GAS TURBINE AXIAL COMPRESSOR FOULING AND WASHING

by

Cyrus B. Meher-Homji

Principal Engineer, Turbomachinery Group

LNG Product Development Center

Bechtel Corporation

Houston, Texas

and

Andrew Bromley

Vice President, Operations

Turbotect (USA) Inc.

Houston, Texas

Cyrus B. Meher-Homji is a Principal Engineer, Turbomachinery Group, LNG Product Development Center, with Bechtel Corporation, in Houston, Texas. His 25 years of industry experience encompasses turbomachinery design, engine development, vibration and failure analysis, and the testing of gas turbines and compressors. He has developed several aerothermal and transient analysis techniques for the condition monitoring of gas turbines. In the past he was Chief Engineer of Mee Industries, Gas Turbine Division, and Boyce Engineering International Inc.

Mr. Meher-Homji has a B.S. degree (Mechanical Engineering) from Shivaji University, an M.E. degree from Texas A&M University, and an MBA from the University of Houston. He is a member and past Chair of ASME's International Gas Turbine Institute's Industrial and Cogeneration Committee. Mr. Meher-Homji was named Fellow of the ASME in 1997. He is a registered Professional Engineer in the State of Texas, and has several publications in the area of turbomachinery engineering.

Andrew Bromley is Vice President of Operations for Turbotect (USA) Inc., in Houston, Texas, where he is responsible for Turbotect's U.S. activities in compressor cleaning technology and gas turbine fuel treatment. He is active in supporting the company's ongoing research initiatives relating to compressor wash systems and the development of specialty cleaners, and assists users with unique compressor cleaning applications. Mr. Bromley's 25 years of experience with gas turbines includes the areas of high temperature corrosion, fuel chemistry, fuel treatment, hot gas path and compressor fouling, and other chemical applications associated with gas turbine performance.

Mr. Bromley holds a first degree (Chemistry) and an M.S. degree (Corrosion Science & Engineering) from City of London Polytechnic. He is a chartered professional member of the Royal Society of Chemistry and the Institute of Metallurgy & Materials (UK), and an active member of the ASME/IGTI Division.

ABSTRACT

The privatization of utilities, intense competition in the petrochemical and gas distribution industries, coupled with increasing fuel costs, have created a strong incentive for gas turbine operators to minimize and control performance deterioration. The most significant deterioration problem faced by gas turbine operators is compressor fouling. The effect of compressor fouling is a drop in airflow, pressure ratio, and compressor efficiency, resulting in a "rematching" of the gas turbine and compressor and a drop in power output and thermal efficiency.

This paper provides a comprehensive practical treatment of the causes, effects, and control of fouling. Gas turbine inlet filtration, fouling mechanisms, and compressor washing are covered in detail. The major emphasis will be on compressor washing approaches, technology, and practical aspects. The complexities and challenges of online washing of large output new gas turbines will also be covered. The treatment also applies to axial air compressors used in the hydrocarbon processing industry.

INTRODUCTION

The use of gas turbines in power generation and other industrial applications has grown significantly in the past two decades. The emphasis on environmental protection and the influence of higher fuel prices, market deregulation, and industry privatization also demand reduced emission levels and higher operating efficiencies. 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 detail. An overview of fouling deterioration may be found in Meher-Homji (1990) and an overall treatment of gas turbine performance deterioration including other sources is presented in Meher-Homji, et al. (2001). A detailed treatment of gas turbine degradation is made by Kurz and Brun (2000). Flashberg and Haub (1992) have provided a treatment of nonrecoverable deterioration.

The fouling of axial flow compressors is a serious operating problem and its control is of supreme importance to gas turbine operators especially in the deregulated and highly competitive power market. It is also significant in the mechanical drive market where a loss in gas turbine output directly affects plant throughput. Foulants in the parts per million (ppm) range can cause deposits on blading, resulting in severe performance deterioration. 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 causing a drop in power output and thermal efficiency. In extreme cases, fouling can also result in surge problems, as it tends to move the compressor surge line to the right, i.e., toward the operating line.

Estimates have placed fouling as being responsible for 70 to 85 percent of all gas turbine performance losses accumulated during operation. Output losses between 2 percent (under favorable conditions) and 15 to 20 percent (under adverse conditions) have been experienced.

Gas Turbine Airflow Ingestion and Fouling

Gas turbines ingest extremely large quantities of air, with larger gas turbines having airflow rates as high as 1500 lb/sec (680 kg/sec). A scatter plot of the airflow rate versus power for 67 gas turbines is presented in Figure 1. A very important parameter in evaluating gas turbines is "specific work"—defined as the power per unit of airflow rate (kW/kg/sec). A scatter plot of specific power versus power output for 67 gas turbines is shown in Figure 2. This indicates a general correlation where higher output machines tend to have higher specific work (i.e., they produce more power per unit of airflow). The linkage between airflow rate and compressor fouling is indicated in Table 1. The data were derived by computer software simulations running at an ambient temperature of 59°F and with typical inlet and outlet losses. This table indicates the ingested amount of foulant assuming an ambient loading of 10 ppm for a variety of gas turbines. To help visualize the huge airflow, the volume of air consumed per year is presented in terms of miles above a traditional football field (360 × 160 ft). As an example, the Frame 9351FA gas turbine, (ISO airflow of 1429 lb/sec), would ingest, in a year of operation, a column of air over a football field *1950 miles high*! At a 10 ppm foulant loading rate, 450,650 lb or 225 tons of foulant would be ingested.

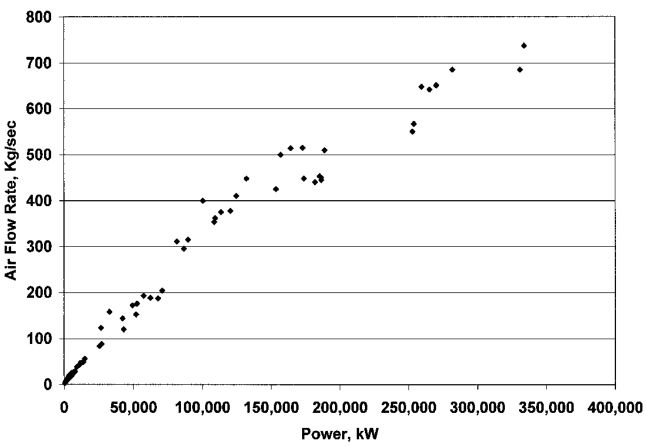

Figure 1. Scatter Plot of 67 Gas Turbines Showing Airflow Rate Versus Power.

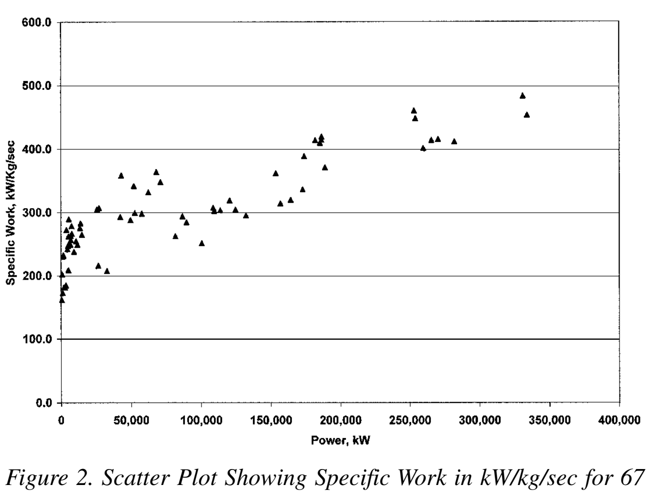

Figure 2. Scatter Plot Showing Specific Work in kW/kg/sec for 67 Gas Turbines.

Table 1. Parameters for 13 Representative Gas Turbines Indicating Amount of Air and Foulant Ingested, and also Calculated Axial Compressor Work (W_c), Turbine Work (W_t), and Compressor Work Ratio (W_c/Wt).

GAS TURBINE	Output kW	ISO air flow rate Lbs/sec	Air ingested/yr miles above a football field	Foulant Ingested/yr at 10 ppm ambient lbs	Compressor Work W_c kW	Total Turbine Work, W_t kW	W_c/W_t
Typhoon	5,106	39	53	12,299	8,104	13,536	0.59
Centaur 50	4,481	41.8	57	13,182	6,263	11,093	0.56
Mars 100	10,463	92	126	29,013	17,938	28,981	0.61
Frame 5371 PA	26,056	269	367	84,832	38,960	66,311	0.59
RB-211	26,678	199	272	62,757	41,222	69,258	0.60
Frame 6581B	41,487	321	438	101,231	50,706	94,229	0.54
Trent 50	50,990	340	464	107,222	84,280	136,478	0.61
GT 8C2	56,295	429	585	135,289	77,627	136,149	0.57
Frame 6101FA	70,116	450	614	141,912	74,237	147,127	0.50
Frame 7121EA	85,206	655	894	206,561	105,264	192,481	0.55
13E2	160,748	1153	1573	363,610	190,196	354,064	0.54
Frame 7241 FA	172,757	971	1325	306,215	164,387	340,671	0.48
Frame 9351 FA	255,400	1429	1950	450,649	229,707	490,325	0.47

In a gas turbine, approximately 50 to 60 percent of the total work produced in the turbine is consumed by its axial compressor. Consequently, maintaining a high compressor efficiency is important for the plant's revenue stream. To quantify this important fact, the last column of Table 1 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 in the model, 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.

Solids or condensing particles in the air and in the combustion gases can precipitate on the rotating and stationary blades causing changes in aerodynamic profile, reducing the compressor mass flow rate and affecting the flow coefficient and efficiency; thus reducing the unit's overall performance. Further, contaminated air can cause a host of problems that include erosion, fouling, corrosion, and, in some cases, plugging of hot section cooling passages. There is also a close co-relation between mechanical reliability and fouling deterioration, and an example is the damaging effects of fouling on blading integrity as discussed in the following sections. This is another important reason to keep the compressor clean. Some typical photos of fouled compressors are shown in Figures 3, 4, 5, and 6.

Figure 3. Fouled Gas Turbine Air Inlet Bellmouth and Blading on a 35 MW Engine Operated in an Industrial Environment.

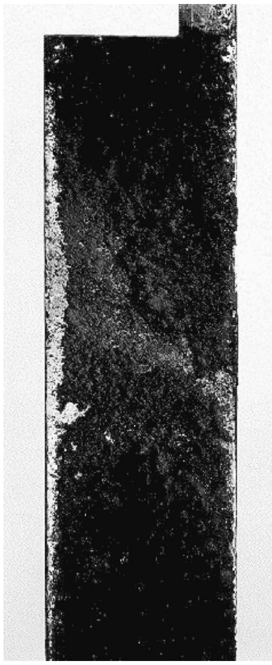

Figure 4. Severe Fouling on a First-Stage Compressor Vane. Deposits are Oily and Carbonaceous.

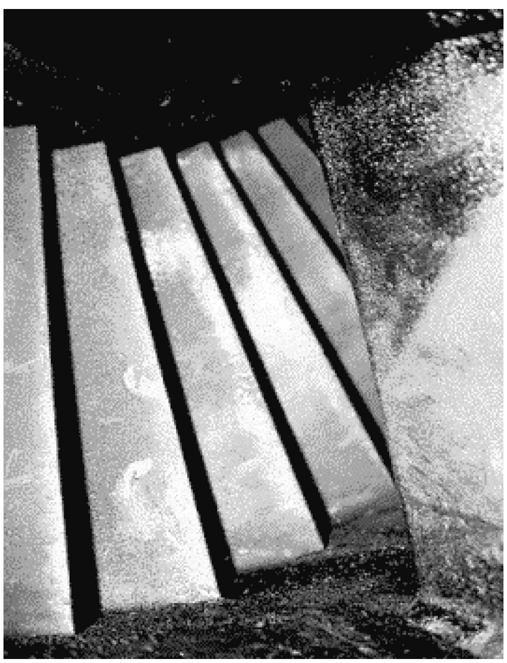

Figure 5. Severe Oily Deposits on Compressor Blading, Caused Mainly by Oil Leaks in the Bearing System.

Fouling and its Impact on Emissions

With the highly competitive nature of the power and hydrocarbon market, performance retention action has to be taken in a scientific manner and has to be tailored to the specific gas turbine and environment under consideration. Emissions are now becoming a major issue in several parts of the world and at times, maintenance activities are driven by emission considerations. Fouling deterioration results in higher turbine temperature for a given power level and this leads to increased hot-end degradation and a possible increase in emissions. If the turbine is temperature limited, then there will be a loss of power.

Figure 6. Fouled Compressor Blading with a Mixture of Salts and Oily Deposits.

The effect of compressor fouling on NO_x emissions for a three-spool gas turbine is shown in Figure 7. This is an often-neglected area with respect to the impact of compressor fouling. In this figure, which is derived from simulation data provided in Singh and Murthy (1989), the increase in NO_x emissions at two ambient temperatures (45°C and 15°C [113°F and 59°F]) is shown for increasing steps of compressor degradation. The degradation steps include drops in mass flow and compressor efficiency as shown on the lower two curves of the graph.

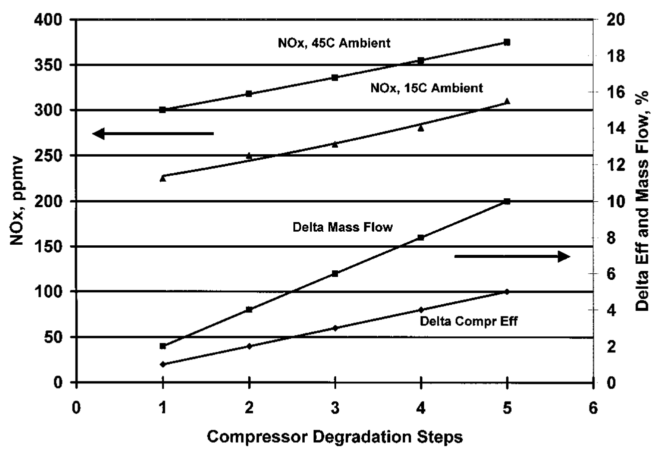

Figure 7. Effect of Compressor Fouling Deterioration on NO_x Level for a Three-Spool Gas Turbine. (Data courtesy Singh and Murthy, 1989)

Economic Impact of Fouling

The cost of performance deterioration can also be estimated as shown in Table 2. This example is based on a realistic model published by Diakunchak (1991), and has been expanded to cover several additional types of gas turbine. The model assumes 8000 hours per year base load operation on natural gas, an extremely modest annual average power loss of only 3 percent, and an equally modest annual average increase in heat rate of 1 percent. Results indicate that the cost of power degradation can range from about \$500,000 for a Frame 5 (26 MW), to over \$5,000,000 per year for a Frame 9FA (255 MW).

Power losses attributable to compressor fouling can be restored through regular compressor cleaning and judicious plant maintenance programs. Overall plant profitability can be significantly

Table 2. Calculated Cost of Power Degradation for Selected Gas Turbines Due to Compressor Fouling, Based on 8000 Hours Base Load Operation.

	Frame 5	W-251B	GT11NM	V94.2	Frame 7FA	Frame 9FA
Power output, MW (ISO)	26.3	49.5	87.9	163.3	171.7	255.6
Heat rate, BTU/kWh (ISO)	11,990	10,450	10,043	9,905	9,360	9,250
Cost of performance degradation, US$ per year	564,191	1,032,303	1,810,239	3,371,025	3,508,119	5,211,428

Note: Assumptions used in the above model are as follows:

- 8,000 hours per year, simple-cycle base load operation on natural gas
- Price of generated electricity US$ 0.07 per kWh
- Cost of natural gas fuel US$ 5.0 per MMBTU
- 3% average annual decrease in power (this is not excessive)
- 1% average annual increase in heat rate

improved for a relatively small cost. The amount of improvement at a given site depends on the type of cleaning program adopted and the thoroughness of its implementation. All elements of the program are important, including the design of compressor cleaning systems such as wash skids and injection nozzles, the choice and use of detergents, the frequency of cleaning, and the actual washing procedure used.

An axial compressor is a machine where the aerodynamic performance of each stage depends on that of 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 often occur when oil and industrial smog or pollen are present and form adherent deposits. The forward compressor stages are usually fouled the worst. If the rear stages foul, this seems to have a smaller impact on performance; but due to higher temperatures, deposits can become baked and difficult to remove. This baking effect is more severe on the high pressure ratio compressors of aeroderivative machines ranging from 18:1 to 35:1 pressure ratio, as opposed to the typical 10:1 or 14:1 pressure ratios found on the heavy duty industrial gas turbines.

Figure 8 shows a fouling process that occurs in large gas turbine engines. This graph shows the changes in compressor efficiency and heat rate over time.

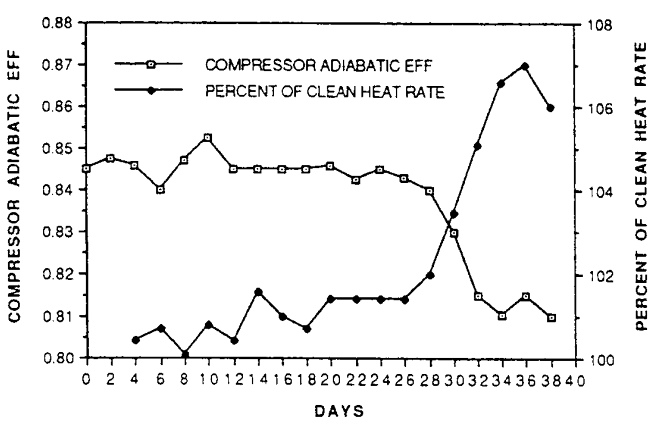

Figure 8. Change in Compressor Efficiency and Heat Rate for a Heavy Duty Gas Turbine.

A typical characteristic curve for an axial flow compressor stage is shown in Figure 9. Under design operating conditions, most stages would operate at design flow coefficient and at a high isentropic efficiency. When the flow coefficient is to the right of the characteristic curve, the stage is lightly loaded and the extreme right point is known as the choke point. To the left of the characteristic curve is a region where aerodynamic stall occurs (surge region).

As fouling drops the mass flow (flow coefficient) in the first stage, this affects the performance of the latter stages as follows: the operating point on the first-stage characteristic moves toward the left, thus increasing the pressure ratio. This causes a higher

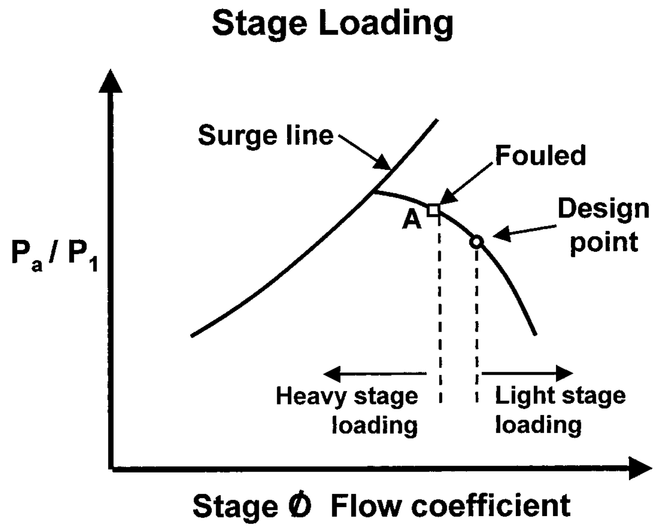

Figure 9. Compressor Stage Characteristics During Fouling.

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 later stage stalls aerodynamically and triggers a surge. Basic velocity triangles indicating how a drop in mass flow causes excessive incidence angles and subsequent aerodynamic stall is shown in Figure 10 (Dundas, 1982).

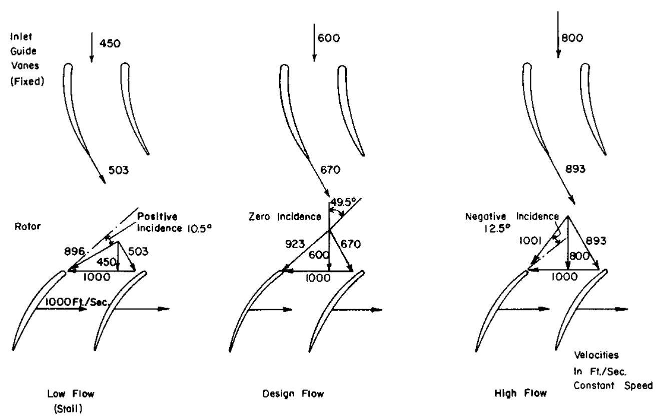

Figure 10. Velocity Triangles under Fouled Conditions. Change in Mass Flow Promotes a Change in Incidence Angle.

Simulation Results of Compressor Deterioration

Simple Cycle Simulation

It is instructive to investigate the effect of compressor fouling deterioration on simple cycle plant performance. To this end, simulation runs using computer software have been made considering a 39.6 MW ISO Frame 6B in simple cycle configuration. Natural gas (21,518 Btu/lb) has been considered and typical inlet and outlet pressure drops (4 and 5 inch water gauge [WG]) for simple cycle were considered. The machine has an ISO pressure ratio of 11.8:1 and a mass flow rate of 304 lb/sec. The firing temperature is 2020°F.

The simulation was run at an ambient temperature of 59°F, and imposing deterioration in the following sequence:

- Step 1: New and clean, mass flow drop = 0 percent, compressor efficiency drop = 0 percent
- Step 2: Mass flow drop = 1 percent, compressor efficiency drop = 0.833 percent
- Step 3: Mass flow drop = 2 percent, compressor efficiency drop = 1.67 percent

- Step 4: Mass flow drop = 3 percent, compressor efficiency drop = 2.5 percent
- Step 5: Mass flow drop = 4 percent, compressor efficiency drop = 3.33 percent
- Step 6: Mass flow drop = 5 percent, compressor efficiency drop = 4.167 percent
- Step 7: Mass flow drop = 6 percent, compressor efficiency drop = 5 percent

The output and heat rate variation with the deterioration steps is shown in Figure 11. The output at the end of the seventh deterioration step has dropped 5.5 MW while the heat rate has increased by 850 Btu/kW hr. The change in mass flow rate, compressor discharge pressure, and compressor discharge temperature corresponding to the simulated deterioration steps is shown in Figure 12. The drop in efficiency causes the discharge temperature to increase by approximately 19°F and the compressor discharge pressure to drop by 10 psia. Figure 13 shows the variation in axial compressor work, turbine section work, and the output after losses. Whereas the axial compressor work is seen to drop slightly due to the reduction in mass flow (middle line of the figure), there is a steep drop in the turbine work (upper line in the figure) resulting in a drop in overall output of the gas turbine of 5.5 MW. The steep drop in turbine section work is due to the drop in mass flow and the reduced expansion ratio available as a result of the loss in compressor discharge pressure.

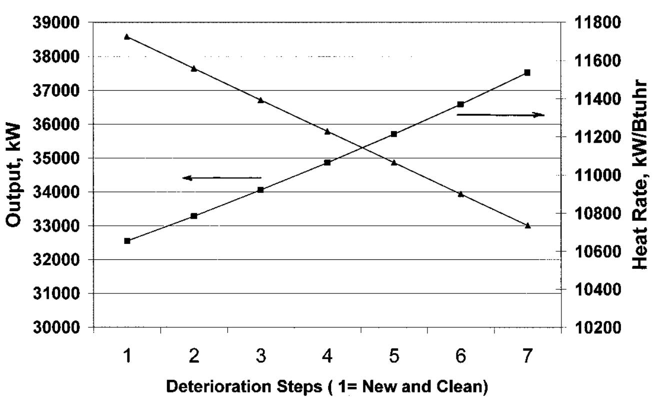

Figure 11. Output and Heat Rate Changes with Compressor Degradation Steps for a Frame 6 Gas Turbine.

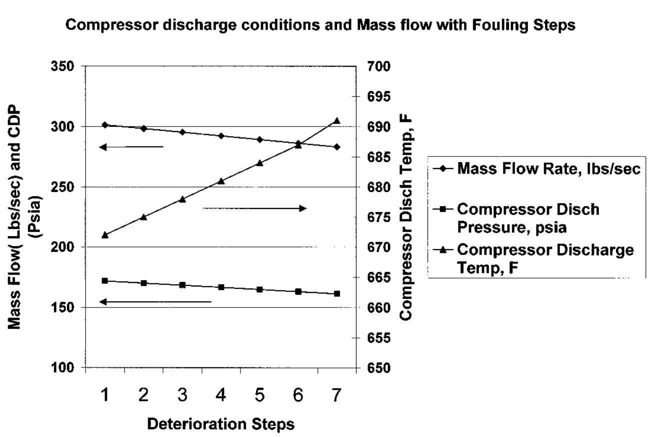

Figure 12. Change in Mass Flow, Compressor Discharge Temperature, and Pressure with Compressor Degradation Steps for a Frame 6 Gas Turbine.

Combined Cycle Simulation

In order to examine the effect of compressor fouling on a combined cycle plant, simulations were conducted on a combined

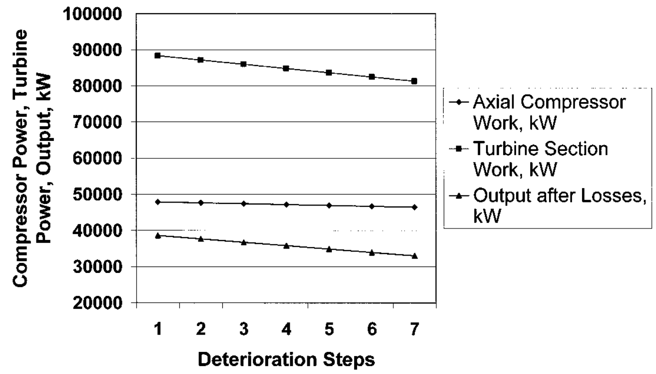

Figure 13. Change in Compressor Work, Turbine Work, and Overall Output with Compressor Degradation Steps for a Frame 6 Gas Turbine.

cycle power plant (CCPP) based on a 7241FA gas turbine operating with a three-pressure level heat recovery steam generator (HRSG) and a reheat condensing steam turbine. The gas turbine is ISO rated at 174 MW and operates at a pressure ratio of 15.5:1 and a firing temperature of 2420°F. The ISO mass flow rate is 988 lb/sec. Typical inlet and outlet losses for a CCPP were considered to be 4 and 10 inches WG.

The same seven deterioration steps were considered as for the simple cycle model above, and the effect of compressor fouling deterioration on CCPP output power and heat rate are indicated in Figures 14 and 15. As can be seen in Figure 14, the drop in gas turbine output and the slight reduction in steam turbine output result in a net power drop of approximately 23.7 MW. As the gas turbine flow drops, the exhaust gas temperature increases, thus resulting in a relatively moderate drop in steam turbine performance of 2.3 MW. The heat rate increase and overall CCPP efficiency drop is depicted in Figure 15, and it can be seen that the drop in the overall CCPP efficiency is almost 1.22 percent.

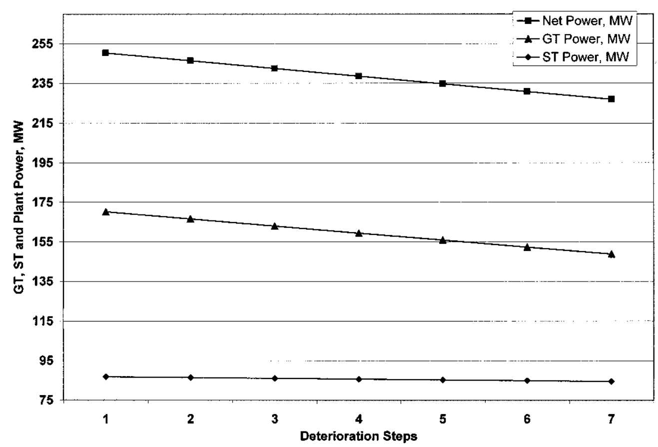

Figure 14. Change in Net Power, Gas Turbine Output, and Steam Turbine Output in a Combined Cycle with Compressor Degradation Steps.

Effect of Turbine Hot Section Fouling

While this paper focuses on compressor fouling it should be noted that fouling of the turbine hot section can also occur. Foulants can enter the combustion zone through the inlet air, via water injected for NO_x control, or as contaminants in the fuel itself. Unburned hydrocarbons can also contribute to hot section fouling.

Fuel contaminants are typically not a problem for natural gas fired units, but liquid fuels often contain trace metals that not only cause ash fouling but also promote serious high temperature corrosion such as sulfidation attack. Trace quantities of sodium, for

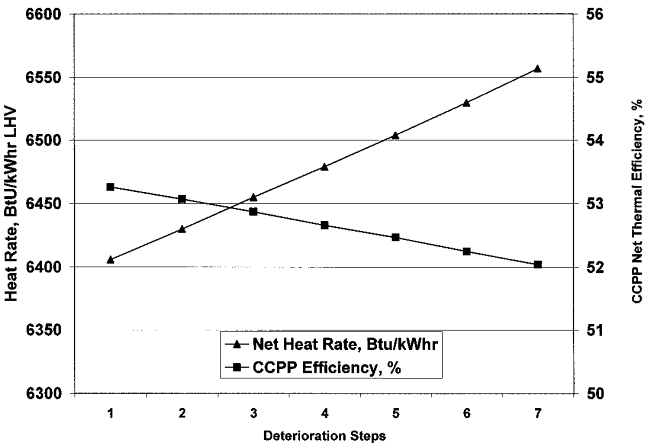

Figure 15. Change in Net Cycle Efficiency and Heat Rate with Compressor Degradation Steps for a CCPP.

example, are commonly found in distillate-grade fuel oils, and may originate from seawater contamination during marine transportation. Other trace metals such as vanadium and nickel are natural components of crude oils and residual-grade oils, and the use of these types of "heavy" fuels always results in significant ash fouling. The associated risk of high temperature corrosion is really a factor of ash composition and ash melting point, and various corrosion mechanisms can occur if ash deposits remain molten on blade surfaces. Thus, the potential for high temperature corrosion is also influenced by gas turbine firing temperature. Chemical additives can be injected to control high temperature corrosion by raising the ash melting point, but additive treatment also increases the quantity of ash generated. Ash deposits can also impair blade and disc cooling, which further increases the risk of corrosion and can lead to premature component failure. A heavily fouled turbine section is shown in Figure 16.

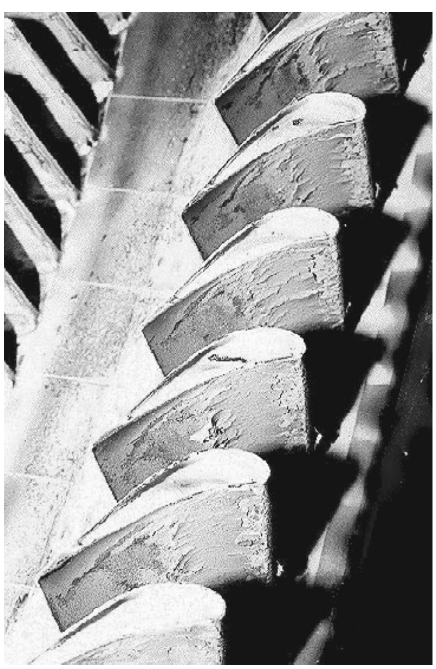

Figure 16. Gas Turbine Operating on Heavy Fuel, Showing Blading Fouled with Ash Deposits.

Especially in coastal locations, airborne sea salt ingested through the compressor inlet can have a serious impact on hot section components—even in the case of gas-fired units. Note also that as the fuel flow rate is typically about 2 percent of the air mass flow rate, 1.0 ppm sodium (Na) entering via the fuel is equivalent to only 0.02 ppm Na entering the airflow as airborne salt. This is a significant requirement considering that most manufacturers call for a maximum of 0.5 ppm Na (or less) in gas turbine fuels.

Even with good air filtration, salt can collect in the compressor section, and will continue to accumulate (together with other foulants) until an equilibrium condition is reached. At this point, large particles will start to break away and are ingested into the combustion section in relatively high concentrations. This ingestion 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 performance. Note also that during compressor cleaning the actual instantaneous rates of salt passage are very high, together with greatly increased particle size.

The effect of hot section fouling is that the nozzle throat area is reduced. As this controls the compressor-turbine match, it causes a movement away from the design match point and results in a corresponding loss of performance. Deposits will also form on the rotating blades causing a further loss in performance.

Also, as the 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 "swallowing" capacity is reduced. Note that in some original equipment manufacturer (OEM) control systems, the compression ratio and exhaust gas temperature are used to determine the turbine inlet temperature (Zaba, 1980). This algorithm is based on an assumption of 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.

UNDERLYING 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.
- Industrial pollution—hydrocarbons, fly ash, smog, exhaust emissions from traffic, 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 (this is especially true of high pressure ratio compressors).
- Ingestion of gas turbine exhaust or lube oil tank vapors.
- Mineral deposits such as limestone, coal dust, and cement dust.
- Airborne materials—soil, dust, sand, chemical fertilizers, insecticides, and plant matter.
- Insects—this can be a serious problem in tropical environments.
- Internal gas turbine oil leaks—leakage from the front bearing of the axial compressor is a common cause. Oil leaks combined with dirt ingestion cause heavy fouling problems.
- Impure water from evaporative coolers (carryover).
- Spray paint that is ingested.
- Vapor plumes from adjacent cooling towers.

Often the inlet struts and inlet guide vanes (IGVs) get severely fouled. Hand cleaning the IGVs and first stage will restore a considerable amount of performance.

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 as follows:

- 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 importance of climatic conditions, rain showers, relative humidity, etc., cannot be overemphasized. Several operators have reported dramatic drops in gas turbine output coincident with rain showers. Often air filters will exhibit a sudden growth in differential pressure as the filters get saturated with water due to high humidity. Under certain conditions, the filter may suddenly unload into the airflow causing rapid compressor fouling.

The susceptibility of different types of gas turbines to compressor fouling (as a function of their design parameters) has been studied by Seddigh and Saravanumuttoo (1990). Aker and Saravanumuttoo (1988) have also provided results pertaining to fouling based on stage stacking techniques. More recent findings by Tarabrin, et al. (1996, 1998) are presented ahead.

Importance of Recognizing Site Specifics for Fouling Control

It is very important to emphasize the importance of site specifics that will impact the extent and severity of fouling and the wash control method. While this may seem axiomatic, there have been several dogmatic positions taken in the gas turbine community regarding compressor washing, based on findings at one site (or with one type of equipment) that are then generalized to global applications. This causes a lot of controversy with respect to topics such as the choice of cleaning fluids, frequency of washes, or the efficacy of online water washing. Several OEM water wash systems in the past were inadequately designed and have resulted in poor performance. This sort of situation often leads to polarized positions within even the same operating companies with respect to choice of cleaners, or use of online washing. Part of the objective of this paper is to provide a comprehensive body of information, pointing out that each application is unique, and an optimal compressor wash strategy has to be worked out.

Marine and Offshore Environment

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), and are 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, and sizes 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, drilling mud, 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.

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 17 (Cleaver, 1988). The National Gas Turbine Establishment (NGTE) "30 knot standard" is often used to define the environment of offshore platforms.

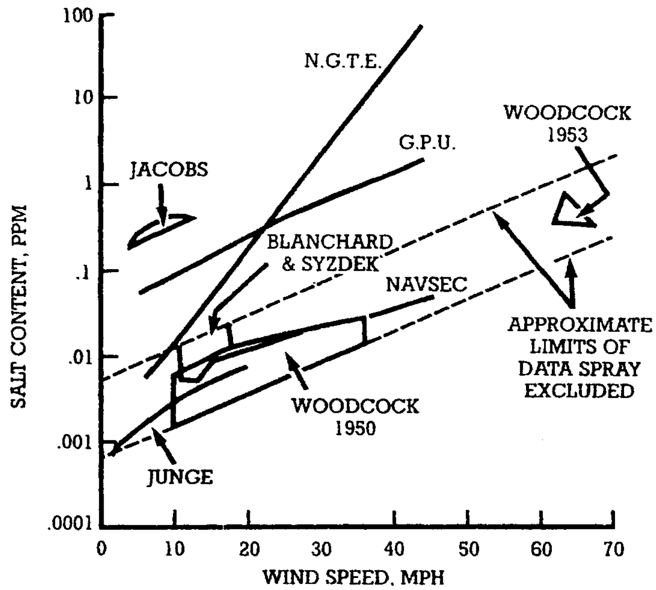

Figure 17. Airborne Salt Concentrations.

FOULING DETERIORATION RATE IN AXIAL FLOW COMPRESSORS

The type of foulants entering the compressor varies widely from site to site, and influences selection of wash detergent as discussed in a later section of this paper. Deposits of oil and grease are commonly found in industrial locations as a result of local emissions from refineries and petrochemical plants. These types of deposits act as "glue" and entrap other materials entering the compressor. Coastal locations usually involve the ingestion of sea salt, desert regions attract dry sand and dust particles, and a variety of fertilizer chemicals may be ingested in agricultural areas.

Compressor foulants are often classified as being "oil soluble," "water soluble," or "water wettable," but experience has shown that they typically are a combination of these types. For example, although sea salt is essentially water soluble, its retention within the compressor may be significantly influenced by trace quantities of oil and grease. In this case, the use of water alone for washing may not be sufficient, and a chemical detergent would be required for effective compressor cleaning. The susceptibility of gas turbine axial flow compressor to fouling is controlled by the following major factors (Stalder, 1998).

- Gas turbine design parameters.
- Location of the plant and the environment.
- Plant design and layout.
- Atmospheric parameters.
- Plant maintenance practices.

Gas Turbine Design Parameters

Tarabrin, et al. (1996, 1998), concluded that the rate of particle deposition on blades increases with a growing angle of attack, and that smaller engines exhibit a higher sensitivity to fouling than larger engines. Further, the sensitivity to fouling increases with increasing compressor stage head ($C_p\Delta T$). Design parameters such as air inlet velocity at the inlet guide vanes, compressor pressure ratio, aerodynamic and geometrical characteristics will determine the inherent sensitivity to fouling for a specific compressor design.

Gas Turbine Sensitivity to Fouling

Tarabin, et al. (1996, 1998), have suggested that the index of axial compressor sensitivity to fouling can be represented by:

$$ISF = \frac{\dot{m} C_p \, \Delta T_{stage}}{\left(1 - r_h^2\right) D_c^3} \times 10^{-6} \qquad (1)$$

where:

ISF = Index of sensitivity to fouling
$\dot{m}$ = Mass flow rate (kg/sec)
C_p = Specific heat (J/kgK)
ΔT_{stage} = Average total temperature rise/stage, °C
r_h = Hub/tip ratio for the first stage
D_c = Tip diameter of axial compressor first stage, m

According to Tarabrin, et al. (1996, 1998), and assuming engines operating under similar environmental conditions and with the same level and quality of air filtration, engines with higher ISF values would exhibit a greater reduction in airflow, pressure, and efficiency than engines with a lower ISF.

This factor is based on the following underlying principles:

• The deposit of particles on the surface of blades takes place under the action of inertia forces acting on the particle and forcing them to move along the curved streamlines. Particles of dirt that collide with the blade can stick to the blade surface. The sticking is increased by the presence of oily hydrocarbons that may have deposited on the blades. The coefficient of cascade entrainment as defined by Tarabrin, et al. (1996, 1998) is:

$$Ec = \left(0.08855 \; Stk - 0.0055\right) \cdot \left(b / t\right) \cdot Sin\left(\Delta B / 2\right) / \; Sin \; B_1 \qquad (2)$$

where:

Stk = Stokes number
B = Blade chord
T = Blade pitch
B/t = Blade solidity
ΔB = Flow turning angle
B_1 = Inlet flow angle

• As the design compressor head (work/stage) increases, the flow turning angle ΔB and the solidity of the cascade typically increases. By using correlations for the solidity and axial flow velocity developed by Howell and Calvert (1978) and as the head per stage is $C_p(\Delta T)$, this results in this term appearing in the numerator of the ISF. The ISF for selected engines as calculated by Tarabrin, et al. (1996, 1998), are provided in Table 3.

Table 3. Index of Sensitivity to Fouling (ISF) for Different Engines. (Courtesy Tarabrin et al., 1996)

Parameters	engine compressors				
	Centaur	LM 2500	GTE-150 LMZ model i=1:4.14	GTE-150 LMZ full scale	V 94.2
engine output, kW	2850	20134	-	150000	150000
air mass flow, kg/s	17.2	65.80	35.47	630.00	500.0
pressure ratio	9:1	17.2:1	12.5:1	12.9:1	10.6:1
number of stages	11	16	14	15	16
hub/tip ratio of the first stage	~0.6	0.480	0.422	0.422	0.520
ΔT^*_{stg} per a stage, K	28.20	25.90	24.60	23.34	19.25
tip diameter of the first stage, m	0.4400	0.7356	0.5712	2.3650	2.1730
nominal rotation speed of a gas generator, rpm	15015	9160	12420	3000	3000
ISF	8.94	5.59	5.72	1.36	1.29

Computational results provided by Tarabrin, et al. (1996, 1998), on a gas turbine operating under ISO conditions and having a mass flow rate of 35.47 kg/sec at a speed of 12,420 rpm and a pressure ratio of 12.5:1, a D_c of 0.5712, and a hub/tip ratio of 0.422 and an ISF of 5.72 are presented in Figure 18. This figure indicates the change in adiabatic efficiency and pressure ratio as successive stages foul for the first six stages. As can be seen, after the sixth progressive stage of fouling, there is a 4.5 percent reduction in mass flow, a 4 percent reduction in pressure ratio, and a 2 percent drop in compressor efficiency.

It is interesting to note that Seddigh and Saravanamuttoo (1990) proposed a fouling factor defined as:

$$Fouling \; Index = kW / \dot{m} C_p \Delta T \qquad (3)$$

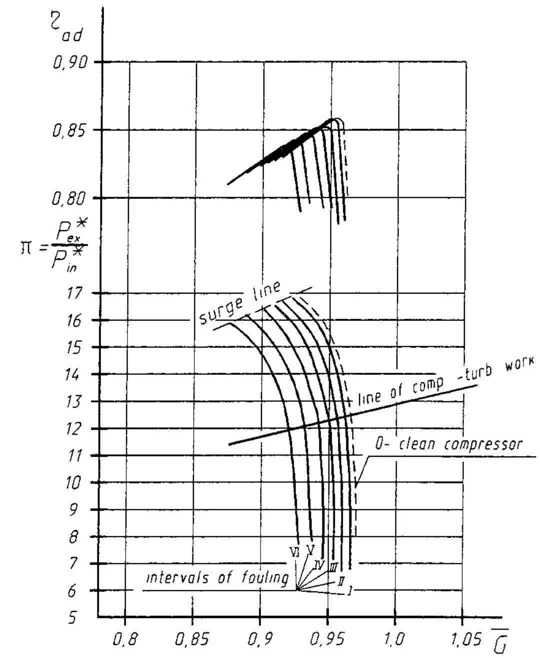

Figure 18. Modifications in Compressor Adiabatic Efficiency and Pressure Ratio with Simulated Fouling Stages. (Courtesy Tarabrin, et al., 1996)

This represents a nondimensional coefficient of the ratio of the specific power output and the enthalpy rise for a stage. The results are not fully coincident with that of Tarabrin, et al. (1996, 1998), however, and in general more work needs to be done to validate these approaches.

Fouling Degradation Rate

Experience has shown that fouling tends to occur during initial operation and roughly follows an exponential law, stabilizing after 1000 to 2000 hours. An empirical formula proposed by Tarabrin, et al. (1996), is:

$$\Delta Power = a\left[1 - e^{-bt}\right] \qquad (4)$$

where:

a = 0.07
b = 0.005

The influence of fouling on changes in compressor efficiency ($\delta\eta$), mass flow rate (δG), and pressure ratio ($\Delta\pi$) as a function of time (Tarabrin, et al., 1988) is shown in Figure 19.

Location of Fouling in an Axial Compressor

Experimental studies conducted by Tarabrin, et al. (1998), have indicated that the first five to six stages of the compressor tend to foul, and the degree of fouling tends to decrease from the front to the back end of the compressor. Deposits on the blades measured from a 16 stage Frame 5322 gas turbine are quantified in Figure 20 for the rotor blades and in Figure 21 for the stator blades. These figures provide the weight distribution on the convex and concave sides of the rotor and stator blades for the IGV and up to stage six. An insignificant amount of deposits were noted on the seventh stage onward.

Relationship Between Airflow Loss Due to Fouling, Pressure Ratio, and Gas Turbine Thermal Efficiency

For single shaft machines, the percent change in mass flow rate due to compressor fouling and pressure ratio are approximately equal. The relationship between compressor efficiency and mass

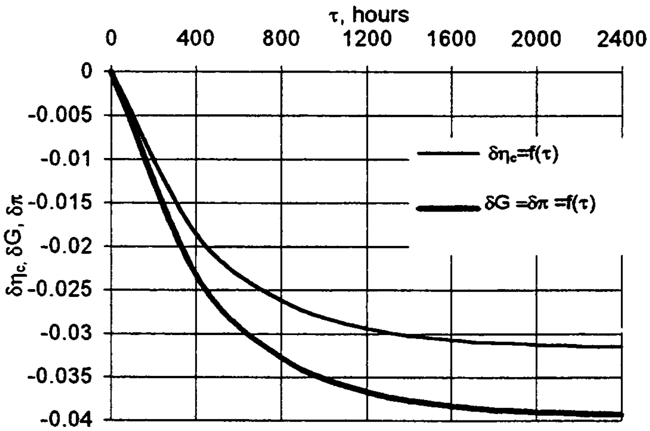

Figure 19. Influence of Fouling on Efficiency, Mass Flow, and Pressure Ratio of Axial Flow Compressor as a Function of Time. (Courtesy Tarabrin, et al., 1998)

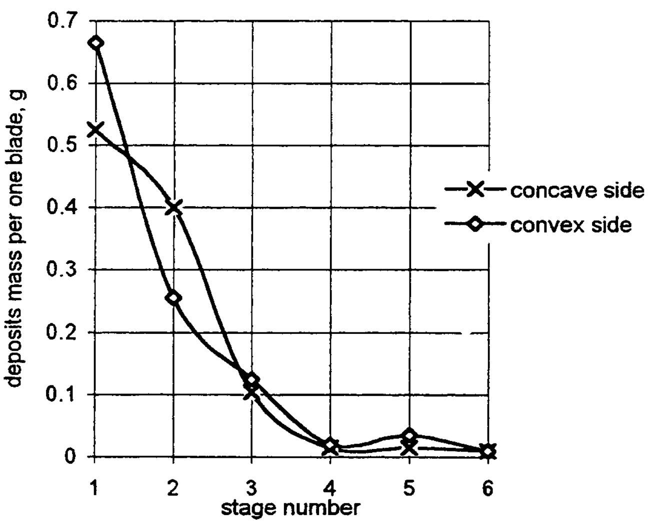

Figure 20. Measured Distribution of Deposits on Axial Compressor Rotor Blades on a Two-Shaft Frame 5 Gas Turbine. (Courtesy Tarabrin, et al., 1998)

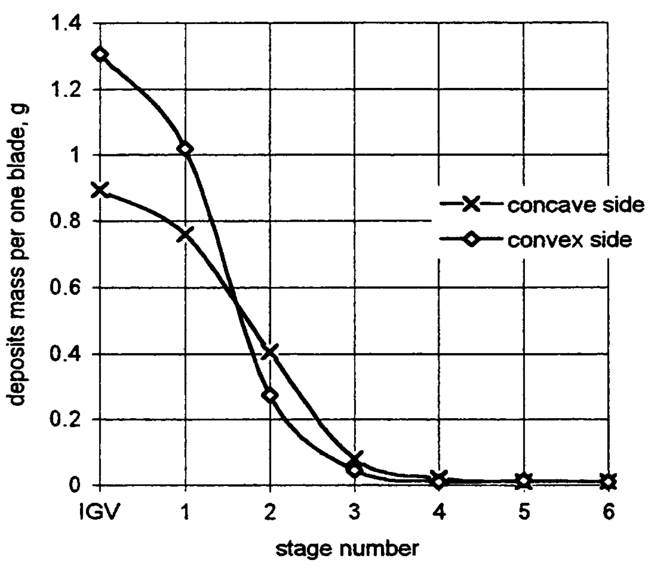

Figure 21. Measured Distribution of Deposits on Axial Compressor Stator Blades on a Two-Shaft Frame 5 Gas Turbine. (Courtesy Tarabrin, et al., 1998)

flow rate change follows the following rules according to Zaba (1980):

- Percent change in mass flow is approximately equal to the percent change in compressor efficiency if all the stages are equally fouled.
- Percent change in mass flow is greater than percent change in compressor efficiency if the early stages are fouled (this situation is most commonly found in practice). Typically, the change in mass flow rate in percent equals 1.25 times the change in compressor efficiency in percent. For example if the mass flow changes 2 percent, the compressor efficiency changes approximately 0.8 percent.

Engine Configuration and Number of Shafts

Analytical studies by Tarabrin, et al. (1998), examined the susceptibility of fouling based on the engine configuration. Three configurations were examined:

- Single shaft gas turbines
- Two shaft gas turbine, i.e., a gas generator and a free power turbine
- Three shaft configuration with a two shaft compressor (low pressure, LP, and high pressure, HP) driven by their respective HP and LP turbines and a free power turbine

For the comparative study, the same cycle thermodynamic parameters of pressure ratio, airflow, and ISF were used. The study examined the effect of a 1 percent drop in compressor efficiency with results being presented in Table 4 and graphically in Figure 22. The table and figure indicate changes in power (δNe), gas thermal efficiency (δηe), airflow rate (δG), pressure ratio (ΔΠ), and compressor speed (δN, δN_{LPC}, δN_{HPC}).

Table 4. Coefficient of Influence of Gas Turbines with a 1 Percent Drop in Compressor Efficiency.

gas turbine unit type	Nominal regime parameters	Conditions of calculation	δN_e	$\delta \eta_e$	δT_3	δG	$\delta \pi$	δn	δn_{LPC}	δn_{HPC}
single-shaft	T_3=1223 K π=12.5	n=const T_3=const	-2.82	-1.41	0	-1.25	-1.25	0	-	-
two-shaft	T_3=1223 K π=12.5	T_3=const	-3.5	-1.55	0	-2.07	-2.07	-1.4	-	-
		n=const T_3=const, δF=varia	-2.82	-1.42	0	-1.25	-1.25	0	-	-
three-shaft	T_3=1223 K π=12.5	T_3=const	-4.32	-2.16	0	-2.16	-2.16	-	-1.16	-0.58

Note: coefficients of influence are given in % for -1% of the axial compressor efficiency change.
LPC- low pressure compressor, HPC- high pressure compressor.

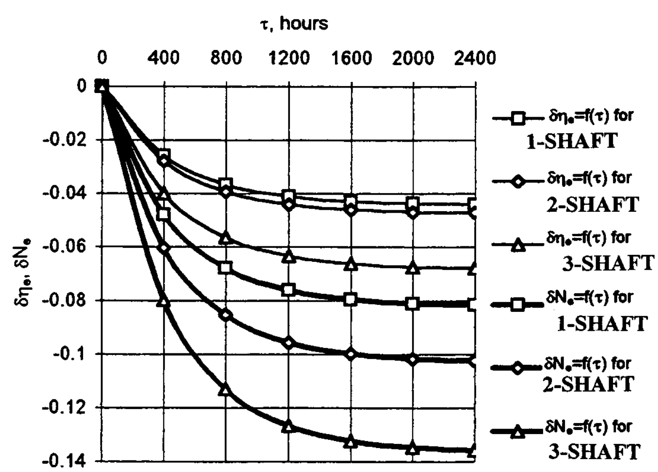

Figure 22. Influence of Axial Flow Compressor Fouling on Efficiency and Power for Different Shaft Configurations. (Courtesy Tarabrin, et al., 1998)

For the case of constant turbine inlet temperature T_3 it can be seen that the single shaft gas turbine exhibits an output drop of 2.82 percent compared to a drop in power for the three-shaft gas turbine of 4.32 percent. The physical explanation of this behavior is that the power match point is of a gas generator compressor, and a gas generator turbine is achieved by a greater change in air mass flow in comparison with the single shaft gas turbine. The two-shaft turbine lies between the three-shaft and single-shaft turbine in terms of power drop (power drop of 3.5 percent).

In the case of a two-shaft turbine with variable power turbine nozzles (as may be found on mechanical drive split shaft gas turbines) the compressor speed can be kept close to constant by varying the power turbine area, thus resulting in a power drop of 2.82 percent.

Simulation Runs with Compressor Degradation on Six Gas Turbines

The computer software simulation used runs assuming an ambient temperature of 59°F by imposing compressor degradation were conducted on the following available gas turbines ranging in power from 38 to 174 MW.

- Advanced technology high output gas turbine, single shaft configuration (designated A)
- Moderate firing temperature gas turbines, single shaft configuration (designated as B and C, with B being a scaled up version of C)
- Moderate firing temperature gas turbine, single shaft approximately half the power of C (designated D)
- Aeroderivative gas turbine with two compressor spools and a power turbine (designated E)
- Aeroderivative gas turbine with three compressor spools (designated F)

As before, the degradation steps imposed on these gas turbines were:

- Step 1: New and clean, mass flow drop = 0 percent, compressor efficiency drop = 0 percent
- Step 2: Mass flow drop = 1 percent, compressor efficiency drop = 0.833 percent
- Step 3: Mass flow drop = 2 percent, compressor efficiency drop = 1.67 percent
- Step 4: Mass flow drop = 3 percent, compressor efficiency drop = 2.5 percent
- Step 5: Mass flow drop = 4 percent, compressor efficiency drop = 3.33 percent
- Step 6: Mass flow drop = 5 percent, compressor efficiency drop = 4.167 percent
- Step 7: Mass flow drop = 6 percent, compressor efficiency drop = 5 percent

The following assumptions were made for all simulations:

- Fuel CH_4, LHV = 21,518 Btu/lb supplied at 77°F
- Gas turbines run at 100 percent rating, inferred turbine inlet temperature (TIT) control mode, control curve limited
- Site ambient conditions 14.7 psia, 59°F and 60 percent relative humidity
- Inlet and outlet losses 4 and 5 inch WG, respectively

In order to compare the gas turbine and their response to fouling, the simulation results were normalized where "1" represents the output and efficiency at Step 1 (i.e., in a new and clean condition with no deterioration imposed). The results in terms of normalized power and normalized efficiency are shown in Figures 23 and 24. The results seem to corroborate Tarabrin's findings in that the multispool machines tend to be more sensitive to fouling.

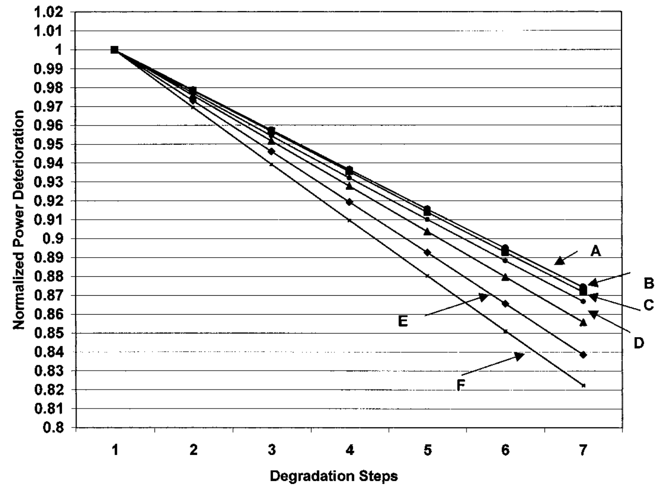

Figure 23. Normalized Power Changes with Fouling Steps for Different Gas Turbine Models.

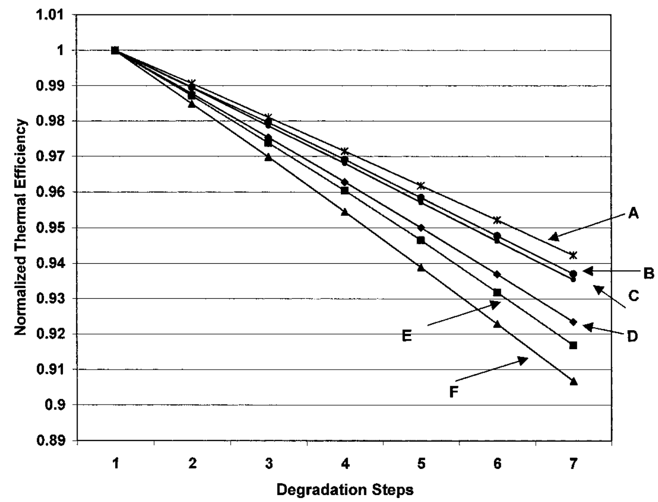

Figure 24. Normalized Efficiency Changes with Fouling Steps for Different Gas Turbine Models.

It is important to note that this analysis provides a sensitivity of performance to fouling that is imposed uniformly on all the simulated gas turbines. It does not address the *susceptibility* of fouling of the subject gas turbines.

Research conducted by Arnulfi and Massardo (1993) states that the distribution of pressure rise in the compressor is also an important determinant in fouling deterioration behavior.

Site Location and Environment

The geographical area, climatic conditions, and the geology of the plant location and its surrounding environment are major factors that influence compressor fouling. Areas can be classified as desert, tropical, rural, artic, offshore, and marine. Salient data for different climatic conditions is provided in Table 5. Airborne contaminants (dust, aerosols) and their type (salts, heavy metals, etc.), their concentration, particle sizes, and weight distribution as well as the vegetation cycles are important parameters influencing the rate and type of compressor fouling.

Plant Layout and Design

The rate of compressor fouling deterioration and foulant type are strongly affected by predominant wind directions. Factors that must be taken into account include:

- Orientation and elevation of air inlet ducts.

Table 5. Comparison of Operating Environments.

ENVIRONMENT	COUNTRY SIDE	COASTAL OFFSHORE	LARGE CITIES/ URBAN (Power Stations) (Chemical Plants)	INDUSTRIAL AREAS (Steel Works) (Petrochemical) (Mining)	DESERTS (Sand Storms) (Dusty Ground)	TROPICAL	ARTIC
Type of Dust; Foulant	Dry-Non Erosive	Dry-Non Erosive Salt Particles. Corrosive Mist. Drilling mud on offshore platforms	Sooty-Oily. May be Erosive, also Corrosive.	Sooty-Oily Erosive. May be Corrosive. Aggressive Chemicals may be present	Dry-Erosive in sand storm areas. Fine talc-like in areas of non-sand storms but dusty ground.	Non-Erosive May cause fouling	Non-Erosive
Dust Concentration Mg/m³.	0.01-0.1	0.01-0.1	0.01-10	0.1-10	0.1-700	<135	0.01-0.25
Typical Particle Size	0.01-10	0.1-3 Salt <5	0.1-20	0.1-50	1-700	0.1-10	0.1-10
Effect on G.T.	*Minimal Fouling*	*Fouling Corrosion*	*Fouling* Sometimes corrosion and fouling.	*Fouling /Erosion* Sometimes corrosion. Acidic conditions can exist at compressor stages	*Erosion* Plugging of filters with sand or with insect swarms.	*Fouling*	Plugging of air intake system with snow and ice.
Temperature Range (°C)	-20 to +30	-20 to +25	-20 to +35	-20 to +35	-5 to +45	5 to 45	-40 to +5
Weather Conditions	Dry and sunny Rain Snow Fog	Dry and sunny Rain, snow, sea mist. Freezing fog in winter.	Dry and sunny Rain Snow Hail stones Smog	Dry and sunny Rain Snow Hail stones Smog	Long dry sunny spells. High winds Sand and dust storms. Some rain.	High humidity Tropical rain Insect and mosquito swarms.	Heavy snow High winds Icing conditions. Insect swarms in summertime in some areas.

- Location of water cooling towers or air to air heat exchangers.
- Possibility of gas turbine exhaust recirculation into the air inlet.
- Orientation of lube oil breathers and vapor extractors.
- Location of highways, and adjacent plants, that can contribute to the fouling problem.

CFD predictions can be used to understand the phenomenon especially in grass roots installations. Other important plant design parameters that affect the rate of compressor fouling are the selection of appropriate air inlet filtration systems (self-cleaning, depth loading, etc.), the selection of filter media, the number of filtration stages, weather louvers, inertial separators, and mist eliminators. Design parameters such as the air face velocity through the filters, filter loading, and their behavior under high humidity and high pressure drops are also critical.

Plant Maintenance Philosophy

The quality and maintenance philosophy of an operating plant will have a significant effect on the fouling deterioration rate. The following plant maintenance activities will have an impact:

- Quality of maintenance of the air filter system. This includes checking the media quality, monitoring the differential pressure, taking maintenance action to avoid leakages, and ensuring proper storage of the spare air filter elements to avoid damage.
- If an evaporative cooling system is present the degree of maintenance becomes even more important. Evaporative media must be checked for signs of deterioration or separation from the steel framework or loss of integrity of the mist eliminator. Unequal wetting of the media can create droplet carryover. With fogging systems, constant checks on the quality of the demineralized water must be made.
- Method of detection of fouling deterioration. This could include some form of performance monitoring or tracking of key aerothermal parameters.
- Schedule and frequency of compressor washing
- Maintenance of the gas turbine bearings and seals to prevent oil leakages into the inlet plenum

In general a plant that is conscious of the fouling problem and the losses associated with it will adopt a maintenance schedule to manage and mitigate the problem. Unfortunately, in several mechanical drive applications the power margins that are designed into the system (excess gas turbine power over compressor absorbed power) may promote a negligent attitude toward fouling control.

Climatic Parameters

Ambient temperature, relative humidity, and climatic conditions such as fog and smog will strongly impact compressor fouling. The change in differential pressure in an inlet system over a day is shown in Figure 25, with the effect of fog (increase in differential pressure) being evident during the early hours of the day. In some cases, turbines have been known to trip due to excessive backpressure caused by morning fogs. In other cases, excessive sandstorms have also been known to create high differential pressure conditions.

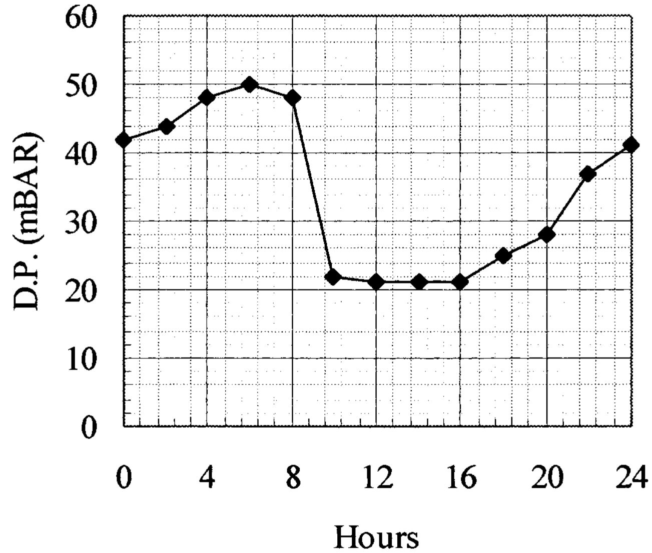

Figure 25. Filter Differential Pressure Changes with Time Due to Fog in Early Morning.

Correlation Between Rate of Power Degradation and Atmospheric Condition

Based on various field tests and observations (Stalder, 1998), a correlation between rate of power degradation and atmospheric conditions prevailing at site has been established. Stalder's studies included extensive testing covering eighteen months and utilizing several types of wash cycles including online and crank washing. The salient results are presented below.

Out of 40 measured continuous operating periods (without shutdowns and startups), a total of 14 operating periods each between 70 to 72 hours can be directly compared in Figure 26. This graph shows the power output measurements were made at the beginning of each period after online washing (100 percent reference point) and at the end of each period, prior to online washing of the next period. One can see that there is a large spread in the power losses over such a short period, and the highest loss in performance was 3.1 percent. Interestingly, the output at the end of one period shows a gain of 0.5 percent power output. This surprising result was obtained on the same unit, with the same air inlet filtration system, the same washing nozzle system, the same washing procedure, and the same detergent.

It is generally assumed that power losses will depend on the amount of humidity in a specific environment. With the data collected during the comparative test periods above, the total quantity of water and vapor mass flow ingested by the compressor was determined. Stalder calculated the compressor air mass flows by means of a heat balance. The average ingested total humidity (water and vapor) amounted to 7.7 metric tons/hour, i.e., a total of 548 metric tons during 70 operating hours. The lowest average value during a period was 4.1 tons/hour and the highest was 11 tons/hour. The distribution shown in Figure 27 shows the measured power losses versus the total quantity of humidity (water and vapor) ingested by the compressor for each of the selected comparative 70 hour operating periods. The results of this test clearly indicated a strong correlation between the mass flow of absolute humidity and output deterioration. The loss in power output on the unit tested increases with increasing mass flow of absolute humidity until it reached a peak at approximately 400 to 450 tons (total over 70 hours) before decreasing again.

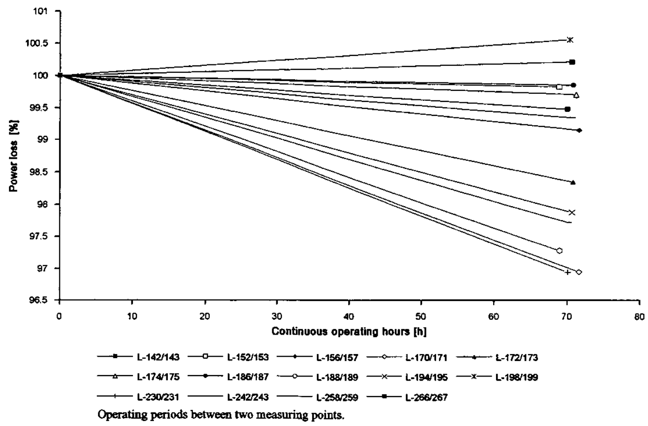

Figure 26. Power Losses over 70 Continuous Operating Hours. (Courtesy Stalder, 1998)

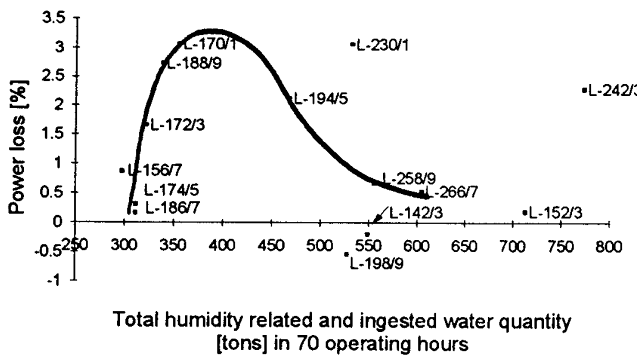

Figure 27. Correlation Between Total Absolute Humidity and Power Loss over 14 Comparative Operating Periods.

Correlation Between Power Loss Gradients and Humidity

Due to a drop in static pressure at the bellmouth caused by the acceleration of the air near the IGVs, the humidity in the air will start to condense. To give an idea of the amount of water condensed, Stalder (1998) provides the following example for a 250 MW gas turbine with an airflow rate of 500 m^3/sec operating at an inlet Mach number of 0.5 at the IGVs:

- At 12°C (53.6°F) gas turbine compressor inlet temperature and 90 percent relative humidity (RH), the total condensing water mass forming droplets will be up to 6.3 tons/hr.
- At 12°C (53.6°F) and 60 percent RH, the total condensing water mass forming droplets will be up to 3.1 tons/hr.
- The latent heat released by the condensing water (i.e., increasing the compressor inlet temperature) will be higher at 90 percent RH. Therefore the final static temperature drop in the air inlet at 90 percent RH is 7°C (44.6°F) whereas it is 10°C (14°F) at 60 percent RH.

Summarizing Stalder's findings with respect to humidity:

- *Influence of humidity:* Surface wetness of compressor blades operating in saturated condition will modify the aerodynamic boundary layer promoting fouling and a subsequent drop in performance.
- *Latent heat release:* The ingested air temperature will increase at the compressor bellmouth entry as condensation occurs and latent heat is released, thus reducing cycle efficiency.
- *Humidity and water wettable/soluble deposits:* With lower amounts of condensed water droplets, ingested foulants will combine with the water droplets and deposit on the vanes and blades. The rate of deposition will increase with the resulting roughness. Above a certain amount of condensed water droplet mass flow that is formed, the blading will be naturally washed and power losses due to fouling will be recovered to some extent. (This explains the phenomenon of "self cleaning" that several operators have experienced but do not understand.)
- *Humidity and hydrocarbon type of deposits:* A similar effect as that above will occur for a small amount of condensed water droplet mass flow. However, as water is a poor hydrocarbon solvent, the natural washing effect as the condensed water droplet mass flow increases will be very limited. Power losses due to fouling will continue until equilibrium is reached.
- *Humidity and combination of water wettable/soluble and hydrocarbon type of deposits:* This type of deposit is very common, and depending upon the mass relationship of hydrocarbon versus water wettable/soluble foulants, and their respective embedment in the deposit layers, the "natural washing" effect when condensation is high can also be limited.
- *Fouling rate at low ambient temperatures:* The flattening of the saturation curve in the low ambient temperature range (below about 10°C, 14°F) in the psychrometric chart shows that the amount of water droplets that can combine with foulants is significantly reduced, resulting in lower fouling.
- *Duration and sequence of operation in a saturated condition:* The changes in rate of power losses noticed over a given operating period will be strongly influenced by the duration and sequences of operation under saturated high humidity conditions (versus a dryer condition).

FILTER SYSTEM INFLUENCES ON FOULING

Humidity Effects on the Fouling of Axial Compressors

As air passes through the intake and filtration system, it proceeds at a very low velocity with filter face velocities being typically around 3 m/sec. 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 (50°F to 59°F). 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. The process is depicted in Figure 28 (Zaba and Lombardi, 1984). In this figure, T_{as} is the static air temperature, T_s is the saturation air temperature, and γ is the relative humidity. Three values of relative humidity are shown (40, 60, and 80 percent). As can be seen in this figure, when the relative humidity is 60 percent, the air temperature T_{as} is lower than the saturation temperature (in the shaded region of the figure), resulting in condensation of the water vapor.

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 and typically causes less fouling in the rear stages. A photograph of salt leeching through a filter is shown in Figure 29.

Stalder and Sire (2001) have conducted detailed analytical and experimental work on salt percolation through gas turbine air filters and some of the salient observations and results are presented below.

- The ingestion of airborne sea salt through a filter can result in high temperature corrosion damage. With the high temperature engines available today this is of significant importance as turbine manufacturers limit the total amount of salt entering the engine to low values such as 0.2 to 0.5 ppmw. Total contaminants that enter the engine must be calculated based on airflow, water injection flow, and fuel flow. The total contaminant entering the engine is given by:

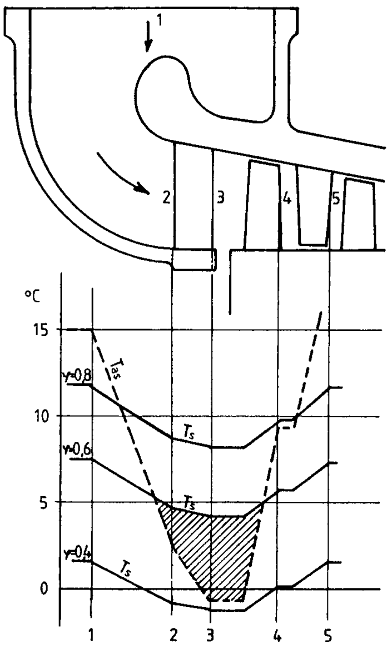

Figure 28. Intake Temperature Depression and Condensation. (Courtesy Zaba, 1984)

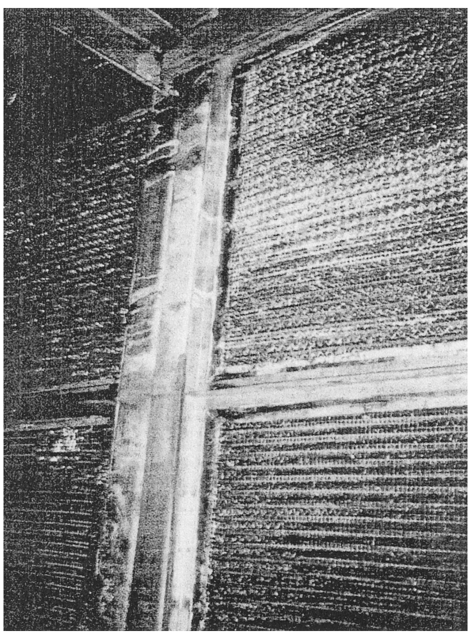

Figure 29. Salt Leeching Through Intake Filter. (Courtesy Altair Filter)

$$TCL = I_f + \left[I_{air} \times A/F\right] + \left[I_w \times W/F\right] + \left[I_s \times S/F\right] \quad (5)$$

where:
TCL = Total contaminant level, ppmw
I_f = Contaminant level in the fuel, ppmv
I_{air} = Contaminant level in the air, ppmv
I_w = Contaminant level in the injection water, ppmv
I_{stm} = Contaminant level in the injection steam, ppmv
A/F = Air to fuel ratio for the gas turbine
S/F = Steam to fuel ratio
W/F = Water to fuel ratio

• With regard to limits of airborne contaminants, the following guidelines are provided: maximum allowable level of sodium chloride (NaCl) in treated air is 0.01 ppmw (which is equivalent to 0.03 ppmw Na in the fuel assuming an air to fuel ratio of 75). Some specifications set an airborne limit for Na + K + V + Pb not to exceed 0.005 ppmw. Specific OEM requirements should be considered, as this is dependant on blade metallurgy, coating technology, and cooling approach. The performance of filters in limiting ingress of salt is therefore extremely important.

• Studies performed on used filters indicate that percolation of salt laden water can occur in both surface and depth loading type of filters. The mechanism is supported by the presence of dissolved acid gases (SO_x and NO_x) found in industrial environments.

• The large area of the air filter media may also act as a reaction site for various chemical processes and reactions. The real challenge is to determine the efficiency of the media under real life operational conditions where aggressive gases may exist in the environment.

• Optimal selection of filter media is of supreme importance and the system must be tailored to climatic conditions.

• Good filtration must be coupled with a carefully planned online and offline washing regime to minimize salt deposits on compressor airfoils.

Stalder and Sire (2001) present results of a very interesting investigation into salt ingress into the combustion air by the means of examining filters. Details of the analysis of one of the filter elements are shown in Table 6. The results show that higher concentrations of chlorides and sodium are found *within* the filter near the clean airside, clearly indicating that percolation or leeching has taken place. These analysis results are for a cylindrical type air filter element after 15,000 hours of operation in a coastal and industrial environment.

Table 6. Analysis of Air Filter Element after 15,000 Hours Operation in a Coastal and Industrial Environment, Indicating Salt Migration to the Clean Air Side.

$\mu g / cm^2$	Front location "upstream" (Dirty air side)	Middle location "middle"	Bottom location "downstream" (Clean air side)	Average of the three sample locations
Soluble fractions:				
$Na^+ + K^+$ cations	12	66	158	79
Total cations$^{(+)}$	80	402	683	388
Chloride anions, Cl^-	18	201	213	144
Total anions$^-$	161	875	1350	795
Total soluble fraction	**241**	**1277**	**2033**	**1183**
Total insoluble fraction	**779**	**656**	**609**	**681**
Sampling locations represent approx. 1/3rd sections along the cylindrical air filter element. Elemental results are expressed in micrograms per square centimeter of filter element surface area.				

Additional details on airborne contaminants are provided in Hsu (1988).

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. (Other vane type designs are also available.

- *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, and air is drawn through the media at a low velocity. At a predetermined pressure drop (about 2 to 3 inches WG) 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. Details of offshore platform filtration systems are provided in Kimm and Langlands (1985) and Schweiger (1983). Zaba and Lombardi (1984) have detailed experiences with different types of filters.

Practical Aspects Relating to Air Filter System Integrity

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.

- The 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. The filter house should withstand 12 inches water gauge pressure. All seams and joints should be airtight. 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 changeout of all filters from the upstream side. Filter change should be possible without turbine shutdown. Filter elements should be designed for quick changeout, 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 air tightness.

- 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 Design and Corrosion

Experience has shown that stainless steel (316L) construction is *by far the most sensible approach* especially in offshore and coastal areas. The life cycle costs 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. A highly corroded floor of an inlet filter is shown in Figure 30.

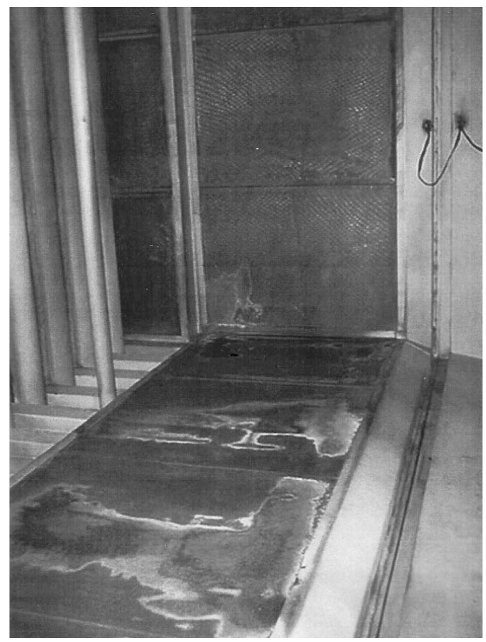

Figure 30. Corroded Floor in Inlet Duct of Large Gas Turbine.

Corrosion of inlet systems is widespread especially when they are made of carbon steel. Corrosion of the filter house downstream of the filters totally defeats the purpose of having air filtration. Painted carbon steel structures need constant painting and, often, poor application of paint and other problems involving dissimilar materials start the corrosion process only 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. Modern technology has allowed the use of high performance SS 316L systems that include relatively light elements, and 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.

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 enable water to be removed from 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).

Filter System Pressure Drop

The effect of high pressure drop in the inlet filter system is a reduction in the inlet density and mass flow of the gas turbine and a reduction in the turbine section expansion ratio. The effect of inlet system pressure drop is provided for a Frame 5 gas turbine in Table 7. Salient cycle conditions for the engine are provided for inlet pressure drops of zero through 5 inches WG. As the inlet system differential pressure increases, the pressure to the turbine section drops and results in a lower expansion ratio and an increase in the exhaust gas temperature (EGT). A power drop of 0.4 percent per inch of WG differential pressure and a heat rate increase of 0.19 percent per inch of WG differential pressure emphasizes the importance of this parameter.

Table 7. Impact of Inlet Filter Differential Pressure on a Frame 6 Gas Turbine.

INLET DIFFERENTIAL PRESSURE SENSITIVITY STUDY SIMULATION

Fuel=CH4, supplied @ 77 F, LHV = 21517.58 BTU/lb
G.T. @ 100 % rating, inferred TIT control model, CC limit
Site ambient conditions: 14.7 psia, 100 F, 60% RH

GE 5371PA	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
INLET DELTA P, inch WG	**0**	**1**	**2**	**3**	**4**	**5**
Computation Result, Thermoflow - STQUIK	OK	OK	OK	OK	OK	OK
GT gross power [kW]	21861	21773	21685	21598	21510	21422
GT gross LHV eff [%]	26.84	26.8	26.75	26.7	26.65	26.6
GT gross heat rate [BTU/kWh]	12711	12734	12758	12781	12805	12829
Compressor inlet mass flow [lb/s]	245	244.4	243.7	243.1	242.5	241.9
Compressor inlet temperature [F]	99.86	99.86	99.86	99.86	99.86	99.86
Turbine inlet mass flow [lb/s]	238.3	237.8	237.2	236.6	236	235.4
Turbine inlet temperature [F]	1761.2	1761.2	1761.2	1761.2	1761.2	1761.2
Turbine exhaust mass flow [lb/s]	248.5	247.9	247.3	246.7	246.1	245.5
Turbine exhaust temperature [F]	939.6	940.3	941	941.8	942.5	943.3
GT fuel HHV input [kBtu/hr]	308341	307663	306985	306306	305627	304947
GT fuel LHV input [kBtu/hr]	277881	277270	276659	276048	275436	274823
GT fuel flow [lb/s]	3.587	3.579	3.571	3.564	3.556	3.548
%Change in Power per 1" WG	**0.40**					
%Change in HR, per 1" WG	**-0.19**					

EFFECTS OF FOULING ON GAS TURBINE OPERATION AND MAINTENANCE

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 flow compressor is a sensitive component that requires smooth aerodynamic surfaces. Fouling causes an alteration in the shape and profile of the blading (increased surface blade roughness), and this reduces air flow rate, pressure ratio (of the overall compressor), and compressor efficiency. Modeling details may be found in Lakshminarasimha, et al. (1994), and Tabakoff (1988).

Figure 31 shows the changes in discharge pressure due to fouling at different ambient temperatures for a heavy-duty gas turbine.

Surface roughness caused by compressor fouling increases the profile losses of the blading. These losses appear as a boundary layer momentum thickness, which increases with increasing roughness in the blade profile. In simple terms, the drag increases and results in an increase of the specific work (kW/unit mass flow) of the compressor.

For example, for the simulated values of the Frame 6 modeled with degradation steps in Figures 11, 12, and 13: the specific compressor work increases from 158.77 kW/lb/sec in the undeteriorated condition to 163.82 kW/lb/sec with the maximum deterioration.

A detailed treatment of losses is provided in Koch and Smith (1976). A treatment of roughness effects on the aerodynamics is provided in Bammert and Woelk (1980). An interesting treatment of the effect of Reynolds number and blade surface roughness is provided in Schaffler (1980).

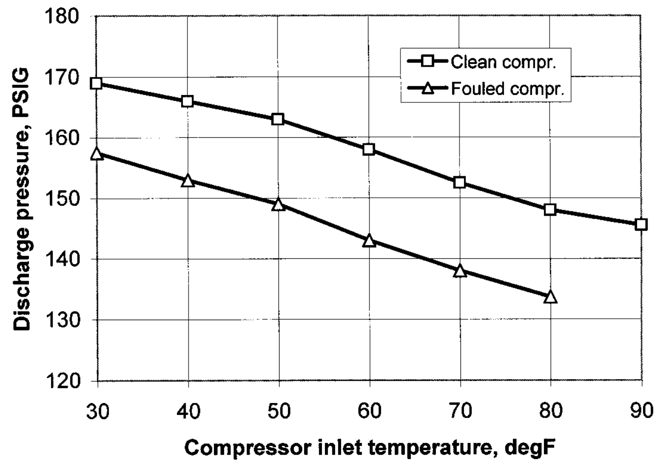

Figure 31. Drop in Compressor Discharge Pressure Before and After Fouling.

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. Details on the effects of stage characteristics on axial flow compressor performance may be found in Stone (1958).

Dundas (1986) has conducted a detailed 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 parameters 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.

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 a 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. Closure of the IGVs during part load operation restricts airflow and this, in conjunction with severe fouling, can promote surge. Steam injection results in a higher back pressure on the compressor, thus moving the operating point closer to the surge line.

There have been several cases where excessive distortion of the inlet airflow has triggered a surge event resulting in compressor damage. Icing, for example, 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 given by 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 imbalance, 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 adds 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. Airborne salt may be ingested as dry particles or as droplets of brine, and moisture will be absorbed during high humidity operating conditions or during water washing. Salts, mineral acids, and aggressive gases (e.g., SO_x, NO_x, Cl_2, etc.) along with water can cause pitting of compressor blades due to electrochemical corrosion mechanisms. This can lead to local stress raisers that can diminish blade fatigue life. Compressor coatings are of value here.

• 5 to 10 micron particle size represents the transition zone between fouling and erosion, and particles causing erosion are normally 10 microns or greater. (Note: 10 microns = 1/15 diameter of human hair.) Erosion impairs aerodynamic performance and can affect the mechanical strength of the blade. Erosion first increases blade surface roughness, thus lowering efficiency slightly. As erosion progresses, airfoil contour changes occur at the leading and trailing edge as well at the blade tip. Severe erosion has also been known to cause changes in blade natural frequency.

• On relatively small gas turbines, a 0.1 mm (.0039 inch) 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 directly to fouling, this subject is mentioned because it can be caused by a loss in filter integrity. Damage is typically caused to the forward compressor stages, although in some cases the foreign object also works its way to rear stages and causes damage. Damage is a function of foreign object size and 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, or by materials and tools left in the inlet plenum.

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 solution that promotes 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 are 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 50-60 percent, and this situation is worsened when notches due to corrosion pitting are present. Even with effective air filtration, the conditions of fog, humidity, or rain can cause migration of the salt through the inlet filter (leeching) and into the compressor. Blading on which corrosion deposits have formed is shown in Figure 32.

Figure 32. Pitting Corrosion on Compressor Blading.

Haskell (1989) states that corrosion is rarely observed beyond the eighth compressor stage, as no moisture will survive at the temperatures beyond this point. As shown in Table 8, sulfur dioxide (SO_2) and hydrochloric acid (HCl) can create very acidic conditions on the compressor blading, even in the parts per billion concentration range. This situation is acute in heavy industrial areas or close to chemical manufacturing plants. Furthermore, even moderate relative humidity conditions of about 50 percent will result in water formation at the compressor inlet due to the intake temperature depression. A detailed treatment of gas turbine blade failures including underlying causes and troubleshooting is provided in Meher-Homji and Gabriles (1998).

Table 8. Effect of Ambient Gases on Acidity at the Compressor.

	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

DETECTION OF FOULING

Gas turbine manufacturers and operators usually develop guidelines to define 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. However, it is the opinion of some operators that the only way to detect a fouled compressor is by visual inspection. Unfortunately, though, with most turbine designs this means shutting down the unit, removing the inlet plenum hatch, and visually inspecting the compressor inlet, bellmouth inlet guide vane, 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)

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

The real problem is to detect fouling in time to prevent a significant power drop, and before a fuel penalty cost has been incurred. Several philosophies are in use. Some operators believe in regular periodic washing of the machine, whereas others base the washing requirement on a certain set of performance parameters. The philosophy utilized is a function of normally expected fouling levels, 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 flow meter. This approach has been successfully used by Scott (1979, 1986).

By means of suitable software, data available in the gas turbine control system can often be used to monitor compressor deterioration. The set of graphs taken on a Frame 7 EA gas turbine engine (Dusatko, 1995) show the correlations very clearly. In Figure 33, 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. The rule of thumb derived from this figure is that a 2 psig loss of compressor discharge pressure is equivalent to one gross MW of power for these particular 7 EA gas turbine engines. The effect of a crank wash on the CDP versus CIT plot is shown in this figure.

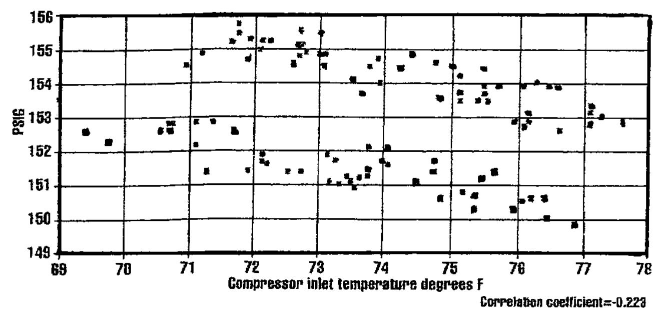

Figure 33. Scatter Plot Showing Effect of Crank Wash on CDP Versus Compressor Inlet Temperature. (Courtesy Power Engineering*)*

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

$$Power\ Capacity\ Factor = MW_{Actual} / MW_{Expected} \tag{6}$$

The MW expected is the corrected "new and clean" performance test output, which is typically 3 to 5 percent higher than the "guarantee" output. Corrections should be made for:

- Inlet temperature.
- Inlet pressure.
- Specific humidity.
- NO_x water injection rates.
- Inlet and outlet pressure drops.
- Speed corrections (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 "under-frequency" problems).

A baseline development process and the use of scatter plots for deterioration analysis of advanced F class turbines is provided in Meher-Homji, et al. (1993).

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 predicted characteristics. 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 on the map. The corrected speed represents the wheel tangential Mach number whereas the corrected flow represents the through-flow Mach number. Practical details of monitoring of compressor performance from a standpoint of detecting problems are provided in Dundas (1982) and Dundas, et al. (1992).

Once a judicious schedule for online and offline washing has been established, it is important to monitor the performance of the gas turbine and track for unexpected events and monitor the efficacy of washing. It is important that all parameters be monitored and appropriate corrections made using corrected speeds, temperatures, and flows that can then be trended. For aeroderivative engines the control approach must be taken into account (i.e., speed or temperature control). For aeroderivatives with multiple compressor spools, the interrelationship between the corrected LP and HP compressor spool speeds can provide a valuable indicator of compressor deterioration. Special performance monitoring considerations when operating at peak loads are provided in Syverud, et al. (2003).

CONTROL OF FOULING BY COMPRESSOR WASHING

Fouling is best controlled by a combination of two methods. The first line of defense is to employ a high quality air filtration system. However, as fouling will inevitably occur, compressor washing should be used to control its impact.

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

Several different methods of gas turbine compressor cleaning have been applied over the years, but "*wet cleaning*" has been found to be by far the most effective and economic technique. However, today's sophisticated large industrial engines and blade coatings require appropriately designed cleaning systems to ensure operational safety, reliability, and optimum efficiency. Two different wet cleaning techniques are generally applied, known as offline (crank wash) and online cleaning. Under extreme fouling conditions, hand washing of the IGVs may have to be conducted if time permits. During overhauls, hand cleaning of the full axial compressor is most effective. There has also been some recent interest in foam cleaning of gas turbines (Wicker, 2004)

Offline washing is almost always carried out with the aid of a detergent, and extremely effective power recovery can usually be achieved. However, it is important that the manufacturer's recommendations are followed with respect to water quality, detergent/water ratio, and other operating procedures. Typically, wheel space temperatures must be below 200°F to avoid thermal shock, and the offline water wash is done with the machine on crank. The downtime for a crank wash depends mainly on the time it takes for cooling the engine. Larger heavy-duty engines can take 8 to 10 hours to cool whereas on light aeroderivative engines only 1.5 to 3 hours may be needed because of the low metal mass. Offline cleaning is most effective when carried out in several steps that involve the application of a soap and water solution, followed by several rinse cycles using water alone. 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 between each wash and rinse cycle is also very important, and allows the soapy cleaning fluid to penetrate into the fouling deposits, thus dissolving salts and emulsifying oil and grease components. A useful method of determining the effectiveness of the offline wash, and perhaps the need for additional wash or rinse cycles, is to collect samples of the effluent water from all available drain ports. The samples can be checked visually for color and clarity (which should improve as dirt is removed), or by a simple conductivity meter to monitor the removal of salts.

Online washing is now very popular as a means to control fouling by avoiding the problem from developing. The primary objective of online washing is to extend the operating period between offline washes by minimizing the buildup of deposits in the compressor, and thereby reducing the ongoing incremental power losses. Online washing is performed with the unit in full operation, and techniques and wash systems have now evolved to a point where this can be done effectively and safely. Outages or shutdown periods are not required. Depending on the nature of the fouling material, online washing is sometimes performed with water alone. In most cases, however, the use of an approved cleaner (detergent) will improve the effectiveness of the washing operation. This is particularly true if the fouling material contains any quantity of oil or grease. Demineralized water quality is almost always specified by the OEMs for online washing, to avoid the possibility of introducing harmful trace metal contaminants (such as Na+K) into the combustion turbine.

Optimal compressor cleaning can normally be achieved by adopting a combined program of regular and routine online washing (for example every few days or weekly), plus periodic offline washing during planned outages.

Two main types of cleaning agents (detergents) are available for compressor washing and are normally classified as "water-based" or "solvent-based" products. Most of the new-generation products contain surfactants, wetting agents, and emulsifiers, and involve either an aqueous- or petroleum-based solvent system. Both type products are normally supplied as concentrates, and are diluted onsite with water (typically one part cleaner with four parts water) to produce the cleaning fluid. Solvent-based cleaners have traditionally been recognized as being more effective in removing oil and grease deposits, but certain new-generation water-based cleaners are formulated to be equally effective. Most water-based products also have the advantage of being biodegradable, which is an increasingly important requirement within this industry.

A comparative summary between on and offline cleaning is provided in Table 9.

OFFLINE (CRANK) WASHING

The basic objectives of offline cleaning are to clean a dirty compressor and to restore power and efficiency to virtually "new and clean" values. When performed correctly, and provided the operating period between offline washing is not too long (site specific), this type of cleaning will typically restore virtually 100 percent of the lost power and efficiency attributed to compressor fouling. However, irrespective of the compressor performance degradation actually encountered, experience has indicated that users of both base load and peaking gas turbines should incorporate a minimum of three or four offline compressor cleanings per year in order to remove the salt laden deposits on the downstream stages.

Table 9. Comparison of Offline and Online Washing.

OFF LINE WASHING	ON LINE WASHING
• <u>Objective</u>: To clean a dirty compressor • Reaches all compressor stages • Virtually full power recovery (approaches new & clean values) • Involves shut-down and cool-down period (12 to 36 hours) • Lost revenue during shut-down • Optimum time for cleaning may not be convenient, especially with base load plants • Effluent water for disposal	• <u>Objective</u>: To keep a clean compressor cleaner for longer • Extends operating period between off-line cleaning, thus enhancing production • About 1% power can be recovered per wash, with a frequent on-line cleaning program • Primarily cleans the IGVs (no effect after water evaporates) • No shut-down, no lost revenue • Optimum cleaning frequency is site specific • No effluent water for disposal • Maintains safe margin to surge line • Reduces risk of blade corrosion

Offline wet cleaning (also known as crank washing) is a typical "soak and rinse" procedure for which the gas turbine must be shut down and cooled. The compressor is rotated at crank-speed while a cleaning fluid is injected via nozzles or jet lances. Hand-held jet lances were widely used in the past and are still fairly popular with some operators. However, permanently mounted offline nozzle systems installed in the air intake plenum are now preferred, and are generally offered as standard by most of the major turbine manufacturers. Nozzle design, system operating pressure, and total mass flow parameters vary widely, however, between the different manufacturers.

The injected cleaning fluid is normally a mixture of chemical detergent and water. Both solvent-based detergents and water-based products are used, depending mainly on the type of fouling material found in the compressor and local plant experience.

After a soaking period the compressor is rinsed with a quantity of fresh water. The amount of rinse water required and the number of rinse cycles vary from site to site, according to the gas turbine model and the amount of dirt removed during the offline wash. Note that demineralized water is usually not specified for offline cleaning and fresh water quality is normally acceptable. Effluent water drained from the compressor has to be disposed of according to local regulations.

Typical water quality requirements for an offline wash are:

- Total solids (dissolved and undissolved) ≤ 100 ppm
- Total alkali metals (Na, K) ≤ 25 ppm
- Other metals that may promote hot corrosion (V, Pb) ≤ 1 ppm
- pH 6.5 to 7.5

This water would be used for cleaner dilution and also for rinsing.

Offline crank washing systems should be designed to achieve the highest washing efficiency with the smallest injection mass flow. This is important for the following reasons:

- Gas turbine users are interested in minimizing the quantity of effluent water to be disposed of.
- Some users claim that offline water effluent is transported up to the exhaust during the wash procedure, and may wet and soak into the expansion joint fabric—resulting in damage of the expansion joint by lowering its insulation properties.
- A lower offline injection mass flow will also reduce the potential risk of trace metal contamination in exhaust systems, where selective catalytic reactors (SCR) for NO_x reduction or carbon monoxide (CO) catalysts are installed.
- A smaller offline injection mass flow will significantly reduce the required size, volume, and cost of washing skids and the overall water and cleaner consumption.

Important Considerations During Offline Washing

Wetting of IGVs: Effective wetting of the IGV's suction area can be achieved by using full cone jet spray nozzles. The number of nozzles will be defined by the area to be wetted, which is usually the area between two intake struts. The necessary offline injection mass flow characteristic will therefore be determined by the area to be wetted and impacted by the jet spray, and the distance between nozzles and the IGVs. The injection pressure is generally between 5.5 to 6 bar (80 to 87 psi). As spray jet trajectory is subject to gravity, the nozzle should be designed to provide an adjustment of up to 5 degrees to compensate for the gravity effect. Crank washing (soaking) and rinsing can be considered as a mechanical "erosion" of the deposit layer, and soaking time will allow the cleaner to penetrate and soften the deposit layers. Systems with high atomization pressure will have no impact pressure on the IGVs, because the spray pressure will have decayed approximately 20 cm (7.87 inches) from the nozzle outlet, and the atomized droplets will need to be carried by the relatively low air flow rate produced at crank speed. Most high pressure systems do not show the same effectiveness in removing salts and insoluble compounds on downstream stages. Typical location of wash nozzles for both online and offline systems is shown in Figure 34.

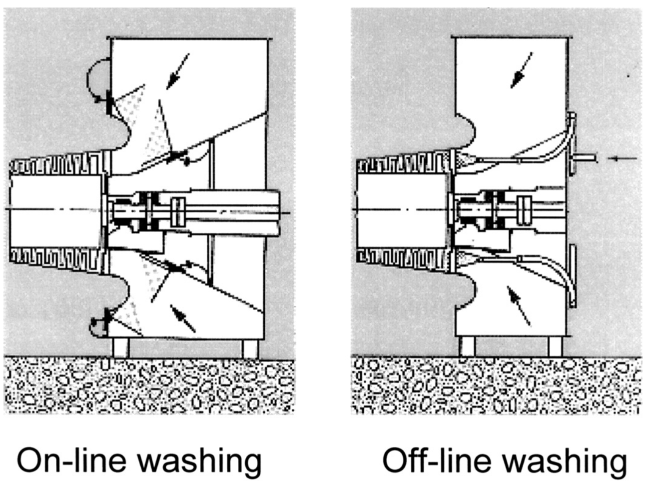

Figure 34. Locations of Wash Nozzles. (Courtesy Turbotect Ltd.)

Drainage: The effluent water collection system to drain and remove dirty water out of the engine is of prime importance. Equally important is the isolation scheme to prevent runoff water from penetrating sensitive areas such as sealing and cooling air systems and instrumentation air systems. The physical location of bleed air taps on compressor casing is also important. If they are located on the bottom they are likely to drain runoff water, and therefore should preferably be located in the upper part of the casing. Drains in air inlet and compressor casing, combustion chamber, and exhaust should be located at the lowest point. The drain diameters should be sized for an easy runoff. Care must also be taken that they do not become plugged with dirt.

Engine speed variation: Offline crank cleaning is very efficient for removing all deposits on all the compressor stages. To enable penetration of the wash and rinse fluid through the entire axial flow compressor, offline washing should be conducted at *variable speeds*; for example by injecting the solutions during coastdown of the shaft, after an acceleration of up to 500 to 600 rpm. By doing this, the pattern of the centrifugal forces on the injected solution through the compressor will decrease and allow better wetting and distribution on the blade and vane surfaces of all stages. By contrast, offline washing at high and constant cranking speed will result in a lower cleaning efficiency. Conductivity measurements and checks on the clarity/turbidity of the drain water will help assess cleaning efficiency.

Procedures and precautions: The OEM's recommendations and checklists should be followed prior to a crank wash procedure. Some typical items of importance include:

- Ensure that OEM's wheel space temperature criteria are met prior to the crank wash.
- Seal and atomizing air pipes should be closed-off to avoid water entry.
- IGVs (if applicable) should be in the open position (maximum airflow) prior to the crank wash.
- All drains should be opened.
- Flame detector valves should be closed.
- Auxiliary air compressor should be disconnected.
- Special precautions may have to be taken for regenerators, if these exist.
- If possible, the plenum should be hand washed to avoid dirt being washed into the compressor.
- Follow the OEM's or wash system supplier's recommendations regarding the duration and amount of fluid injection. These guidelines may need to be modified by evaluating the results.
- Approved anti-icing agents will be required if offline cleaning is performed at ambient temperatures below or near to freezing. Even at crank speed, some degree of temperature depression will occur at the bellmouth.

ONLINE WASHING

The basic objectives of online cleaning are to *maintain* the cleanliness of a compressor after offline washing, to *maintain* power and efficiency by minimizing ongoing losses, and to *extend* the operating period between shutdowns required for offline (crank) washing. Online washing for fouling control has become increasingly important with base load combined cycle plants and combined heat and power production (CHP) plants. It is also important for gas turbines in mechanical drive service, where little or no redundancy is installed and where the downtime associated with crank washing must be minimized.

Online wet cleaning is performed while the gas turbine is in operation and at load. The procedure involves the injection of a mixture of water and chemical detergent via atomizing spray nozzles positioned around the compressor air intake plenum. This is followed by a flushing period using pure water. With online cleaning it is mandatory to use demineralized water for preparing the cleaning fluid and for flushing. This is because the turbine is in operation, and high temperature corrosion damage may occur if sodium or other contaminant metals enter the combustion path.

The water specification of the particular OEM should be followed, but typical values are presented below:

- Total solids (dissolved and undissolved) ≤ 5 ppm
- Total alkali and other metals that may promote hot corrosion (Na, K, Pb, V) ≤ 0.5 ppm
- pH 6.5 to 7.5

An online washing program should always be started on a clean engine, after an overhaul or crank wash. It is not recommended to perform online washing on a heavily fouled engine, because large quantities of dirt removed from the front stages would instantaneously pass through the compressor. Therefore, after starting an online wash program, the time intervals between subsequent washings should be kept short: approximately every three days to weekly, depending on the local conditions. Also, depending on the type of deposits (i.e., portion of water-insoluble compounds), detergent cleaners may be used for every online wash, or for every second or third online wash, but not less frequently than once per week. Note that the longer detergent washing is not done, the greater the risk of downstream contamination due to large portions of insoluble compounds suddenly being removed when the next detergent wash is performed. Thus, frequent online washing using detergents is advisable to minimize the accumulation of insoluble foulants.

The duration of each online wash can also be varied according to the degree of fouling, engine size, and plant experiences, etc. Typical online cleaning cycles are in the order of 10 to 20 minutes, and a flushing or rinsing cycle (using only demineralized water) of about the same duration should be applied after each cleaning cycle with detergent—for example, 10 to 20 minute cleaning cycle with detergent, followed by 10 to 20 minute flushing cycle without detergent. This type of regular online wash regime will extend the operating period between outages required for offline cleaning, which is particularly important for base load plants.

The design of an online washing system should attempt to obtain the highest possible cleaning efficiency with the lowest injection mass flow rate, and this *can* be achieved in combination with an optimum washing regime as discussed above. Frequent online washing (to keep a clean compressor clean) enables the use of low injection mass flow rates, and this can only be considered as "good" for the gas turbine. To summarize:

- Frequent online cleaning with a low injection mass flow system minimizes the risk of deposits being suddenly washed from the front stages onto downstream compressor stages. This also addresses concerns that removed dirt may enter airfoil film cooling systems of turbine blades.
- Low injection mass flow reduces blade loading and creates less stress on downstream blading.
- High water mass flow may interfere with flame detector intensities on units with dry low NO_x (DLN) combustors, and may also create a trip by causing fogging of flame detector lenses, etc.
- Higher CO emission levels may be observed during online washing, which becomes more significant with higher water injection rates.
- Typically, high mass flow nozzles create larger sized droplets. This means they are more influenced by gravity, and have a greater tendency to fall and be deflected onto inlet plenum surfaces. This also increases the risk of blade erosion.
- Effective online cleaning requires that the IGVs are thoroughly wetted with appropriately sized water droplets (typically 50 to 250 micron range). Any excess water is likely to "stream" over inlet plenum surfaces and struts, etc., and serves little or no purpose in the washing process. As mentioned above, excess water impacting the root area of IGVs may also initiate erosion of the rotating blades.
- Lower injection mass flow rates reduce demineralized water and cleaner consumption, and also reduce the size, volume, and cost of the wash skids.
- Experience has shown that a low water wash mass flow does not impair online cleaning efficiency.

Online Compressor Cleaning Efficiency

Fouling of the first stage guide vanes is the primary cause of reduced air mass flow through the compressor. Online cleaning is most effective in removing such deposits, and therefore restoring design air mass flow and lost power. Regular online cleaning will keep inlet guide vanes clean and free from deposit buildup. Droplets of cleaning solution or water may survive up to the sixth stage, but most will have vaporized by then. Online cleaning has no effect on downstream stages.

Figure 35 shows a compressor that was washed online every four days with detergent during one month of continuous operation. The first stage guide vanes of the same compressor are shown in Figure 36. Note that this photograph shows fairly heavy, black deposits still remaining at the tip and root areas of the IGVs, and this illustrates the severity of fouling that would have occurred over the entire vane length had regular online cleaning not been performed.

Figure 35. Axial Compressor Online Washed Every Four Days During 744 hours Operation. (Courtesy Turbotect Ltd.)

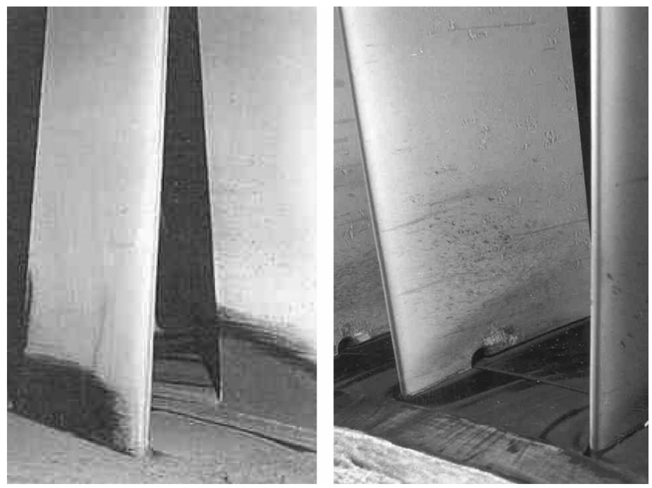

Figure 36. Compressor First Stage Guide Vanes Shown in One-Third Blade Length from the Tip (Left) and from the Root (Right). (One month continuous operation, washed online every four days. Deposits on the tip would have, in all probability, covered the entire vane if online washing had not been performed.) (Courtesy Turbotect Ltd.)

Blade Wetting Considerations with Online Washing

Effective wetting of the IGVs is obtained by a uniform and finely distributed atomized cleaning solution. Droplets must be stable in size and small enough that they do not cause blade erosion (due to the high blade speeds during operation). Droplets are subject to gravity, so they must also be light enough that they do not drop out of the air stream before they reach the compressor blade surface. Nonuniform wetting of the IGVs will result in spot cleaning and heavier droplets will most likely fall to the bottom, wasting some of the injected cleaning solution.

Nozzle Location for Online Washing

Correct positioning and location of online injection nozzles is of prime importance to achieve uniform wetting and efficient blade cleaning. Nozzles should be designed to inject a small quantity of finely atomized cleaning solution into the air stream where it will be thoroughly mixed and carried uniformly into the compressor bellmouth. Compared to offline cleaning these design factors are also extremely important in ensuring operational safety, because insufficient atomization and/or nonuniform injection flow can result in blade erosion and vibration. A relatively high number of online nozzles positioned in the air inlet casing both upstream and downstream of the bellmouth ensures a better distribution of the injected fluid into the air stream and provides better wetting.

The design and configuration of air inlet systems varies according to different gas turbine models, and this is also an important consideration in selecting appropriate nozzle locations. For example, some inlet systems direct the entire incoming air stream from the top of the plenum, down toward the bellmouth. This creates a higher air velocity within the plenum, and correct positioning of the online nozzles is more challenging. Other inlet systems split the air stream so that it enters the plenum equally from the right- and left-hand sides. This design lowers the air velocity and creates fewer disturbances to the airflow pattern. Hence, positioning of the online nozzles is somewhat easier. A typical online nozzle manifold is shown in Figure 37.

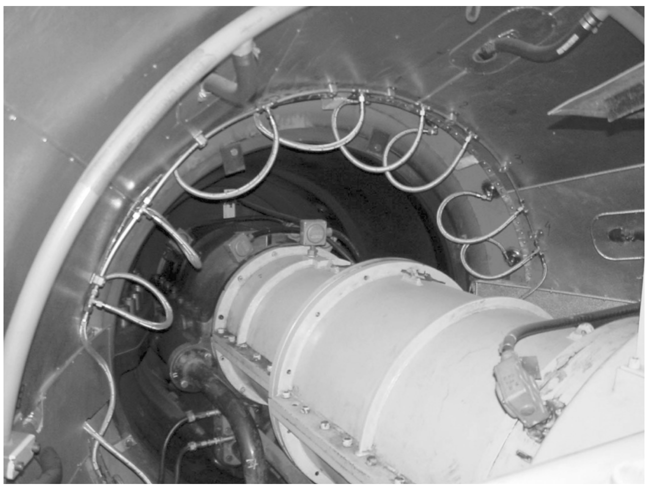

Figure 37. Photograph of Online Nozzle Manifold.

Nozzle Designs for Online Washing

The online injection nozzle is the most critical component in the washing system, and its design determines the overall effectiveness of the cleaning process. Important issues such as nozzle location, injected mass flow, wash frequency, and the use of detergent, etc., can be rendered immaterial if the installed nozzles do not perform. A nozzle is essentially an atomizing device, and they are designed to create droplets in a certain size range and with a specific spray profile. However, in order to achieve the design droplet size and spray pattern, the nozzle must be operated at its design flow rate and operating pressure. Changing the design parameters of a particular nozzle will affect its spray characteristics. For example, droplet size will usually increase by reducing operating pressure below the design point.

Many different types and designs of online injection nozzle are used within this industry; some are available directly from the gas turbine OEMs and others are manufactured and marketed by independent suppliers. Design differences between the various types of nozzles are quite significant, and this results in a wide range of mass flow rates, operating pressures, spray patterns, droplet size ranges, and, not surprisingly, performance characteristics. However, accepting that different design philosophies do exist (and probably will continue to exist), there is fairly common agreement within the industry that online droplet size should be within the range of 50 to 250 microns. If droplets are too small they are easily deflected by the air stream, and may not survive to perform wetting of the IGVs. If droplets are too large they are more influenced by gravity, and tend to fall toward the lower surfaces of the inlet plenum. Large sized droplets are also more likely to cause erosion problems.

General design factors for online wash nozzle systems are as follows:

- Nozzles must provide excellent atomization characteristics, and create a stable distribution of droplets within the recommended size range of 50 to 250 microns.
- Nozzles should be positioned to enable uniform wetting of the IGVs. (Note that high mass flow designs can create too much spray that collects and "streams" on plenum surfaces.)
- A sufficient number of nozzles must be installed to provide the required total mass flow for the particular gas turbine model.
- Flush-mounted, low profile nozzle designs create less disturbance to the air flow.
- All components must be corrosion resistant (stainless steel) and designed for maximum safety.
- The system should be easy to install on new units and on retrofits.
- The system should be operated at its design flow rate and pressure.

Figure 38 shows one particular design of online nozzle that is adjustable, and can be rotated in two dimensions to optimize the spray direction. This nozzle is a low pressure, low mass flow design, and produces a 90 degree cone-shaped spray pattern. The nozzle's low profile ensures minimal penetration into the air stream to avoid airflow disturbance. It also prevents misuse of the nozzles as climbing supports during compressor intake inspections.

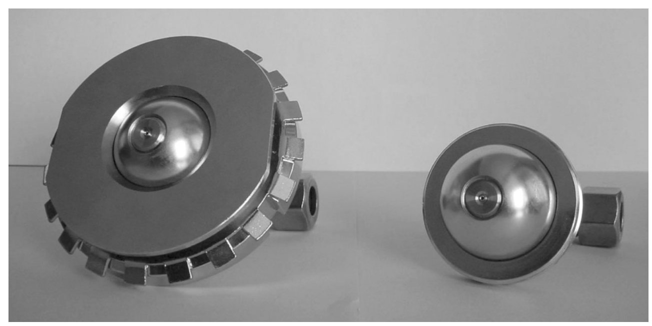

Figure 38. Adjustable Online Injection Nozzle, for Flush Mounting in Air Inlet Plenum. (Courtesy Turbotect Ltd.)

A useful review of gas turbine online wash systems has been published by Mund and Pilidis (2004). The different types of nozzles and online wash system designs used in this industry are surveyed and categorized according to operating pressure and mass flow rate.

Special Considerations for Large Gas Turbines

As gas turbines have become larger in output, and physical dimensions have grown considerably, the ability to effectively distribute droplets across a large inlet plenum and uniformly wet the blading has become a very important consideration. Due to the longer distances involved, gravity forces on the droplets become more significant and may create a deviation in the desired trajectory. Details are provided in Jeffs (1992, 2003) and Chellini (2004).

A new type of online nozzle that minimizes gravity effects and air mass flow distortion is shown in Figure 39. The nozzle produces a flat profile water spray that is "sandwiched" between two high velocity flat profile air sprays. These dual air sprays protect the water spray trajectory, enabling it to break through the boundary layer into the main air stream and thereby preventing premature deflection of the droplets. The nozzle is a low pressure, low mass flow design. It operates at 4 bar pressure (58 psi) and produces droplets in the size range of 50 to 250 microns. A nozzle in operation at base load on a Frame 9E gas turbine is shown in Figure 40.

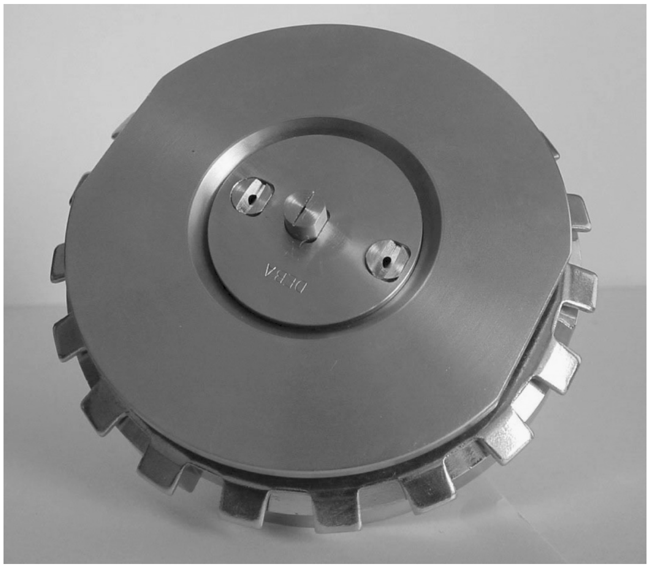

Figure 39. Online Wash Nozzle with Air Curtain, Developed for Large-Sized Gas Turbines. (Courtesy Turbotect Ltd.)

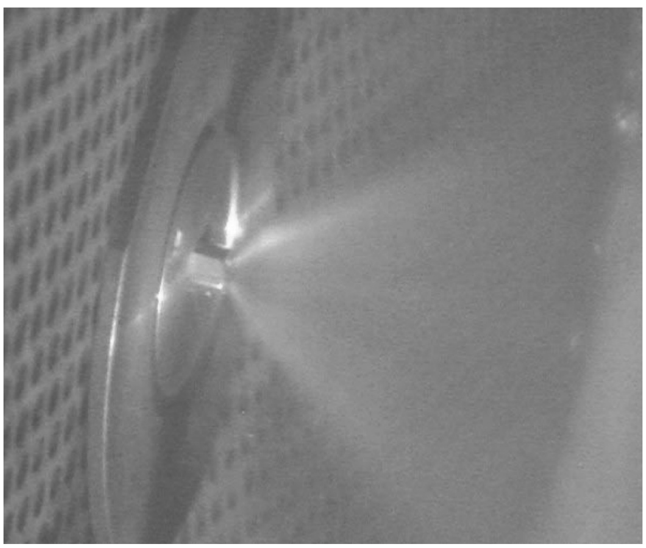

Figure 40. Online Wash Nozzle with Air Curtain, in Operation at Base Load on a Large 123 MW Gas Turbine.

In order to observe and verify the behavior of wash trajectories, it is a good idea to install a viewing window in large gas turbine ducts. One such window and associated lighting arrangement is shown in Figure 41.

Some new, large, advanced gas turbine models utilizing transonic zero stage blading have experienced erosion problems on the leading edge root area of R-0 blades. The erosion problem has been attributed to online washing, and has resulted in the OEM restricting the total number of "wash hours" that can be accrued. The type of online wash system involved can be classified as a "high mass flow" system, with a design operating pressure of 6.9 bar (100 psi). A corrective action program is underway and may possibly result in a redesign of the online water wash system. As an interim remedial measure, water injection mass flow rate and system operating pressure have been reduced for these particular gas turbine models.

Compressor Cleaning with Demineralized Water

As mentioned previously, compressor-fouling behavior is highly plant specific, and it can be very misleading to generalize about the

Figure 41. Illuminated Viewing Window on a 123 MW Gas Turbine.

efficacy of a particular wash program. For example, the use of demineralized water without detergent may prove to be an effective way to clean the compressor in some specific locations, but it would be wrong to state that this is a recommended method for all gas turbine sites. Using water as the online cleaning media will be effective if fouling deposits are totally water-soluble and/or water-wettable. However, if fouling deposits contain some quantity of oil or grease, water alone will not be effective. Foulants containing even trace quantities of hydrocarbon need the surfactant action of a detergent to break surface tension forces and lift the deposit from blade surface.

As a practical matter, and at the risk of "generalizing," the authors believe that there are few engines in the overall population that will have totally water-soluble or water-wettable deposits, and most operating environments are likely to involve some quantity of oil and grease. Using water exclusively as the cleaning media will typically result in very short time intervals between washings, in order to avoid any deposit buildup at all. Also, the use of demineralized water alone can be detrimental in cases where deposits are water insoluble, or where deposits are a combination of water-soluble and insoluble compounds. This is because the insoluble material will not be removed, which allows a greater buildup of deposits on the front stages.

Environmental Impact of Online Washing

It is recognized that online washing will create a small increase in CO emissions due to disturbance of the combustion conditions associated with the injection of water. This should not, however, be misinterpreted as a reason *not* to perform online cleaning. Any emission increase will be short-lived (only for the duration of the wash and rinse cycle), and is normally classified as a "transient condition" that can be addressed in the operating air permit in a similar way to startups and shutdowns. Low mass flow online injection systems will have less impact on CO emissions than high mass flow designs, because less water is injected. Also, the use of a detergent during online washing can be expected to have an insignificant impact on emissions, compared to the effect of water alone and the quantity of hydrocarbon fuel consumed.

COMBINED ON AND OFFLINE WASHING APPROACHES—FIELD TEST FINDINGS

Optimum Washing Regime for Axial Flow Compressors

An optimum-washing regime would combine on and offline cleaning to minimize power deterioration and extend the operating period between outages required for crank washing. Depending on local site conditions, the number of offline washes may be reduced to two or three per year, compared to four or more per year without online cleaning. Information on some important field studies involving combined online and offline wash programs is presented below.

Combination of Online and Offline Washing—
Tests on 2× 30 MW Gas Turbines

While much of the information on the best approach to on and offline washing is anecdotal, the first detailed and scientific investigation covering a long operating period was conducted by Stalder and van Oosten (1994) and Stalder (1998). This study ran for over 4000 operating hours in the mid 80s at a 100 MW combined cycle plant in the Netherlands (2× 30 MW gas turbines + 1× 40 MW steam turbine). Salient environmental and operational conditions included:

- The plant is situated 60 km (37.3 miles) from the sea.
- A very busy motorway crosses over the canal near the plant.
- Local industries include chemicals and food processing. Air pollutants include dust, salts, and fine aerosols.
- The gas turbines ran on natural gas (no fouling of the hot section).
- All power measurements were made with the gas turbines running in temperature control mode at base load.

Results showing power output changes over time (corrected to new and clean guaranteed conditions) are shown in Figure 42.

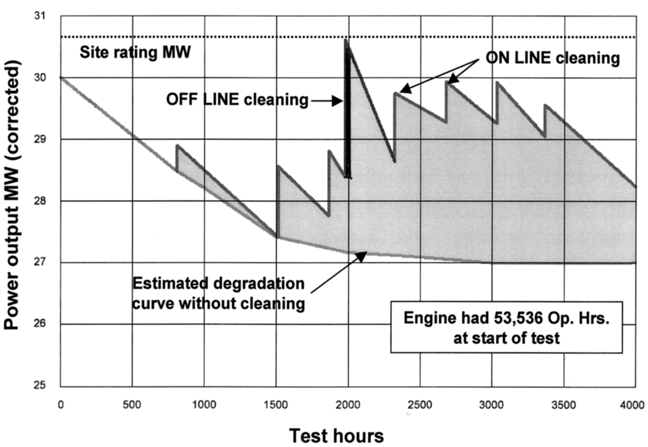

Figure 42. 4000 Hour Operating Test with Online and Offline Cleaning on a 30 MW Gas Turbine. (Courtesy Turbotect Ltd.)

Key findings of this study included:

- Without cleaning, the power output degradation tended to stabilize itself with increasing operating hours. It was confirmed, in the unit tested, that the power degradation stabilized at 90 percent base load (reference new and clean).
- Power recovery after an offline crank wash (soak and rinse wash) was significantly higher than after an online wash.
- Online washes were performed at time intervals in the range between 700, 350, and 120 operating hours. Results showed that plant performance is significantly improved with shorter online washing intervals as these prevented incremental power degradation.
- The combination of both crank and online washing methods was found to be the most effective and economical.

Improvements with Shorter Online Washing Intervals
(Six Month Field Study on 66 MW Gas Turbine)

In the Spring of 1990, a second long-term field test was jointly conducted on a 66 MW natural gas fired turbine operating in a combined cycle plant located at the same site. The tests were conducted over 18 months under the combined online and offline wet cleaning regime, from 18 May 1990 to 18 November 1991. During the entire test period, the gas turbine unit operated for a total of 8089 hours. An outage for a major overhaul (at 26,408 operating hours) took place after 3915 operating hours since beginning the test. The tests aimed to give some comparative indications of the effectiveness of more frequent online washing as applied to a new machine, and to one that operated for several years. The air inlet filtration system consisted of a weather louver, a first-stage coarse filter, and a second-stage fine filter. Details are reported in Stalder and van Oosten (1994) with salient points highlighted below:

- Online compressor washes were performed on average every four days at base load, with the gas turbine on temperature control mode. Gas turbine performance was measured before and after each wash. The pattern of the corrected power output is shown in Figure 43.

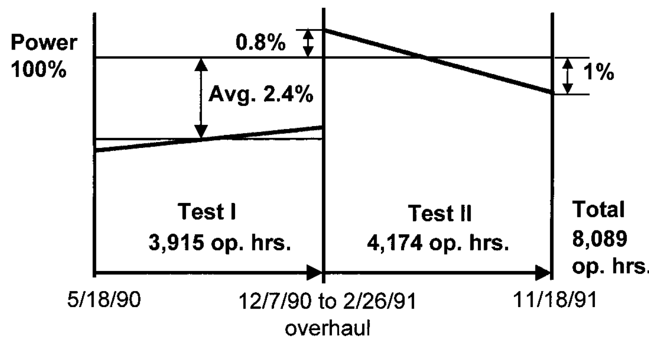

Figure 43. Summary of Corrected Power Output with 8000 Hour Test on a 66 MW Gas Turbine with Online and Offline Washing. (The gas turbine was overhauled after 3915 operating hours.) (Courtesy Turbotect Ltd.)

- Observations showed a sustained high output level close to the nominal guaranteed rating, despite severe atmospheric conditions.
- In the first evaluation block of Figure 43, the gas turbine was operated at a load factor of 97.6 percent or 2.4 percent below the original guaranteed site power output at new and clean conditions. During this period 38 online compressor washes were performed, at an average frequency of one every four days. In addition, three offline washes were performed by taking the opportunity whenever the gas turbine plant was shut down for a few days. This was after intervals of 760, 2435, and 605 operating hours. The average increase in power output after an offline wash was approximately 1800 kW. The trend (power versus time) of the performance tests made during this period is nearly horizontal; showing that ageing due to mechanical wear and tear of the gas turbine (nonrecoverable deterioration) had already stabilized.
- In the second evaluation block of Figure 43, the gas turbine started in a practically new and clean condition as the result of work done during the major overhaul. The corrected results of the compressor wet cleaning regime in the second evaluation block show that the gas turbine operated for 4174 hours at a load factor of 100.16 percent or 0.16 percent above the original guaranteed site power output at new and clean conditions. At the end of this period the number of operating hours of the gas turbine was 30,725. During this second period, 45 compressor online washes were performed, again, at an average of one every four days. In addition, two offline washes were performed, one after 1143 and the second 1381 operating hours later. The average power output increase after each offline wash in the second period was approximately 1 MW.

Findings of the Study

The following findings were reported by Stalder (1998):

- Out of the 83 online washes made during the total testing period covering 8089 operating hours, 87 percent (i.e., 72 online washes) demonstrated positive power recovery.
- The average power output recovery measured after an online wash was 712 kW. This relative small amount represents approximately 1 percent of the nominal gas turbine power output.
- No definitive results on efficiency improvements could be demonstrated because of incomplete data over the testing period with regard to fuel gas analysis and densities to determine the lower heating value, which is required for accurate heat balance calculations to derive corrected efficiencies and turbine inlet temperature.
- Power recovery due to offline cleaning if the unit's performance is close to nominal guaranteed values is not as significant as an online wash under deteriorated conditions.
- Online and offline cleaning is complementary.
- Shutdowns and startups can positively affect compressor fouling by causing deposits to spall-off. The deposits may soak up humidity during a standstill and the swollen material will partly spall-off as the compressor is accelerated during gas turbine start up.
- The test program confirmed that frequent online cleaning extended the time interval between offline cleaning operations. This is a real benefit to the operator, because the scheduled down time allowed for maintenance can be reduced if the frequency of offline cleaning with its associated cooling down time can be reduced.
- The results demonstrated that a combined online and offline washing regime can effectively be applied to new or old engines. The unit performance measured in the second evaluation period shows that the power output trend most probably followed the ageing of the gas turbine.

In terms of power recovery and economic return, the optimum program is likely to be a combination of regular online and periodic offline wet cleaning, coupled with the use of appropriately designed equipment systems and effective chemical detergents. Many base load power plants now adopt this combined cleaning regime and experience proves that the concept is highly successful in terms of cleaning efficiency and restored power.

To summarize: an optimum cleaning program is site-specific, and will be influenced by the severity of compressor fouling, the magnitude of associated power losses, and the operating schedule of the particular plant. Program parameters and options to be considered and evaluated include the following:

- Cleaning technique, i.e., online, offline, or combined online/offline wet cleaning. For most plants the combination technique appears to be optimal.
- Frequency of each online and offline cleaning cycle.
- Duration of each cleaning cycle and total mass flow of injected cleaning fluid.
- Duration of the rinsing or flushing cycle and the amount of rinse water used.
- Selection of the most appropriate water-based or solvent-based chemical detergent.

The possibility of ice formation during cold weather washing operations should not be overlooked, and the use of an approved anti-icing agent can be considered.

COMPRESSOR WASH SKIDS

Wash skids come in a wide variety of designs, engineering concepts, and sizes. Some utilize pumps and others use compressed air. Skids can be designed for only offline washing, or for combined offline and online duty. It is important that the design of the wash skid be integrated with the nozzle systems installed on the gas turbine, so that wash fluid and rinse water are delivered at the correct pressure, temperature, and flow rate. It is also important that the user make a design review of the skid from an operability and reliability standpoint.

Some important considerations are as follows:

- Skids should be designed to provide sufficient storage capacity of cleaning fluid (i.e., water-detergent mixture) and rinse water to enable at least one complete offline or online washing operation on one gas turbine, without the need to refill the storage tanks. In most cases the volumes required for offline cleaning will exceed those for online cleaning, so the crank wash volumes will normally determine tank sizing.
- Skids can be designed to serve a single gas turbine or multiple units on the same site. If multiple gas turbines are involved, outlet manifolds for connection to offline and online piping to each gas turbine should be included. If possible, skids should also be located so that piping runs to each gas turbine are approximately equal.
- Skids can be designed with a variety of control options—for example, "local manual" control for offline washing, and fully automatic control (local and/or remote) for online washing. Note that for automatic online washing, a control system can be provided so that the wash sequence is initiated via a signal from the plant control room.
- The complexity of the wash skid control system can be a fairly major cost factor, and users should determine up front whether options for fully automatic operation are really required.
- Certain gas turbine models need an external compressed air source for opening and closing shutoff valves and drain valves, as required for offline washing. In this case, wash skids can also be specified to include an air compressor system.
- Transfer of wash fluid and water to the nozzle systems can be facilitated by an electrically driven pump, or by compressed air. In the case of compressed air systems, the equipment must be designed/sized to enable complete wash and rinse cycles to be performed without interruption and without pressure fluctuations. It is also extremely important that compressed air used for washing systems is absolutely oil-free. It is not uncommon to find situations where oil is being introduced into the compressor via an air-driven wash system.
- Some OEMs recommend that offline washing be performed with hot water, and in this case the skid must be designed with a heating system, which could be steam or electrical. This requirement greatly increases the physical size and cost of the wash skid. The purpose of heating the offline wash fluid is to reduce gas turbine cool down time, but it is debatable as to whether cleaning efficiency is actually improved.
- In cases where several gas turbines cannot be served by one centrally located stationary wash skid, and as a less expensive alternative to installing multiple skids, a mobile wash cart is often considered. Wash carts, however, have limited storage capacity for cleaning fluid, and usually have no storage tank for rinse water, so more filling and handling time is required. As a general rule, most operators consider wash carts to be inconvenient, and certainly more man-hours are required for compressor washing than with stationary skids.
- Other considerations that apply to both wash skids and mobile carts include: materials of construction, safety features, reliability, pressure testing (especially for high pressure systems), hose couplings, interconnecting piping, maintenance and operating parameters, etc. The handling, storage, and mixing of chemical detergents (usually supplied as concentrates) also need to be considered.

A typical online/offline wash skid designed to accommodate 5× heavy-duty gas turbines is shown in Figure 44.

Figure 44. Typical Compressor Wash Skid.

COMPRESSOR WASH FLUIDS AND DETERGENTS

Detergents greatly enhance the removal rate of compressor deposits, compared to the use of water alone. Their function is primarily to reduce surface tension forces between deposits and blade surfaces (and between individual deposit particles), and also to emulsify oil and grease components. Actual removal mechanisms are likely to involve both dissolution and emulsification (depending on the nature of the deposit), coupled with the centrifugal forces and air mass flow effects within the compressor.

Most fouling deposits are mixtures of water-soluble, water-wettable, and water insoluble materials. Water-soluble compounds can cause corrosion, since they are usually hygroscopic and will absorb moisture and acidic airborne pollutants such as SO_x and NO_x. In fact, pH values of 4 and lower can often be measured in compressor blade deposits. Soluble chloride salts are often found, and can promote pitting corrosion. They can be rinsed, but they can also be embedded within water insoluble compounds. Water insoluble compounds are mostly organic such as hydrocarbon residues, and these materials act as "glue" to entrap and hold other foulants. All compressor deposits become more difficult to remove if left untreated, as the ageing process bonds them more firmly to the airfoil surface and thus reduces cleaning efficiency.

The benefits of using a detergent during offline cleaning are well established throughout the industry, and almost all crank wash procedures specify the use of a cleaning agent during the wash cycles. However, the benefits of using a detergent for online cleaning are less well appreciated, and in fact many of the gas turbine OEMs send "mixed signals" to the industry when confronted with this question. It can also be quite misleading to generalize on these type questions, because the nature and degree of compressor fouling is extremely site specific, and a cleaning program that is effective in one location may not work in others. Water alone can indeed be effective in removing fouling material that is completely water-soluble and/or water-wettable, but it is unlikely to have any real effect on deposits containing oil and grease. For these situations, and probably in fact for the majority of cases, a chemical detergent is needed to counteract oil and grease components so that complete removal of all the fouling material can be ensured. In fact, the authors believe that there are few engines in the overall population that will have totally water-soluble or water-wettable deposits, and most operating environments are likely to introduce at least some quantity of oil and grease. As discussed in previous chapters, the authors also believe that using demineralized water alone can be detrimental in cases where deposits are a combination of water-soluble and water-insoluble compounds. This is because the insoluble material will not be removed, thus allowing a greater buildup of deposits on the front stages. Provided a detergent meets the strict quality specifications established by the OEM for online compressor cleaning, the authors believe there is no technical reason to limit or restrict its use.

There are many different brands of compressor cleaners available on the market, but generally they are classified as being either "*water-based*" or "*solvent-based*" products.

Traditionally, solvent-based cleaners have been recognized as being more powerful in terms of oil and grease removal, but their popularity within the industry has now declined because of environmental and safety considerations. During an offline crank wash, for example, the discharged effluent water will usually require treatment before disposal and may be classified as hazardous waste. Also, due to the presence of hydrocarbon solvents, this type of cleaner may be subject to transportation restrictions and/or higher freight rates.

Water-based cleaners are now more widely applied within this industry, and are generally preferred because of their safe handling properties and high level of biodegradability. Many water-based products can be highly effective in removing a wide variety of foulants, and provide acceptable cleaning performance in most situations. New-generation water-based formulations are also available that can match the cleaning efficiency of solvent-based products.

Compressor cleaners must comply with various industry standards (for example military (MIL) -specs and environmental regulations), as well as quality and use specifications established by the various gas turbine manufacturers. General requirements include:

- Low trace metal impurity content (extremely important for online applications).
- Thermal decomposition properties that ensure gums or sticky residues do not form during the cleaning operation. (Again, this is particularly relevant for online cleaning applications.)
- Approximately neutral pH value.
- High cleaning performance, which can be evaluated via several established testing procedures.
- Long-term storage stability.
- Compatibility with alloys, coatings, paints, and elastomers within the gas turbine engine.

Many detergents are sold as concentrates, and should be used at an appropriate dilution ratio as recommended by the manufacturer. A typical dilution ratio is one part detergent to four parts water.

Online and Offline Compressor Cleaners

Cleaners available on the market today are generally nonionic, and designed to fulfill gas turbine engine manufacturer's specification for both online and offline cleaning. This also simplifies stock keeping and onsite handling. The main constituent of a cleaner is its surfactant (surface active agent), the purpose of which is to reduce surface tension of the cleaning fluid to enable it to wet, penetrate, and disperse the deposits. The surfactant also reduces surface tension of oil and grease films, which otherwise have strong adherence to blade surfaces and to other fouling materials. Such rapid penetration and dispersion of the deposits cannot be achieved with water alone. Surfactants are therefore needed for water insoluble deposits (both liquid and particulate types), to enable their removal from compressor blade surfaces and to prevent redeposition.

Foaming of Compressor Cleaners

Foam is an indication of the degree of activity of the compressor cleaner, and therefore of the effectiveness of the surfactants used in the cleaner formulation. Foam helps to achieve a better distribution and penetration of the cleaning solution into the deposits during offline washing, and it extends contact time by holding moisture. Water films alone will tend to drain off the blades more rapidly, thus reducing contact time during the offline soaking period. In addition, the foam also acts as a dirt carrier, to help transport the removed foulants from the compressor. Foam should be quickly displaced during rinsing, but if it dissipates ("collapses") *too* rapidly it can release and redeposit dirt within the compressor. So-called "low-foaming" cleaners can reduce the time required for rinsing after an offline wash, but insufficient foam or foam that collapses too rapidly can also impair the cleaning process.

Corrosion Inhibitors

The concept of including a corrosion inhibitor in compressor cleaner formulations originated primarily in the aero industry, and may be important for jet engines that are frequently not reoperated (i.e., dried) immediately after compressor crank washing. This philosophy spread to the industrial gas turbine market, and as a result, many compressor cleaners used for stationary gas turbines are also required to include a corrosion inhibitor in order to comply with certain industry specifications. Conductivity or pH measurements on offline effluent water have shown that corrosive salts are often left behind to some extent after the cleaning cycle with detergent, and in this regard the corrosion inhibitor can provide some degree of protection. However, rinsing with water alone will also be beneficial in removing residual salt solution, and it is important that sufficient rinse cycles are performed. The authors also recommend that a "blow out" and "dry out" run is conducted after completing the rinse cycles, even if the cleaning agent contains a corrosion inhibitor.

Corrosion inhibitors have a high affinity to surfaces and will tend to form a film on compressor blading. However, there is a potential risk during online cleaning that this film may decompose in the midstage temperature range of 200°C (about 390°F). The authors therefore recommend that a demineralized water rinse of the same duration as the cleaning cycle is performed to eliminate any salts left behind. As discussed in previous sections, this does not pose a risk to the gas turbine provided online washing is not applied to heavily fouled units.

Use of Hot Water for Crank Washing

Some OEMs recommend the use of hot water for offline washing, and primarily this is to reduce the cool down time before crank washing can commence, so that thermal shock is prevented. However, this means that compressor wash skids need to be equipped with a heating system, insulated tanks, and insulated piping, etc., and this significantly increases acquisition costs. Equipment maintenance costs and operating costs are also increased, and offline washing must be scheduled in advance to enable heat up time.

In addition to reducing cool down time, it is conceivable that hot water may improve cleaning efficiency by helping to soften deposits and increase solubility. It is arguable, however, as to whether or not this improvement is significant, and probably more frequent washing (combined online and offline regime) and the use of a good detergent will more than compensate for these possible benefits.

Engdar, et al. (2004), recently addressed this subject, and conducted CFD studies on hot water injection for offline compressor washing. The authors concluded that, regardless of its inlet temperature, injected wash fluid is cooled down to ambient air temperature well before the spray reaches the inlet guide vanes. This study indicates that heating the wash fluid may serve little benefit.

COLD WEATHER COMPRESSOR WASHING

The ability to perform compressor washing during cold weather periods has become increasingly important for many gas turbine operators and OEMs. Occasions when it may be required are as follows:

- Offline cleaning prior to an important performance test, regardless of ambient conditions.
- Scheduled outage for offline cleaning may occur during winter months and cannot be postponed.
- Plant decisions to continue an online cleaning program during winter months.
- Plant decisions to perform online cleaning in order to delay an outage for offline cleaning .

In the case of online cleaning it is important to remember that ice formation can occur in the compressor at ambient temperatures significantly above the freeze point. This is due to a static temperature depression in the bellmouth area, resulting mainly from air acceleration effects. The amount of temperature depression is related to specific design parameters of the compressor and can be as high as 18°F (10°C). Thus, depending on the particular model of gas turbine, conditions at the IGVs can be 18°F (10°C) lower than actual outside air temperature. The influence of static temperature depression is less significant with offline washing due to relatively low crank speeds, but a few degrees of safety margin should still be allowed.

Various chemicals have been used as anti-icing agents—such as alcohols, ketones, and glycols. However, not all are suitable for gas turbine applications, and a summary of issues and selection criteria are provided in Table 10. Antifreeze agents based on propylene glycol offer significant safety and handling advantages (nonhazardous, high flash point, biodegradable, etc.), and can be used for both online and offline cleaning applications. In essence they are used as a substitute for water during the cleaning and rinsing cycles. Using this type of anti-icing agent, online cleaning can be performed in ambient temperature conditions down to about 14°F (−10°C) and offline cleaning can be conducted at ambient temperatures down to about −8°F (−22°C). Actual product freeze points can vary, however, depending on the specific formulation, so manufacturers' product information should always be consulted. Cold weather washing procedures and minimum allowed ambient temperatures also vary according to the different gas turbine manufacturers and for different gas turbine models, so OEM recommendations should always be observed.

Table 10. Issues with Anti-Icing Agents for Low Temperature Compressor Washing.

ANTI-ICING AGENT	ISSUES AND COMMENTS
Alcohols (methanol, ethanol, isopropanol)	• Main problems are safety related; i.e. low flash point, volatility and evaporation risk • Explosion-proof equipment and area rating is required
Ketones (acetone, methyl ethyl ketone)	• Even greater safety risks due to low flash point
Ethylene glycol	• Does not present a flash point problem, but forms sticky deposits on compressor blades
Propylene glycol	• Non-hazardous, high flash point material • Non-toxic and biodegradable • Does not form sticky deposits during on-line cleaning • Generally recognized as being the most suitable for gas turbine applications • Propylene glycol antifreeze agents are available with freeze point down to -18°F (-28°C) • Can be mixed with certain compressor cleaners to provide the wash fluid, or used "as delivered" for compressor rinsing

CLOSURE

This paper provides a comprehensive overview of compressor fouling and washing for gas turbine engines and presents, in a single document, the state-of-the-art regarding the causes, effects, and control of axial compressor fouling. The paper has covered numerous practical aspects of direct applicability to users.

With increasing fuel costs and a highly competitive market, the understanding, measurement, and control of fouling deterioration is an imperative. Fouling rates can vary from plant to plant and are highly environment and machine specific. Furthermore, fouling behavior is influenced by inlet air filter selection and maintenance. Site weather patterns also have a dramatic impact on fouling. A judicious combination of offline and online cleaning usually provides the best results in helping operators fight this common and insidious operating problem. Close monitoring of compressor performance can help optimize compressor-washing regimes and improve plant profitability.

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ACKNOWLEDGEMENTS

The authors acknowledge the important work of Jean-Pierre Stalder of Turbotect Ltd., Switzerland, on his series of seminal papers on compressor fouling and washing, including his 1998 ASME paper that won the John P. Davis Award. His extensive and careful experimental studies on compressor fouling and washing on operating gas turbines is unique as to date, most of the experience and pronouncements relating to fouling and washing are anecdotal. Many of his key technical findings and results are provided in this paper. The authors also acknowledge the work of Dr. Tarabrin for his work in conjunction with J.P. Stalder on sensitivity factors for gas turbine compressor fouling. They thank Terryl Matthews of the Turbomachinery Symposium Advisory Committee who acted as paper monitor, and Joanne Burnett, Editor and Technical Sessions Coordinator of the Turbomachinery Laboratory for her help and support in getting this paper to print. Opinions stated in the paper are the personal opinions of the authors and may not necessarily represent positions of their respective companies.