increases the losses, but also reduces the work capability (i.e., the turning angle) of the blade.

Brun et al., (2013) report about a project to provide a thorough experimental evaluation on the effectiveness of online turbine cleaning with various cleaning agents, including high-purity deionized (DI) water. Dirt was sampled from actual gas turbine axial compressor blades to characterize the composition and consistency of typical blade surface fouling. Samples of dirt were analyzed to determine the chemical composition. A representative dirt formula and blade coating procedure was developed for repeatable testing of blade cleaning and redeposit. Dirt was applied to the compressor blades. The blades were then installed in a wind tunnel with an upstream spray nozzle that simulated compressor operating conditions during online washing. Washing was performed with five different liquids (three grades of water, a commercial compressor cleaning product, and a laundry detergent) to determine the effectiveness of any of the liquids. Once this testing was completed, a similar test setup was utilized to inject a mixture of the formulated fouling dirt with various online cleaning liquids upstream of the blade into the wind tunnel to assess redeposit characteristics. Separate residue experiments were studied to determine the effect of high-purity water versus regular water on fouling dirt.

Within the parameters/ fluids tested in this study, the results indicate that:

1. Spraying cleaning fluid into a flowing air stream is a viable means of cleaning a compressor blade. Each of the fluids was able to clean the test blade at both low and high air velocities and at different blade incident angles. However, for all tested cleaning cases, there was always an area of the blade where some fouling deposits remained.
2. The blade cleaning is primarily a mechanical (droplet impact) function and does not depend on the fluid used for cleaning. Test results show that

most of the cleaning occurs shortly after the cleaning fluid is introduced into the flow stream. The type of fluid used did not have a significant impact on the cleaning effectiveness.

3. Dirt removed from the blades will redeposit in downstream stages as the cleaning fluid is evaporated. Redeposit occurred in flow recirculation zones during the cleaning tests, and heated flow tests demonstrated dirt deposit in the presence of a cleaning fluid. The type of fluid used for cleaning has no effect on the redeposit characteristics of the dirt.
4. Blade erosion was not found to be a significant issue for the short durations that online water-washing is performed. However, uncontrolled water-washing (or overspray) for extended periods of time does result in measurable leading and trailing edge blade erosions.
5. The results suggest that it may be beneficial to the cleaning process to slow the compressor speed or vary the cleaning fluids spray rate while the online wash is being performed.
6. The results show that most of the cleaning occurs shortly after the cleaning fluid is introduced into the flow stream. Field tests with online water-washing should include tests that vary the cleaning time. The issue of redeposit should be addressed during field testing; if possible, inspection of downstream compressor stages should be used to assess the transport and redeposit of dirt from early to later compressor stages.

The objectives and results of the referenced study can be summarized with the previously posed four questions and answers:

1. Does online cleaning work? Yes, wind tunnel blade test results indicated up to 95 percent removal of blade fouling is possible.
2. Is there any difference between any of the online cleaning liquids? No, there was no clear evidence that any of the liquids or detergent mixes improved the overall blade washing efficiency.
3. Will dirt be removed during online cleaning redeposit once the cleaning liquid has been evaporated? Yes, redeposit tests showed that a significant fraction of the dirt will redeposit on downstream blades. The actual quantity of the redeposit depends strongly on the local flow field and the type of particles that are being carried in the free stream.
4. Do any liquids reduce the redeposit of dirt in the online washing process? No, testing showed that redeposit occurred with all liquids tested, and there was no clear evidence that any mixtures or detergents reduced particle redeposit.

In a study by Flesland (2010) the deterioration rates of four different gas turbines (two-shaft aeroderivative engines in offshore service, Sleipner field) were evaluated. When choosing gas turbines, it was emphasized to select gas turbines operating under equal conditions but with different washing procedures. In addition to offline washing, two of the gas turbines had daily online washing routines, and one of the gas turbines ran an idle wash every 1,000 hours between each offline wash.

The evaluation showed that an overall trend was that the gas turbine that had been running with online washing continuously over a long period of time had higher performance than the reference engine. For the second gas turbine, a daily online washing procedure has recently started. The advantage with the evaluation of this gas turbine was that a good reference engine was available. The two engines were operating under quite similar conditions at the same location in addition to having equal filter systems. Some deterioration trends were detected. For the first period, both engines seemed to have quite equal deterioration trends. During the second period, no clear trends were seen in corrected CDP and corrected EGT, when evaluated for constant GG speed. The compressor efficiency had decreasing trends for both engines during the second period as well, but the compressor efficiency for machine 1 was overall higher during the period with online washing than the previous period. The borescope pictures taken after the first period with online washing showed good visual results. However, it is too premature to make a final decision regarding the exact performance gain of online washing. At the time the study was performed, the engine had only been running online washing for one operating interval. More investigation over longer time is recommended.

For the engine running with idle wash, it was not possible to conclude on the basis of the collected data. No clear deterioration trends were detected, and investigations over longer time and several operating intervals are recommended. It is also important to be aware of the fact that the performance gain of idle wash needs to be much higher than for online washing in order for idle wash to be economically profitable.

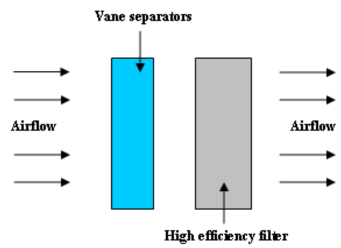

Figure 29: Filter System A (Flesland, 2010)

Sleipner currently has two different filter systems in operation on the gas turbines. Filter system A, illustrated in Figure 29 has one set of vane separators located in front of the high efficiency filter. These filters are dependent on removing the majority of water and moisture from the airflow before it enters the high efficiency filter. The high efficiency filter is specially designed to withstand moisture. Seals are installed on the bottom of each filter bag to make sure that the water reaching the filter elements are drained out upstream of the filter.

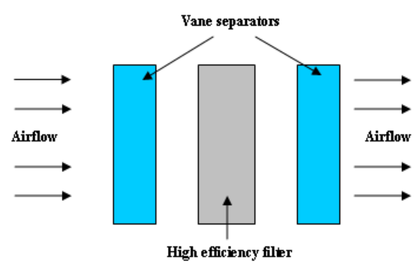

Figure 30: Filter System B (Flesland, 2010)

Figure 30 shows filter system B which has two sets of vane separators, one each before and after the high efficiency filter. The first stage of vane separators removes the majority of water and humidity from the airflow before it enters the high efficiency filters. The greater part of the pressure loss takes place over the high efficiency filters, and these filters determines the final filtration efficiency. The last vane separator is designed to remove the remaining water droplets from the airstream. This filter type also allows for the use of pre-filters, but pre-filters are normally not used on Norwegian oil platforms today (Flesland, 2010).

In Flesland's (2010) study, Machine 1 and 2 use Filter System A, while Machine 3 is equipped with filter system B. All three gas turbines are of the same type. All machines are installed on the same platform, Machines one and 2 on the same level, and Machine 3 on a higher deck. Offline washing is performed every 3,000 operating hour on all the gas turbines evaluated in this study. In addition, machine 1 and 3 have daily online washing. The data from Machine 1 could not be verified. While machine 3 showed very little deterioration of compressor efficiency over the period of a year, machine 2 shows clear compressor efficiency deterioration between off line washes. In other words, in this case the filtration system with water removal after the filter, or the fact that the machine was higher up on the platform, showed to be advantageous.

CONDITION MONITORING

Condition monitoring to evaluate gas turbine degradation must be viewed in a holistic manner and consequently should be a blend of

- Performance analysis - evaluation of aerothermal parameters, trends, efficiencies, exhaust temperature spreads, etc.
- Mechanical analysis - evaluation of bearing temperatures, vibrations, lube oil characteristics etc.
- Examination of transient behavior of the turbine - both aerothermal and mechanical – this would include startup/ run down profiles, transient vibration analysis, etc.

PERFORMANCE ANALYSIS

In the past decade, there has been considerable interest in the application of online performance monitoring of gas turbines. Dundas (1994) has provided an interesting statistical study of the causes of gas turbine failures, some of which may have been prevented by performance monitoring. New developments in the area of sensors that provide detailed information on the gas path, tip clearances, blade metal temperatures, oil conditions and even exhaust gas debris analysis can make performance monitoring an important management tool for a gas turbine installation. An excellent treatment covering the use of performance monitoring for gas turbine problem prevention was made by Dundas (, 1992).

DEVELOPMENT OF PERFORMANCE BASELINES

In order to determine and monitor performance deterioration, it is imperative to have a baseline of gas turbine performance. The baseline forms a datum of reference to compare changes that occur over time. There are wide ranges of performance parameters that can be used to develop a baseline. Some of the baseline curves can have qualitative value. For example, with a two-spool gas generator without variable geometry, the relationship of the corrected N_1 (LP Compressor) and N_2 (HP Compressor) speeds can be of value to see if something is amiss with the gas turbines. Figure 31 shows the N_1 and N_2 speed relationships for a 2-spool gas generator. Any sudden deviation from the normal relationship of the spool speeds is an indicator of a potential problem. Much of the curves shown here can be developed by hand, assuming that the data is available. The task is much simpler with modern control systems and DCS units that allow rapid assimilation of data.

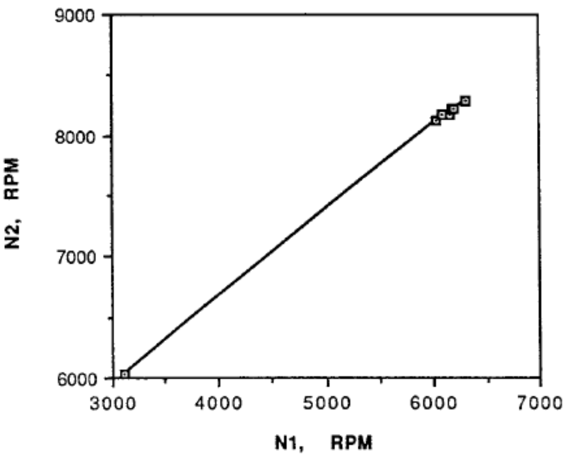

Figure 31: Relationship between N1 and N2 in a two-spool gas generator

A number of graphs developed for a 2,300°F, 160 MW advanced gas turbine are shown (Meher-Homji, et al., 1993). Figure 32 depicts the pressure ratio as a function of the corrected power and is a baseline plot developed for this machine. The relationship between firing temperature and exhaust gas temperature (EGT) is shown in Figure 33. The effect of IGV movement can be clearly seen in this graph

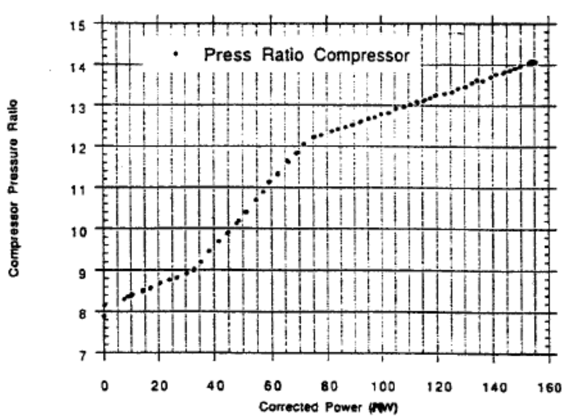

Figure 32: GT pressure ratio as a function of corrected power- large 165 MW single-shaft GT

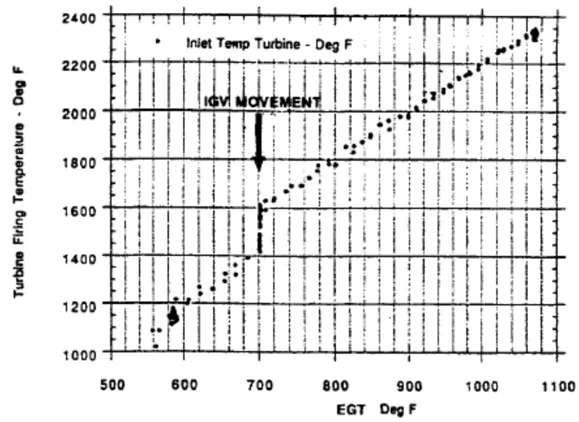

Figure 33: Calculated Firing Temp vs EGT

While baselines are usually thought of as relating to steady state operation, it is also valuable to create a baseline of *transient* operating data so that this can be compared with later runs to examine deterioration in the gas turbine. Wheel space temperature behavior during a start transient is shown in Figure 34 and the radial bearing temperature transient behavior during startup is shown in Figure 35.

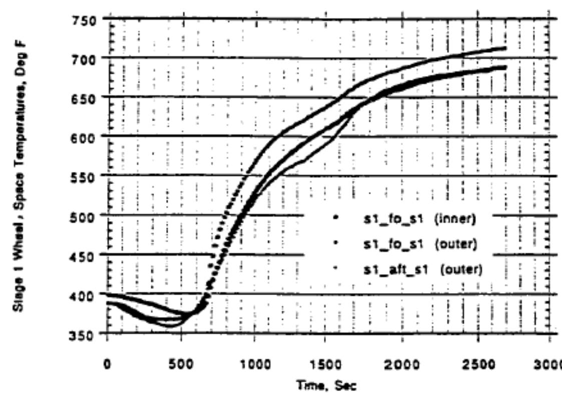

Figure 34: Wheel space temperature behavior during a start transient

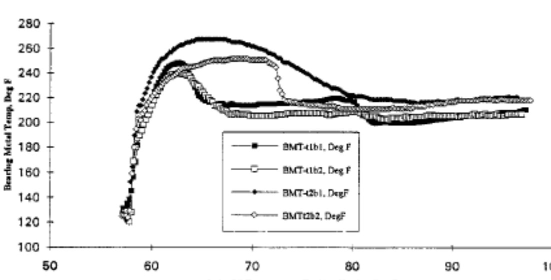

Figure 35: Transient bearing metal temperature during startup

Parameters that may have a static alarm point such as wheel space temperatures may be shown to be direct functions of other parameters. The relationship between wheel space temperatures and ambient temperature can be seen in Figure 36.

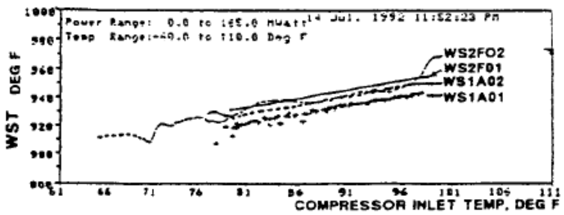

Figure 36: Wheel space temperatures as a function of compressor inlet temperature

A common approach to trend gas turbine output is to use a power capacity factor defined as

Power Capacity Factor = $MW_{Actual}/MW_{Expected}$

Corrections are made for

- Inlet temperature
- Inlet pressure
- Specific humidity
- NO_x water injection rates
- Inlet and outlet pressure drops
- Speed corrections (Note that this is not a major issue for power generation applications with single-shaft gas turbines unless significant off frequency operation is experienced. This happens in some countries due to grid "underfrequency" problems).

EXHAUST TEMPERATURE MONITORING FOR HOT SECTION DISTRESS

The importance of EGT monitoring has been covered in Meher-Homji and Gabriles (1998), and is most important for gas turbines regardless of the type of combustor employed. There is always some circumferential temperature distortion that may, if excessive, indicate problems with fuel nozzle malfunction and other combustor issues.

These problems can be detected either by monitoring exhaust gas temperature distribution or, on two-shaft gas turbines, by monitoring the circumferential temperature distribution at the power turbine inlet.

TORQUE MONITORS AND TORQUE MONITORING

Unlike with generator drives, it is difficult to determine the power output of mechanical drive gas turbines. While in theory, measurement of the flow and head of the driven compressor or pump can be used for a horsepower computation, there are many complexities and uncertainties involved in this computation when the compressors are side stream designs or when gas composition varies. The use of torque meters on mechanical drive gas turbines makes economic sense as it allows an accurate measurement of output power allowing degradation to be controlled and load balancing to be accomplished.

Unfortunately, there is still a mistaken perception in the market that torque measurement devices are unreliable. Torque measurement units have successfully and reliably accumulated millions of operating hours and are routinely specified by several large petrochemical and energy companies. They are seeing increasing utilization in LNG, ethylene, and ammonia plants and accuracies are now within 1 percent. Several of the newer products also provide for dynamic torsional analysis which is valuable for VFD drives and long strings with low natural torsional frequencies.

TRENDING OF PERFORMANCE DATA

For meaningful trending and to get a visual picture of the degradation, the following basic facts have to be recognized

- Several performance parameters both analog input and computed will vary with operating regimes of the machine (i.e., with power level, flow, etc.)
- With "off design" operation, there is a change in several performance parameters that occurs not because of deterioration, but because of the fact that the machine is operating "off design." This causes changes in the aerodynamics and match points in the engine and very often a drop in component efficiency. The effect is less severe with split-shaft machines and variable geometry engines.

Mathematically, where DET represents Deterioration,

$$DET_{Total} = DET_{off\ design} + DET_{actual}$$

- Vibration parameters also have variations with operating conditions (this is particularly true of amplitudes at gear mesh, blade pass frequencies). The presence of resonances, interaction effects and several other factors makes trending even more complicated.

There are several approaches to trending meaningfully. One approach to this problem by means of segmenting data in terms of a parameter ranges for parameters such as power, flow, etc. Thus trends can be called up for a given power range and deterioration can be observed qualitatively on the screen. Another approach is to provide *multiple* trends to permit a set of trends to be viewed and for the observer to determine deterioration by looking at different power settings. Another approach is the baseline transpose trend approach. Note that for the baseline trend approach, a basic requirement is that a unique performance curve exist that shows one performance parameter versus the baseline parameter. If such a curve exists (or can be generated during operation of a clean machine) then this method can be applied. In this approach, the off design point is "transposed" back to the design point, and the differential at this point is trended. Thus, the trend shows an indication of deterioration at the "design condition" even when data has been acquired at off-design conditions. There are several assumptions inherent in this procedure:

- The off design point is transposed to the base point via a parallel (or similar shaped curve) to the original curve.
- The trajectory of the transposition is assumed independent of other operating problems that may cause a deviation from this smooth trajectory.

The above approaches can be implemented as follows:

A performance model for the engine has to be built (either acquired from the manufacturer, or created using data) that allows to predict the performance of a reference engine under the prevailing ambient conditions (Kurz and Brun, 2000). A test data point for a parameter (for example heat rate, output power, compressor discharge pressure etc.) P can thus be compared to the reference engine performance for this parameter, P_{ref}. A relative performance $P_{rel}=P/P_{ref}$ can be calculated. The changes in relative performance P_{rel} are good indicators of engine deterioration and engine health. Figure 37 shows how a performance parameter (power in this case) is plotted against a predicted curve.

The above-mentioned methods are applicable to performance related data. For vibration data, the use of adaptive modeling can be applied wherein vibration can be characterized on a statistical basis as a function of a large number of operational parameters.

Besides describing a promising method to use past operational data to predict future engine behavior, Venturini et al., (2013) also provide field data for large heavy duty gas turbines in power generation service. The data provides information about the effectiveness of off- line washing, as well as engine overhauls. Non-recoverable degradation can be clearly identified. In all cases, the amount of power degradation is significantly larger than the amount of compressor efficiency degradation. The data indicates that the compressors are mostly subject to recoverable degradation, but show very little non-recoverable degradation over time. However, since only compressor efficiency is reported, no statements about the effects on compressor mass flow can be made.

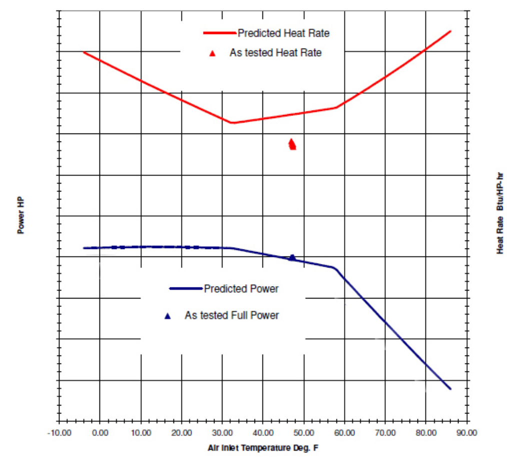

Figure 37: Data point plotted against predicted engine power and heat rate. The predicted curves are based on site ambient pressure, power turbine speed, humidity, fuel, and inlet and exhaust losses.

Even with these approaches, there is a class of problem that may not show up as a long term trend. In cases such as these, *transient behavior* will often be the best descriptor of the problem.

TRANSIENT ANALYSIS AS AN INDICATOR OF PERFORMANCE DETERIORATION

Most engine health monitoring systems used for land based systems are in essence, based on steady state operation, i.e., data in which time based variation is, at a minimum. Traditionally, diagnostic analysis has been conducted under steady state conditions and online systems have tended to concentrate on map based performance diagnostics using pattern analysis or fault matrices. There is however, significant diagnostic content in turbine startup and shutdown data as well as data obtained when power or speed changes are made. The importance of transient analysis for gas turbine peaking applications is self-evident. There are currently some large high technology machines that will be used for peaking service. Transient analysis will be of importance here, especially as the durability aspects of these units have to be proven. Details of transient analysis for the condition monitoring of gas turbines is provided in Meher-Homji and Bhargava (1992).

Steady state operation is defined as that condition wherein parameters are at relatively steady state with only random scatter effects occurring. In a steady state situation, the mass balances in the turbine are totally satisfied, and there is no accumulation of mass. This means that the mass flow from one component equals the mass flow to the adjoining component. A transient condition is said to occur when condition parameters such as speed, firing temperature, and load vary with time. Obviously, startup and shutdown are transient events as is a change in load or an acceleration event. During transient conditions:

- Shaft inertia will either demand or produce power (depending on whether it is being accelerated of decelerated).
- Pressure and temperature gradients occur, causing changes in the mass flow rates into and out of components.
- Heat balances are not satisfied with heat being either absorbed or rejected by engine components.
- Dimensions of various components can change due to temperature effects and centrifugal effects. Tip clearances can be affected. Several aero gas turbines employ active tip clearance control to limit this effect.

In large critical turbomachines, problems often develop under transient conditions due to factors such as increased loading, thermal stresses, changes in tip clearances and changes in thrust position. In a certain sense, gas turbine operators have historically used transient analysis informally when they measure coast down times or plot startup curves using strip chart recorders and trending packages.

MECHANICAL TRANSIENT ANALYSIS

Several parameters can be monitored to gain insight into items that may be causing performance or mechanical deterioration in the gas turbine engine.

- For startup, it is possible to prepare maps of mechanical parameters (lube oil temperatures, and pressure) as a function of rotor RPM.
- Detection of bleed valve problems - Several gas turbines utilize bleed valves for surge protection during startup and shutdown. In some machines, these are butterfly valves that may stick in a partially open position during a start event. The effect of this is to reduce the compressor discharge pressure (hence pressure ratio). With a machine on temperature control, this means that the unit will run at a higher exhaust gas temperature. Examining a startup transient of the CDP as well as wheel space temperatures is therefore of value.

CONCLUSIONS

The goal of this tutorial was to provide a non-mathematical treatment of performance deterioration to help plant operators to understand the underlying causes, effects, and countermeasures as well as the measurement of gas turbine performance degradation.

To understand the impact of degradation, basic concepts of gas turbine matching and off design operation have been addressed, not just for simple cycle, but also for combined cycle installations.

The three major types of performance deterioration, recoverable degradation, non-recoverable degradation, and permanent deterioration were covered. In addition, mechanical deterioration was discussed.

Methods to reduce the impact and extend of engine degradation as well as condition monitoring approaches focused on the detection of deterioration have been explained.

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