GAS TURBINE PERFORMANCE DETERIORATION

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Mr. Motiwala has built and patented a novel rheometer design used for measuring visco-elastic properties of fluids. He has B.S. and M.S. degrees (Mechanical Engineering) from the University of Houston.

ABSTRACT

With privatization and intense competition in the utility and petrochemical industry, there is a strong incentive for gas turbine operators to minimize and control performance deterioration, as this directly affects profitability. The area of gas turbine recoverable and nonrecoverable performance deterioration is comprehensively treated in this paper. Deterioration mechanisms including compressor and turbine fouling, erosion, increased clearances, and seal distress are covered along with their manifestations, rules of thumb, and mitigation approaches. Permanent deterioration is also covered. Because of the importance of compressor fouling, gas turbine inlet filtration, fouling mechanisms, and compressor washing are covered in detail. Approaches for the performance monitoring of steady-state and transient behavior are presented along with simulations of common deterioration modes of gas turbine combined cycles. As several gas turbines are used in cogeneration and combined cycles, a brief discussion of heat recovery steam generator performance monitoring is made.

INTRODUCTION

With the introduction of high performance gas turbines and increasing fuel costs, maintaining of the highest efficiency in gas turbine systems becomes an imperative. A large-scale 934 MW combined cycle plant can have an annual fuel bill of 203 million dollars. Over its life cycle of 20 years, the fuel would cost over four billion dollars even assuming a flat fuel cost of $4/MM BTU. With a moderate fuel cost escalation, this can easily total six or seven billion dollars. With the deregulation of the electric utility market fostering competition between power producers and an environment in which fuel costs have dramatically escalated, plant operators have to understand and control all forms of gas turbine performance deterioration.

There are several excellent papers that treat this important subject either in a highly mathematical fashion or focus on a particular subset of the deterioration phenomenon. The goal of this paper is to provide a nonmathematical treatment of the physics of performance deterioration covering the subject matter so that plant operators and engineers can grasp the underlying causes, effects, and measurement of gas turbine performance deterioration. To this end, rules of thumb and checklists are provided. The APPENDIX will provide some simple models and equations that can be used for day-to-day performance deterioration monitoring. Details on gas turbine performance are found in Walsh and Fletcher (1998). This reference is particularly valuable because of its comprehensive treatment of gas turbine performance and its appendix providing all the required equations.

The area of performance retention is also one of considerable interest to commercial airlines. The techniques for measurement and control used for civil aircraft engines are worth studying as the industrial sector can benefit from some of the techniques employed.

Gas Turbine Design Aspects and Performance Deterioration

In order to understand gas turbine and deterioration issues, some basic aerodynamic and mechanical design issues are covered.

Three main limiting factors in gas turbine design and development are:

- Restriction on the material temperatures.
- Stress levels and (therefore) the rotor speeds.
- Fluid dynamic behavior—choking, separation, stall, and mixing.

To attain a high thermodynamic efficiency, heat addition (in essence the turbine inlet temperature) should be at as high a temperature as possible. This means that cooling has to be employed. There is an inherent tradeoff between increasing the cycle temperature and the cooling penalties in the cycle. Cooling flows can impact the overall thermal efficiency of the gas turbine and, therefore, losses that increase cooling flow can have a cycle penalty. Several forms of deterioration can directly impact the cooling flow and impact in turn the reliability of the gas turbine. For example, dirt in the compressor can find its way into the small cooling holes of the hot section blading, impair cooling, and result in blade distress. This is an example where mechanical behavior and reliability of the gas turbine are linked to performance degradation. As a practical matter, availability is closely linked with performance deterioration as will be shown in several case studies ahead.

With higher firing temperatures, pressure ratios have also steadily increased. Highly optimized gas turbine designs (with high pressure ratios, stage loadings, cycle temperatures, and tight tip clearances) may be more susceptible to performance deterioration.

The following issues link mechanical aspects to the performance:

- *Rotor speeds*—The blade stresses vary with the square of the tip speed and linearly with the mass flow rate. The equation providing the specific stress on the blading is given by:

$$\frac{\sigma}{\rho_b} = \frac{\omega^2}{2\pi} \times A_m \tag{1}$$

where A_m is the axial flow area. The stress levels increase with the square of the speed and linearly with the mass flow.

The blade Mach number (also commonly known as the U/C ratio), is defined by:

$$Ma = \frac{u}{\sqrt{\gamma R T_0}} \tag{2}$$

The blade Mach number provides a better understanding of the situation. This blade Mach number does not differ a lot between different engines. In a turbine with a choked nozzle, it is often around 0.6. The ideal value for an impulse stage with a sonic nozzle is about 0.47 to 0.5, and for a reaction stage is around 0.8.

The specific stress then can be expressed in terms of the hub/tip ratio (v_{ht}) by the equation:

$$\frac{\sigma}{\rho_b} = \frac{Ma^2}{2}\left[\frac{1}{v_{ht}} - 1\right]\gamma R T_0 \tag{3}$$

In high-pressure turbines, the hub/tip ratio is limited by the clearances and the associated leakage and losses to values less than 0.9. The specific stress in the blades will increase with turbine inlet temperature (TIT). Consequently the materials have to be constantly improved.

- *Mass flow*—Several of the newer turbines are uprated by increasing the flow rate by “zero staging.” This approach increases the output with minimal cost and risk. The early stage is often transonic, which requires special transonic blading that is less tolerant of incidence losses than subsonic blading. The higher relative velocities increase the possibility of erosion if hard particles are present in the airflow.
- *Compressor stall and surge* are an important issue often linked to performance deterioration. Part speed operation results in the earlier stages not making the design pressure ratio and consequently the density in the latter stages drops. This results in a choke condition in the rear of the compressor that can force the early stages into stall. As will be shown ahead, compressor deterioration can often result in surge damage.
- *Turbine design considerations*—Turbines have not been as difficult to design as compressors because of the fact that diffusing flow is not present. The turbine is still the component that will define the gas turbine operating line and any change in nozzle area or turbine fouling will affect the match point of the gas turbine and therefore its output and efficiency.

Simplified Physical Understanding of Gas Turbine Matching

In a gas turbine engine, the operating point is defined by the match between the compressor and the turbine. A match point is simply a set of operating conditions (pressures, temperatures, and flows) where the compressor and turbine can work in unison and in equilibrium.

Matching is based on the compatibilities of flow, work, and speed. This means:

- The compressor work must match the work output of the turbine that drives it. In aeroderivative engines, the gas generator turbine drives the compressor. In the case of a three-spool engine, the low-pressure (LP) turbine work must match the LP compressor work and the high-pressure (HP) turbine, the HP compressor work. In a single shaft gas turbine, the turbine work must equal the sum of the output and the compressor work. Of course this would take into account mechanical losses.
- The flow rates must be compatible because gas turbines are continuous flow machines. Air bleeds and fuel flow must be taken into account.
- The speeds of the compressor and driving turbine must, by definition, be the same. In multiple spool engines, the free spools must be in equilibrium.

A typical compressor and turbine map are shown in Figure 1. In the case of a single shaft gas turbine (typically found in power generation applications), the operating point at a given temperature is along a corrected speed line as the mechanical speed is locked to the grid. After attaining full speed no load conditions (FSNL), the turbine is loaded by increasing fuel flow (i.e., increasing turbine inlet temperature), causing the operating point to move up the speed line along the trajectory depicted by the cross marks in the figure. In the case of a split shaft gas turbine as is typical of mechanical drive applications, load changes involve changes in the speed of the gas generator spool and therefore the trajectory of operating points is along the dotted line.

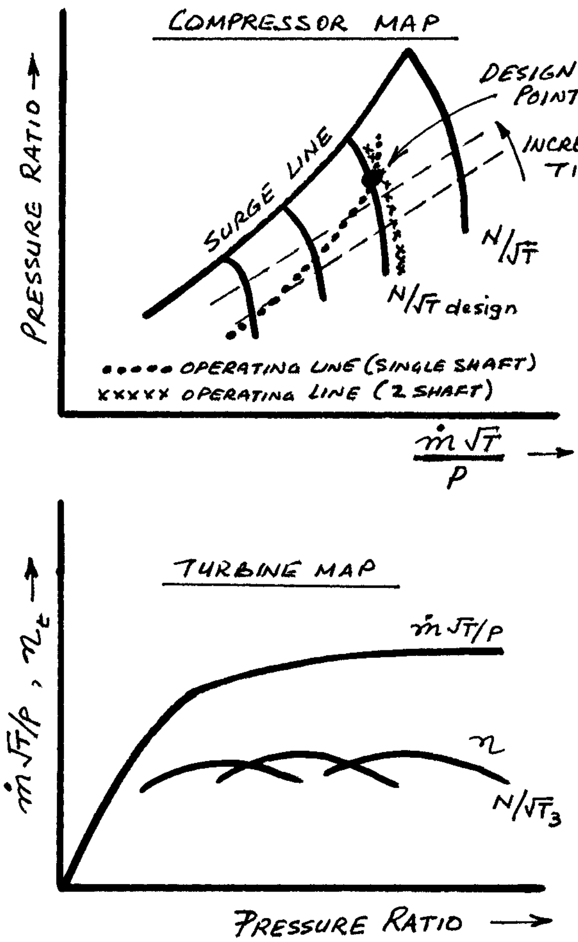

Figure 1. Typical Compressor and Turbine Maps.

In the case of the turbine map shown, the flow chokes at a certain expansion ratio (for a given turbine inlet temperature) and thereafter the turbine flow function (m√T/P) stays flat. The swallowing capacity of the gas turbine is Mach number limited and therefore a function of the turbine inlet temperature. Most gas turbines run choked with a choked nozzle and this defines the operating point on the compressor characteristic. Thus, for a given speed, the compressor operating pressure ratio can be calculated from the turbine "resistance." This calculation requires an iterative computation knowing both the turbine and compressor maps. Obviously as the flow is choked, the quantum of flow will be defined by the turbine inlet temperature (due to the influence of temperature on the Mach number).

The optimum operation at design conditions will be obtained with a certain hot effective nozzle area. For a power turbine to match a gas generator, the overall turbine effective area has to be equivalent to the hot effective nozzle area. Variations in effective area from the standard will modify performance. If the effective area is oversize by 1 percent, the gas generator cycle temperature is reduced at constant gas generator speed, and a power reduction of between 2 and 4 percent is typical. If the effective area is undersize by 1 percent, the gas generator speed is reduced at constant temperature and a similar reduction in power occurs (Blauser and Gulati, 1984). Details on gas turbine matching may be found in Cohen, et al. (1987).

Turbine and Compressor Losses

Efficiency is one of the most important parameters for gas turbines, fundamentally because the net output is the difference between turbine output and the work consumed in the compressor. With the ratio of compressor work to total turbine work being approximately 0.5, clearly component efficiencies are of critical importance and will create a large change in gas turbine output. The deterioration of efficiency in the compressor and turbine is of critical importance from a performance retention standpoint.

With advances in gas turbine technology, it is common to find compressor and turbine efficiencies at 90 percent and above. These have been obtained by the use of sophisticated computational and experimental studies. Some of the earliest studies on turbomachinery efficiency were made by Howell (1945) and Ainley and Mathieson (1951), which were widely used in the 60s and 70s. In the late 70s and onward, the use of advanced laser-based instrumentation allowed the flow in cascades and actual turbomachinery to be better visualized and understood, and allowed the incorporation of 3D analyses and the effect of unsteady flow. An excellent review of turbomachinery losses is made by Denton (1993).

The three forms of losses can be conveniently classified as:

- *Profile losses*—Profile losses are the losses generated in the blade boundary layer well away from the end walls.
- *Endwall losses*—Also referred to as "secondary losses," these losses arise partly from the secondary flows generated when the annulus boundary layers pass through the blade row.
- *Tip leakage losses*—These arise from the leakage of the flow over the tips of rotor blades and the hub clearance of the stator blades. The losses depend on whether the blades are shrouded or unshrouded.

While these are often considered as discrete losses, the loss mechanisms are really interrelated.

The losses in an axial flow compressor with respect to the flow coefficient are shown in Figure 2. In this figure, the stage efficiency at the design point is 90 percent, which is typical of today's turbomachinery. Secondary losses are associated with 3D flows, and profile losses relate to blade surface effects. Problems relating to compressor fouling and deposits on the blading worsen the profile losses.

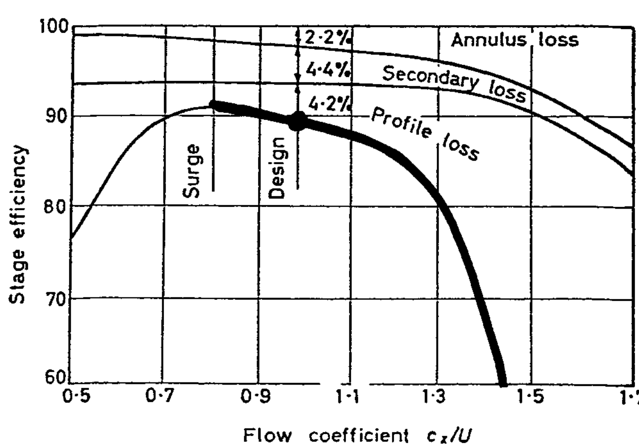

Figure 2. Losses in an Axial Flow Compressor with Respect to the Flow Coefficient.

Special Consideration for Split Shaft Mechanical Drive Gas Turbines

In mechanical drive split shaft gas turbines, there is an optimal power turbine speed (optimal U/C ratio). Operating away from this speed will result in a loss in terms of power and efficiency. The power turbine optimal speed is a function of ambient temperature. Assuming the design is for ISO conditions, as the ambient temperature increases the power turbine optimal speed drops. If adjustment of turbine geometry is available (variable PT inlet vanes), then the effect can be mitigated, resulting in a flatter efficiency curve.

Economic Considerations of Performance Deterioration

Industrial Power Plants

Petroleum refineries consume about 3 to 4 percent of their hydrocarbon throughput and pipelines are often themselves the largest consumer of energy. In most petrochemical plants, energy costs can be between 10 to 20 percent of the total production cost. The break up of costs for a gas turbine based combined cycle plant in terms of fuel, operation and maintenance, and capital costs is shown in Figure 3. In examining this figure, it is evident that any focus put on controlling performance deterioration will affect the plant's profitability. A typical set of long-term deterioration curves over 60,000 hours of operation for power and heat rate is shown in Figure 4. This figure is based on natural gas operation. Most plants will deteriorate somewhere between the limits shown. In plants that operate on distillate fuels, deterioration over time can be more severe and can jeopardize long-term contracts. Over a 30,000 hour period some gas turbines operating on liquid fuels can experience, in extreme cases, nonrecoverable degradation rates of up to 20 percent.

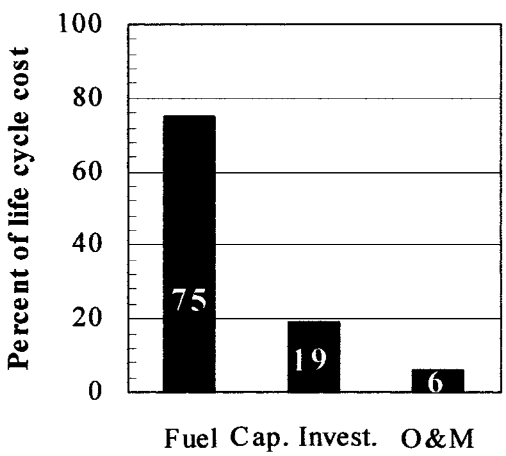

Figure 3. Break Up of Life-Cycle Costs for a Gas Turbine Based Combined Cycle Plant.

With deregulated industry and the highly competitive nature of the market, performance retention action has to be taken in a scientific manner and often has to be tailored to the specific gas turbine under consideration. There are always some projects where a pronouncement is made that efficiency is not of importance because fuel is "free." This is a mistaken notion because:

- There is always an opportunity cost associated with excessive fuel usage.
- Performance deterioration results in higher turbine temperature for a given power and this leads to increased hot end degradation. If the turbine is temperature limited, then there will be a loss of power.
- There is a close linkage between mechanical reliability and performance deterioration. An example would be the damaging effects of fouling on blading integrity that is discussed ahead.

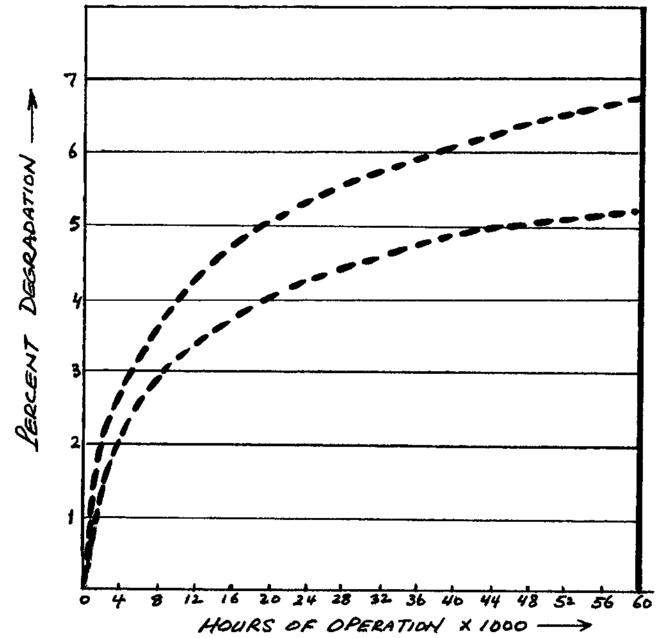

Figure 4. Long-Term Deterioration of Power and Heat Rate Showing Zone of Deterioration Versus Operating Hours.

Airline Industry

The annual fuel cost for a 30 aircraft fleet of wide-body aircraft is of the order of $250,000,000. Improving the efficiency of aircraft gas turbines has been a major technological driver over the past 50 years. Several of the technology approaches used on the aircraft engines will see application in the industrial gas turbine arena. Under the new Department of Energy program entitled the Next Generation Turbine Program (NGT), which is a follow up of the Advanced Gas Turbine Project, the gas turbine design targets a significant increase in efficiency. There are four competitors in this design effort—two of which have indicated that they will pursue intercooled aeroderivative designs. Rolls Royce will develop a derivative of the three shaft RB-211 and Trent engines. Pratt and Whitney will study a derivative of the FT60K IC, an intercooled aeroderivative gas turbine using a high-pressure spool that is the same size as a 60,000 lb thrust engine, operating at an estimated pressure ratio of 50:1. As fuel costs represent a major proportion of the operating costs of an airline, performance retention is of supreme importance.

The reasons for deterioration in modern high bypass turbofans are erosion 45 percent, blade and vane radial clearance 22 percent, seal radial clearance 5 percent, and the remaining 20 percent being due to miscellaneous causes. The airlines have done significant work in the performance monitoring of turbofan engines and much of this work can be transferred to the monitoring of industrial gas turbines.

TYPES OF PERFORMANCE AND MECHANICAL DETERIORATION IN GAS TURBINES

The causes of gas turbine deterioration fall essentially into two categories:

- Performance degradation
- Mechanical degradation

A further classification of performance deterioration into recoverable and unrecoverable deterioration can also be made:

- *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

The classification of performance deterioration is shown in Figure 5 and major factors are discussed below. An excellent treatment of performance degradation is made by Kurz and Brun (2000).

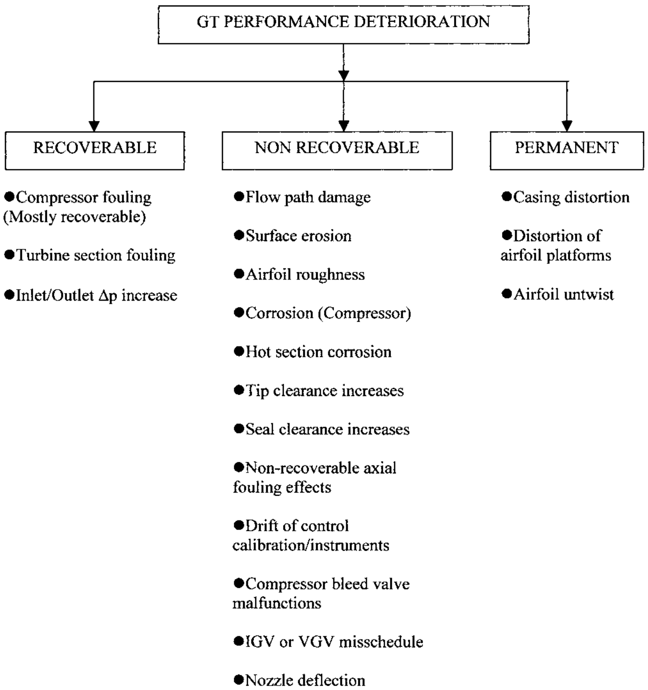

Figure 5. Classification of Gas Turbine Performance Deterioration.

Dirt Deposits and Fouling

Deposits form mainly on the compressor blading causing loss of flow capacity and efficiency. This mechanism is a predominant form of recoverable deterioration. Fouling deterioration is described by Meher-Homji (1990). The turbine section can also foul. The effects of fouling are a drop in output and a worsening of heat rate. There are several peripheral problems that may be created such as cooling hole blockage, imbalance, blade root lockup, and hot corrosion. Because this is a topic of major importance, it will be treated in a dedicated section ahead.

Erosion

Erosion changes airfoil shape, contours, and surface finish. This typically does not pose a major problem to land-based turbines due to the air filtration system present, but is a serious factor in certain environments. It contributes to 45 percent of the deterioration in modern turbofans. Erosion is a form of permanent unrecoverable deterioration. Bammert and Woelk (1980) have provided a detailed treatment of the influence of blade surface roughness on compressor performance.

Increases in Clearances in Vanes, Blades, and Seals (i.e., Wear)

Wear is a significant problem especially during the early stages of engine operation. It can be caused by thermal growth problems and centrifugal growth. Tip clearance increase of blading is a serious cause of deterioration as it reduces the stage efficiency. Tip stalls can also be caused by combinations of blade tip erosion and consequent loss of clearance. Increase in clearances is a form of unrecoverable deterioration.

Hot Section Problems

This can include detached liners, cracks, or unbalanced nozzles and is a form of unrecoverable deterioration. Problems of this nature may, in most instances, be detected by means of evaluation of exhaust gas temperature (or inter turbine temperature) spreads. Problems such as hot section corrosion, erosion, and fouling also exist and may be recoverable during a refurbishment.

Manufacturing Deviations

Deterioration can, at times, be induced due to manufacturing variations. Details of the deviations as they affect axial compressor blading and performance are covered in Marson (1992). The main sources of deviation include:

• Steps along compressor flow path. For example, a small step caused by the inlet casing being out of round will induce flow separation as the flow enters the bellmouth. Other sources include deformed inner shrouds or wavy outer shrouds of compressor diaphragms.

• Stagger angle deviations. Small variations in compressor blade stagger angles can cause a large spread in the mass flow.

• Excessive surface roughness and irregular coating.

• Excessive radial clearance between blade tip and casing and between seal diameter of diaphragms and the disk seal land. While minimal clearances provide high efficiency, from a mechanical point of view there is a limit to the tightness, and provisions have to be made for transient conditions to avoid rubs under perturbed condition.

• Reductions in blade chord and thickness.

• Improper axial position (distance between vane and blade is smaller than design value).

Other Sources of Deterioration

In addition to the items mentioned above, other sources of deterioration include:

• Excessive drop in inlet filter differential pressure.

• Excessive back pressure—this can be of importance when heat recovery steam generators (HRSG) are present.

• Increased mechanical losses (gearboxes, bearings, couplings, etc.).

• Internal losses.

• Stator nozzle plugging (or turbine fouling).

• Overboard leakage.

• Change in high-pressure flow function ($m\sqrt{T}/P$).

Details on gas turbine degradation and detection methods are presented by Saravanamuttoo and Lakshminarasimha (1985) and Diakunchak (1991, 1993). Modeling details of gas turbine performance deterioration may be found in Lakshminarasimha, et al. (1994). Williams (1981) has detailed the common performance related faults in pipeline applications.

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. 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, and 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 exhaust gas temperature (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).

EFFECTS OF SITE CONDITIONS ON GAS TURBINE PERFORMANCE

It is important to understand the effects of site conditions on gas turbine performance as they should not be confused with performance deterioration. Included here are:

- Effect of ambient temperature.
- Effect of altitude (i.e., pressure).
- Inlet filter pressure losses.
- Exhaust system pressure losses.
- Ambient humidity effects.
- Influence in changes in fuel heating value.

Influence of Ambient Temperature

As the density of air increases with a drop in inlet temperature, the airflow rate will be inversely proportional to the ambient temperature. In practice, the mass flow at a given rpm and inlet guide vane (IGV) setting will be proportional to the inverse of the temperature raised to a power between 1 and 0.5 because of the Mach number that the machine operates at. At colder temperatures, the increase in mass flow causes the power to increase. As the nozzle runs choked, the increased mass flow causes an increase in the pressure ratio of the machine assuming that the TIT was held constant. Further, as was discussed in Meher-Homji and Mee (1999), the compressor specific work drops as the inlet temperature drops compared to the turbine specific work (which is essentially governed by TIT).

With aeroderivative gas turbines, the gas generator spool speed would accelerate when the ambient temperature increases as the power developed by the turbine would stay the same, but the specific work of the compressor would drop. Thus the spool speed would attain equilibrium with a higher mass flow rate and pressure ratio. Usually there is a limit to the extent that power can increase with cooler temperatures. The limitation being mechanical speed or compressor discharge pressures or excessive Mach numbers at the inlet. It is important to note that because turbine blade cooling air is derived from compressor bleed, turbine blades will tend to run hotter during higher ambient temperatures.

As a rule of thumb, there is an estimated power drop of approximately 0.3 to 0.5 percent per degree Fahrenheit increase in ambient temperature. Unless a gas turbine inlet cooling technique is used, there is little that can be done regarding the ambient temperature. Inlet cooling techniques include traditional refrigeration, though its high first cost has made inlet evaporative fogging popular. Inlet fogging is described by Meher-Homji and Mee (2000). Chaker, et al. (2001), have shown that even though this is an evaporative cooling technique, it can be applied in humid areas due to the reduction in humidity that occurs at high dry bulb temperatures.

Influence of Altitude (i.e., Ambient Pressure)

High altitude reduces the air density and consequently the mass flow rate resulting in a drop in power with increasing altitude. As a rule of thumb, the power drop is approximately 3 to 4 percent per 1000 ft altitude. There is also a drop in mass flow rate similar in magnitude to the loss of power. Nothing can be done about this loss other than supercharging the gas turbine inlet with a blower, a solution that is not popular in practice.

Influence of Inlet Filter Pressure Losses

This is an important parameter as it is controllable to some extent and will affect turbine engine performance. The pressure drop across the inlet filter reduces the density to the inlet of the gas turbine, thus reducing the mass flow rate. The lower pressure at the turbine inlet implies a lower expansion ratio across the turbine and a consequent elevation in EGT.

As a rule of thumb, a 1 inch water gauge (WG) inlet pressure loss will cause a 0.48 percent drop in power and a 0.12 percent increase in heat rate.

Influence of Exhaust System Pressure Losses

An increase in the backpressure due to excessive stack losses or the presence of silencers causes a backpressure on the turbine section, which decreases the expansion ratio across the turbine. This drop in expansion ratio results in an increased EGT.

As a rule of thumb, a 1 inch water gauge exhaust pressure loss will cause a 0.15 percent drop in power and a 0.12 percent increase in heat rate.

Influence of Ambient Humidity

As a practical matter, the effect of ambient humidity is very small and is dependent on the control philosophy. For most performance deterioration evaluations, it can be neglected. A gas turbine operating at fixed firing temperature would produce more power with high humidity. This is because of the increased C_p value due to the presence of the water. However, the situation is not so straightforward due to control system complexities.

Influence in Changes in Fuel Heating Value

Gas turbines operate on a wide range of gaseous and liquid fuels. A useful number to understand the effect of gaseous fuel is the Wobbe number defined as:

$$Wobbe\ No. = LHV / \sqrt{SG} \tag{4}$$

Some OEMs define a temperature compensated Wobbe number by including the fuel temperature in degrees Rankine as a multiplier to the specific gravity (SG) term in the denominator. Fuel with significant heavy hydrocarbons present will have a higher Wobbe number, while the presence of inerts such as nitrogen and carbon dioxide will lower the Wobbe number.

In general, the use of low BTU fuel (i.e., low heating value) will cause an increase in power because of the extra mass flow through the turbine. A secondary effect is that the increased turbine mass flow causes an increase in pressure ratio. Changes in heating value become significant in gas turbines that operate on multiple fuels in that the effects on performance have to be distinguished from deterioration.

A treatment of performance impact due to variations in fuel heating value is made by Cloyd and Harris (1995). An interesting graph showing correction factors as a function of the C/H ratio of the fuel is shown in Figure 6 (Cloyd and Harris, 1995).

Effect of Turbine Fouling on Control Algorithms

As a turbine section of a gas turbine fouls, there will be a drop in the turbine flow coefficient, and the compression ratio of the compressor will increase as the turbine sections swallowing capacity is reduced. In some OEMs control systems, the compression ratio and exhaust gas temperature are used to determine the turbine inlet temperature (Zaba, 1980). This

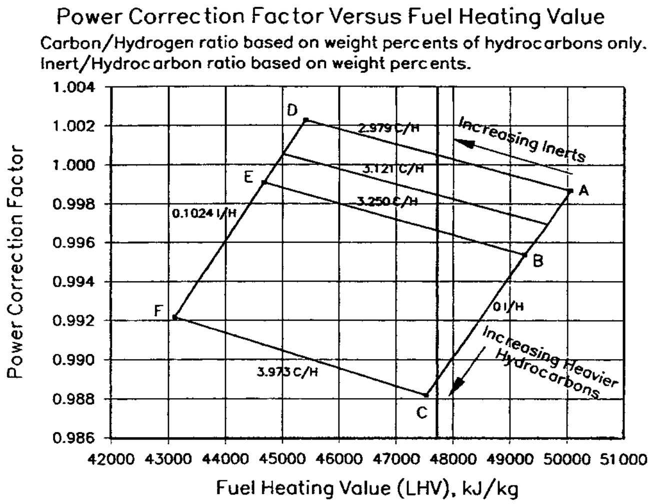

Figure 6. Performance Impact Due to Variations in Fuel Heating Value. (Courtesy ASME)

algorithm is based on a constant turbine efficiency and if this efficiency decreases due to turbine fouling, then the control system will indicate a higher turbine inlet temperature than is really present. Therefore the output of the turbine is further reduced.

RECOVERABLE DETERIORATION

Compressor Fouling

Compressor fouling and its control is by far the most important topic in the area of gas turbine performance deterioration and consequently is dealt with in some detail. Details on the topic may be found in Meher-Homji, et al. (1989), and Meher-Homji (1990).

The fouling of axial flow compressors is a serious operating problem with gas turbines. Its control is of supreme importance to gas turbine operators especially in the current deregulated and highly competitive power market. Foulants even in the ppm range can cause deposits on the blading, resulting in severe performance decrement. The effect of compressor fouling is a drop in airflow and compressor isentropic efficiency, which results in a "rematching" of the gas turbine and compressor resulting in a drop in power output and thermal efficiency. In some extreme cases, fouling can also result in surge problems as its effect is to move the compressor surge line to the right, i.e., toward the operating line.

Gas turbines ingest large quantities of air with the larger gas turbines having airflow rates of as high as 1500 lb/sec. The solids or condensing particles in the air and in the combustion gases can precipitate on the rotating and stationary blading causing changes in aerodynamic profile, dropping the compressor mass flow rate and affecting the flow coefficient and efficiency. This has an adverse effect on the unit's performance. The output of a gas turbine can drop by as much as 10 percent. Moreover, contaminated air can cause a host of problems that include erosion, fouling, corrosion, and, in some cases, plugging of the hot section cooling passages.

Some estimates have placed fouling as being responsible for 70 to 85 percent of all gas turbine performance loss accumulated during operation. Output losses between 2 percent (under favorable conditions) and 15 to 20 percent (under adverse conditions) have been experienced. In a gas turbine, approximately 50 to 60 percent of the total work produced in the turbine is utilized by the compressor, hence maintaining high compressor efficiency is an important contributing factor for the plant's revenue stream. To quantify this important fact, computer software simulations were run for a variety of gas turbines running at 59°F and with typical inlet and outlet losses, and the results are depicted in Table 1. The last column of this tabulation provides the ratio of compressor section work, W_c, to the total turbine section work, W_t. Parasitic losses and mechanical efficiency have also been considered, which accounts for the output being somewhat less than the difference between the turbine and compressor work. It is useful to present this type of data to operators to convince them of the importance of maintaining clean axial compressors.

Table 1. Computer Simulation Giving Amount of Work Consumed in the Compressor for Several Gas Turbine Models. (Inlet Temperature 59°F, Inlet and Outlet Losses Taken at 3 and 4 inch WG, Respectively, Natural Gas Fuel.)

GAS TURBINE	Output, kW	Compressor Work, W_c kW	Total Turbine Work, W_t kW	W_c/W_t
Typhoon	5,106	8,104	13,536	0.59
Centaur 50	4,481	6,263	11,093	0.56
Mars 100	10,463	17,938	28,981	0.61
Cyclone	12,585	15,344	28,629	0.54
Frame 5371 PA	26,056	38,960	66,311	0.59
RB-211	26,678	41,222	69,258	0.60
Frame 6581B	41,487	50,706	94,229	0.54
Trent 60	50,990	84,280	136,478	0.61
GT 8C2	56,295	77,627	136,149	0.57
Frame 6101FA	70,116	74,237	147,127	0.50
Frame 7121EA	85,206	105,264	192,481	0.55
Alstom 13E2	160,748	190,196	354,064	0.54
Frame 7241 FA	172,757	164,387	340,671	0.48
S-W 501G	247,391	241,350	493,292	0.49
Frame 9351 FA	255,400	229,707	490,325	0.47
Frame 9001H	326,606	324,650	657,922	0.49

Simulation Results of Compressor and Turbine Section Deterioration

It is instructive to examine the effect of compressor and turbine deterioration on overall plant performance. To this end some simulation runs using computer software have been made considering a Frame 7EA in simple cycle configuration and a Frame 7FA in combined cycle operation. Natural gas has been considered and typical inlet and outlet pressure drops for simple and combined cycles were used.

Salient performance parameters for a Frame 7EA gas turbine at an ambient temperature of 59°F and operating with natural gas fuel are shown in Table 2. The compressor work is 105,264 kW and the turbine total work is 192,481 kW. The net output is 82,206 (considering mechanical and parasitic losses). The effect on cycle efficiency with a reduction in compressor efficiency is shown in Figure 7. A 5 percent change in compressor efficiency causes the thermal efficiency to drop from 33 percent to 30.7 percent. The effect on the plant output is shown in Figure 8.

A simulation of a combined cycle comprising 1× Frame 7FA (TIT of 2373°F), a three pressure level HRSG, and reheat condensing steam turbine was made to examine the effects of turbine section efficiency drops. The effect on the plant output is shown in Figure 9. The corresponding effect on the overall combined cycle heat rate is shown in Figure 10. Salient gas turbine parameters for the 7FA when operating under the maximum deteriorated condition are provided in Table 3.

Causes of Fouling

Experience has shown that axial compressors will foul in most operating environments, be they industrial, rural, or marine. There are a wide range of industrial pollutants and a range of environmental conditions (fog, rain, humidity) that play a part in the fouling process.

Compressor fouling is typically caused by:

- Airborne salt.

Table 2. Salient Performance Parameters for a Frame 7EA Gas Turbine at an Ambient Temperature of 59°F and Operating with Natural Gas Fuel.

GT PRO 10.0.1 Cyrus B Meher-Homji
1135 02-25-2001 23:07:51 file=c:/TFLOW4/MYFILES/GTPRO.GTP

Plant Configuration: Simple Cycle Gas Turbine(s)
One GE 7121EA Engine, GT PRO Type 0, 0 Drum, Subtype 0

ESTIMATED G.T. SITE PERFORMANCE

Fuel=CH4, supplied @ 77 F, LHV = 21518 BTU/lb
G.T. @ 100 % rating, inferred TIT control model, CC limit
Site ambient conditions: 14.7 psia, 59 F, 60% RH
Total inlet loss = 4 inch H2O, Exhaust loss = 5 inch H2O
Duct & stack = 5, HRSG = 0 inch H2O

# 104 Model	PR	TIT F	TET F	Mair lb/s	kW	H.R.LHV BTU/kWh	Mex lb/s	N2+Ar %	O2 %	CO2 %	H2O %
GE 7121EA	12.6	2035	1000	644	85206	10302	655	75.82	13.97	3.10	7.12

Fuel compressor = 579.9 kWe, Q rejected = 110.9 BTU/s, exit temperature = 135.1 F
Fuel molecular weight = 16.04; LHV @ combustor = 21549 BTU/lb
G.T. auxiliary power = 433.2 kWe.

ESTIMATED G.T. CYCLE

STREAM	TEMP. F	PRESS. psia	MASS FLOW lb/s	M.W.	MOLE COMPOSITION % N2+Ar	O2	CO2	H2O
Ambient air in	59	14.70	644.00	28.86	78.22	20.74	0.03	1.01
Compr. inlet	59	14.55	644.00	28.86	78.22	20.74	0.03	1.01
Turbine coolant	misc.		39.41					
Combustor inlet	690	183.38	604.59	28.86	78.22	20.74	0.03	1.01
Fuel flow	135	247.57	11.33127					
Turbine inlet	2035	176.05	615.92	28.44	75.67	13.54	3.29	7.50
Turbine coolant			39.41					
Turbine exhaust	1000	14.88	655.33	28.46	75.82	13.97	3.10	7.12

Compressor = 105264 Turbine = 192481 kW
Turbine coolant = 6.12% compr in
Mech loss = 709.7 kW Generator loss = 1301.7 kW
Mech eff. = 99.19% Generator eff. = 98.5%
GT specific power @ gen term = 132.3 kW per lb/s
GT efficiency @ gen term = 29.852% HHV = 33.12% LHV

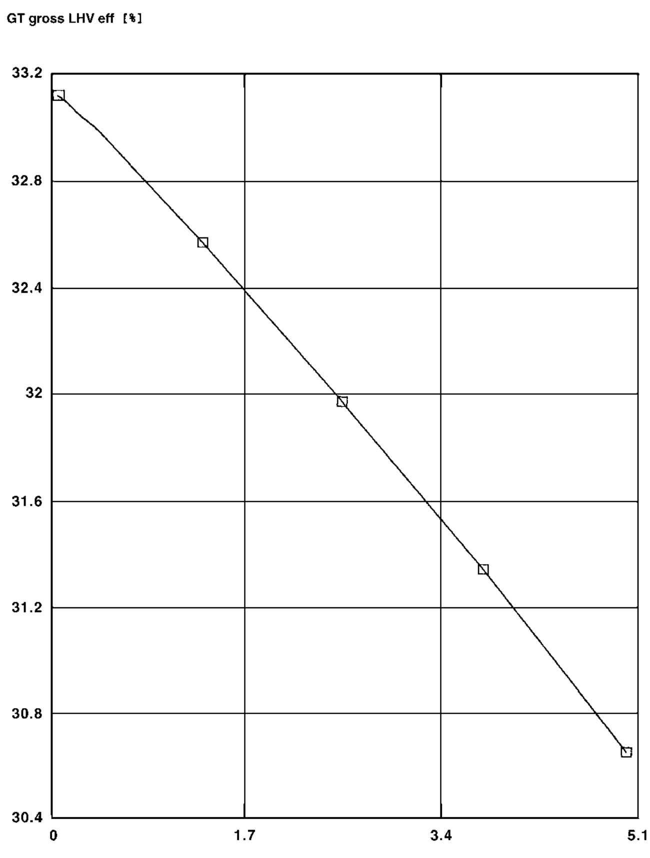

Figure 7. The Effect on Reduction in Cycle Efficiency with Changes in Compressor Efficiency on a Frame 7EA Gas Turbine in Simple Cycle Configuration.

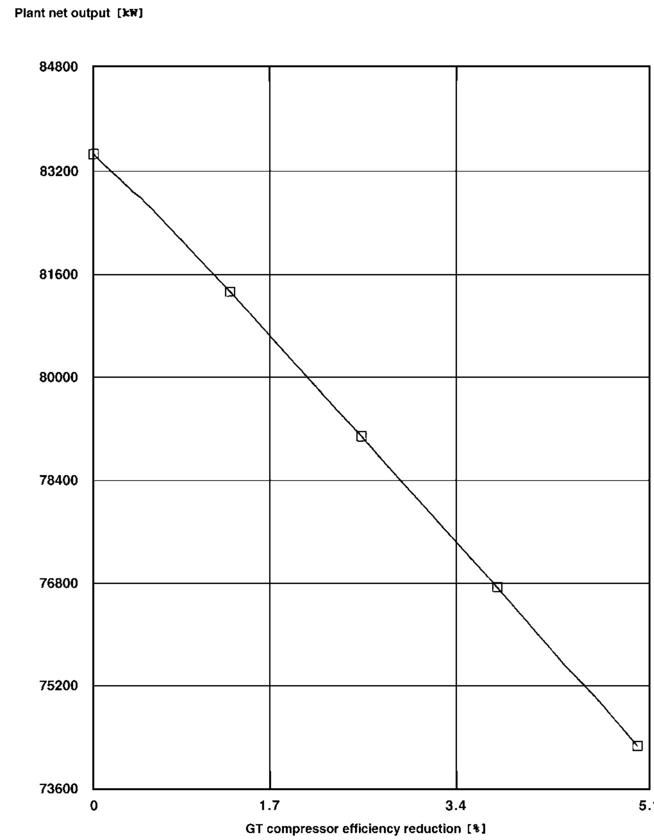

Figure 8. The Effect on Reduction in Power with Changes in Compressor Efficiency on a Frame 7EA Gas Turbine in Simple Cycle Configuration.

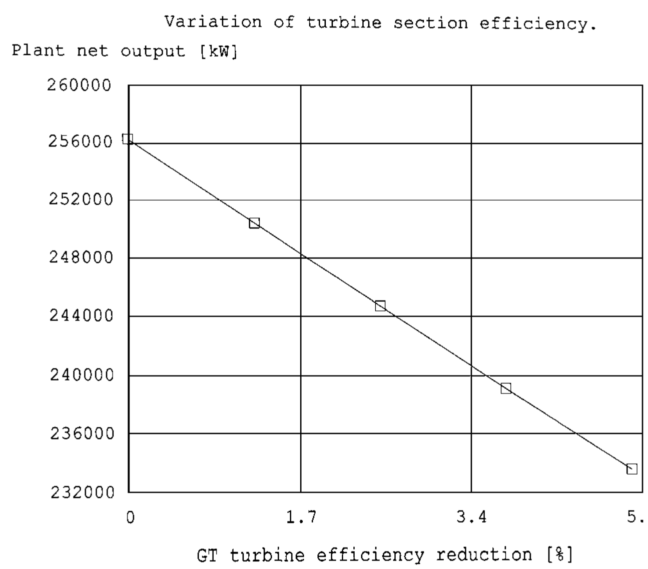

Figure 9. Simulation Results of Effect on Plant Output of a Drop in Turbine Section Efficiency for a Combined Cycle Comprising 1× Frame 7FA (TIT = 2373°F), Three Pressure Level HRSG, and Reheat Condensing Steam Turbine).

- Industrial pollution—Fly ash, hydrocarbons, smog, etc. This causes a grimy coating on the early compressor stages and can get "baked on" in the latter stages because of the high compressor discharge temperatures (especially in high-pressure ratio compressors).

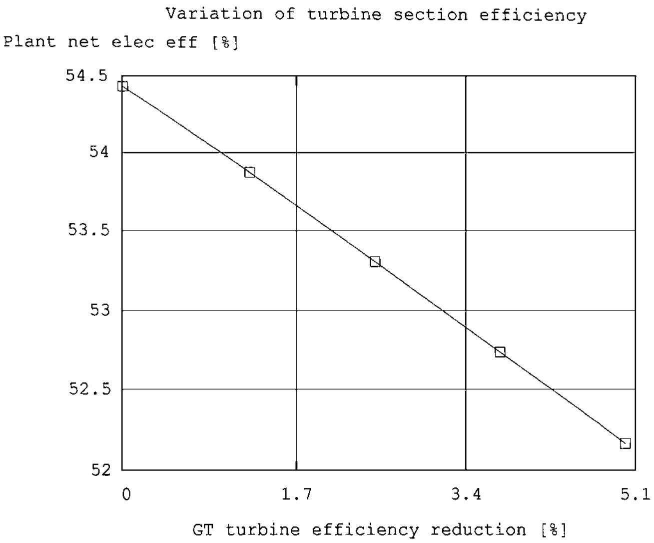

Figure 10. Effect on the Overall Combined Cycle Efficiency with Drop in Turbine Section Efficiency.

Table 3. Salient Gas Turbine Performance Parameters for 7FA Gas Turbine when Operating Under Maximum Turbine Section Deteriorated Condition.

ESTIMATED G.T. CYCLE

STREAM	TEMP. F	PRESS. psia	MASS FLOW lb/s	M.W.	MOLE COMPOSITION % N2+Ar	O2	CO2	H2O
Ambient air in	59	14.70	961.25	28.86	78.22	20.74	0.03	1.01
Compr. inlet	59	14.55	961.25	28.86	78.22	20.74	0.03	1.01
Turbine coolant	misc.		139.96					
Combustor inlet	702	220.12	821.29	28.86	78.22	20.74	0.03	1.01
Fuel flow	157	297.16	19.72499					
Turbine inlet	2371	209.11	841.02	28.33	74.98	11.60	4.17	9.25
Turbine coolant			139.96					
Turbine exhaust	1146	15.49	980.98	28.40	75.44	12.88	3.59	8.09

Compressor = 159042 Turbine = 309113 kW
Turbine coolant = 14.56% compr in
Mech loss = 1056.3 kW Generator loss = 2109.5 kW
Mech eff. = 99.3% Generator eff. = 98.58%
GT specific power @ gen term = 152.8 kW per lb/s
GT efficiency @ gen term = 29.566% HHV = 32.81% LHV

- Ingestion of gas turbine exhaust or lube oil tank vapors.
- Mineral deposits.
- Airborne materials—Soil, dust and sand, chemical fertilizers, insecticides, and plant matter.
- Insects—This can be a serious problem in tropical environments. Moths with wingspans of as high as 18 cm (7 inches) have been known to clog up intake systems. Figure 11 shows one such insect.
- Internal gas turbine oil leaks—Axial compressor front bearing is a common cause. Oil leaks combined with dirt ingestion cause heavy fouling problems.
- Impure water from evaporative coolers (carryover).
- Coal, dust, and spray paint that is ingested.

A fouled compressor is shown in Figure 12. Often the inlet struts and IGVs get severely fouled. A photo of a fouled intake strut is shown in Figure 13. Hand cleaning the IGVs and first stage will restore a considerable amount of performance.

It is interesting to note that severe fouling was caused on the high bypass engines of the Fairchild A-10 "Warthog" aircraft due to the ingestion of gun propellant when the high performance Gatling gun was fired. This was a serious problem for the aircraft causing a severe loss of performance and necessitating compressor washes after every mission until a solution could be found. Figure 14 shows the A-10 aircraft with the Gatling gun firing (3000 rounds/minute) and propellant being ingested into the turbofans. The solution involved the use of a specially designed deflector at the gun barrel.

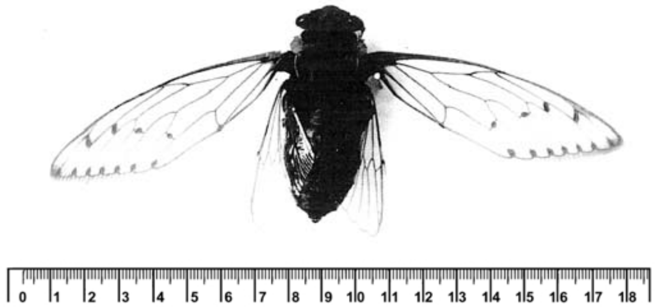

Figure 11. Moth with Wingspan of 18 CM (7 Inches) That Can Clog Intake Systems in Tropical Environments. (Courtesy Altair Filter Technologies)

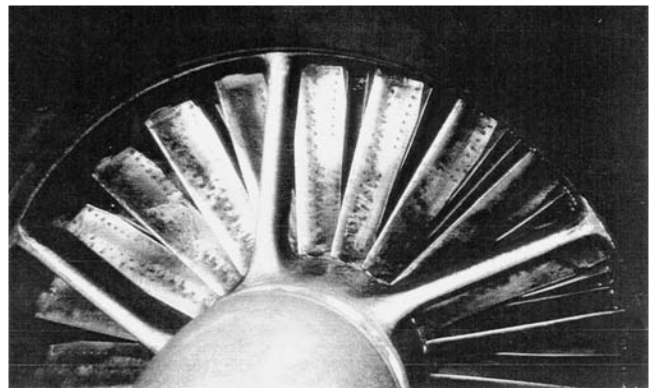

Figure 12. Fouled Compressor. (Courtesy Altair Filter Technologies)

Figure 13. Fouled Intake Strut.

Figure 14. Fairchild A-10 "Warthog" Aircraft with Gatling Gun Firing (3000 Rounds/Minute) Showing Used Propellant Being Ingested into the T-34 High Bypass Turbofans Causing Severe Compressor Fouling.

The fouling rate for a compressor will be a strong function of the environment, the climatic conditions, the wind direction, and the filtration system. A 75 MW unit located in an industrial environment with air loading of 10 ppm will ingest 594 lb of particulates in a day. Ambient air can be contaminated by solids, liquids, and gases. Air loadings can be defined in mg/m^3, grains/1000 ft^3, or ppm (mass of contaminant per unit mass of air). In general, particles up to 10 microns cause fouling, but not erosion. Particles above 10 to 20 microns cause blading erosion. Some typical air loadings, are:

- Country—0.01 to 0.1 ppm by weight.
- Coastal—0.01 to 0.1 ppm by weight.
- Industrial—0.1 to 10 ppm by weight.
- Desert—0.1 to 700 ppm by weight.

Felix and Strittmatter (1979) have detailed the type of analysis that should be done at a gas turbine plant site. In most industrial areas, the air quality can create quite acidic conditions in the axial compressor.

The susceptibility of different types of gas turbines (as a function of their design parameters) has been studied by Seddigh and Saravanamuttoo (1990). Aker and Saravanamuttoo (1988) have provided results pertaining to fouling based on stage stacking techniques.

Marine and Offshore Environment

There is considerable variance in data regarding the amount of salt present in an offshore environment. Airborne salt concentrations provided by different researchers are shown in Figure 15 (Cleaver, 1988). The NGTE 30 knot standard is often used to define the environment of offshore platforms.

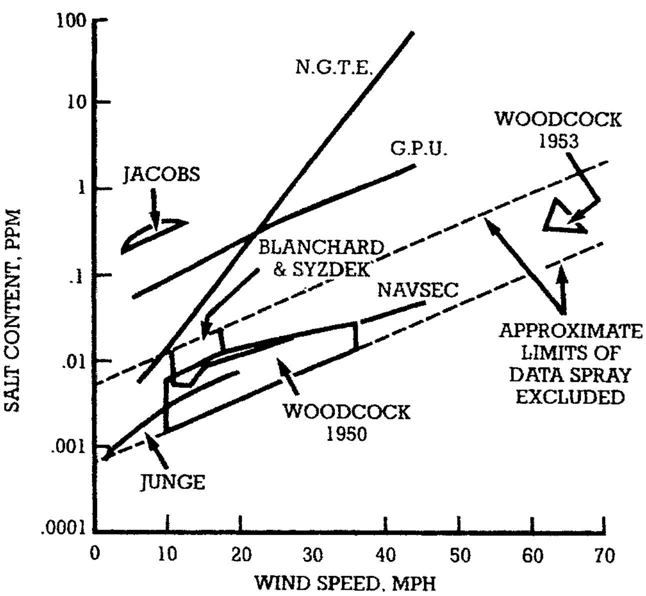

Figure 15. Airborne Salt Concentrations Provided by Different Researchers. (Courtesy Altair Filter Technologies)

The offshore environment is particularly challenging. Some key issues are:

- Airborne salt can exist in three basic forms: aerosol, spray, and crystal. Aerosols can range in size from 2 microns to 20 microns (1 micron = 10-6 m). Aerosol is generated by bubbles shattering on the sea surface. Sea spray generates large droplets sized 150 to 200 microns and these tend to drop out due to gravity. Sea salt crystals absorb moisture under appropriate relative humidity conditions. The size of these peak in the range of 2 microns. The relative humidity offshore was found to be almost always high enough to ensure that salt was in its wet form. Studies by Tatge, et al. (1980), concluded that salt would stay as supersaturated droplets unless the relative humidity dropped below 45 percent.
- The environment on offshore platforms is not "dust free" and can include flare carbon and mud burning foulants, drilling cement, and other dusts. These can pose a problem with poorly positioned flare stacks and with sudden changes in wind direction. Grit blasting has also been a serious problem.

Humidity Effects on the Fouling of Axial Compressors

As air passes through the intake and filtration system, it proceeds at a very low velocity. As it approaches the compressor face, the air accelerates to a high velocity (0.5 to 0.8 Mach number). This results in a static temperature reduction of about 10 to 15°C. The saturation air temperature also drops. If the relative humidity is high enough, it is possible that the static air temperature falls below the saturation air temperature. This causes condensation of water vapor, which is a common occurrence in most gas turbines when ambient relative humidity is high. Filters tend to unload salt (leeching effect) under high ambient humidity conditions and this is a factor that is often neglected. It is this factor that causes the sudden fouling of compressors during periods of ambient fog. Particles then form nuclei for the water droplets and start to adhere to the blading. As the air progresses to the rear compressor stages, it gets hotter and drier typically causing less fouling in the rear stages. A photograph of salt leeching through a filter is shown in Figure 16. Typical static temperature drops that might be expected in a gas turbine are shown in the graph in Figure 17. The fouling effect of early morning fog on filter systems is shown in Figure 18, where during the cold morning hours, the differential pressure across the filter increases. There have been some instances where the differential pressure has grown to levels that have actually caused turbine trips.

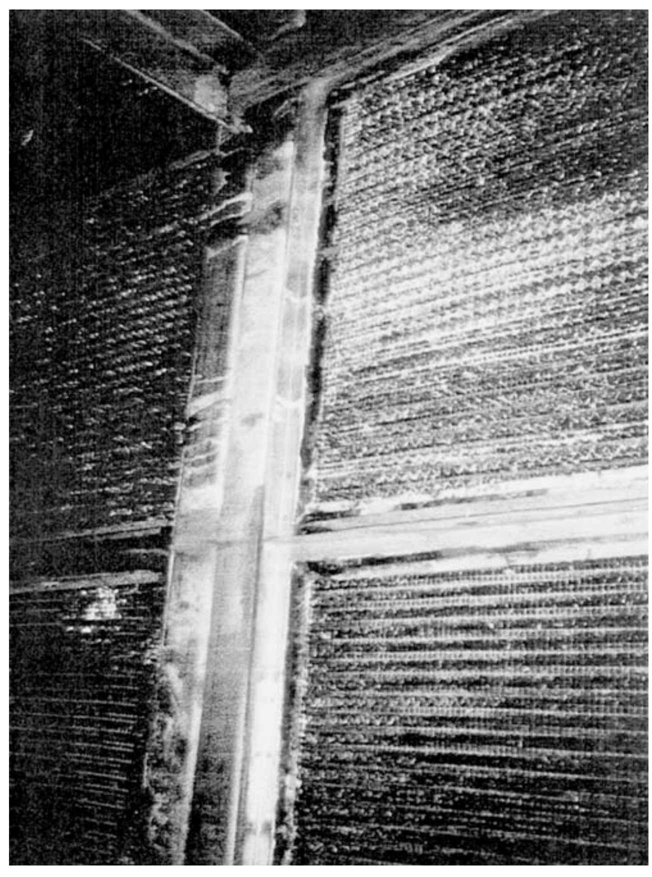

Figure 16. Photograph of Salt Leeching Through a Filter. (Courtesy Altair Filter Technologies)

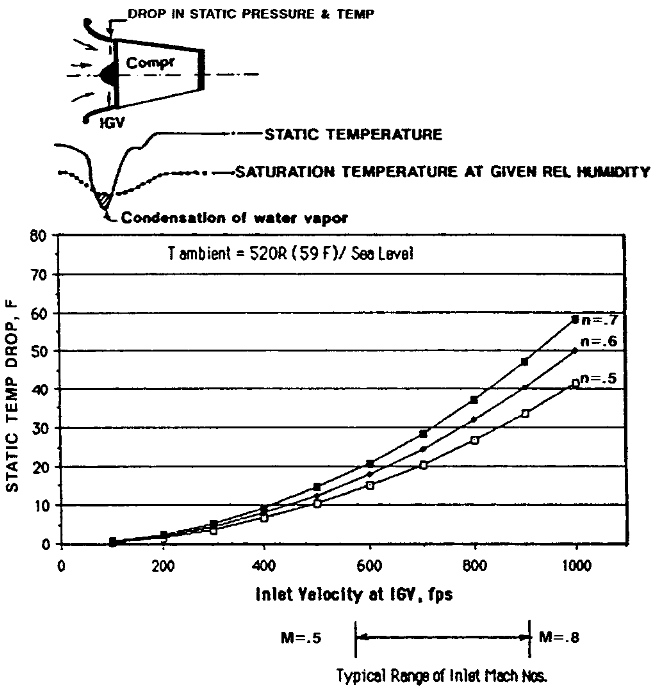

Figure 17. Typical Static Temperature Drops Expected in a Gas Turbine Intake.

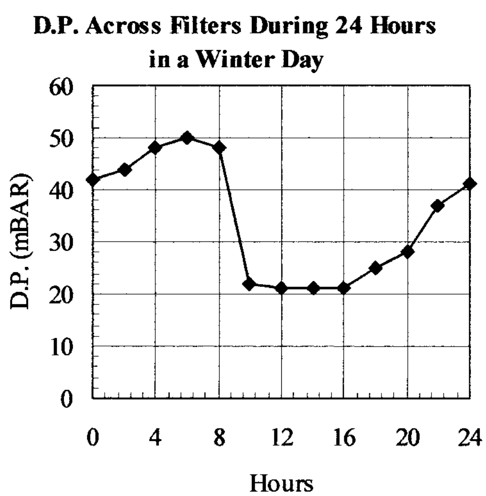

Figure 18. The Fouling Effect of Early Morning Fog on Filter System Differential Pressure.

Filter House Design

Experience has shown that stainless steel (316L) construction is *by far the most sensible approach* especially in offshore and coastal areas. The life-cycle cost when one considers the constant painting that has to be done to carbon steel filter systems is several times the small incremental first cost increase for the use of SS 316L construction. It is indeed unfortunate that the desire to minimize first cost often results in filter systems that cause great harm to the project in terms of gas turbine deterioration, and even subsequent damage that can be caused by severe rusting and other distress over the project life-cycle.

Corrosion in Inlet Filter Houses

Corrosion of inlet systems especially when they are made of carbon steel is widespread. Corrosion of the filter house downstream of the filters that totally defeats the purpose of having air filtration is shown in Figure 19. Painted carbon steel structures need constant painting and, often, poor application of paint and other problems involving dissimilar materials start the corrosion process weeks after the painting is completed. Proper painting requires the use of shot blasting to fully clean the parent surface prior to the painting process. Corrosion at the discharge side of a silencer is shown in Figure 20.

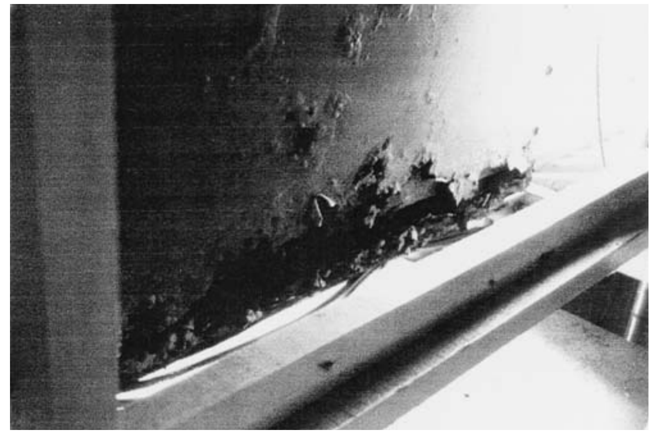

Figure 19. Corrosion of the Filter House Downstream of the Filters. (Courtesy Altair Filter Technologies)

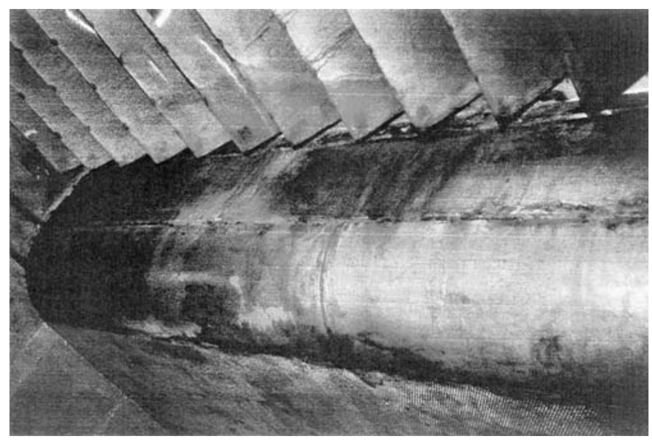

Figure 20. Corrosion at the Discharge Side of a Silencer. (Courtesy Altair Filter Technologies)

Modern technology has allowed the use of high performance SS 316L systems that include relatively light elements, which are very effective against driving rain and high humidity. These filter systems use special high velocity vane separators that have proven very effective for offshore applications. A filter house for an Avon gas turbine is shown in Figure 21.

The choice of material is also very important as SS 304 or 321 steels do not have sufficient corrosion protection. The presence of chromium (up to 18.5 percent) in SS 316L helps to build a passive protective film of oxide and prevents corrosion. Along with the 10 to 14 percent nickel content, 316L can be easily welded. In offshore and hostile environments, it is best to have all the system components, including vane separators and door hinges, made of 316L stainless steel. Proper attention must also be paid to drain systems to get the water out of the system. An excellent treatment of air filtration is made by Cleaver (1990). Details regarding the testing of gas turbine air filters are provided in Gidley, et al. (1993).

Aerothermodynamic Effects of Compressor Fouling

The observable effect of compressor fouling is a drop in thermal efficiency (increase in heat rate) and a drop in output. The axial

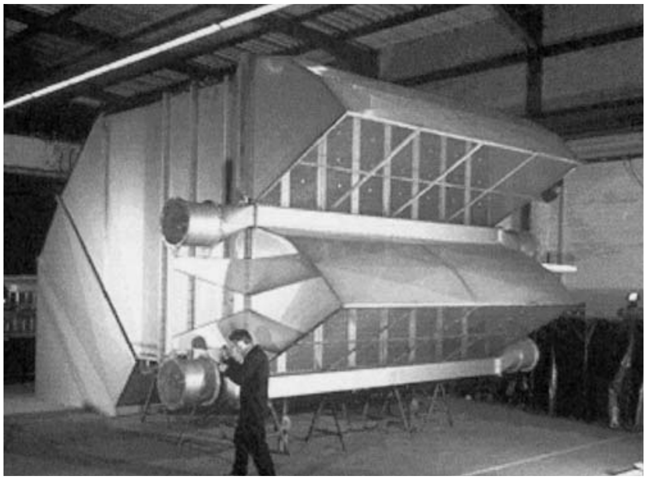

Figure 21. 316 L Filter House for an Avon Gas Turbine. (Courtesy Altair Filter Technologies)

flow compressor is a sensitive component that requires smooth aerodynamic surfaces. Fouling causes an alteration in the shape of the blading, which reduces air flow rate, pressure ratio, and compressor efficiency.

An axial compressor is a machine where the aerodynamic performance of each stage depends on the earlier stages. Thus, when fouling occurs in the inlet guide vanes and the first few stages, there may be a dramatic drop in compressor performance. This can occur when oil and industrial smog or pollen are present and form an adhesive wetting agent. The early stages are often the worst fouled. If the rear stages foul, this seems to have a smaller impact on performance; but due to higher temperatures, deposits can get baked on and become difficult to clean. This baking effect is more severe on the high pressure ratio compressors, e.g., 18 to 30:1 pressure ratio of typical aeroderivative gas turbines as opposed to the typical 10:1 or 14:1 pressure ratios found on the heavy duty industrial gas turbines.

Figure 22 shows a fouling process that occurs in a heavy duty gas turbine engine. This graph shows the changes the compressor efficiency and gas turbine heat rate over time.

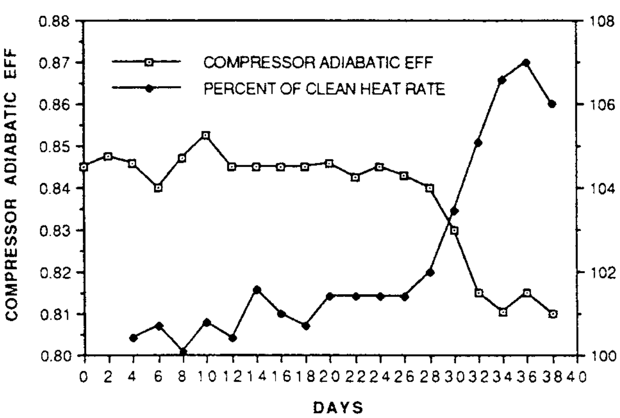

Figure 22. Fouling Process on a Large Gas Turbine Engine Showing Changes in Compressor Efficiency and Heat Rate Over Time.

Fouling, Airflow Distortion, and Compressor Surge

As fouling reduces the mass flow (flow coefficient) in the first compressor stage, the performance of the latter stages is affected and the operating point on the first stage characteristic moves toward the left. The first stage pressure ratio is thus increased. This causes a higher density at the inlet to the second stage. Thus there will be a further reduction in second stage flow coefficient. This effect progresses through successive stages until a rear stage stalls triggering surge. The velocity triangles and effect of fouling on incidence losses is shown in Figure 23 (Meher-Homji, et al., 1989). Details on the effects of stage characteristics on axial flow compressor performance may be found in Stone (1958).

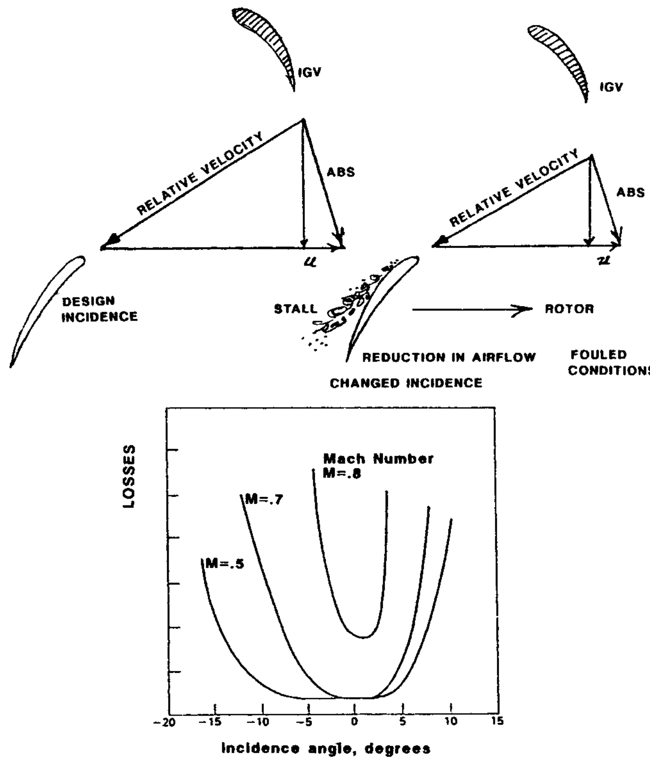

Figure 23. Velocity Triangles and Effect of Fouling on Incidence Losses.

Dundas (1986) has conducted an excellent analytical investigation into the deterioration of turbine operation including drop in compressor efficiency, fouling, first stage nozzle distortion, internal bleed seal deterioration, drop in turbine efficiency, inlet filter fouling, and low fuel heating value. These were examined to study the effect on the turbine operating line. His study concluded that compressor fouling had a pronounced effect on the operating line. Figure 24 (Dundas, 1986) indicates the effect of compressor fouling on the compressor operating line of a single shaft gas turbine. Being a constant speed machine, the traditional "speed lines" on the compressor map become constant temperature lines ($N/\sqrt{T}$).

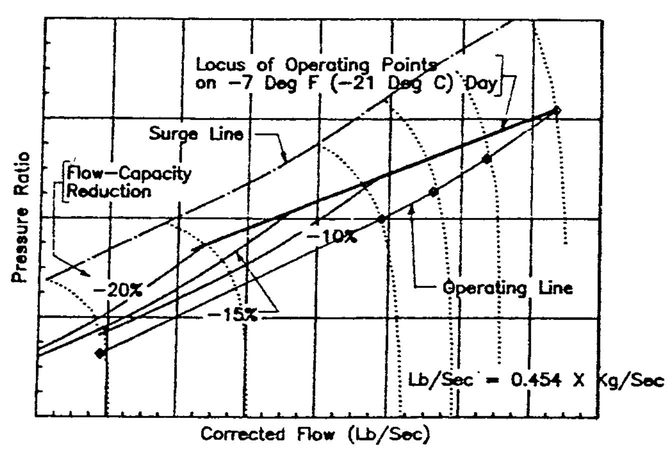

Figure 24. Effect of Compressor Fouling on the Compressor Operating Line of a Single Shaft Gas Turbine. (Courtesy ASME)

Whereas this effect causes the movement of the operating line toward the surge line, there are other factors that can cause movement of the surge line itself. Erosion of compressor blading can affect boundary layer development and increase the tendency toward separation. Stall can therefore occur at lower incidence angle than with smooth compressor blading. Heavy erosion can also reduce blade tip chords, thereby reducing blade tip solidity, which would adversely affect stage stability.

The importance of considering fouling effects on surge becomes more important with the use of gas turbines in combined cycle cogeneration applications and with IGV control and steam injection applications.

There have been several cases where excessive distortion of the inlet airflow has triggered a surge event resulting in compressor damage. Icing, causing uneven inlet circumferential distortion or uneven clogging of filters, possibly due to a bend in the inlet duct before the filter or improper inlet system design, can create distortion effects that could result in surge. Studies and results relating to the flow in inlet ducts are in Manfrida, et al. (1988).

Effects of Fouling on Compressor Blading Integrity

While fouling cannot be said to be a major cause of blading failure, it can contribute to blading problems as indicated below:

- By promoting surge or rotating stall that may have a dangerous effect on blades.
- In some cases, blading natural frequencies can be affected by the increase in mass due to dirt buildup on the blading. Blade roughness, and therefore efficiency and performance, can be adversely affected by corrosion, erosion, and fouling. Excessive dirt on the blades can cause unbalance and a consequent increase in running speed vibration. In some cases dirt can get between the bearing surfaces of the blade root, causing the blades to operate in an abnormal position, which would add to the stresses. If the root constraint is changed due to buildup in the fir tree region, a change in natural frequency could result (as the boundary condition changes). The blocking (or partial blockage) of cooling passages of hot section stators and blades can be caused by fine foulants (typically less than 5 microns). As the cooling air is bled from the compressor, foulants can enter the cooling system. Cement dust, coal dust, and fly ash can be responsible for this problem. The effects can be improper cooling and accelerated thermal fatigue though typically the effects are gradual in nature.
- Foulant buildup on compressor blading can lead to a serious corrosion problem, especially when humidity is high (galvanic action can be set up). Pitting of the blading can lead to local stresses that reduce the blade's fatigue life. Some airborne salt may be 1 to 3 microns in size. This problem will occur if salt water or salt particles are ingested in the compressor. The dry salt or brine will absorb moisture during high humidity operation or during water washing. This is caused due to chemical reactions between the engine components and airborne contaminants. Salts, mineral acids, and aggressive gases (e.g., chlorine) along with water can cause wet corrosion and compressor blade pitting. This can lead to local stress raisers that can diminish blade fatigue life. Compressor coatings are of value here. An important corrosion process in compressors is known as electrochemical or "wet corrosion."
- Particles causing erosion are normally 10 microns or greater. Five to 10 microns represent the transition zone between fouling and erosion. (Note: 10 microns = 1/15 diameter of a human hair). Erosion impairs aerodynamic performance and can affect the blade mechanical strength of the blade. Erosion first increases blading surface roughness thus lowering efficiency slightly. As erosion progresses, airfoil contour changes occur at the leading and trailing edge as well as at the blade tip. Severe erosion has also been known to cause changes in blade natural frequency.
- On a relatively small gas turbine, a 0.1 mm coating applied to the blading can cause a flow reduction of 10 percent and a reduction of compressor efficiency of 5 percent.

Foreign Object Damage (FOD)

Although not linked to fouling directly, this is mentioned as it could be caused due to a loss in filter integrity. Damage is typically caused to the early compressor stages, although in some cases the foreign object works its way to later stages also and causes damage. Damage is a function of size, foreign object composition, blade construction, and impact location. It can lead to direct or secondary failure. Foreign object damage can be caused by ice, failed intake section components, materials, and tools left in the inlet plenum.

Special Considerations for Traditional Evaporative Cooling Relating to Deterioration

Traditional evaporative coolers that use media for evaporation of the water have been widely used in the gas turbine industry, especially in hot arid areas. The basic principle of media type of cooling is that as water evaporates, it consumes 1160 BTUs of heat (latent heat of vaporization (LHV)) and in doing so reduces the ambient air temperature.

For a typical industrial gas turbine with an air mass flow rate of 250 lb/sec, the water evaporation rate can range from 200 to 800 gal/hr depending on the relative humidity. A detailed treatment of evaporative cooling as applied to gas turbines may be found in Johnson (1988).

There are several issues relating to media evaporative cooling that can affect performance deterioration.

- *Blowdown issues*—There are two types of evaporative coolers: circulating and noncirculating. The coolers used for most gas turbine operations are of the recirculating type. Consequently there is a requirement for blowdown in order to avoid the accumulation of minerals in the water. Thus make up water will equal the blowdown water plus the water evaporated. The blowdown rate is dependent on the hardness of the water. The blowdown rate should equal 4× the evaporation rate.
- *Mist eliminator*—As the water is not treated, it is imperative that *none enters the compressor* and so a mist eliminator is provided on the downstream side to ensure that the air entrained water droplets are removed. If water were to enter the compressor, severe fouling could occur.
- *Water flow rates*—Water flow rates are typically between 1 to 2 gal/min for each square foot of surface area of the distribution pad but this number can be higher for larger evaporative coolers. Higher flow rates minimize the potential of mineral buildup but increase the risk of entrainment of the water in the air stream. Thus the amount of water should be carefully adjusted during commissioning and should not be "tuned" constantly as this often leads to excessive dry spots or the other problem of water carryover.
- *Water carryover*—Even with the use of mist eliminators, water carryover must be minimized. Johnson (1988) has listed the causes of water carryover, some of which are provided below.
    - Incorrect media polarity
    - Damaged media (can occur after field reinstallation and can result in improper alignment and cracks between media). Poor handling often crushes media.
    - Improperly aligned media strips. If strips are not properly aligned, the resulting gap may allow water carryover.
    - Poor media seal against retainers
    - Excessive water flow. Media flooding can cause carryover.
    - Uneven water distribution from the header. This is often caused by improper initial design of the holes or clogging of the holes resulting in an imbalance of flow over the media.
    - Uneven or distorted airflow throughout the evaporative cooler
    - Scale deposits on the media

Media type evaporative coolers do not normally require demineralized water; in fact demin water can damage the media. However, some operators have reported compressor fouling caused by carryover of water with high levels of dissolved minerals. This can be avoided by installing mist eliminators downstream of the cooler or by ensuring that the air velocity through the media is not so high as to cause carryover. Some form of water treatment may be required in order to deal with potential problems of microbiological fouling, corrosion, and of course scaling. It is also advisable to clean and flush the header on a regular basis.

It is very important that the drains from the air filtration system have seal legs or valving to prevent the sucking in of air into the air filter. The arrangement shown in Figure 25 shows a drain and an overflow piping of an evaporative cooler on a 20 MW gas turbine without a seal leg.

Figure 25. Drain and Overflow Piping without a Seal Leg.

Further the floor of the inlet plenum must have the appropriate draining arrangement to get rid of the water that is used for crank washes. A floor of a filter house with paint damage caused by a lack of drainage is shown in Figure 26.

Figure 26. Floor of a Filter House with Paint Damage Caused by a Lack of Drainage.

The presence of a media type evaporative cooler inherently creates a pressure drop and this will create a drop in turbine output. This inlet pressure drop exists year around regardless if the media type evaporative cooler is used or not. This factor is important when considering the application of any inlet cooling technology such as evaporative systems, refrigeration, etc., and the effect of the increase in inlet differential pressure must be factored into the evaluation.

Detection of Fouling

Gas turbine manufacturers and operators usually develop guidelines as to when fouling deterioration calls for corrective action. This is based on a combination of load and exhaust gas temperatures. Users also monitor compressor discharge pressure and compressor efficiency. Graphs can be plotted to show expected (clean) versus measured parameters. It is the opinion of some operators that the only way to detect a fouled compressor is by visual inspection. With most turbine designs, however, this means shutting the unit down, removing the inlet plenum hatch, and visually inspecting the compressor inlet, bellmouth inlet guide vane (IGVs), and visible early stage blading. The following factors can be used as indicators of fouling:

- Drop in compressor mass flow rate on fixed geometry engines.
- Drop in compressor efficiency and pressure ratio (or discharge pressure). A graph giving compressor discharge pressure versus compressor inlet temperature can be formed as a baseline as shown in Figure 27.

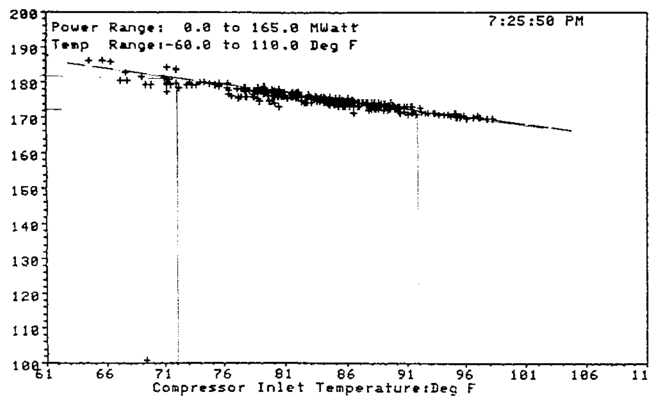

Figure 27. Compressor Discharge Pressure Versus Compressor Inlet Temperature for a 165 MW Gas Turbine.

The most sensitive parameter of the above factors is the mass flow rate.

The real problem is to detect fouling at an appropriate time before a significant power drop has occurred and a fuel penalty cost has been paid. Several philosophies are in use. Some operators believe in periodic washing of the machine while others base washes on condition (i.e., some set of performance parameters). The philosophy utilized is a function of normally expected fouling level, its severity, washing effectiveness, and plant operation criteria. Measurement of air intake depression is a practical and economical method for fixed geometry machines. The technique involves measuring intake depression as an analog of airflow rate. In this approach, the gas turbine inlet bellmouth is utilized as a flowmeter. A typical graph of the intake depression of an aeroderivative engine as a function of the gas generator speed is shown in Figure 28. It is best to create such a baseline plot for each installation because of the differences that exist in the intake geometries for different engines. A decision line where washing is to be triggered may be set at a 3 percent drop in intake depression. This approach has been successfully used by Scott (1979, 1986).

Control of Fouling

Fouling is best controlled by a combination of two methods. The first line of defense is to employ a high quality air filtration system. If fouling occurs (and it usually will) then the compressor can be washed.

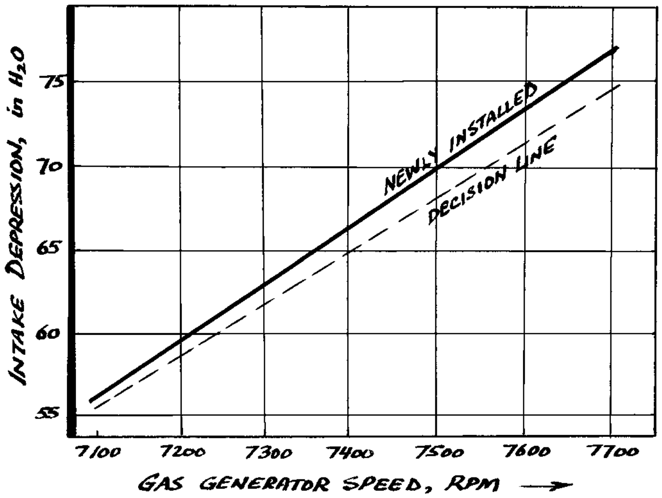

Figure 28. Typical Graph of the Intake Depression as a Function of the Gas Generator Speed for an Aeroderivative Engine.

Filtration

There are several types of filters that can be categorized in the following groups:

- *Inertial filters*—The objective here is to make the air change direction rapidly causing separation of dust particles. These filters are permanently fixed and require minimal maintenance. Inertial filters typically operate at face velocities of 20 ft/sec.
- *Prefilters*—These are medium efficiency filters made of cotton fabric or spun fiberglass. They are relatively inexpensive and serve as "protection" for high efficiency filters.
- *Coalescers*—These are constructed by the use of wire mesh that acts as an agglomerator. The mist in the inlet air is agglomerated and the moisture is thus removed.
- *Louvers and vanes*—These are typically used in the first stages along with coalescer filters to remove water droplets.
- *High efficiency filters*—These filters remove smaller particles of dirt. They are typically barrier or bag type filters.
- *Self-cleaning filters*—These consist of a bank of high efficiency media filters. Air is drawn through the media at a low velocity. At a predetermined pressure drop (about 2 to 3 inches water gauge), a reverse blast of air is used to remove dust buildup.

Goulding, et al. (1990), have provided a detailed treatment of the technical considerations in the selection of gas turbine filters. Stalder and Sire (2001) have provided an excellent study of the migration of salt through filters showing that airborne salt ingestion is a significant issue. Details on offshore platform filtration systems are provided in Kimm and Langlands (1985) and Schweiger (1983). Zaba and Lombardi (1984) have detailed experiences with different kinds of filters.

Air tightness is a must for any gas turbine inlet system as even the most efficient filtration system will be useless if unfiltered airflow leaks in and enters the compressor. Some common causes of leakage are:

- Bypass door leakage.
- Poor gaskets and seals at flanges.
- Modifications made on the inlet ducting. Over the years, personnel may add structures or devices to the inlet system that may cause problems.

Some important considerations in intake filter design are:

- Aerodynamic design should be such as to keep intake velocities uniform across the entire filter area.
- Filter housing should be of a bolted and welded design fabricated of steel no less than 3/16 inch thick and reinforced by steel members. Filter house should withstand 12 inch water gauge pressure. All seams and joints should be air tight. All nuts and bolts used inside the clean air plenum should be welded after assembly to prevent air leaks and foreign object damage to the turbine.
- Design should facilitate change-out of all filters from the upstream side. Filter change should be possible without turbine shutdown. Filter elements should be designed for quick change-out, avoidance of blind assembly and loose retaining nuts, ungasketed washers, etc.
- The filter design should ensure that the inlet air is drawn at least 10 ft above grade level. (In some locations a greater height may be required.)
- A stainless steel trash screen with 1 inch square mesh should be provided in the transition section between the clean air plenum and the compressor intake.
- Avoid the use of gravity weighted bypass doors. Bypass doors are designed to permit emergency airflow to the engine when intake pressure drop rises above a critical value. Bypass doors are typically gravity operated or power operated. The gravity type has earned a reputation for unreliability. Poor sealing, hinge corrosion, and improper operation have made the bypass door a weak link in inlet filter design.
- All filter seal points should be reviewed during the design phase. Poor intake sealing has allowed leaks through bypass doors, access doors, and flanges on the intake filter. In several situations, flange distortion has allowed air ingress. Users should specify types of seals required and call for a filter house integrity test under specified depression to ensure airtightness.
- System design, in the case of pulsed cleaning systems, should be such as to minimize flow distortions and pressure pulsations due to pulse cleaning. More than 5 percent of the total filter elements should not be cleaned simultaneously.
- Filter system pressure drop is an important parameter affecting gas turbine performance. It is important to consider both new filter pressure drop and the pressure drop increase over time.

Filter house construction should also be carefully reviewed. In practice, all kinds of unusual things can occur. The filter house shown in Figure 29 shows that the foundation bolts are missing. A sister unit located a few feet away had the foundation bolts as shown in Figure 30. The absence of foundation bolts can create problems. Further the house sits on the foundation slab and there is a layer of grout that has been painted to seal it. Over time, differential expansion will cause cracking of the grout and a loss of sealing, allowing rain and dirty air to enter the gas turbine.

Figure 29. Filter House Where Foundation Bolts Are Missing.

Figure 30. Sister Unit with Foundation Bolts.

At times, filter systems are preceded by inertial separators that cause the ambient air to swirl, imparting a centrifugal force to the particulate matter forcing it to the outside of the tube. The scavenged air is pulled into the scavenge plenum beneath the tubes and then into the scavenge ducts by the use of blowers. This particulate matter can at times collect on the ducting as reported by Gunsett (2001). As reported by Gunsett, a higher than normal amperage draw of these blower motors should call for the inspection of the plenum area shown in Figure 31. In this particular case, the duct was lined with foam rubber and is held in place with a layer of mesh wire. In the location where the vertical duct enters the horizontal duct (Figure 32), the OEM removed the foam rubber from that area but left the wire mesh, which is a perfect place for buildup of dust. In this case, the buildup was 1 inch as shown in Figure 33. This problem was solved by removing the wire mesh from the bottom of the vertical duct as shown in Figure 34.

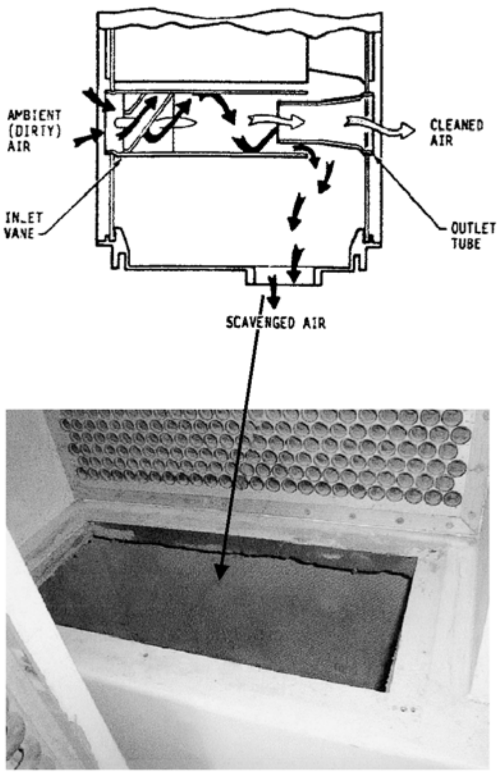

Figure 31. Inertial Separator Sketch and Vertical Duct Connecting to Plenum (Courtesy J. Gunsett, Yuba City Cogeneration)

Figure 32. Location Where the Vertical Duct Enters the Horizontal Duct—Wire Mesh is a Perfect Place for Buildup of Dust. (Courtesy J. Gunsett, Yuba City Cogeneration)

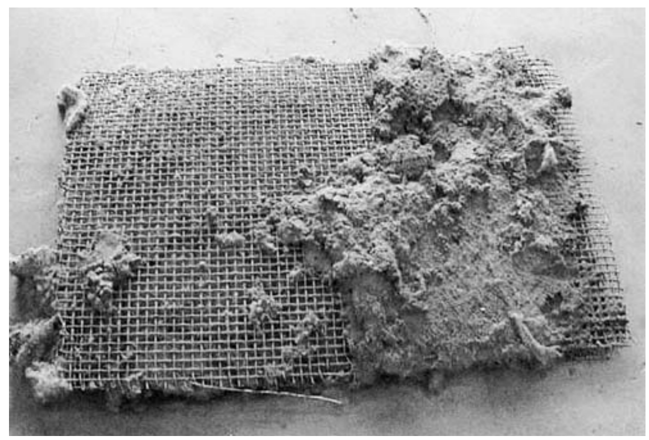

Figure 33. Buildup of Dust to a Thickness of Over One Inch Causing Fan Motors to Trip on Overload. (Courtesy J. Gunsett, Yuba City Cogeneration)

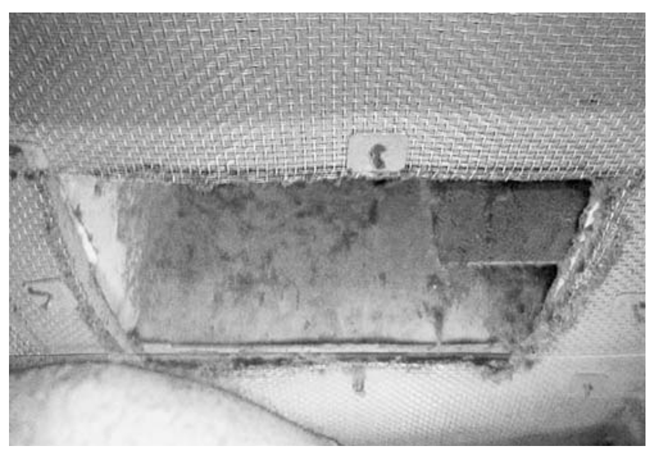

Figure 34. Solution by Removing Wire Mesh. (Courtesy J. Gunsett, Yuba City Cogeneration)

Filter Life

A question often asked by gas turbine operators relates to what the average life of a filter system should be. Unfortunately, there is no simple answer to this as the life is dependent on the local climate, the pollutants/contaminants in the environment, and the intake design itself. Environmental conditions can often be dynamic and change with changes in the surroundings. Very often drilling operations, painting, or the addition of cooling towers or even a major road by the plant can change the life of the filter.

Most commonly the media or gasket life is what limits the life of the filter. Gas cracks and pinhole leaks can be examined by a light test on site, and general filter condition can be observed visually. There is however a strong case to be made to have a sampling of elements being sent back to the filter OEM for closer examination. Filter samples can be analyzed in the laboratory. The lab analysis can yield information regarding:

- Estimate of media efficiency.
- Analysis of the foulants.
- Analysis of media conditions and extent of deterioration.

Filter Storage

Several excellent filter systems have been damaged by improper storage approaches when they are stored outdoors for long lengths of time. It is important to store these indoors for any extended duration to avoid damage due to humidity, mildew, and bacterial growth. If filters have to be stored outdoors, they should not be on the ground where rain can penetrate the boxes, and the boxes should be covered by tarps.

Installation is an important consideration. Prior to installation of the media, the filter system should be thoroughly cleaned, and in the case of pulse type filters, the full compressed air system should be fully tested, purged, and cleaned. Figure 35 shows a blade that experienced coating erosion as the compressed air filter system had not been purged of the grit blast material that was in the manifolds. When the unit started up, the grit was ingested into the gas turbine. Installation should be carefully done, avoiding any action that could cause deformation or tearing of the media.

Figure 35. Axial Compressor Blade with Coating Erosion.

While the above may seem obvious, much gas turbine deterioration is due to the poor storage and installation of filters.

Compressor Washing

This is an area in which strong and, at times, divergent opinions exist. Washing efficacy is so site specific that approaches that work for one site may not be appropriate for another. Controversy is often caused by polarized opinions relating to wash approaches, wash media, and techniques. Some of the highlights are presented below in an attempt to present the overall picture. Operators must determine the best approach for their gas turbines by trial and error in terms of wash technique, use of online washing, what should be used, and the frequencies of wash. This is a complex technical-economical problem also depending on the service that the gas turbines are in. Independent power producer operators and merchant power plants may need to be more aggressive in controlling fouling not having the ability to shut down for crank washes. A useful set of papers relating to compressor washing has been provided in Stalder (1998), Stalder and van Oosten (1994), and Bagshaw (1974).

Two approaches to compressor cleaning are abrasion and solvent cleaning. As abrasive cleaning has diminished in popularity, liquid washing will be focused on.

Water washing (with or without detergents) cleans by water impact and by removing the water soluble salts. It is most important that the manufacturer's recommendations be followed with respect to water wash quality, detergent/water ratio, and other operating procedures. Typically, wheel space temperatures must be below 200°F to avoid thermal shock, and the water wash is done with the machine on crank. On a heavy duty machine, the outage time for a crank wash can easily be between 8 to 10 hours. Water washing using a water-soap mixture is an efficient method of cleaning. This 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. Each rinse cycle involves the acceleration of the machine to approximately 50 percent of the starting speed, after which the machine is allowed to coast to a stop. A soaking period follows during which the soapy water solution may work on dissolving the salt. The method recommended for determining whether or not the foulants have a substantial salt base is to soap wash the turbine and collect the water from all drainage ports available. Dissolved salts in the water can then be analyzed.

Online washing is now very popular as a means to control fouling by avoiding the problem from developing. Techniques and wash systems have now evolved to a point where this can be done effectively and safely. Washing can be accomplished by using petroleum-based solvents, water-based solvents, or by the use of surfactants. The solvents work by dissolving the contaminants while surfactants work by chemically reacting with the foulants. Water-based solvents are effective against salt but fare poorly against oily deposits. Petroleum-based solvents do not effectively remove salty deposits. With solvents, there is a chance of foulants being redeposited in the latter compressor stages.

Details showing the approaches to water washing covering the different injection methods available are provided in Faddegon (1999). A typical water wash cart is shown in Figure 36.

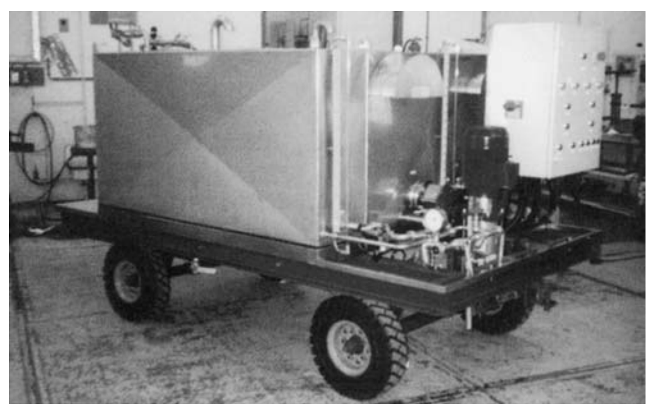

Figure 36. Typical Compressor Wash Cart. (Courtesy F.P. Turbomachinery)

Practical Considerations for Compressor Washing

Some areas to look out for during compressor washing are provided below.

- Ensure that nozzles do not create wakes that could disturb compressor airflow. It is advantageous to locate them in lower velocity areas to ensure effective misting of cleaner. Wash system manufacturers have valuable insight into how and where nozzles should be placed.

- If online washing is used, it is best not to wait till foulants get a chance to build up. The optimal schedule is machine/environment dependent and no firm guidelines can be given. It could range from a daily wash to much longer intervals.

- Examine the spray nozzle design to ensure that there is no chance of it coming loose and creating an FOD incident.
- Tanks, nozzles, and manifolds must be made of stainless steel to reduce corrosion problems.
- If the commercial cleaner being used requires dilution, the quality of the dilution water is important. Strength and wash intervals should be adjusted based on performance improvements.
- Prior to washing a very dirty compressor, attempt to clean the inlet plenum and bellmouth. Silencers have also been known to store lots of dirt.
- If excessive water is found at the inlet, consider increasing the crank speed of the gas turbine to draw more air in.
- Carefully follow manufacturers' requirements relating to drains, valving, protection of piping, and flushing.

Crank Washes Utilizing Borescope Ports

The quest for high efficiency washing has led some operators of aeroderivative units to conduct crank washes by introducing water via compressor borescope ports. This technique has attained some success in keeping sections of the compressor clean that may not be effectively cleaned by traditional crank wash approaches.

Online Interstage Injection of Water

A novel method for the cleaning of multistage compressors has been described by Ingistov (2001) and applied to a Frame 7EA gas turbine. The technique involves injecting water at interstage locations. Several equally spaced injection nozzles were drilled into the casing and the water mist flows into the coaxial spaces formed by the compressor housing inner radius and the compressor hub outer radius. The mist is directed in the region of the stator blades in the plane perpendicular to the rotor centerline. Ingistov (2001) reported that this approach has now been in operation for one year and has allowed significant extension of the time between offline crank washes.

Case Studies in Compressor Fouling

- Case 1—A Single shaft 20,000 hp turbine driving a process centrifugal compressor experienced fouling deterioration. Even with frequent cleaning with an industrial abrasive, rapid performance deterioration was found to occur. The problem was worsened by the fact that during the summer months the process throughput suffered as the turbine was exhaust temperature limited. The underlying problem was found to be oil leakage from the No. 1 compressor bearing.
- Case 2—A Process axial flow compressor located in the Gulf Coast region got fouled creating a drop in the surge margin. The unit was reported to have surged for numerous cycles before corrective action was taken. As a result of the prolonged surge, the casing was severely distorted (due to the high temperatures experienced).

Turbine Section Fouling

While this is not typically a serious problem with natural gas fired gas turbines, contaminants that cause turbine fouling can enter the gas turbine through the inlet air, or the fuel (if liquid), fuel additives, or NO_x control injection fluid.

In the hot turbine section, and in the presence of hot gases, low melting point ashes, metals, and unburned hydrocarbons can be deposited in the form of scale. As hot combustion products pass through the first stage nozzle, they experience a drop in static temperature and some ashes may be deposited on the nozzle blades. As the throat area of the nozzle controls the compressor-turbine match, a reduction in throat area causes a movement away from the design match point. This then causes a loss in performance. Deposits will also form on the rotating blades causing a further loss in performance. Blade and disk cooling can also be impaired by foulants causing a reduction in component life or even failure. As the fuel flow rate is typically about 2 percent of the air mass flow rate, 1 ppm sodium (Na) entering from the fuel would have the same effect as just 20 ppb airborne salt entering the airflow. This is a significant requirement considering that most manufacturers call for not more than 0.5 ppm of Na. Turbine section fouling is accentuated when heavy fuels are used, and inhibitors used to counteract the presence of vanadium cause ash buildup. A photo of a fouled turbine section is shown in Figure 37.

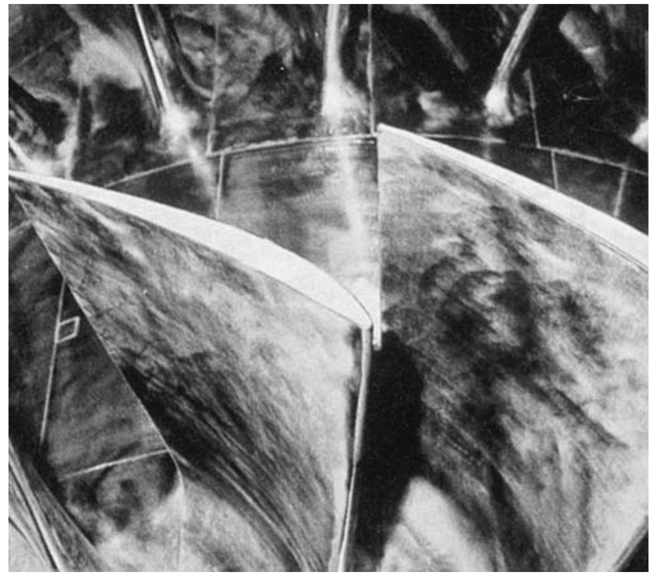

Figure 37. Fouled Turbine Blading.

Even with good filtration, salt can collect in the compressor section. During the collection process of both salt and other foulants, an equilibrium condition is quickly reached, after which reingestion of large particles occurs. This reingestion has to be prevented by the removal of salt from the compressor prior to saturation. The rate at which saturation occurs is highly dependent on filter quality. In general, salts can safely pass through the turbine when gas and metal temperatures are less than 1000°F. Aggressive attacks will occur if temperatures are much higher. During cleaning, the actual instantaneous rates of salt passage are very high together with greatly increased particle size.

NONRECOVERABLE DETERIORATION

Unrecoverable performance deterioration is the residual deterioration that will exist and detract from gas turbine performance even after the gas turbine has been washed and cleaned. Damage that occurs to the flow path due to erosion, corrosion, and increased tip and seal clearances or due to distortion of the casing itself will contribute to this.

During a major overhaul, deteriorated gas turbine components are cleaned or replaced. Airfoils are recoated (compressor and turbine section as needed) and tip and seal clearances restored as close as possible to the initial condition. Even after the overhaul however, there will be some performance loss due to:

- Casing distortion.
- Increased flow path surface roughness.
- Airfoil untwist.
- Increased leakage areas.

Per Diakunchak (1993) the extent of this unrecoverable deterioration is usually less than 1 percent as shown in Figure 38 (Diakunchak, 1993). As is shown in this figure, the extent of the loss will be greater if heavy fuels are being utilized.

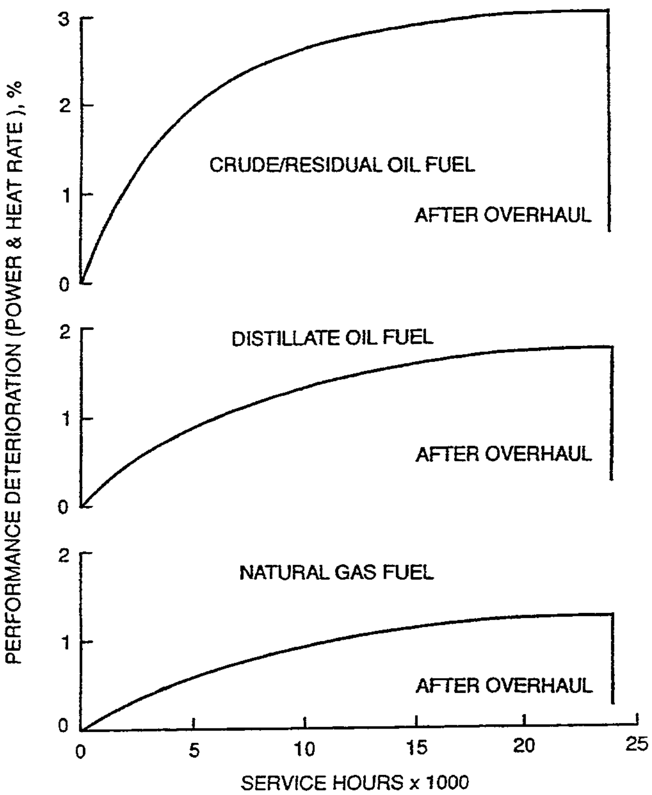

Figure 38. Nonrecoverable Performance Deterioration. (Courtesy ASME)

Erosion

Due to the presence of air filtration systems, erosion is not a major problem with industrial gas turbines, though it has been a problem in areas where the filtration system could not cope with very large dust and sand loading such as exists during a sandstorm.

In aircraft engines however, erosion is a serious problem as abrasive dust may be ingested during low altitude flight and also during takeoff and landing. Because of the high power setting during takeoff, suction vortices often form into the engine inlet and these can suck up gravel, dust, and puddles of water.

Compressor Section Erosion

Erosion is the abrasive removal of blade material by hard particles such as sand and fly ash, usually greater than 5 to 10 microns in diameter. Erosion impairs blade aerodynamic performance and mechanical strength. The initial effect of erosion is an increase in surface roughness and a lowering of compressor efficiency. As it progresses, airfoil contour changes occur at the leading and trailing edges and at the blade tip. Thinning of the trailing edge is detrimental to the fatigue strength and can result in blade failure. A significant loss in tip solidity can promote compressor surge. The typical area of metal loss for a rotor blade is at the tip while for a stator it is near the root. A highly eroded compressor stator blade is shown in Figure 39. Typically, the erosive particles are centrifuged to the outer diameter of the compressor. As a rule of thumb, blade replacement should be considered when loss of cross sectional area exceeds 10 to 15 percent. Tabakoff (1988) has provided details on erosion studies.

Hot Section Erosion

Erosive particles entrained in the air or fuel can cause turbine section erosion. The damage is particularly severe if cooling hole blockage occurs, which can lead to excessively hot blades and

Figure 39. Highly Eroded Compressor Stator Blade.

ultimate creep rupture. A reduction in the blade section size furthercompounds the stress problem. Erosive particles in the fuel can result in nozzle wear resulting in a distorted temperature profile and severe hot spots at the turbine inlet.

Hot Gas Erosion

Apart from *particulate* erosion, there is also the phenomenon of hot gas erosion that results from localized overheating and thermal cycling due to intermittent loss of cooling or a breakdown in the coating. After several cycles, damage takes place and the increased roughness (erosion) worsens the problem. This problem can occur in the first stage nozzle segments at the platforms. Typically the most severely affected parts are those in the hottest gas path (e.g., central to the transition piece).

Corrosion-Compressor Section

Experience has shown that deposits on compressor blades often contain sodium and potassium chlorides. These combine with water to form an aggressive acid causing pitting corrosion of the blades (typically a 12 percent chrome steel such as Type 403 or 410). Water condenses due to the acceleration of the air prior to the IGV and the salt particles get dissolved and pass through the compressor. The water evaporates as it moves through the compressor and, at times, salt is found deposited on compressor blading. In a salty environment, the fatigue strength of steel can drop 50 to 60 percent and this situation is worsened when notches due to corrosion pitting are present. Even with good air filtration, the right conditions of fog, humidity, or rain can cause migration of the salt through the inlet filter (leeching) and into the compressor. Donle, et al. (1993), have described an inlet air treatment approach that limits ingress of salt. Blading on which rust-corrosion deposits has formed is shown in Figure 40.

Haskell (1989) states that corrosion is rarely observed beyond the eighth compressor stage as no moisture will survive at the temperatures present at this point. As shown in Table 4 (Haskell, 1989), SO_2 and HCl even in the ppb ranges can create very acidic conditions on the compressor blading. Even moderate relative humidity conditions of about 50 percent will result in water formation at the compressor inlet due to the intake depression. This situation is acute in heavy industrial areas or where close by industrial cement or chemical plants contribute to the pollution. A photo of a large industrial plant adjacent to a power plant is shown in Figure 41.

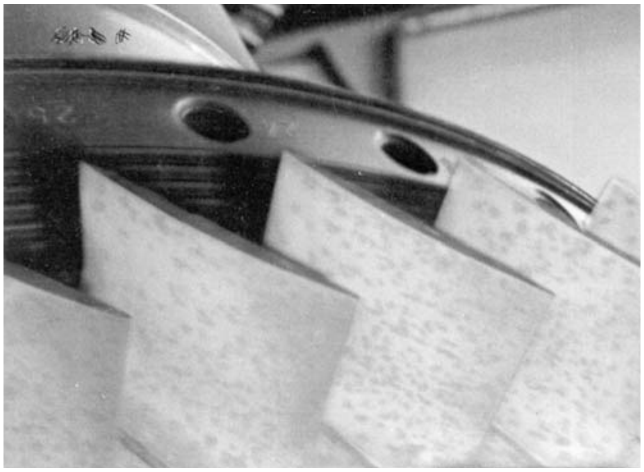

Figure 40. Blading with Rust Corrosion Deposits.

Table 4. Effect of Ambient Pollution on Compressor Inlet pH. (Courtesy ASME)

	Sulfurous acid	
Ambient SO_2 (ppb)	**Dissolved SO_2, ppm**	**pH**
1	0.2	5.5
10	0.64	5.0
100	2.0	4.5
1000	6.4	4.0
10000	19.8	3.5
	Hydrochloric Acid	
Ambient HCl (ppb)	**Dissolved HCl, ppm**	**pH**
1	1600	1.44
10	5500	0.94
100	17600	0.44

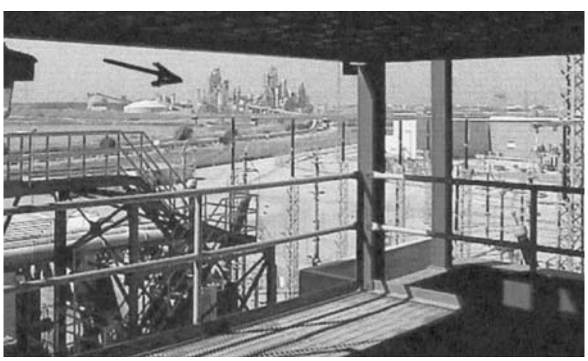

Figure 41. Photo of a Large Industrial Plant Adjacent to a Power Plant Taken from Intake Filter of Gas Turbine.

Standby Corrosion

This is a problem that commonly afflicts peaking gas turbines. It occurs during a turbine shutdown and is the result of air moisture and corrosives being present in the machine. Crevice corrosion occurs when corrosion products that accumulate in the blade attachment areas act as abrasives and increase clearances. In the presence of corrosives possibly from airborne salt, uncoated airfoils frequently develop corrosion pits that may then develop into cracks. Blade fatigue strength is significantly reduced by corrosion. Blade failures caused by crevice corrosion will show symptoms typical of stress corrosion fatigue or stress corrosion.

Foreign Object Damage

While foreign object damage (FOD) is not as serious a problem in industrial gas turbines as it is in aircraft engines, there have been instances where ingestion has occurred. Metallic items such as nuts, washers, etc., can cause both primary and secondary domestic object damage to compressor stages. It is good practice to install a screen prior to the compressor inlet. Some engines require the use of a nylon mesh sock that can be added to the inlet screen. Small ingested items may cause dents or tears that could then be the initiating point of fatigue cracks. Small marks can be removed by filing and polishing following the OEMs guidelines. Nicks are particularly a problem if they occur below the pitchline. With the ingestion of an external object into an axial flow compressor, the first few stages will bear the brunt of the impact. The level of damage tolerance that is built into an engine involves a trade off between aerodynamics and mechanical strength. Axial compressors operate at high tip speeds up to 1500 ft/sec (500 m/sec) with early stages often operating transonic. The sharp transonic blade profiles (much thinner than the rugged NACA 65 series blades) are more susceptible to FOD.

While in most cases, FOD will occur via the compressor inlet, there have been cases of entry of components via variable bleed valves used to unload the compressor during start or under emergency trip situations. It is therefore important that these be closed during maintenance shutdowns.

The ease with which FOD can be detected is a strong function of the *extent* of the FOD. In some cases, a step change in vibration (typically synchronous, i.e., 1× rpm) may be noticeable. In others, a drop in performance may be noted with FOD.

Hot Section Problems

High Temperature Oxidation

High temperature oxidation occurs when nickel-based superalloys are exposed to temperatures greater than 1000°F (538°C). Oxygen in the gas stream reacts with the nickel alloy to form a nickel-oxide layer on the airfoil surface. When subjected to vibration and start/stop thermal cycles during operation, this nickel oxide layer tends to crack and spall. This phenomenon may also occur on the inner surfaces of blade cooling passages and result in blade failure. Coatings are available to mitigate this effect and can be applied both on the blade surface and internal cooling holes.

Sulphidation

Sulphidation is a reaction that occurs when sulphur (from the fuel) reacts with the protective oxide layer and attacks the base metal. The air ingested by a gas turbine can contain impurities such as SO_2, SO_3, sodium chloride (salt), and chlorine. As these impurities pass over the airfoil, droplets (slag) of liquid sodium sulfate (Na_2 SO_4) are formed. Under this slag (and above a threshold temperature that is about 100°C (212°F) higher for cobalt-based superalloys than for nickel-based ones) the protective oxide layer is broken down permitting attack of the parent superalloy and causing very serious damage. Sodium sulfate is highly corrosive causing deep stress riser pits in the airfoil. Sulphidation is of particular concern when it is found in the blade root region or along the leading or trailing edges, or under the blade shroud. Details on sulphidation are presented in DeCrescente (1980).

Hot Corrosion

Hot section parts are often subject to combined oxidation-sulphidation phenomena referred to as hot corrosion. Hsu (1988) discusses the means to avoid hot corrosion problems in gas turbines. Two types of hot corrosion have been identified. Type 1 (high temperature) corrosion occurs at temperatures approximately between 1517 to 1742°F (825 to 950°C). A denuded zone of base metal is often found along with intergrannular attack and sulfide spikes. Type 2 (low temperature) hot corrosion occurs at between 1292 to 1472°F (700 to 800°C) and displays a layered type of corrosion scale. Typically, no intergrannular attack or denuded zone of the base metal is found. Details on Type 2 hot corrosion may be found in Goward (1985). Coatings are commonly used to mitigate hot corrosion problems. Hot corrosion on a turbine blade is shown in Figure 42.

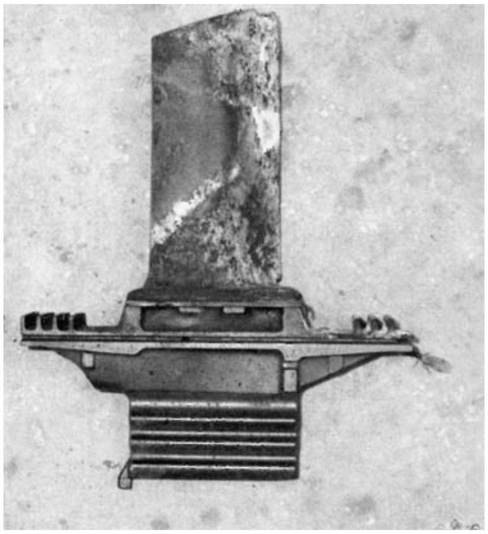

Figure 42. Hot Corrosion on a Turbine Blade.

Reduction in Turbine Nozzle Area Due to Deflection

Due to the high operating temperatures after the combustor, the first stage nozzles are subject to distortion as a result of creep. In most cases, the throat area increases due to this distortion, but there have been some cases where it may decrease. If the gas turbine is operating on heavy fuels, ash deposits can cause blockage and reduce nozzle throat area. If there is nozzle vane distortion causing a reduction in flow area, this will cause a change in the matching in the machine and will cause the operating line to move closer to surge. Simulation result of a 10 percent reduction in turbine nozzle area on the operating line of a single shaft gas turbine is shown in Figure 43 (Dundas, 1986).

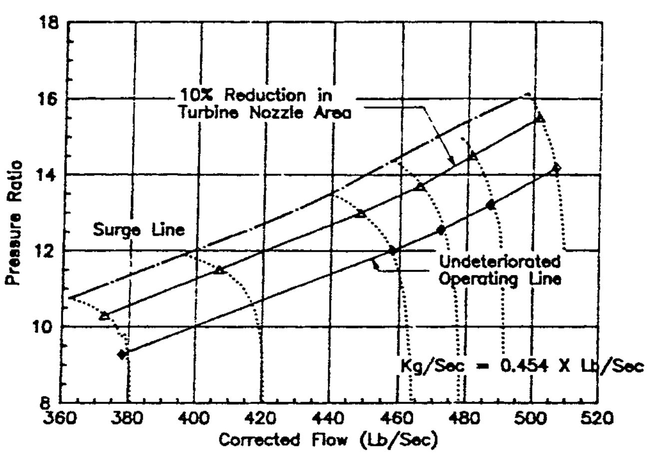

Figure 43. Simulation Effect of a 10 Percent Reduction in Turbine Nozzle Area on the Operating Line of a Single Shaft Gas Turbine. (Courtesy ASME)

Casing Distortion

This is a long-term problem often caused by time and temperature. Frequent and rapid starts and loadings would add to the problem as would sudden transient events such as full load trips. The manifestations would include:

- Eccentricity in clearances and increased leakages.
- Flange and horizontal joint leakages.

Tip Clearance Increases

Increase in blade tip clearance would cause a drop in compressor efficiency and the extent of this loss would depend on several aerodynamic parameters such as the blade loading, blade hub/tip ratios, etc. As a rough rule of thumb, a 1 percent increase in tip clearance would cause approximately a 1 percent decrease in compressor efficiency and a drop in power of more than 1 percent and a drop in engine efficiency of approximately 1 percent (Diakunchak, 1993).

Seal clearance increase or a compressor with a shrouded stator construction will cause a flow recirculation effect and have a similar effect as blade tip clearance. Rubs can cause damage to the close clearances and will contribute to flow and efficiency losses. A casing rub of an axial flow compressor is shown in Figure 44.

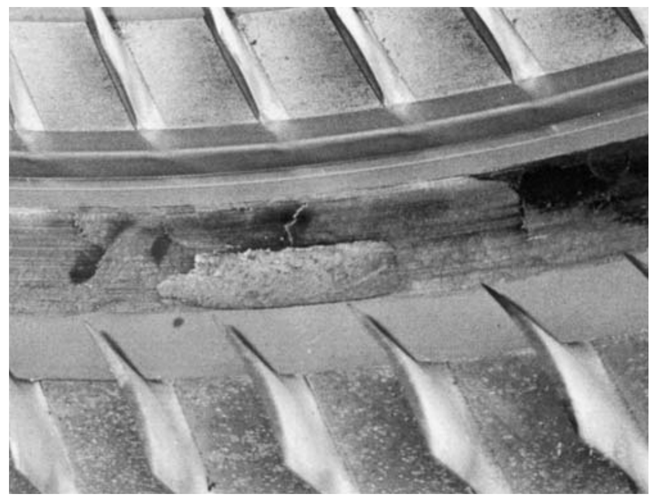

Figure 44. Casing Rub of an Axial Flow Compressor.

Utilization of Brush Seals in Gas Turbine Engines for Reduction of Compressor Leakages in Three Bearing Gas Turbines

While widely used on aircraft gas turbine engines, brush seals are now seeing applications on industrial turbines for minimizing overboard leakages and limiting performance deterioration. Brush seals are designed to come in contact with the rotor to provide a positive seal. The flexibility of the hairlike wires enables the seal to automatically adjust to rotor excursions. In jet engine applications, the use of brush seals can reduce leakages by up to 5 percent when compared to labyrinth seals. The APPENDIX E provides a basic equation to calculate the leakage flow through traditional labyrinth seals.

Several single shaft gas turbines are of the three bearing design where the middle bearing provides additional support to control rotordynamic problems. The third bearing is located between the compressor discharge and turbine inlet locations and is exposed to elevated temperatures as it is essentially surrounded by compressor discharge air as shown in Figure 45 (Ingistov, et al., 2000). Some of the compressor discharge air leaks through the seals into the space around the third bearing housing and into the third bearing. Labyrinth seals are provided to impede air leakage from the pressurized cylindrical space to the bearing cavity. This air then mixes with cooling air (derived from a location on the compressor) and is routed to the atmosphere. The utilization of brush seals as shown in Figure 46 (Ingistov, et al., 2000) has resulted in a reduction of irreversible air losses by 45 percent compared with the traditional labyrinth seals. This represented a savings of 340 kW.

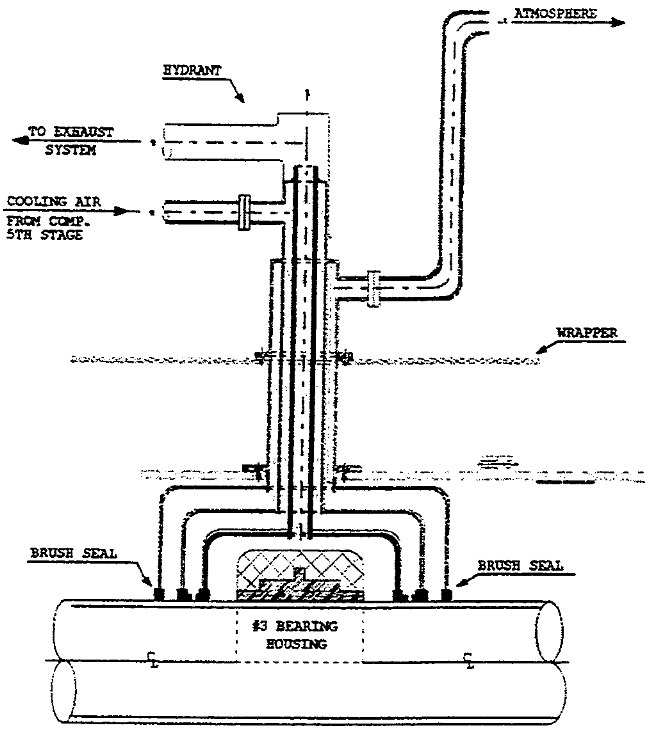

Figure 45. The Third Bearing Located Between the Compressor Discharge and Turbine Inlet Locations. (Courtesy ASME)

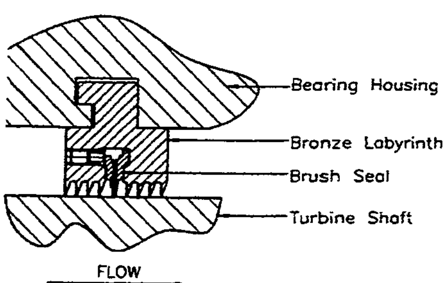

Figure 46. Modified Bronze Labyrinth with Brush Seal at Midpoint Location. (Courtesy ASME)

Deterioration Caused by Fuel System Problems

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 overfiring 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 exceed 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 to 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 80°F (45°C), 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 get 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 5 to 10 minutes.

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

- Transition piece failures (greater cyclic stresses due to liquids being introduced intermittently causing pressure pulsations). At times, seals rupture allowing compressor discharge air to enter at the first stage nozzle. This can cause severe temperature distortion that has a deleterious effect on the blades. Unburned liquid droplets can ignite creating a flame near the stator nozzle.
- Thermal distress of blading (caused due to a distorted temperature profile).
- Compressor surge (due to rapid increase in back pressure).

It is therefore exceedingly important for natural gas fuel to be properly treated *and have a superheat temperature of at least 50°F above the hydrocarbon or moisture dew point*. Details of the design of fuel systems may be found in Wilkes and Dean (1997).

Low Fuel Nozzle Pressure Ratio
and Excessive Pressure Fluctuations

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.

Liquids in natural gas fuel can result in a distorted gas temperature profile and hotter metal oxidation temperatures. Liquids following through the gas side of the fuel nozzle do not follow the gas streamlines, and tend to concentrate along the axial centerline of the combustor. The temperatures are also considerably hotter and combustion occurs farther downstream than with dry gas. The gas does not get cooled with air from the combustion liner cooling holes. As the products of combustion go through the transition piece, the profile distorts and the peak temperature is near the outer radial edge.

Slugs of liquid hydrocarbons in the gas stream can promote compressor surge. This is because the liquid hydrocarbon has a volumetric heating value of 1500 times the heating value of a comparable volume of fuel gas. The slug of liquid hydrocarbon thus results in a rapid temperature rise (additional caloric input) and an accompanying rise in combustion pressure. This transient back pressure increase moves the operating line toward the compressor surge line.

Problems in the fuel system that can cause blockage of the fuel nozzles 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, which will cause the flame to move downstream in the combustor.
- Flow and temperature distortion that can damage the turbine.

Balancing of nozzle flow to better than 4 percent 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 the debris derived from the fuel filters. It is most important to have a means for detecting such problems by an EGT monitoring system that examines qualitative pattern changes and not just absolute limits. A blocked up fuel nozzle is shown in Figure 47. The results of a fire caused by a faulty fuel valve that allowed fuel to catch fire and destroy a turbine as reported by Dundas (1988) is shown in Figure 48.

Figure 47. Blocked Fuel Nozzle.

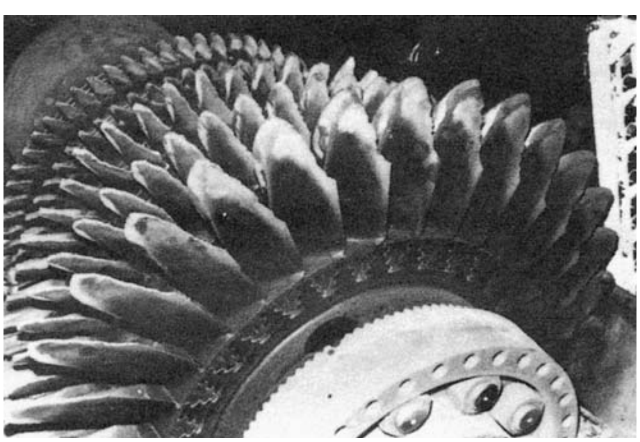

Figure 48. Results of a Fire Caused by a Faulty Fuel Valve That Allowed Fuel to Leak and Catch Fire. (Courtesy ASME)

MECHANICAL DETERIORATION

There are numerous causes of mechanical deterioration including:

- Gas turbine component creep or thermal ratcheting. The creep deformation of a nozzle is shown in Figure 49 (Dundas, 1988). Blade rings can also be distorted due to thermal ratcheting caused by repeated cold starts as shown in Figure 50 (Dundas, 1988).

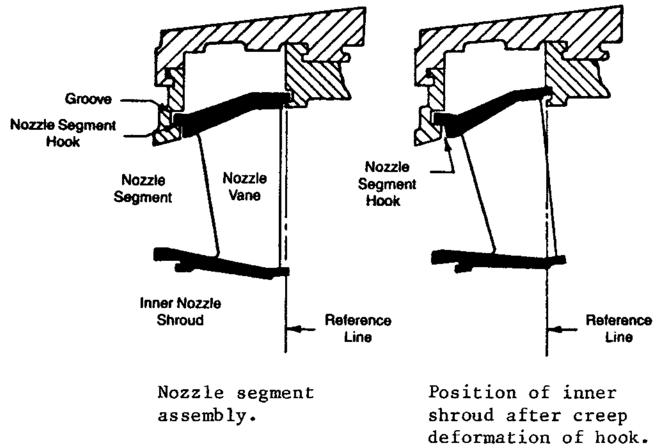

Figure 49. Creep Deformation of a Nozzle. (Courtesy ASME)

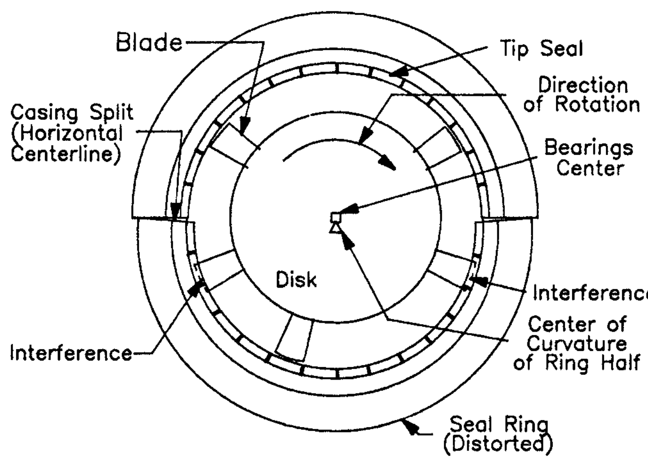

Figure 50. Blade Rings Distorted Due to Thermal Ratcheting Caused by Repeated Cold Starts. (Courtesy ASME)

- 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).

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 51. These were found in both the drive and nondrive end bearings when the gearbox was opened up after an extended run.

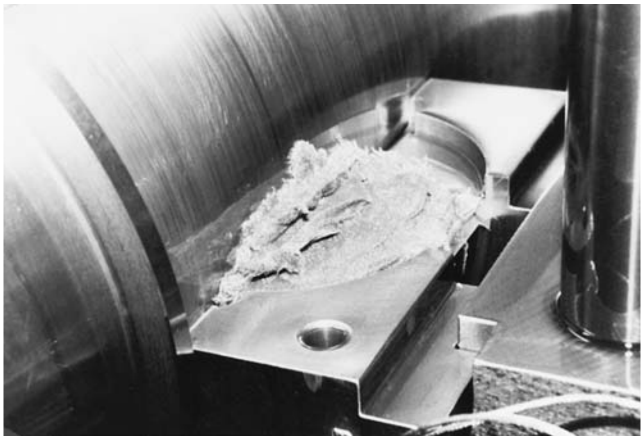

Figure 51. Rags Found in the Oil Galley of a Journal Bearing on a High-Speed Gearbox of a 20 MW Gas Turbine Generator Train after an Extended Run.

PERFORMANCE MONITORING

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 (1982, 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 52 shows the N_1 and N_2 speed relationships for a two-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 are available. The task is much simpler with modern control systems and data collection units that allow rapid assimilation of data.

A number of graphs developed for a 2300°F, 160 MW advanced gas turbine are shown in Figures 53 to 58 (Meher-Homji, et al, 1993). Figure 53 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 EGT is shown in Figure 54. The effect of IGV movement can be clearly seen in this graph. A trend of compressor efficiency is shown in Figure 55.

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 56 and the radial bearing temperature transient behavior during startup is shown in Figure 57.

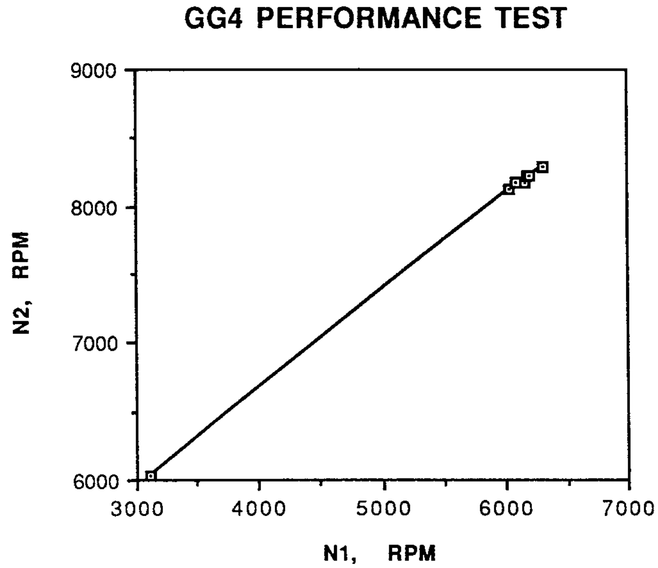

Figure 52. N_1 and N_2 Speed Relationships for a Two-Spool Gas Generator.

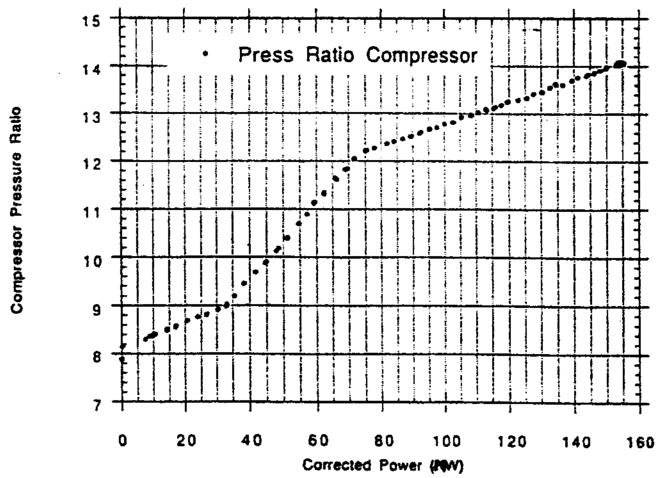

Figure 53. Baseline Plot of Pressure Ratio as a Function of the Corrected Power for a 160 MW Advanced Gas Turbine. (Courtesy ASME)

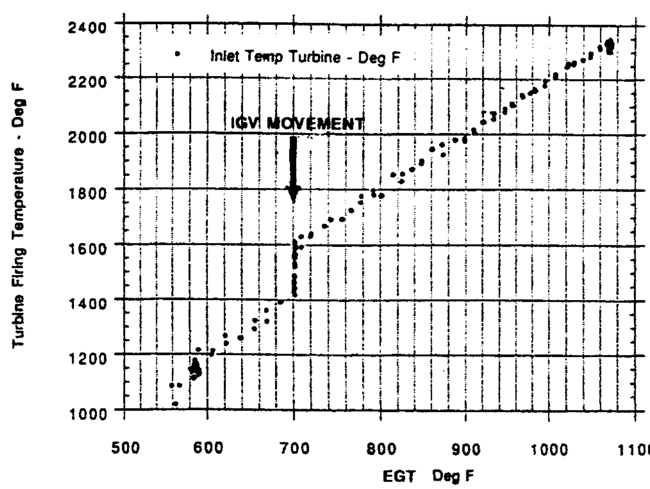

Figure 54. Relationship Between Firing Temperature and Exhaust Gas Temperature. (Courtesy ASME)

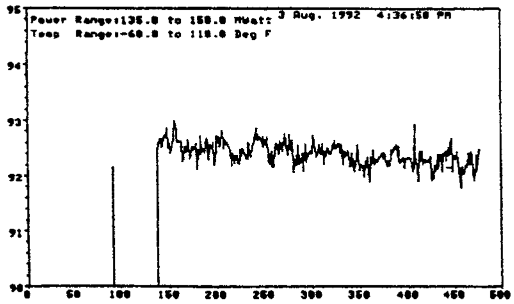

Figure 55. Trend of Compressor Efficiency. (Courtesy ASME)

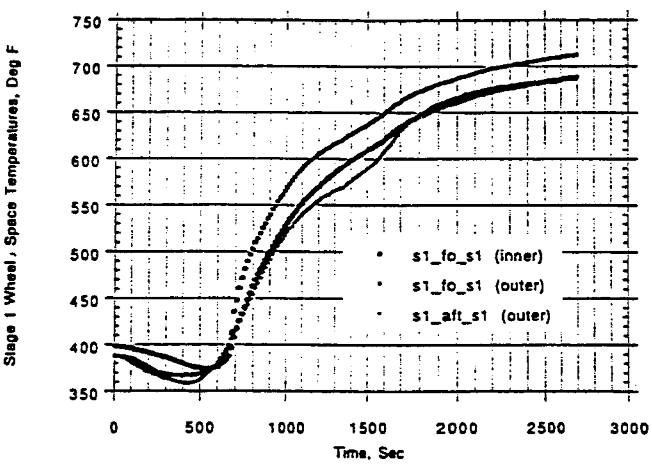

Figure 56. Wheel Space Temperature Behavior During a Start Transient. (Courtesy ASME)

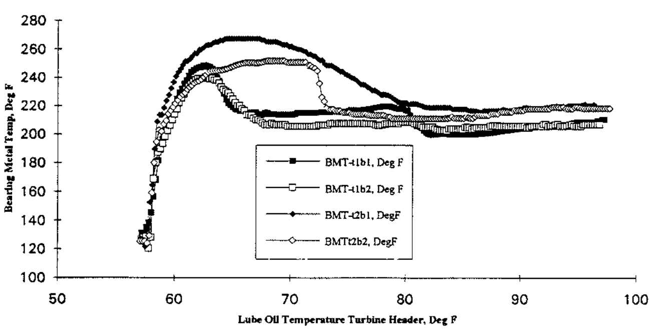

Figure 57. Radial Bearing Temperature Transient Behavior During Startup. (Courtesy ASME)

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 58.

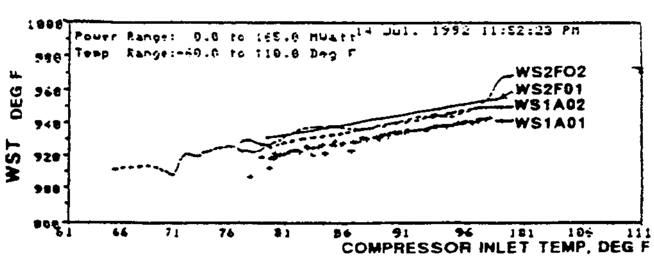

Figure 58. Relationship Between Wheel Space Temperatures and Ambient Temperature. (Courtesy ASME)

Compressor Monitoring Using Scatter Plots

Often data available in the gas turbine control system can be used by means of suitable software to monitor compressor deterioration. The set of graphs taken on a Frame 7EA gas turbine engine (Dusatko, 1995) shows the correlations very clearly. In Figure 59 (Dusatko, 1995), the general relation between compressor discharge pressure (CDP) versus compressor inlet temperature (CIT) can be seen. These data were taken over several hundred hours of operating data at full load conditions. A similar plot of the gross power versus compressor discharge pressure is shown in Figure 60 (Dusatko, 1995). The rule of thumb that can be derived from this figure is that 2 psig of compressor discharge pressure is equal to one gross MW of power for these particular 7EA gas turbine engines. The effect of a crank wash on the CDP versus CIT plot can also be shown in Figure 61 (Dusatko, 1995).

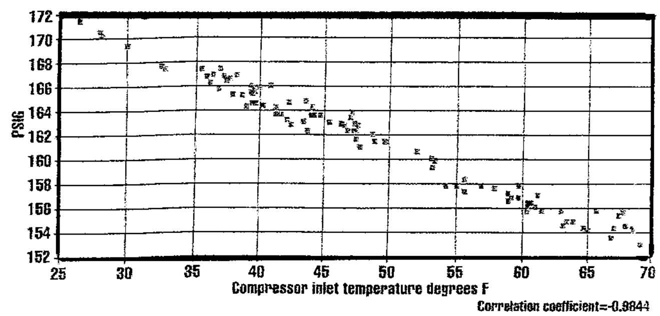

Figure 59. General Relation Between Compressor Discharge Pressure and Compressor Inlet Temperature for Frame 7EA. (Courtesy Power Engineering)

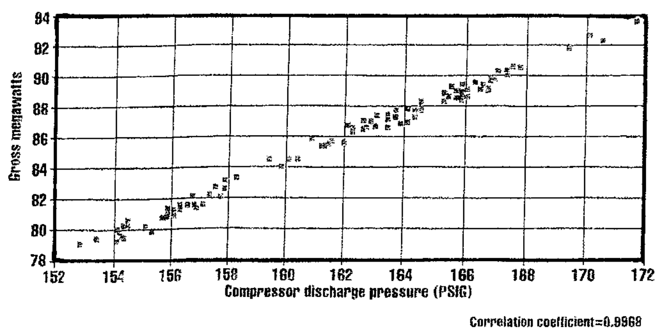

Figure 60. Gross Power Versus Compressor Discharge Pressure. (Courtesy Power Engineering)

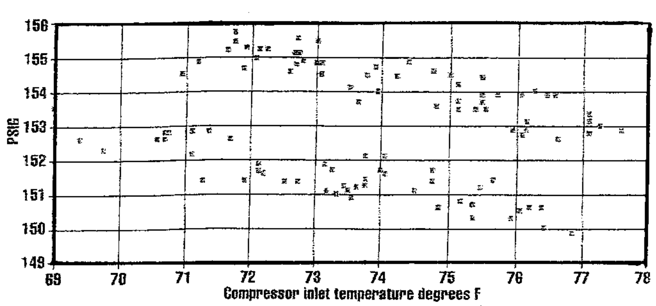

Figure 61. The Effect of a Crank Wash on CDP Versus CIT Plot. (Courtesy Power Engineering)

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

$$Power\ Capacity\ Factor = MW_{Actual}\ /\ MW_{Expected} \tag{5}$$

The MW expected is the corrected day one performance test output and is typically 3 to 5 percent higher than the "guarantee" output. 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.)

Mapping of Compressor Performance to Monitor Deterioration

It is very valuable to monitor compressor performance by creating a compressor map and comparing the performance behavior to the expected characteristics. A typical compressor map is shown in Figure 62 (Dundas, 1982). Generally, the mass flow rate of a gas turbine is not measured but can be derived by computational means (refer to APPENDIX).

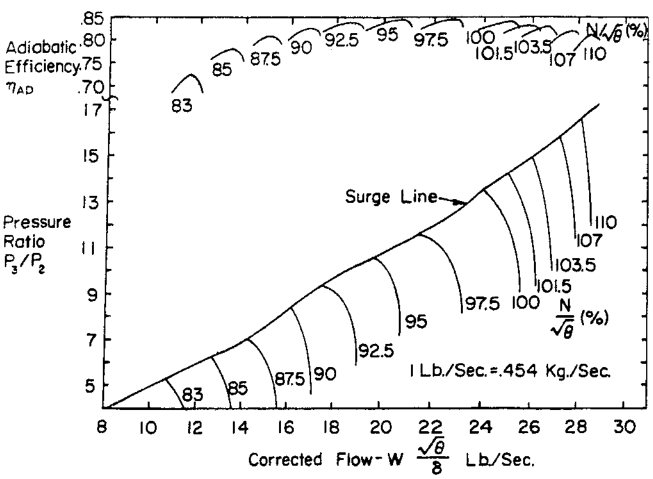

Figure 62. Typical Compressor Map. (Courtesy ASME)

Corrected speed and flow are the best correlating parameters as they allow variations in ambient temperature and pressure drop through an inlet filter to be incorporated in the map. The corrected speed represents the wheel tangential Mach number while the corrected flow represents the through-flow Mach number. The use of the map for performance monitoring is shown in Figure 63 (Dundas, 1982). A significant drop in the characteristic as shown by the dotted lines is indicative of a potential problem.

Exhaust Temperature Monitoring for Hot Section Distress

The importance of EGT monitoring has been covered in Meher-Homji and Gabriles (1998), Knowles (1994), and Wisch (1993) 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.
- Combustor burnthrough or structural damage. Local burnthrough or structural failure and heat distortion will all cause distortion in the EGT pattern.

An exhaust temperature profile plot as combustor basket rupture progressed (note the dramatic change in profile) can be seen in Figure 64 (Dundas, 1992). A situation on a silo combustor (massive damage in the liner) where EGT pattern monitoring was not done is shown in Figure 65.

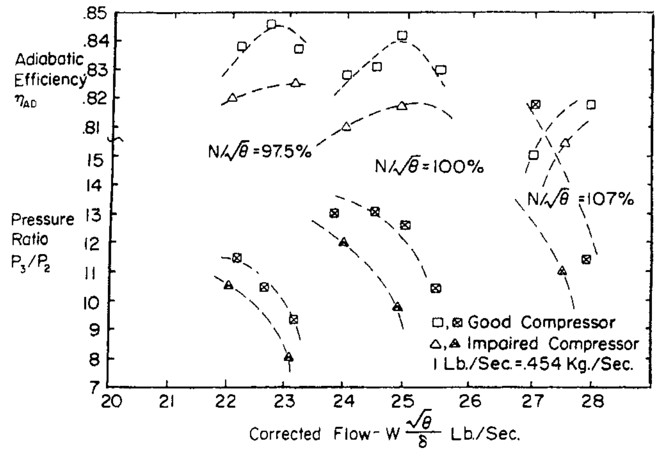

Figure 63. Drop in the Characteristic as Shown by the Dotted Lines Indicative of an Impaired Compressor. (Courtesy ASME)

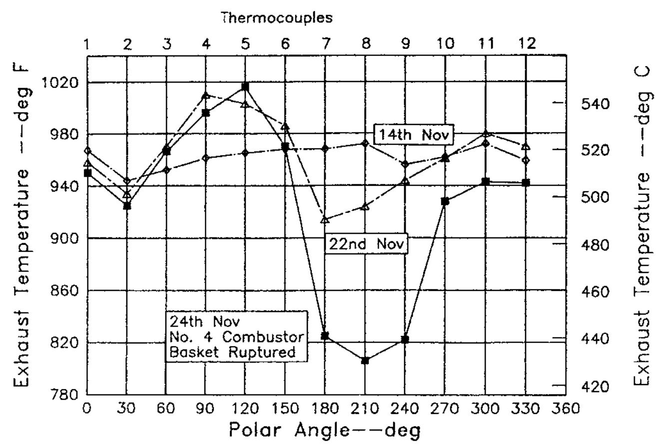

Figure 64. Exhaust Temperature Profile Plot Showing Dramatic Changes as Combustor Basket Rupture Progressed. (Courtesy ASME)

Figure 65. Major Rupture of Liner in a Silo Combustor.

An EGT temperature plot (at the exit of a two-spool gas generator) is shown in Figure 66. The temperature limit is 1380°F and the test data show a consistent profile below this number.

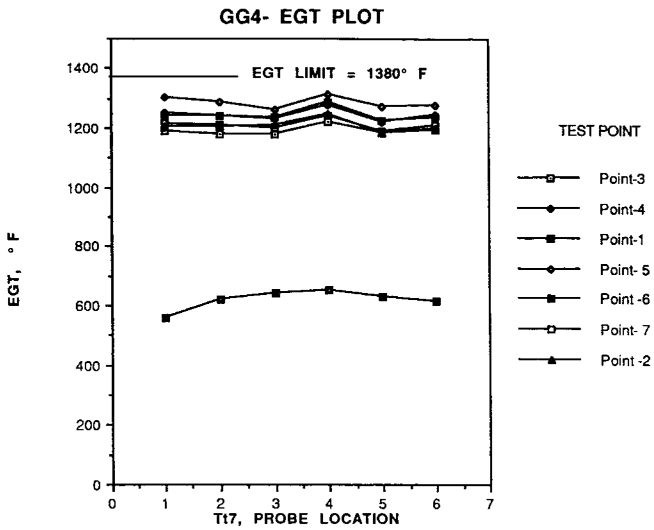

Figure 66. EGT Temperature Plot at the Exit of a Two-Spool Gas Generator Showing a Consistent Profile. The Temperature Limit is 1380°F.

Use of Torque Monitors

Unlike 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 sidestream designs or when gas composition varies. The use of torquemeters 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 liquified natural gas (LNG), ethylene, and ammonia plants.

Torquemeter Operating Principle

While there are several designs on the market, the operating principle of a phase shift model, which the first author is familiar with, is described here. The construction of the torquemeter is shown in Figure 67.

The system shown has been proven to be exceedingly reliable and very accurate and works by measuring the twist of the coupling spacer made with integral teeth. The system has multiple pickups in the form of internally toothed rings. As the shaft rotates, its teeth cut the flux and generate a sinusoidal voltage in each coil. The torque required to twist the shaft through one tooth pitch would correspond to a 100 percent phase displacement. The shaft stiffness constant is determined experimentally by statically torquing the shaft using lever arms and weights in the shop. An electronic readout device converts the speed and phase measurements into torque. Figure 68 shows an installation on a 20 MW gas turbine compressor package.

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.).

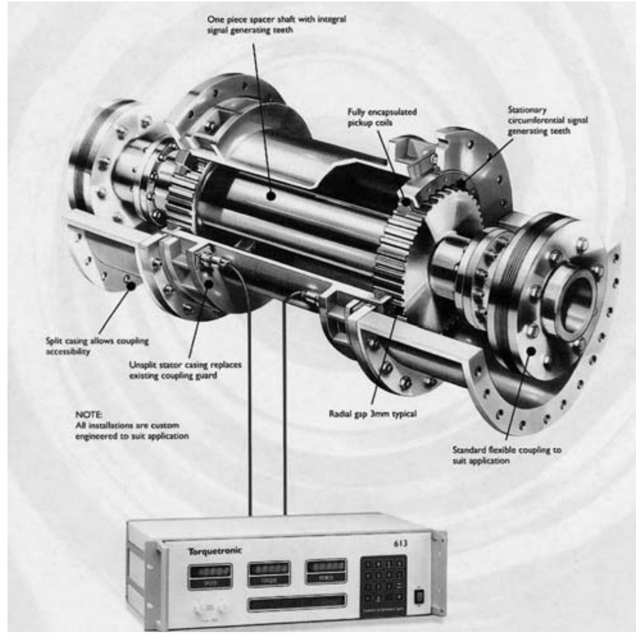

Figure 67. Construction of Torquemeter. (Courtesy Torquetronics Inc.)

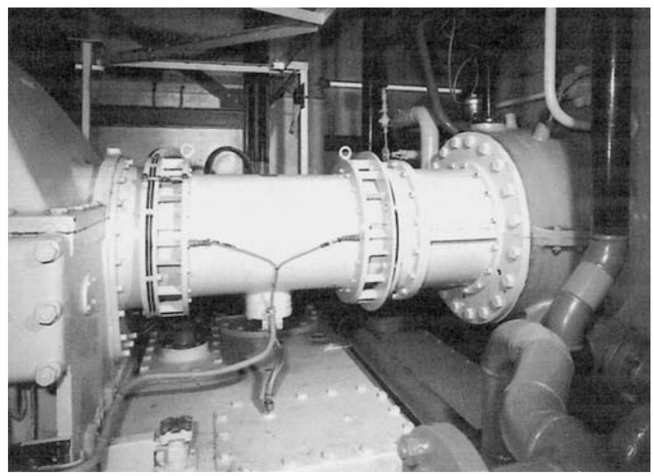

Figure 68. Installation of a Torquemeter on a 20 MW Gas Turbine Compressor Package. (Courtesy Torquetronics Inc.)

- 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:

$$DET_{Total} = DET_{off\ design} + DET_{actual} \tag{6}$$

- 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 is by means of segmenting data in terms of a parameter range 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 exists, showing 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 have 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-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.

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.

Trending Relative Difference

In trending performance parameters, the main objective is to monitor deterioration and estimate losses by observing changes, both abrupt and gradual, in a parameter's value. However, trending a parameter value alone may not be enough to identify deterioration especially in situations where operating conditions vary considerably. To get a proper and accurate measure of change/deterioration, it is necessary to first compute the expected value under current operating conditions and then calculate the deviation from the expected value. This deviation from expected value trended as a percentage (showing relative deviation) will be a better indicator of actual deterioration as the effects of off-design operations will be accounted for.

To compute the expected value of a performance parameter, it is necessary to have a set of performance data/curves recording its variation at different operating conditions. Such curves are typically generated during the performance tests conducted by the manufacturer as part of commissioning. A performance curve for a variable speed gas compressor of polytropic head versus flow is shown in Figure 69. The small "+" sign marks the current operating point. The dotted line shows the curve of expected values at current operating speed. This curve is obtained by doing a 3D interpolation using the family of curves at different operating speeds. The small circle represents the expected value at the current speed and current flow. Thus the relative deviation is computed as:

$$Yr = \frac{Ym - Ye}{Ye} \tag{7}$$

where,

Xm = Measured value of X parameter
Ym = Measured value of Y parameter
Ye = Expected value of Y parameter obtained by 3D interpolation
Yr = The relative difference

A trend of power turbine inlet pressure (actual and relative) for an LM-2500 gas turbine is shown in Figure 70. The relative curve at the bottom indicates that the value of the parameter is between 6 to 8 percent below the expected value. A trend of inlet filter differential pressure and load is shown in Figure 71. The inlet filter differential pressure (which is a function of the air flow rate and consequently power) can be seen to track the power. When examining the relative curve (located at the bottom of the figure), one can see that there is no cause for concern as there is a relatively small deviation from the expected value. A trend of gas turbine compressor discharge temperature (CDT) and load is shown in Figure 72. Again the correlation between the CDT and the load can be easily visualized. The relative value is shown at the bottom of the figure.

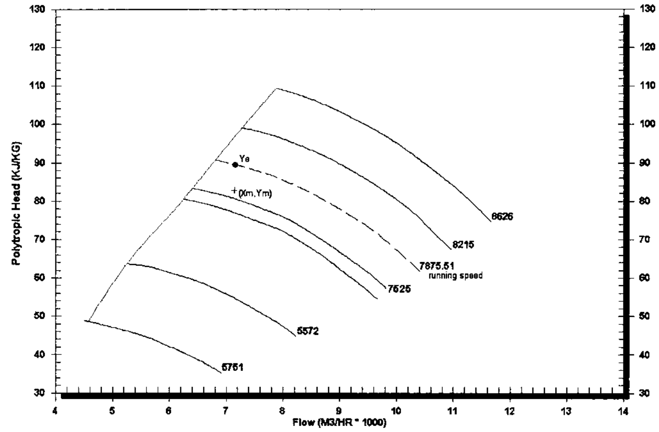

Figure 69. Performance Curves for a Variable Speed Gas Compressor Showing Polytropic Head Versus Flow. (Courtesy Monimax Inc.)

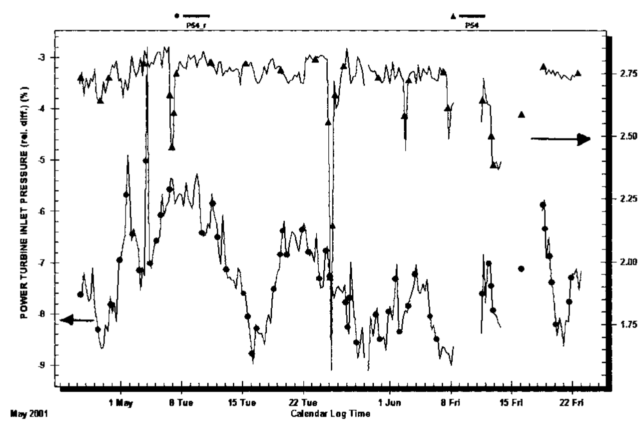

Figure 70. Trend of Power Turbine Inlet Pressure (Actual and Relative) for an LM-2500 Gas Turbine. (Courtesy Monimax Inc.)

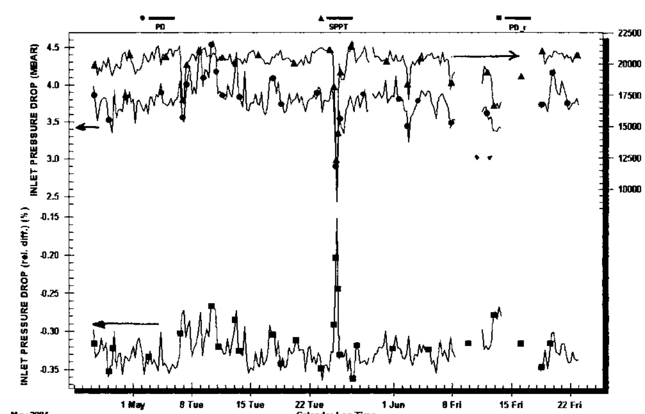

Figure 71. Trend of Inlet Filter Differential Pressure and Load. (Courtesy Monimax Inc.)

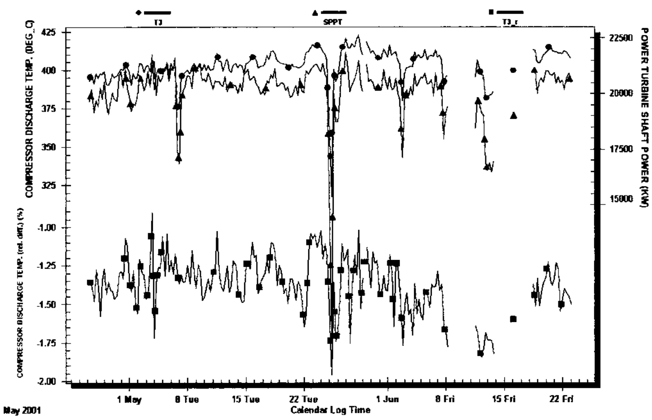

Figure 72. Trend of Gas Turbine Compressor Discharge Temperature and Load. (Courtesy Monimax Inc.)

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 are provided in Meher-Homji and Bhargava (1992).

Definition of Steady-State and Transient Operation

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 coastdown times or plot startup curves using strip chart recorders and trending packages.

A compressor map with lines of constant temperature (T_{03}/T_{01}) is shown in Figure 73. In accelerating a turbine, the fuel injection rate is increased. The increased working fluid temperature raises the turbine work capacity. The increased torque accelerates the shaft. If excessive overfueling is done, the temperature line moves upward before the speed increases, thus pushing the operating point toward the surge line. This is of particular concern with steam injected gas turbine units or cogeneration machines using variable IGV control. These factors, along with severe compressor fouling can significantly (and dangerously) reduce the surge margin.

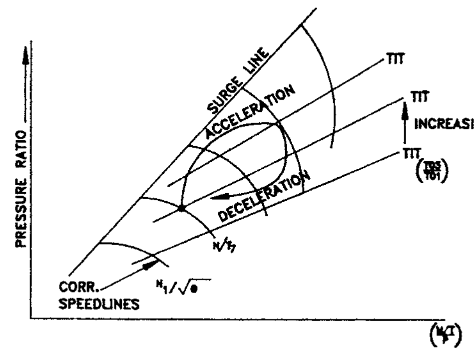

Figure 73. Compressor Map with Lines of Constant Temperature (T_{03}/T_{01}).

Online systems that conduct transient analysis (indeed even steady-state analysis) must have the software capability to report excursions of this nature *even when the parameter does not cross an "alarm" threshold.* What is important here is that a change in pattern or level occurred and this has significant diagnostic content. This is of particular importance when analyzing performance deterioration.

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. Examples of this prepared for a three-spool engine are shown in Figure 74 (Meher-Homji and Bhargava, 1992).

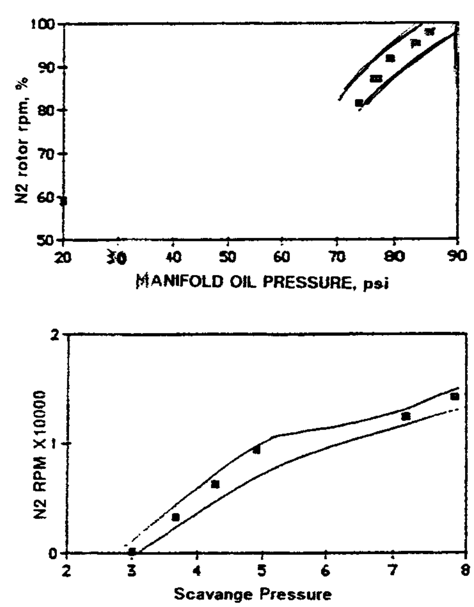

Figure 74. Mechanical Parameters as a Function of Spool Speed for a Three Shaft Gas Turbine. (Courtesy ASME)

• The variable stator vane (VSV) schedule of a 30,000 hp two shaft gas turbine with variable geometry is shown in Figure 75 (Meher-Homji and Bhargava, 1992). By the use of a VSV measurement system, it is possible to plot the actual response on the map.

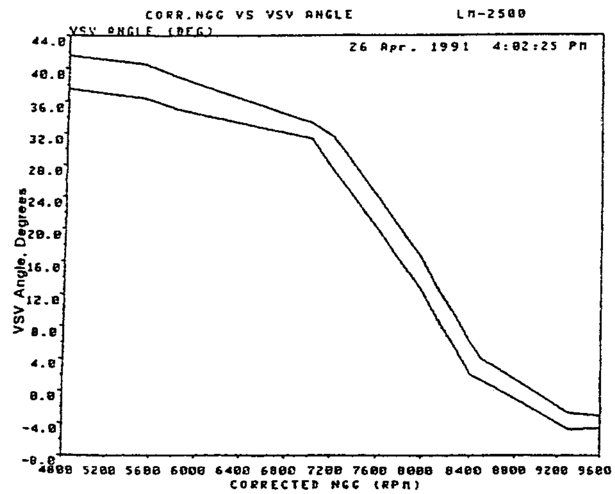

Figure 75. Variable Stator Vane Schedule of a 30,000 HP Two Shaft Gas Turbine with Variable Geometry. (Courtesy ASME).

• 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.

Problems detectable by mechanical transient analysis include:

• *Bearing wear and damage*—Can be detected by evaluating the transient data for changes in amplification factor (Q factor).

• *Thermal bows occurring in the rotor*—Startup data on a 20 MW aeroderivative gas turbine is shown in Figure 76 (Meher-Homji and Bhargava, 1992). The effect of temperature changes (EGT) on the vibration behavior can be clearly seen. The high vibrations die out when some heat soak has occurred.

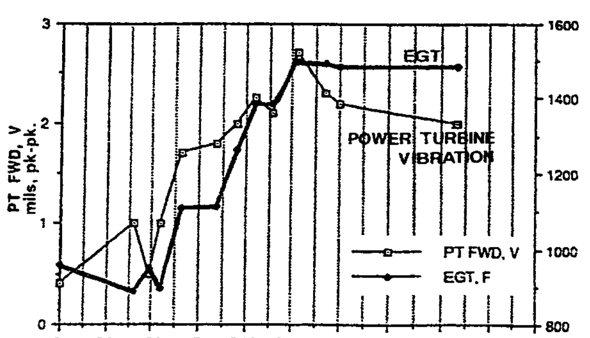

Figure 76. Startup Data on a 20 MW Aeroderivative Gas Turbine Showing EGT and Vibration. (Courtesy ASME)

• *Rotor rubs*—Can be detected by noise detection using a microphone located in the intake duct during coastdown.

• *Alignment changes/locked coupling*—This is best detected by vibration and correlation with axial movements. Online hot alignment systems are now available. Locked couplings can be detected by changes in phase angle that occur from startup to startup.

Diagnostics on an Aeroderivative Gas Turbine Based on Light Off-Transient Behavior

The startup profile of an engine is a valuable health indicator. The pattern shown in Figure 77 (Meher-Homji and Bhargava, 1992) is a valuable picture to trend and compare from startup to startup. It is useful to trend the ΔT shown on the startup curve and examine its behavior.

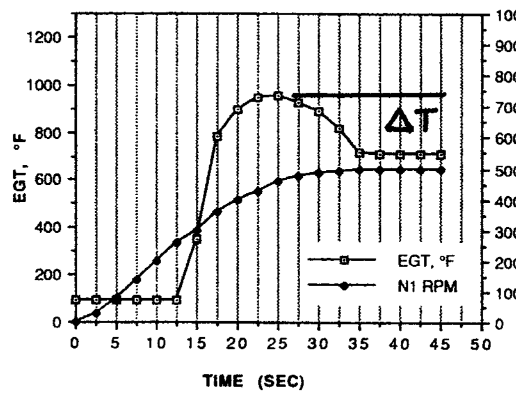

Figure 77. Startup Profile of a Gas Turbine Showing EGT Excursion That Can Be Trended for Successive Starts. (Courtesy ASME)

Transient Behavior of Parameters on a Heavy Duty Gas Turbine

The startup behavior of a large heavy duty gas turbine is shown in Figure 78 (Meher-Homji, et al., 1993). Two startups are indicated and the difference in the profile is significant and can be used for diagnostic purposes. If the EGT is in the normal range, but acceleration is slow, the problems typically reside in the starter assist system. An excessive EGT and higher than normal acceleration rate implies an overfueling condition. Low speed compressor rotating stall or surge can create a situation of low acceleration but higher than normal EGT. Rotating stall can be caused by:

• Degraded or fouled axial compressor.

• Inlet blockage or intake distortion.

• Ignition delays causing pulsations in the combustors.

• Improper operation of the bleed valves.

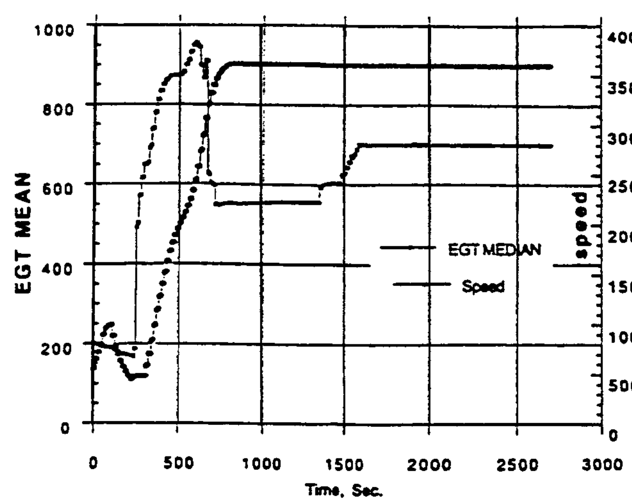

Figure 78. Startup Profile of Heavy Duty Gas Turbine. (Courtesy ASME)

The rundown of a gas turbine is a valuable time to listen for audible noises from gearboxes and bearings. In generator drives, the generator inertia is itself so large that a condition may occur where the generator (inertia) drives the turbine. When reduction gearing is present as is the case with several machines operating at above 50 or 60 Hz, the gearing may mesh on the backside of the teeth. It is during shutdowns that the maximum thermal transients occur.

If it is noted from a plot of transient parameters that the set speed is exceeded after acceleration, then this is indicative of control system or valving problems.

Detection of Damaged Bearings Based on Transient Temperature Monitoring upon Shutdown

Bearing temperature readings can be used to indicate the presence of severely scored bearing journals (i.e., one in which some babbitt wipe has occurred). Due to the fact that the surface has broken down, the journal will ride closer to the bearing and under certain low speed operation will cause the bearing temperature to be more elevated than is expected with an undamaged bearing. During transient operation, the hydrodynamic film is thinner (this is a function of speed) and, at a certain speed, a transition from hydrodynamic to boundary layer lubrication occurs, causing film breakthrough and consequent metal temperature spikes. Figure 79 (Meher-Homji and Bhargava, 1992) shows the nature of such an excursion.

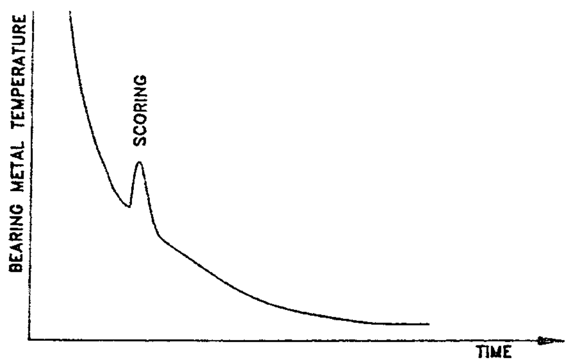

Figure 79. Transient Bearing Temperature Plot Versus Time During Coastdown as an Indicator of Bearing Scoring Distress. (Courtesy ASME)

Speed Decay Transients

It is important that speed decay transients be reviewed (review both the time for decay *and* the profile) to obtain corroborative evidence that damage has occurred. Figure 80 shows a rotor rundown profile as an indicator of bearing condition. It is a good practice to routinely measure the rundown time of rotors and trend the times.

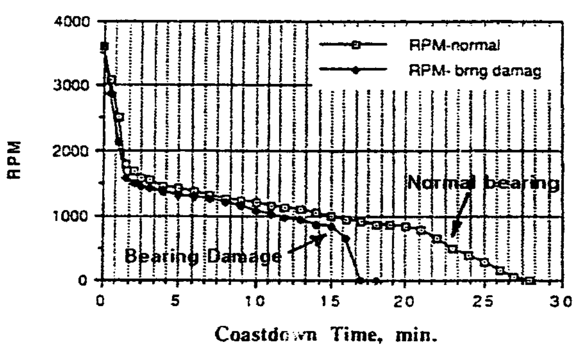

Figure 80. Rotor Coastdown Profile and Indicator of Bearing Health.

HEAT RECOVERY STEAM GENERATOR DETERIORATION MONITORING

As gas turbines are being increasingly used in combined cycles and cogeneration plants, a brief discussion of HRSG performance is made here. It is often important for plant personnel to get a quick feel for the performance of the HRSG without resorting to a full blown ASME PTC type test. Several excellent publications exist in this area including Pasha, et al. (2001). Wambeke and Doore (2001) provide the following simplified approach.

Monitoring of Approach Temperature

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, so does the approach, and the economizer effectiveness decreases. 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.

Monitoring of Pinch Temperature

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.

Monitoring of Stack Temperature

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.

CLOSURE

This paper has provided a comprehensive overview of performance deterioration in gas turbine engines. With increasing fuel costs and a highly competitive environment, the understanding, measurement, and control of performance deterioration is an imperative. The paper has covered the main causes of gas turbine performance deterioration from a practical perspective and has provided several rules of thumb to evaluate losses. As HRSGs are included in most new power plants and cogeneration systems, a brief overview of HRSG monitoring has also been provided. In spite of sophisticated performance monitoring tools that are available, the most important ingredient in controlling performance deterioration is *operator understanding of the gas turbine, its components and performance deterioration mechanisms*. This paper has attempted to meet this need.

APPENDIX A— DESIGN CHECKLIST OF INLET FILTRATION SYSTEMS TO AVOID PROBLEMS

In order to assist gas turbine users in avoiding and troubleshooting fouling problems, a checklist is provided here. The checklist covers a design review and other considerations that would be used when trying to solve a field problem. The checklist is not comprehensive but is intended to provide gas turbine users with some practical pointers.

Axial Compressor Fouling Checklist

General Design Considerations

- Conduct air analysis. Talk to turbine users in the vicinity.
- Consider effects of ambient fog, humidity, unusual conditions such as sandstorms, acid rain, etc.
- Consider wind direction when selecting gas turbine orientation. Are there nearby flares, exhaust ducts, diesel exhausts, hydrocarbon pollutants?
- Review gas turbine blading material and coatings. Small impurities in the air can create very acidic conditions in the first few stages of the compressor.
- Consider possibility of pollutants being generated by maintenance activities including painting, sandblasting, and from operations such as drilling (drilling mud).
- Have operators been trained to recognize the importance of fouling penalty, filter pressure drop, regular inspection of inlet systems?
- Filter change out should be based on inlet delta P and not on appearance.
- If evaporative cooling is used (media type) and water contains Na+K > 40 ppm, has particular care been taken to avoid carryover? Ensure appropriate blowdown rate (approximately 4× evaporation rate). Check if mist eliminator system has been employed. Eliminator efficiency should be 99.9 percent on deposits greater than100 microns.
- Inspect media-type evaporative coolers for media polarity, damaged media—poor alignment, cracks, crushed media, uneven water distribution, clogging of water supply holes, scale deposits on media. Are media retainers that secure media pack to the housing gasketed? Problems such as these can cause carryover and compressor fouling and distress.
- If inlet fogging systems are in use, ensure that there is no galvanized material present and that the duct condition is good.
- Specify SS 316L construction during design phase. If carbon steel is used, use the best quality paint system possible.
- If steam injected cycle, is surge margin adequate with a *deteriorated* compressor?
- If IGV control is used for part-load operation (combined cycle power plant application (CCPP)), is surge margin adequate with a deteriorated compressor?

Air Filter System

- Ensure that the filter system design (type, number of stages, etc.) is based on life-cycle costing and *not* first cost. This is of critical importance as cheap intake systems will result in *major* problems down the road.
- Check for suitability of the proposed system based on inlet conditions, e.g., self-cleaning filters may not be appropriate for high humidity regions.
- Consider air quality testing.
- Have proposed filter systems been checked for face velocities?
- If blow-in doors are specified, consider motor operated doors and review design to ensure airtight behavior.
- Ensure intake velocities are uniform and minimize the number of bends.
- If barrier-type filters are used, has a maintenance and renewal plan been considered?
- Review air filter system for maintainability—ease of access for filter installation, presence of handrails, access ways, etc.
- Ensure that items requiring periodic inspection (solenoid valves, pulse clean controls, etc.) are not placed inside the clean air plenum.
- Have acceptance test procedures been specified?
- Review the quality control plan for the supplier and define the hold points for inspection during construction.
- If severe filter distress exists, consider the use of filter socks or even enlarging the filter elements to reduce the face velocity.

Ducting and Intake System

- Is inlet duct at minimum 10 ft above grade? A greater height may be required depending on the environment.
- Check duct dimensions and construction so that it does not promote vortices.
- Has full system been checked for airtightness? Conduct a thorough inspection of all joints.
- Review online filter change procedures to ensure safety.
- Consider the use of a rain hood unless special coalescer type stages are installed.
- Use SS 316L construction. This single feature can reduce problems and result in huge savings over the life cycle.
- If aluminum construction is used, allow construction check for anodic corrosion problems if dissimilar metals are used.
- If SS wash nozzles are deployed, ensure that proper care is taken of the dissimilar materials if a carbon steel duct is being used.
- Inspect and check that no tools and debris are left in the duct after the construction is complete. While this may seem obvious, it *does* happen.
- Check gasket and seal conditions on all flanged points. Pay particular attention if seals on doors and hatches are attached by adhesives.
- Can filter house withstand 12 inch H_2O gauge pressure? Thickness should be minimum 3/16 inch.
- Are all bolts and nuts welded to prevent FOD? Use special lock washers on all bolting.
- Consider the use of a SS trash screen.
- Ensure that means for visual inspection downstream of the filters exists—use bolted gasketed door for access.
- Pay particular attention to oil ingress from the gas turbine No. 1 bearing. Oil leaks from couplings can also find their way into the compressor.
- Check plenum housing walls for gaps.
- Check for the presence of rain hoods and gutters.
- Instruct operators to look for telltale clues if filter system is not airtight—presence of dust streaks, light "leaks" around grouting at the bottom. Recaulk/weld any suspected problems.

Fouling Detection and Control

- Are operators trained and aware of the indicators that will be used to detect fouling?
- Has intake depression measurement been considered?
- Wash systems should be *all* stainless steel construction—tanks, nozzles, manifolds, etc.
- Review location of online nozzles to avoid wakes.
- Consider hand cleaning of the bellmouth IGV and first stage blades if very highly fouled.
- Define a series of tests to optimize the wash procedures—crank/online and optimal intervals.
- Review wash system spray nozzles and flow rate used—check with wash system supplier as to drop size, spray angles, adjustability of nozzles, etc.
- Work with wash system supplier to optimize chemicals used for wash.

APPENDIX B— PERFORMANCE CORRECTION FACTORS

Performance is usually corrected to ISO conditions (more common) as shown in Table B-1, or NEMA conditions.

ISO 2314 Conditions

Total pressure = 101.3 kPa abs (14.69 psia); total temperature = 15°C (59°F); relative humidity = 60 percent. This implies no inlet and exit pressure drops.

NEMA Conditions

National Electrical Manufacturers Association (NEMA) conditions are *not* currently in use, but some older machines have documentation based on this standard, hence they are listed here. NEMA conditions are: inlet temperature = 80°F, inlet and exhaust pressure = 14.17 psia. This rating essentially relates to an installation at an altitude of 1000 ft, with no inlet or exit pressure drops (Table B-1).

Table B-1. ISO Correction Factors.

Observed	Corrected	Units
Rotor speed	$\frac{N}{\sqrt{\Theta}}$	rpm
HP	$\frac{HP}{\delta\sqrt{\Theta}}$	hp or kW
Temperature	$\frac{T}{\Theta}$	°R or °K
Pressure	$\frac{P}{\delta}$	psia or kPa
Flow rate	$\frac{\dot{m}\sqrt{\Theta}}{\delta}$	lb/sec or kg/sec

where:

$$\theta=\frac{T}{520^\circ R} \qquad \delta=\frac{P}{14.7} \tag{B-1}$$

APPENDIX C— SIMPLIFIED COMPUTATIONAL PROCEDURE FOR A SINGLE SHAFT GAS TURBINE ENGINE

The basic equations ahead provide a simple mathematical model of a single shaft gas turbine engine and are derived from Dundas, et al. (1992). Modern codes can provide a more sophisticated analysis but the beauty of this model is that it provides a physical understanding of the thermodynamics of the gas turbine engine. Figure C-1 provides the schematic of the engine.

Assumptions

1. Turbine runs at design point
2. $\eta_t = 0.9$
3. LHV fuel = 19,300 BTU/lb
4. First stage: Nozzle runs choked
5. Cooling air metering seals run choked
6. Combustor: ΔP = 5 percent
7. 50 percent cooling air chargeable
8. Assumed gas properties (shown in Table C-1)

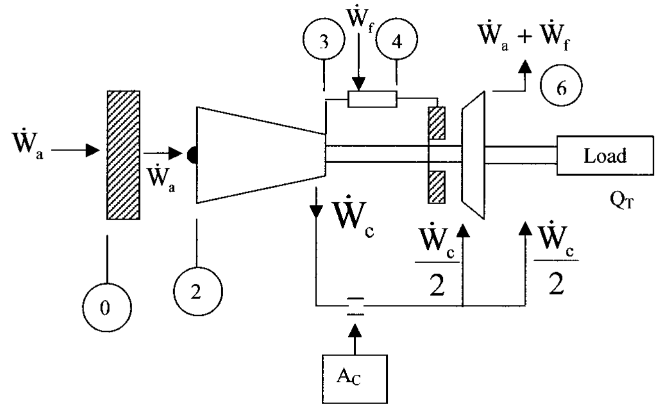

Figure C-1. Single Shaft Gas Turbine Configuration and Stations.

Table C-1. Assumed Gas Properties.

	Cp	γ = Cp/Cv
Station 2	0.24	1.4
Station 3	0.25	1.4
Station 4	0.28	1.33
Station 6	0.25	1.33

Enthalpy Balance Around the Combustor

$$\left(C_{p3}T_3'-C_{p4}T_4'\right)\dot{W}_a-0.5\left(C_{p3}T_3'-C_{p4}T_4'\right)W_c+\left(HV-C_{p4}T_4'\right)W_f=0 \tag{C-1}$$

Equate Compressor Work + Output Work = Turbine Work

$$\left(C_{p4}T_4'-C_{p6}T_6'\right)\left(\dot{W}_a-0.5\dot{W}_c+\dot{W}_f\right)+\left(C_{p2}T_2'-C_{p3}T_3'\right)\dot{W}_a=Q_T\left(\eta\, mech\right) \tag{C-2}$$

Adiabatic Compression

$$T_3'=T_2'\left[\frac{\left(\frac{P_3}{P_2}\right)^{\frac{\gamma_c-1}{\gamma_c}}-1}{\eta_c}+1\right] \tag{C-3}$$

Adiabatic Turbine Expansion

$$\frac{\eta_t-1}{\eta_t}T_4'+\frac{T_6}{\eta_t}=T_6'' \tag{C-4}$$

where T''_6 is given by:

$$T_6''=T_4\left(\frac{P_6}{P_4}\right)^{\frac{\gamma_t-1}{\gamma_t}} \tag{C-5}$$

Choked Flow Through First Stage Turbine Nozzles

$$\frac{\left(\dot{W}_a-\dot{W}_c+\dot{W}_f\right)\sqrt{T_4'}}{P_4}=A_t \tag{C-6}$$

Choked Flow Through Cooling-Air Metering Seal

$$\frac{\dot{W}_c\sqrt{T_3'}}{P_3}=A_c \tag{C-7}$$

Corrected Airflow (lb/sec)

$$W_{corr} = 0.645W_a \frac{\sqrt{\theta}}{\delta}$$

$$EGT = \frac{\left[\left(\dot{W}_a - 0.5\dot{W}_c + \dot{W}_f\right)C_{p6}T_6' + 0.5\dot{W}_c C_{p3}T_3'\right]}{\left[\left(\dot{W}_a - 0.5\dot{W}_c + \dot{W}_f\right)C_{p6} + 0.5\dot{W}_c C_{p3}\right]} - 460 \quad \text{(C-8)}$$

APPENDIX D— COMPUTATIONAL PROCEDURE FOR TWO SHAFT GAS TURBINE

A simplified algorithm for a two shaft gas turbine is provided in Figure D-1. This algorithm allows the computation of the key health parameters—the TIT and the air mass flow rate. The algorithm can be applied for units having typical station instrumentation. While the approach is based on certain assumptions, it can be refined further but is generally good enough for detection of deterioration.

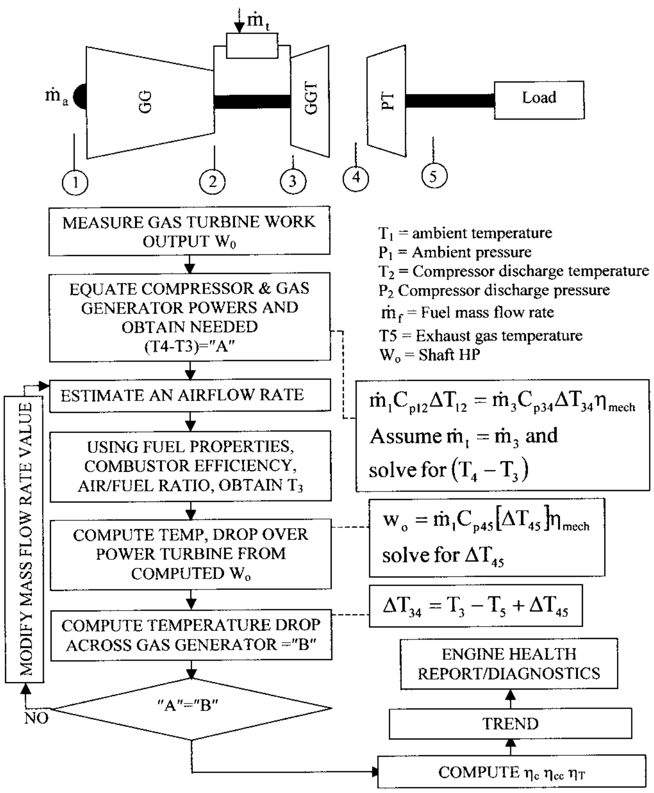

Figure D-1. Computational Procedure for Two Shaft Gas Turbine.

APPENDIX E— AIR LEAKAGE IN LABYRINTH SEALS

Calculation of leakage through labyrinth seals is provided here. This can be used to determine if brush seal retrofits may be beneficial.

Several single shaft gas turbines can benefit from the use of brush seals, especially three bearing gas turbines where the No. 3 bearing is completely surrounded by hot air.

Air leakage flow over a traditional labyrinth seal can be calculated with reasonable accuracy:

$$\dot{W} = 25KA\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^2}{N - \log_e\left(\frac{P_2}{P_1}\right)}}\sqrt{\frac{P_1}{v_1}} \quad \text{(E-1)}$$

where:
$\dot{W}$ = Labyrinth air flow, lb/sec
K = Labyrinth factor (Table E-1)
A = Leakage area, in^2. A is given by:

$$A = \frac{\pi}{4}\left(D_i^2 - D_t^2\right) \approx \pi\delta Di, \left(\delta = radial\ gap\right) \quad \text{(E-2)}$$

N = Number of labyrinth teeth
P_1 = Initial pressure, psia
P_2 = Final pressure, psia
V_1 = Initial air specific volume, ft^3/lb

To find K, use Table E-1.

Table E-1. Determining Labyrinth Factor.

Radial clearance, mils	K
10	75
20	60
30	50
40	45
50	40

APPENDIX F— RULES OF THUMB FOR STEAM TURBINE DETERIORATION

Some rough rules of thumb for typical steam turbines found in cogeneration or CCPP applications are provided here:

- 1 inch Hg increase in condenser backpressure will increase steam turbine heat rate by approximately 200 BTU/kW hr
- 1°F increase in turbine throttle temperature will increase heat rate by 1.2 to 1.3 BTU/kW hr
- If nozzle trailing edges are worn by erosion, then for an impulse type stage a 10 percent change in nozzle throat area will result in a 3 percent loss in stage efficiency.
- If nozzle fouling occurs, the major effect is a reduction in power output. A 10 percent reduction in first stage nozzle area will result in a 3 percent drop in capacity.

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