FOULING OF AXIAL FLOW COMPRESSORS— CAUSES, EFFECTS, DETECTION, AND CONTROL

by

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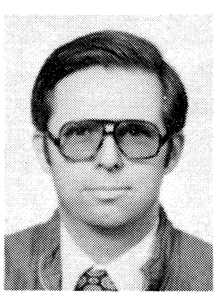

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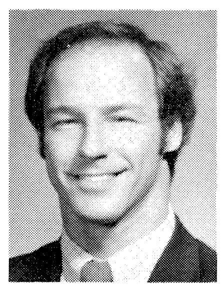

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ABSTRACT

The fouling of axial flow compressors is a serious operating problem in gas turbines and in process axial flow compressors. Gas turbines are being increasingly used in cogeneration applications and with the large air mass flowrate (e. g., 633 lb/sec for a 80 MW gas turbine) foulants even in the ppm range can cause deposits on the blading, resulting in a severe performance decrement. This is a common operating problem experienced by almost all operators of gas turbines. The effect of compressor fouling is a drop in airflow and compressor isentropic efficiency, which then manifests itself as a drop in power output and thermal efficiency. In some cases, fouling can also result in surge problems as its effect is to move the compressor surge line to the right, i. e., towards the operating line. The mechanisms are discussed for fouling, the aerodynamic and thermodynamic effects, types of foulants, detection methods, and control techniques. A brief discussion on turbine fouling is also made.

INTRODUCTION

The fouling of gas turbines is perhaps the most prevalent and perplexing problem faced by gas turbine operators. The economic consequences of fouling can severely impact operation and also affect project economic viability through lost generating ability and poor thermal efficiency. There are also peripheral problems such as blading failures and surge that can be caused by fouling.

Gas turbines being air breathing machines, ingest large quantities of air. The solids or condensing particles in the air and in the combustion gasses can precipitate on the rotating and stationary blading causing changes in aerodynamic profile, dropping the compressor mass flowrate and affecting the turbine flow coefficient and efficiency. This has an adverse effect on the unit's performance. The output of a gas turbine can drop by as much as 10 percent. Moreover, contaminated air can cause a host of problems that include erosion, fouling, corrosion, and, in some cases, plugging of the hot section cooling passages. Because of these factors, it is important for users to understand the mechanism of fouling and its aerodynamic and thermodynamic effects.

The purpose of this presentation is to provide a comprehensive, unified treatment of gas turbine fouling and to put in one source results of a wide variety of studies. An earlier but less detailed review has been reported [1]. Information provided herein is based both on experiences and on the excellent work of several professionals and researchers in this area.

The axial flow compressor has been one of the most challenging turbomachines to design. From an early 1884 patent by Charles Parsons, these compressors have been plagued by operating problems. The early machines suffered due to poor aerodynamics resulting in exceedingly low efficiencies. Parsons made an eighty-one stage machine in 1899 (η_{isen} = 70 percent). A 48 stage machine sold in 1904 is shown in Figure 1. A high spacing/chord ratio used by Parsons caused stall to occur over a large part of the operating range. The complexity of axial compressor design arises due to: (a) the flow being *against* the pressure gradient, i. e., diffusing flow; (b) a susceptibility for retardation in the boundary layer causing eddies and backflow. In the 1920-1930 decade, there were significant developments in axial flow compressor design. By 1927, a pressure ratio of about 2:1 in five stages was obtained by Betz and Encke. Also in 1927, Jackob Ackert designed a 13 stage axial flow compressor with a pressure ratio of 2:1 and an efficiency of 80 percent. This machine was constructed by BBC and used for a supersonic wind tunnel. By 1935, German scientists had a clear lead in high speed turbocompressor phenomena. While the early Whittle & Von Ohain jet engines used centrifugal compressors, by 1939, Junkers was working on the remarkable Axial Flow (JUMO 004B) Turbojet for the ME262 fighter. This unit had an eight stage axial compressor operating at a pressure ratio of 3:1 and a compressor efficiency of 80 percent. Since the Second World War, gas turbines have experienced rapid development.

Compressor fouling has been a problem affecting gas turbines right from the early days. With the new generation of highly

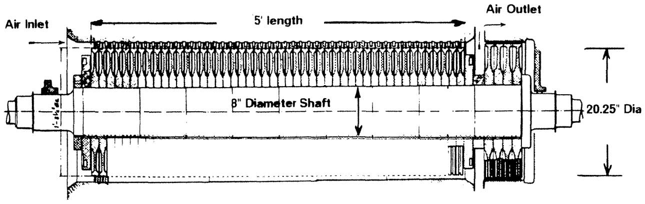


Figure 1. 48 Stage Axial Flow Compressor Designed by Parsons in 1904. Parsons also made a 81 stage machine in 1899, with exceedingly low pressure ratio/stage. The blade spacing/chord ratio was so high that stall occurred over much of the operating range. Modern day high performance compressors can operate at efficiencies in excess of 90 percent.

loaded, high temperature units, susceptibility to fouling becomes an even more important factor. The inexorable movement in fighter jet engine technology towards higher turbine entry temperatures and pressure ratios results in spinoffs of industrial gas turbines that are far more sensitive to fouling than the earlier low pressure ratio machines. The expected advancement in technology is typified by short term goals of jet engine thrust/weight ratio of 10:1 (compared to today's state-of-the-art of 8:1) and long term goals of 20:1. The use of controlled diffusion airfoils will proliferate. These can deal with endwall losses which account for a major proportion of the losses. Firing temperatures will approach stoichiometric levels. The introduction of new generation high performance gas turbines, like the GE Frame 7F (135 MW) and the Frame 9F (200 MW) and interesting combined cycle development programs, such as the Japanese Moonlight project (gas turbine rated at 122 MW ISO, pressure ratio 55:1 with intercooler and reheat features) are indicative of these trends. With large gas turbines entering combined cycle utility service, possibly operating on coal derived fuels in IGCC applications, *it will be exceedingly important to operate with the axial compressors as clean as possible.* Several of the larger machines in the market utilize transonic early stages as a means for increasing the air mass flow.

These new developments, coupled with the economic need for efficient operation, will call for an increased emphasis on fouling detection and control. Fouling is a major cause of performance and cycle efficiency loss in gas turbines. Some estimates have placed fouling as being responsible for 70 to 85 percent of all gas turbine performance loss accumulated during operation. Output losses between two percent (under favorable conditions) and 15 to 20 percent (under adverse conditions) have been experienced. It is important to note that in a gas turbine about 60 to 65 percent of the total work produced in the turbine is utilized by the compressor. Hence, high compressor efficiency is an important contributing factor for high cycle thermal efficiency.

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—fly ash, hydrocarbons, smog, etc. This causes a grimy coating on the early stages and can get "baked on" in the latter stages (especially in high pressure ratio compressors).
- ingestion of gas turbine exhaust or lube oil tank vapors.
- mineral deposits.
- airborne materials—soil, dust and sand, chemical fertilizers, insecticides, insects, and plant matter.
- internal gas turbine oil leaks—axial compressor front bearing is a common cause. Oil leaks combined with dirt ingestion cause heavy fouling problems.
- impure water from evaporative coolers.
- coal, dust and spray paint that is ingested.

The fouling rate for a compressor will be a strong function of the environment, the climatic and humidity conditions, the wind direction, and the filtration system.

A scatter plot showing the high air ingestion rates for 81 gas turbines is featured in Figure 2. Taking an average of say 33 lb/kW/hr, pollutants even in the ppm range will account for several

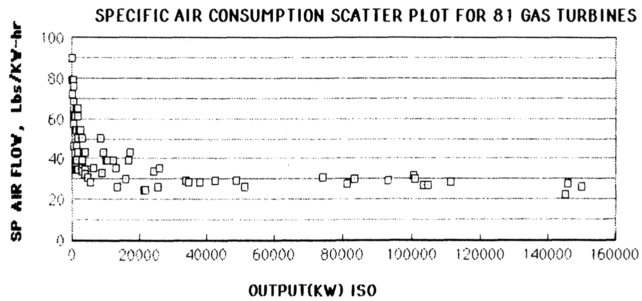


Figure 2. Scatter Plot Showing Specific Air Flow (lb/KW-hr) for 81 Gas Turbines. Data was plotted based from published gas turbine data. Specific air flow is around 30 lb/kW-hr for turbines in the 20-100 MW range. There is significant clustering in the low power regions (points overlayed).

hundred pounds being ingested in 24 hours. For example, a 75 MW unit located in an industrial environment with air loading of 10 ppm will ingest 594 lb of particulates in a day.

Ambient air can be contaminated by solids, liquids and gases. Air loadings can be defined in mg/m^3, grains/1000 ft^3 or ppm (mass of contaminant per unit mass of air). Results of an EPA survey taken at 400 locations in the U. S. are shown in Figure 3 [2]. Curve A shows the percentage of sites exceeding a given dust loading 50 percent of the time. Curve B shows the percentage of test locations exceeding a particular dust load 10 percent of the time. It can be seen that most locations have loadings less than 0.1 ppm. Site specific conditions can, however, vary significantly. In general, particles up to 10 microns cause fouling, but not erosion. Particles above 10 to 20 microns cause blading erosion.

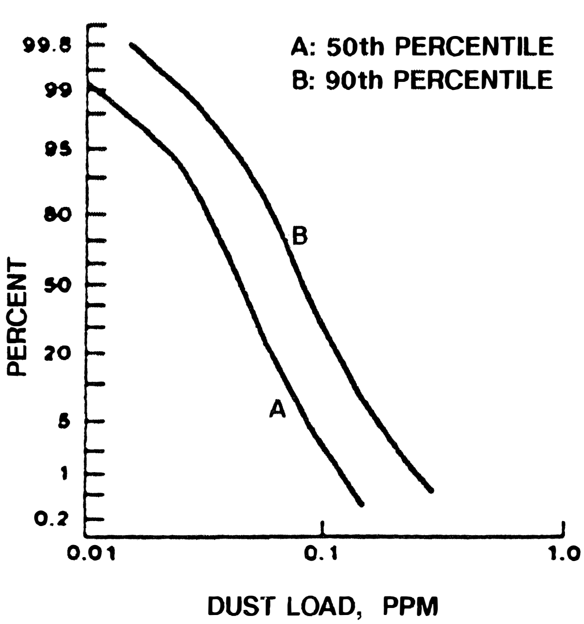


Figure 3. Percentage of U.S. Sited Exceeding a Given Dust L oading. This is based on a survey. Site specific conditions must always be considered.

Some typical air loadings as reported by Upton [3] are:

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

The tabulation provided in Table 1 shows environmental factors. A chart provided in Figure 4 shows typical particle sizes (diameter in microns) that can be present in the environment. Even aeroengines operating at "clean" cruise altitudes of 40,000 ft have incurred flameouts as a result of volcanic ash; although this is a rare occurrence. Similarly, the Fairchild A-10 ground attack aircraft gas turbines experienced severe fouling and compressor stall, due to ingestion of its 4200 round/minute Gatling gun exhaust! This problem required very frequent washing, which interfered with aircraft readiness. The problem was solved using special deflector fittings which routed the offending foulants away from the engines.

Marine and Offshore Environment

The offshore environment is particularly challenging. Some key findings as published in [4] are:

- Airborne salt can exist in three basic forms: aerosol, spray, and crystal. Aerosols can range in size from two microns to 20 microns (1 micron = 10^{-6}m). Aerosol is generated by bubbles shattering on the sea surface. Sea spray generates large droplets sized 150 to 200 microns and these tend to drop out due to gravity. Sea salt crystals absorb moisture under appropriate relative humidity conditions. The size of these peak in the range of two 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. [5], 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: (a) Flare carbon and mud burning foulants—these can be a problem with poorly positioned flare stacks and with sudden changes in wind direction. (b) Drilling cement and other dusts. These can be blown around a rig. Grit blasting has also been a serious problem. On one platform, 700 tons of grit blast was used in one year!

Evaporative Cooler Considerations

With evaporative coolers, the quality of water is important. Evaporative coolers are typically located after the filtration system and water quality is important both from blow down considerations and gas turbine ingestion. Water flowrates are typically one to two gallons per square foot of surface area of the distribution pad. Ingestion (entrainment) of water in the airstream can be contained by proper attention to the orientation of media packs, uniform distribution of water over the media surface and proper drainage. When water exceeds 40 ppm sodium and potassium, particular care should be taken to avoid carryover. Concentration of these elements in air should be less than 0.005 ppm. Vane type mist eliminators located downstream of the media are effective in preventing carryover. A good review of airborne contaminants and their impact on air, fuel, and water management has been provided by Hsu [6].

Humidity Effects On The Fouling Of Axial Compressors

Several turbine operators have noted that humidity plays an important part in fouling. An excellent series of investigations has been conducted by Zaba [7].

As air passes through the intake and filtration system, it proceeds at a very low velocity. As it approaches the compressor face, the air accelerates to a high velocity (0.5 to 0.8 Mach number). This results in a static temperature reduction of about 10°C to 15°C. It is this effect that causes icing problems even when ambient temperatures are above freezing. The saturation air temperature also drops. If the relative humidity is high enough, it is possible that the static air temperature falls *below*

Table 1. Environmental Factors.

ENVIRONMENT	*COUNTRY SIDE*	*COASTAL*	*LARGE CITIES* (Power Stations) (Chemical Plants)	*INDUSTRIAL AREAS* (Steel Works) (Petro-Chemical) (Mining)	*DESERTS* (Sand Storms) (Dusty Ground)	*TROPICAL*	*ARTIC*	*MOBILE INSTALLATIONS*
Types of Dust	Dry-Non Erosive	Dry-Non Erosive but Salt Particles. Corrosive Mist.	Sooty-Oily. May be Erosive, also Corrosive.	Sooty-Oily Erosive. May be Corrosive.	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	Dry-Erosive Sooty-Oily Corrosive
Dust Concentration								
mg/cu.m.	0.01-0.1	0.01-0.1	0.03-10	0.1-10	0.1-700	0.01-0.25	0.01-0.25	0.01-700
Grains/cu.ft. 10^3	0.004-0.0436	0.004-0.436	0.01-0.13	0.043-4.35	0.04-300	0.004-0.10	0.004-0.10	0.04-300
Particle Size	0.01-3	0.1-3 Salt <5	0.1-10	0.1-(50)[1]	1-(500)[2]	0.1-10	0.1-10	0.1-(500)[3]
Effection on G.T.	*Minimal*	*Corrosion*	*Fouling* Sometimes corrosion and fouling.	*Erosion* Sometimes corrosion and fouling.	*Erosion* Plugging of filter with insect swarms.	*Fouling*	Plugging of air intake system with snow and ice.	*Fouling Erosion Corrosion*
Temperature Range °C	−20 to +30	−20 to +25	−20 to +35	−20 to +35	−5 to +45	5 to 45	−40 to +5	−30 to +45
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.	All possible weather conditions

NOTES: 1. In emission areas of chimney.
2. During severe sand storms.
3. At track level and during dust storms.

the saturation air temperature. This causes condensation of water vapor. Dust particles then form nuclei for the water droplets and start to adhere to the blading. As the air progresses to the rear stages, it gets hotter and drier, typically causing less fouling in the latter stages. The process is depicted in Figure 5. The relationship between total and static pressure is given by:

$$T_{total} = T_{static} + (n)\ C_1^2/2C_p g_c\ J \tag{1}$$

where:

T_{total} = Total Temperature °R

T_{static} = Static Temperature

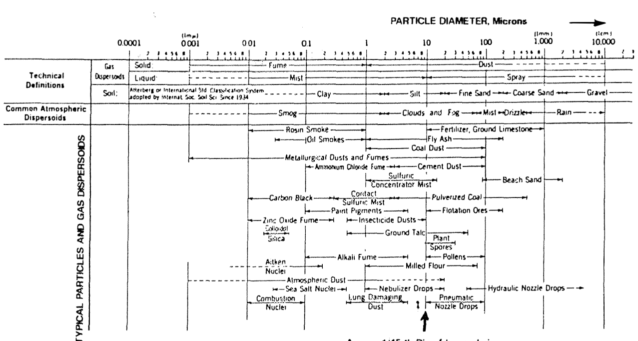


Figure 4. Characteristics of Particles in Atmosphere, Showing Their Size Range in Microns.

n = Recovery factor (0.5-0.7)

C_1 = Air Velocity (fps)

C_p = Specific Heat (BTU/lb °R)

g_c = Gravitational Constant (32.2)

J = Mechanical Equivalent of heat (778 ft lb_f /BTU)

EFFECTS OF FOULING

Fouling of axial flow compressors can have several effects:

- Thermodynamic/aerodynamic effects
- Air flow distortion and compressor surge
- Blading integrity effects
- Associated problems—erosion, corrosion, cooling air passage blockage, unbalance, foreign object damage

Thermodynamic/Aerodynamic Effects

The observable effect (manifestations) 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 of the blading which reduces air flowrate, pressure ratio and compressor efficiency.

The power developed by a turbine is given by:

$$\text{Power} = \dot{m}\ C_p\ T_3 \left[1 - \frac{1}{(P_3/P_4)^{\frac{\gamma-1}{\gamma}}} \right] \tag{2}$$

where,

$\dot{m}$ = Mass Flow Rate Through Turbine ($= \dot{m}_{air} + \dot{m}_{fuel}$)

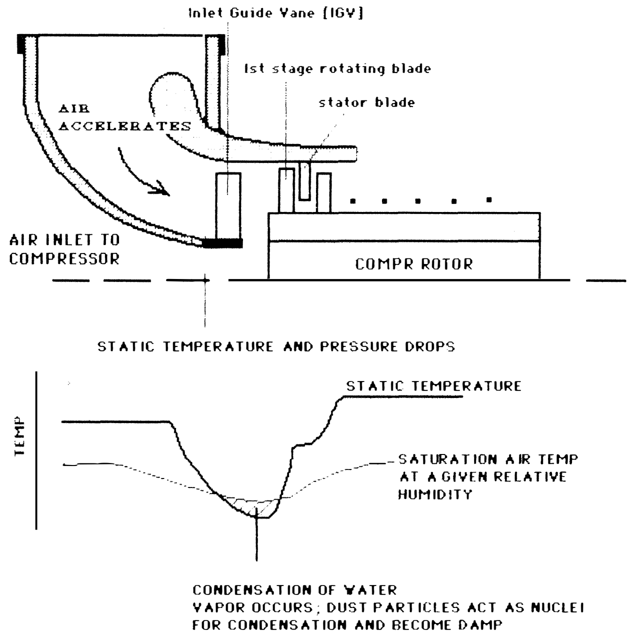


Figure 5. Humidity Effects as air Accelerates and Experiences a Static Temperature Drop at Axial Compressor Inlet.

T_3 = Turbine Firing Temperature (TIT)
P_3/P_4 = Turbine Expansion Ratio
C_p = Specific Heat in Turbine Section
γ = Ratio of Specific Heats = (C_p/C_v)

The pressure P_3 entering the turbine is the compressor discharge pressure less the pressure drop in the combustion system. As the mass flow drops and the discharge pressure from the compressor drops, (i. e., P_3 drops as $P_3 = P_2 - \Delta P$), the work from the turbine drops. This drop in power will be masked (to a point) by the control system, increasing turbine inlet temperature (T_3). Since 60 to 65 percent of the power developed in the turbine goes into driving the compressor, this leaves 35 to 40 percent for useful shaft output. As the compressor efficiency drops, it will consume even more power, thus *further* lowering the gas turbine's output. The firing temperature T_3 can be increased only up to a limit, because of hot section life reduction considerations. This process is schematically shown in Figure 6.

An axial compressor is a machine where the aerodynamic performance of each stage depends on the earlier stages. Thus, when fouling occurs in the inlet guide vanes and the first few stages, there may be a dramatic drop in compressor performance. This can often occur when oil and industrial smog or pollen are present and form an adhesive wetting agent. The early stages are often the worst fouled stages. If the rear stages foul, this seems to have a smaller impact on performance; but due to higher temperatures, deposits can get baked on and become difficult to clean. This baking effect is more severe on the high pressure ratio compressors, e. g., 18 to 30:1 pressure ratio of aeroderivative machines, as opposed to the typical 10:1 or 12:1 pressure ratios found on the heavy duty industrial gas turbines.

A fouling process such as that shown in Figure 7 occurs in a large gas turbine engine. This graph shows the changes in compressor efficiency and heat rate over time. Scatter plots taken on a large, constant speed process compressor in Figure 8 shows degradation in the flow characteristics. The lowering of the compressor characteristic line is typical of fouling.

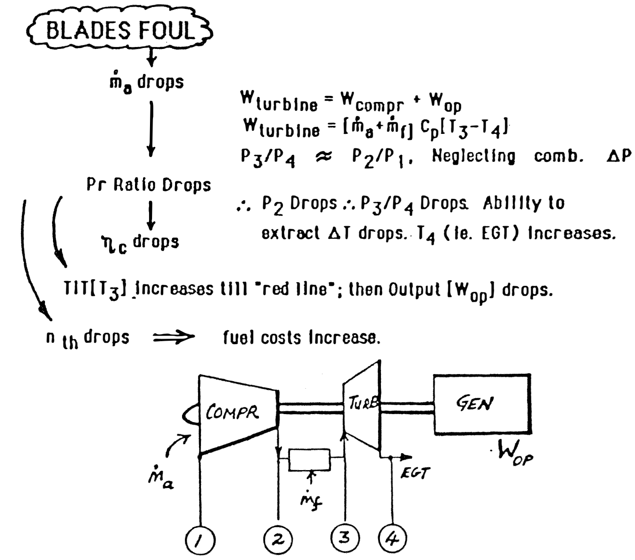


Figure 6. Schematic Representation of Fouling Process. Fouling causes a change in the compressor-turbine match point.

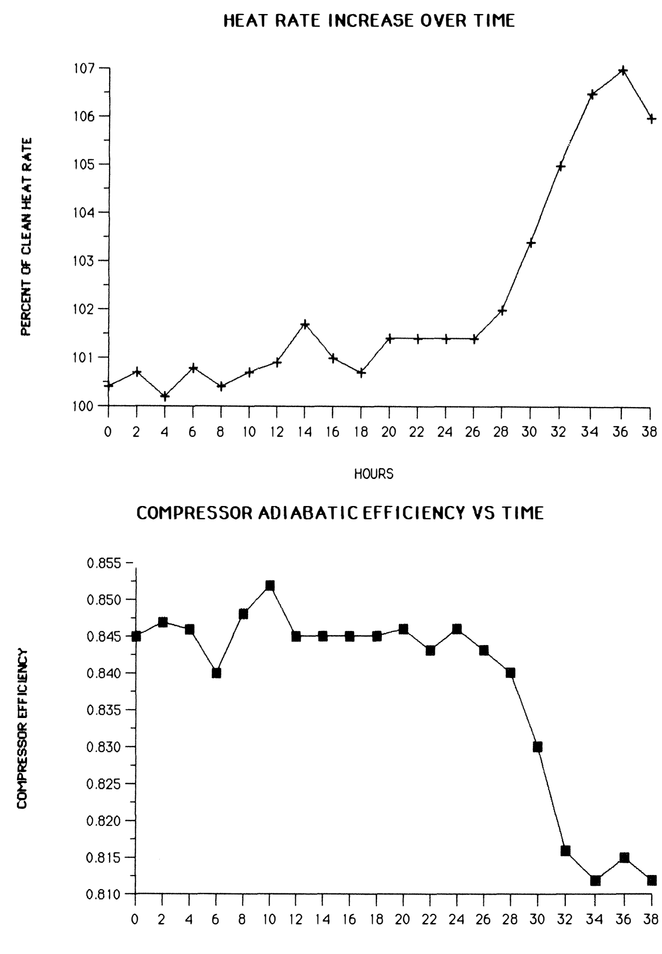


Figure 7. Effects of Fouling on an Industrial Gas Turbine.

Modelling Of Compressor Fouling Behavior—Foulants that adhere to compressor blading affect compressor aerodynamic behavior. Deposits on the blade pressure side are caught by im-

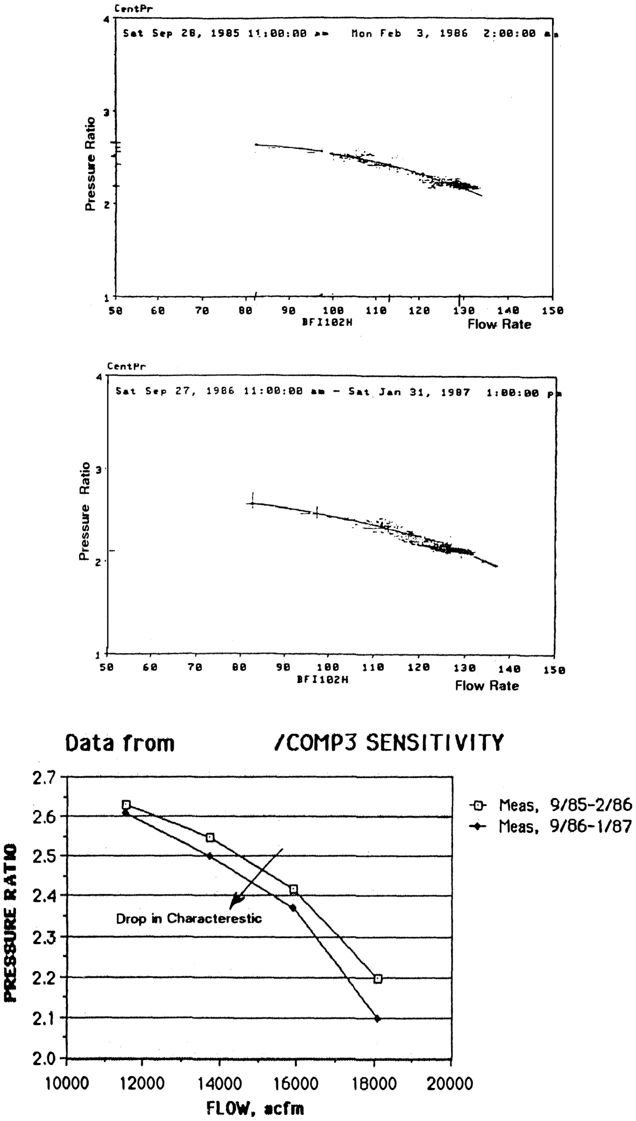


Figure 8. Drop in Characteristic Curve for a Large Compressor Due to Fouling.

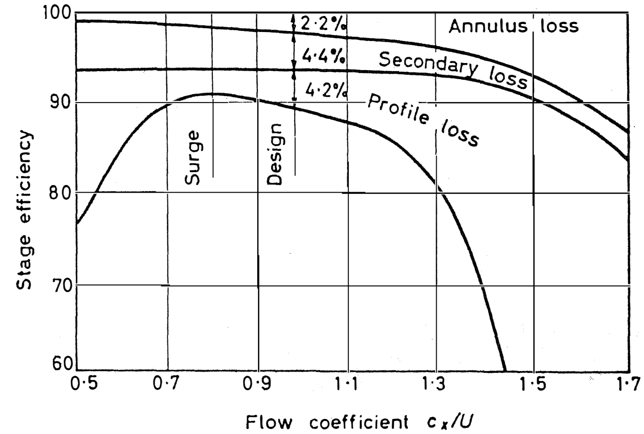


Figure 9. Losses in an Axial Flow Compressor.

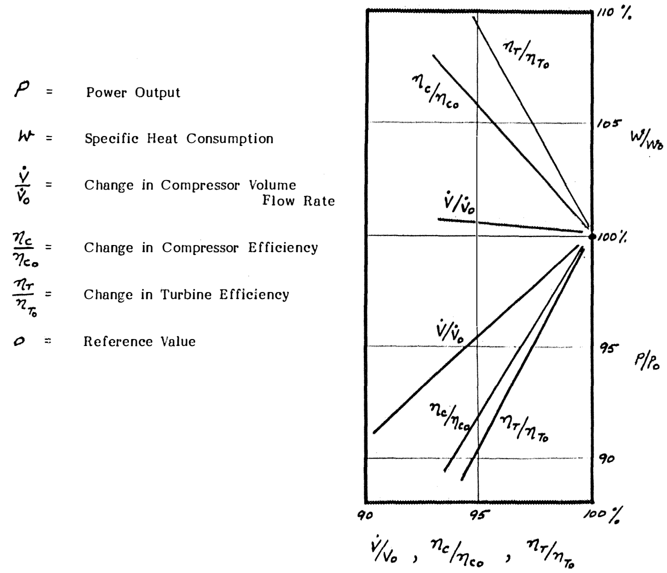


Figure 10. Change in Parameters and Effect on Power Output and Specific Heat Consumption [9].

pact, but foulants also adhere to the suction side. As air forms a boundary layer on the airfoil, there is an aerodynamic force propelling the particle to the surface causing it to deposit. Deposits on the blading leading edge can cause a reduction in operating range as the angle of attack on the airfoil is affected.

A representation in Figure 9 shows some of the losses experienced in an axial flow compressor. The losses can be broken into profile losses on the surface of the blades, skin friction losses on the annulus walls and secondary losses (associated with three dimensional flows). Thus, the deterioration of the rotor and stator profiles due to fouling will have an impact on compressor efficiency. Bammert and Woelk [8] have investigated blading surface roughness effects on axial compressor aerodynamics. Salient points and findings of the excellent theoretical and experimental work of Zaba [9] and Saravanamuttoo [10, 11, 12] are presented here as they provide insight into the fundamental aerothermodynamics of compressor fouling.

The diagram in Figure 10 was developed by Zaba [9] and shows where the change in output and specific heat consumption of a gas turbine was examined as a function of compressor volume flowrate and efficiency (of compressor and turbine). This figure, which provides a comprehensive picture of fouling behavior, is for a gas turbine with pressure ratio of 10:1 and operating at a turbine inlet temperature of 1832°F (1000°C).

Zaba relates the change in efficiency and intake volume caused by fouling by:

$$\frac{\Delta \dot{v}}{\dot{v}} = Z_c \frac{\Delta \eta_c}{\eta_c} \tag{3}$$

The factor Z_c is highly dependent on the nature and distribution of deposits and also the *distribution* of fouling in the different compressor stages. Tests with uniform fouling on all stages yielded values of $Z_c \approx 1$.

In practice, the earlier stages of an axial flow compressor are normally heavily fouled (due to humidity effects). Results shown in Figure 11 were obtained by an investigation by Zaba for a variety of fouling distributions indicated as:

0 = clean blades
1 = uniform fouling in all stages

2 = decreased fouling in later stages
3 = increased fouling in later stages [9]

The influence of compressor fouling is shown in Figure 11 for aerodynamic stage loading and efficiency. In fouling variants one (uniform fouling) and two (decreased fouling in later stages), the front stages are heavily loaded. This means that the distance of the operating point from the surge (stall) point is reduced. In fouling variant three (increased fouling in later stages), the high aerodynamic loading of the *middle* stages can cause flow separation in these stages. The latter stages themselves are relieved of load. For these calculations, it was assumed that for each individual stage $Z_c = 1$. Defining $\bar{Z}_c$ as a mean value for the full compressor given by:

$$\bar{Z}_c = \left(\frac{\Delta\dot{V}}{\dot{V}}\right) / \left(\frac{\Delta\eta_c}{\eta_c}\right) \tag{4}$$

Depending on the fouling distribution, different values of $\bar{Z}_c$ are obtained. Zaba's results (Table 2) show changes in compressor flowrate and efficiency for the different fouling variants. It can be seen that fouling of the early stages has a greater influence on the volume flowrate than the later stages. $\bar{Z}_c$ is, therefore, greater than unity.

Zaba presents a case study of a gas turbine operating in an industrial environment with the deterioration effects shown in Figure 12. In this case, $\bar{Z}_c$ has a value of approximately 2.5, indicating high fouling of the earlier stages.

Table 2. Compressor Flowrate and Efficiency for Various Fouling Distributions Compared to Clean Condition [9].

FOULING VARIANT— —	0 Clean	1 Uniform fouling all stages	2 Decreased Fouling, Latter Stages	3 Increased fouling in latter stages
Compressor Vol Flow rate %	100	95	95	99.5
Compressor Efficiency %	100	95	97.2	97.2
$\bar{Z}_k$ Factor	-	1.0	1.54	.24

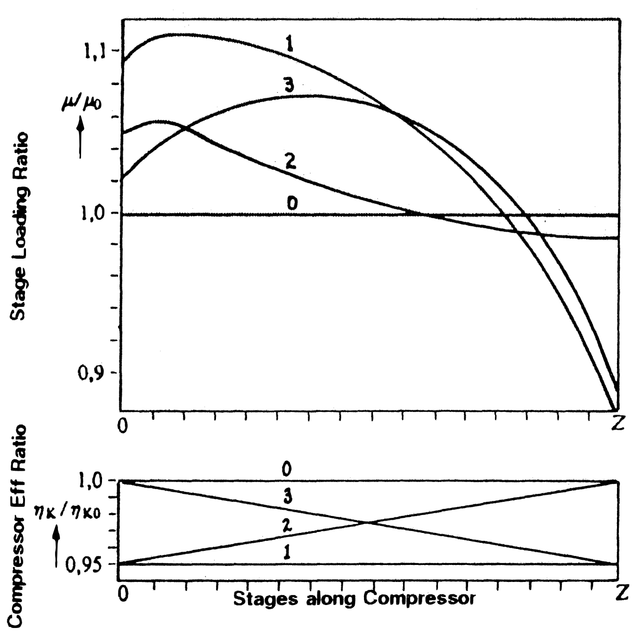


Figure 11. Specific Work Per Stage of an Axial Flow Compressor for Different Fouling Patterns [9].

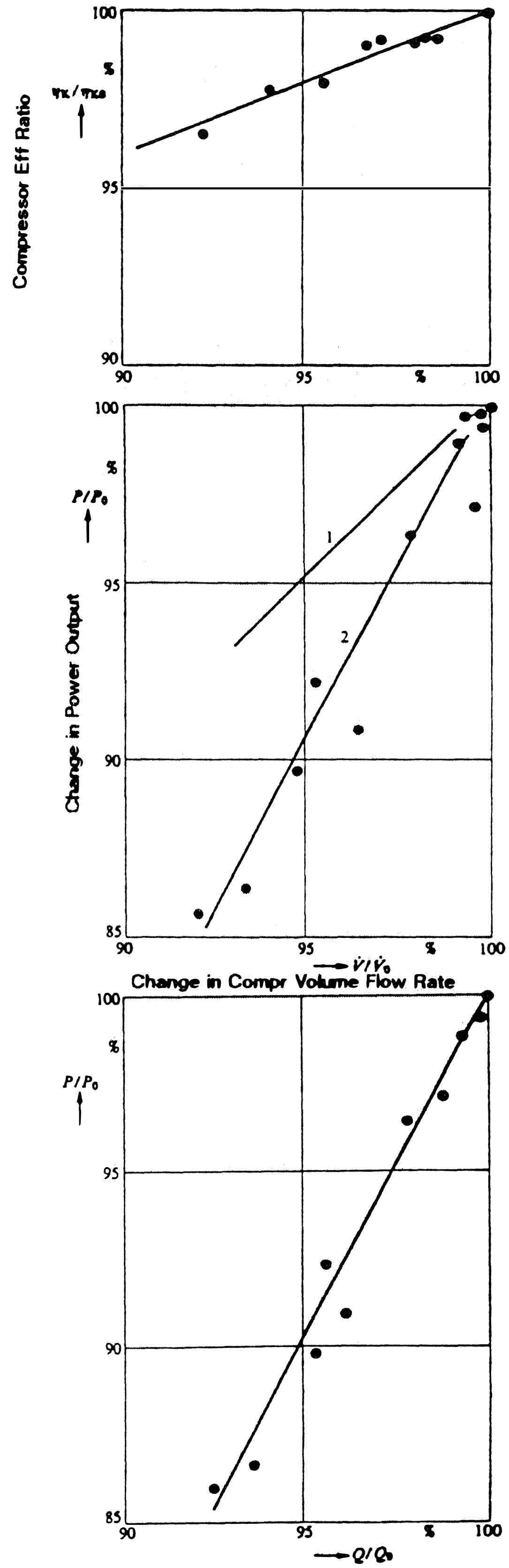


Figure 12. Measured Values for Changes in Compressor Efficiency, Power Output and Heat Consumption Due to Fouling [9].

Saravanamuttoo and his associates have applied a stage stacking model to simulate the performance of a LM2500 gas turbine and a Solar Centaur [12]. The use of stage stacking permits the performance of individual stages to be modified to model different types of deterioration.

The performance of an axial flow stage can be presented as a plot of pressure ratio and isentropic efficiency against nondimensional mass flow for a constant nondimensional speed. For a defined flowrate, inlet conditions and rotor speed, an inlet velocity triangle can be defined. Then by assuming stage performance parameters of pressure ratio and efficiency, a mean line velocity triangle can be obtained at the rotor exit. This process can then be applied to the next compression stage until all compressor stages are "stacked" together.

Fouling of compressor blading affects the profile and the amount and distribution along the blade span will affect stage performance. Obviously, a mean line stage stacking method cannot take this into account. An inlet velocity triangle is shown in Figure 13 for a clean and fouled stage. A flow reduction (effect of fouling) is represented by a change in magnitude (but not direction) of the inlet relative velocity vector. To solve the fouled velocity triangle, Lakshminarasimha and Saravanamuttoo [11] express the shift in the relative velocity vector ($\Delta\beta_{if}$) in terms of the reduction in the flow coefficient ($\Delta\phi/\phi$) and quantities of the velocity triangle of the clean stage. This expression is given as:

$$\Delta\beta if = \tan^{-1} \frac{\phi(\frac{\Delta\phi}{\phi})}{(1-\frac{\Delta\phi}{\phi})\ \left[\phi^2+(1-r)^2\right]\ +\frac{\Delta\phi}{\phi}\ (1-r)} \tag{5}$$

Similarly, in the exit triangle a reduction in flow is reflected as a change in magnitude (not direction) of the relative exit velocity. The efficiency characteristic is altered using:

$$\eta_f = \eta - K_f\, \eta_{ref} \tag{6}$$

where,

r = blade mean radius

K_f = fouling factor of efficiency

ref = design point

β = relative flow angle

ϕ = flow coefficient = C_a/U

Thus, in order to simulate fouling, two factors can be modified. These are the fouling parameter ($\Delta\phi/\phi$) and the efficiency fouling factor (K_f). Thus, by defining these two values for a stage, stage characteristics can be developed that simulate a fouled condition. Various degrees and types of fouling can be set up by choosing different combinations of factors. The results of these simulation studies carried out by Saravanamutto, et al., are summarized here.

- The effect of different first stage levels of fouling on decrease of overall compressor mass flow is shown in Figure 14.
- When fouling was applied to the latter stages (thought to occur because of "baking on" effects) the effect on compressor performance was not noticeable. The effect of fouling location on compressor flow is shown in Figure 15. The strong dependence of change in mass flow over speed is also shown.
- In general, fouling was found to reduce compressor mass flow, efficiency and pressure ratio. The effect of fouling was very dependent on its location, being greater in the early stages.
- The effect of fouling was found to be proportional to the pressure ratio of the stage at which it occurs. A compressor with

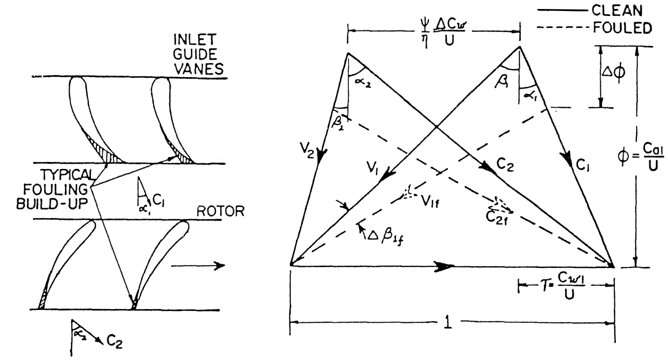


Figure 13. Axial Flow Compressor Velocity Triangles Showing Fouling Effects. Change in absolute inlet velocity results in an incidence angle change [11].

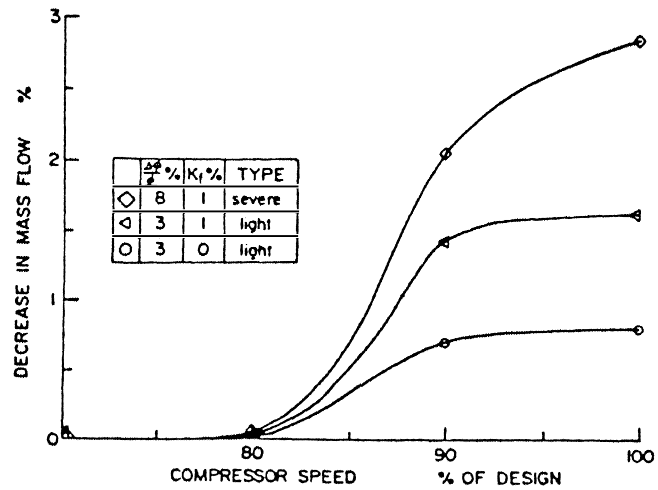


Figure 14. Effect of First Stage Fouling Severity on Compressor Mass Flow [11].

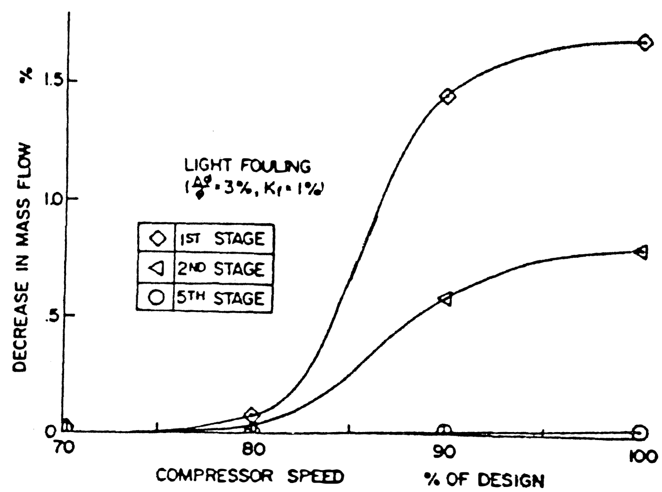


Figure 15. Effect of Fouling Location on Compressor Mass Flow. The effect is minimal for fifth stage fouling but severe for first stage fouling. Three percent drop in flow coefficient and one percent drop in efficiency was assumed [11].

first stage fouling, but with a stage pressure ratio distribution peaking in the center (or rear stages) is less susceptible to suffer from fouling than a similar compressor designed with peak stage pressure ratio coinciding with the first stage.

- The shift of constant speed lines on the compressor map produced by fouling was proportional to rotor speed and the drop in pressure ratio at a given speed varied with mass flow. For fouling at a particular stage, upstream stages are loaded as their flow coefficients are reduced by a combination of reduced mass flow and increased upstream stage density ratios, while downstream stages are progressively unloaded as their flow coefficients increase due to lower density ratios across the fouled and subsequent stages.

Additional investigations conducted by Aker and Saravanamuttoo [12] have resulted in interesting findings relating to the onset and progression of fouling. The simulation studies were conducted for LM2500 and Solar Centaur gas turbines.

Each curve on the following figures [12] has a notation to indicate the fouling schedule used. The notation used for stage efficiency is (1:×). The "1" refers to a one percent drop in flow characteristic assigned to each progressive fouling step and × is the drop in stage efficiency (in percent) per fouling step. Results are shown in Figure 16 for the Centaur engine which clearly indicate that drop in mass flow is a good indicator of fouling. The effect of stage efficiency change does not affect the picture. The use of inlet depression as an analog of mass flow to detect fouling as suggested by Scott is presented later. The way the compressor discharge pressure is affected by the fouling is shown in Figure 17. Inter turbine temperature (ITT) is not such a good indicator of fouling (Figure 18), because it is a function of stage efficiency. Only with significant reductions in stage efficiency was an increase in ITT noted. The effect of fouling on power degradation is reflected in Figure 19. The effect of fouling on heat rate is shown in Figure 20. Based on these curves, Acker and Saravanamuttoo [12, 10] suggest that for operators who require full power from operations are probably best served by periodic washes, while users who require maximum efficiency (typical of pipeline companies) can be well served by "on-condition" washing. Results showing the percent change in mass flowrate upon

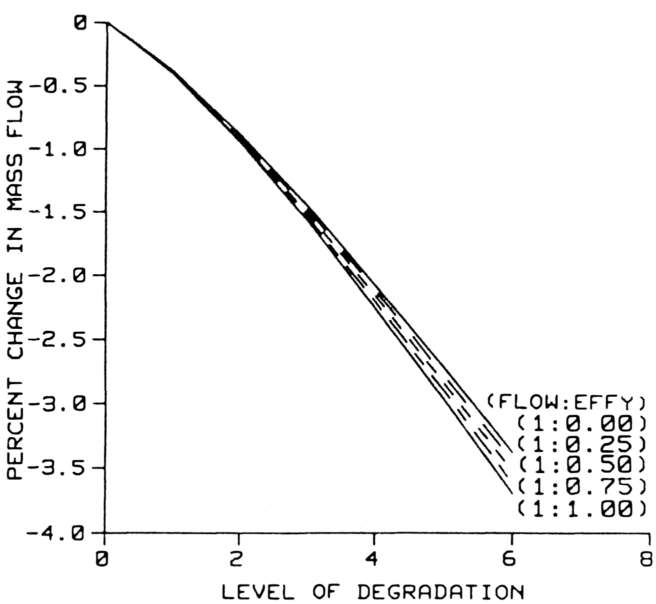


Figure 16. Results of a Simulation Study For Percent Change in Mass Flow for a Centaur Engine Due to Fouling [12].

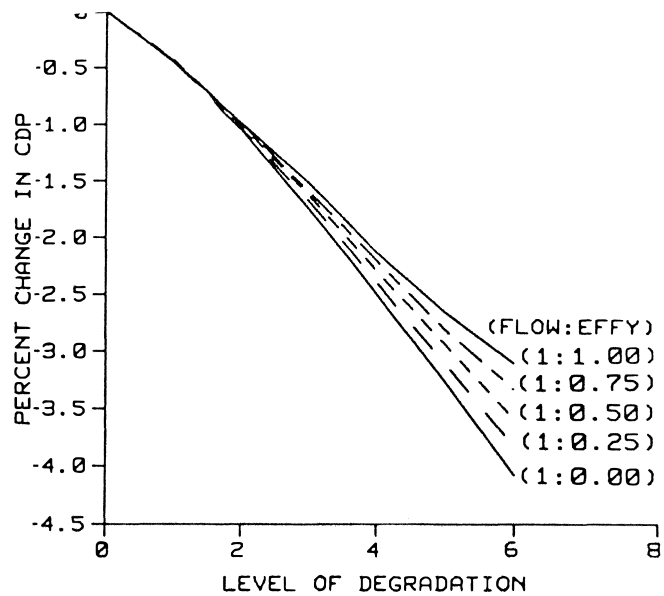


Figure 17. Simulation Results of a Drop in Compressor Discharge Pressure Due to Fouling [12].

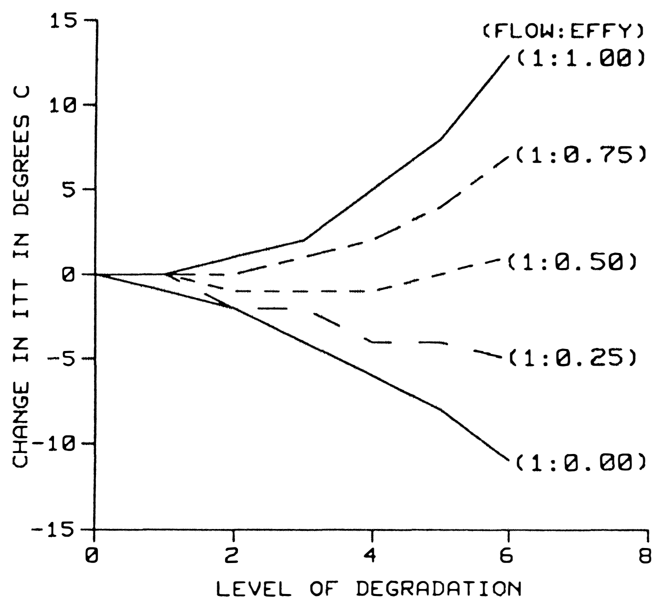


Figure 18. Simulated Results of Change in Inter Turbine Temperature for a Centaur Engine [12].

fouling as a function of the number of fouled stages is shown in Figure 21. They conclude that the effect of fouling on a stage is a function of stage loading and that it is possible that the high performance aeroderivatives are more susceptible to fouling than industrial gas turbines.

Fouling, Air flow Distortion & Compressor Surge

A typical characteristic curve for an axial flow compressor stage is shown in Figure 22. Under design operating conditions, most stages would operate at design flow coefficient at a high isentropic efficiency. When the flow coefficient is to the right of the characteristic 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).

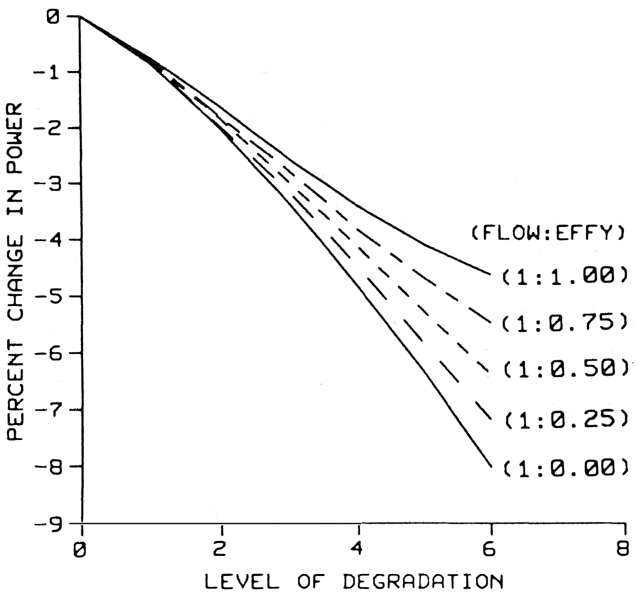


Figure 19. Simulated Results for Change in Power for a Centaur Engine Due to Fouling [12].

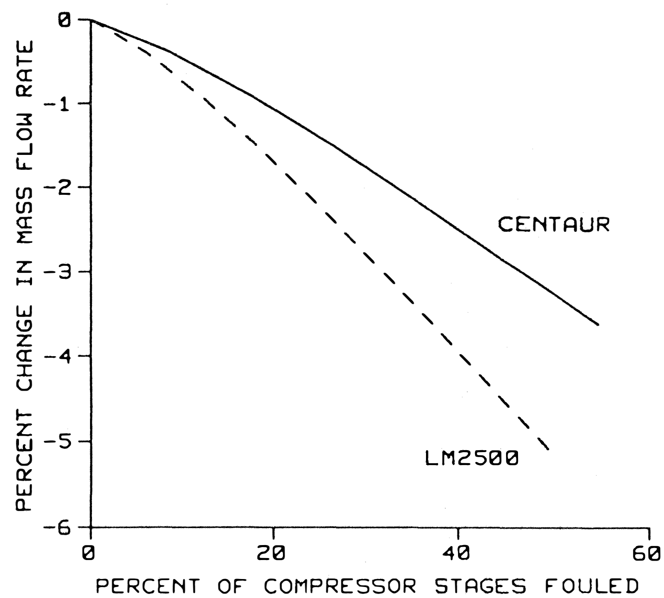


Figure 21. Simulated Results of Fouling Effects on Two Gas Turbine Engines [12].

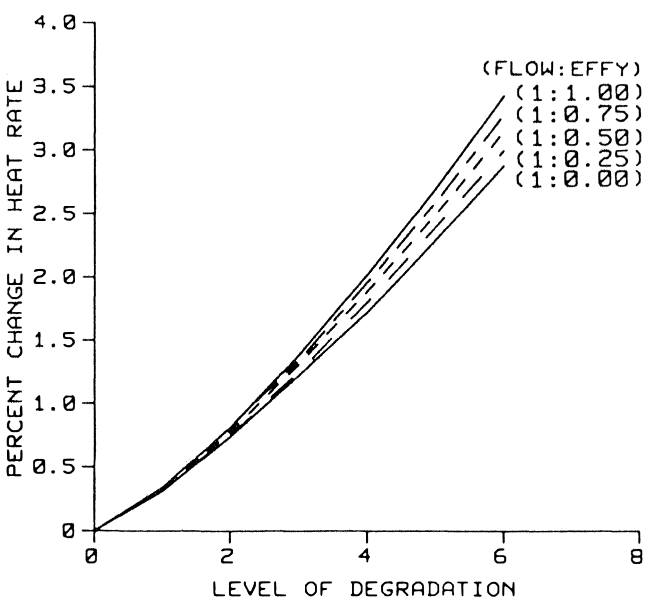


Figure 20. Simulated Results of Change in Heat Rate for a Centaur Engine [12].

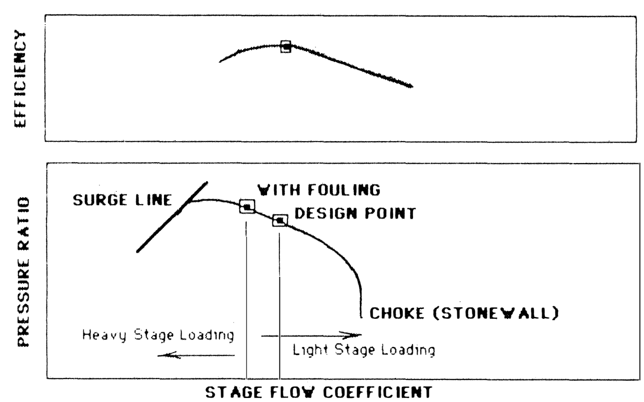


Figure 22. Stage Characteristics Showing Effect of Fouling (drop in flow coefficient). Drop in mass flow causes a rise in first stage pressure and an increase in discharge density.

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 towards 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 later stage stalls triggering a surge. Some basic velocity triangles are shown in Figure 23, indicating how a drop in mass flow causes excessive incidence angles and subsequent aerodynamic stall. The effect of off-design incidence on losses is also shown in this figure.

There are three distinct stall phenomena. Rotating stall and individual blade stall are aerodynamic phenomena. Stall flutter is an aeroelastic phenomenon.

Rotating Stall—Rotating Stall (Propagating Stall) occurs when large stall zones cover several blade passages and propagate in the direction of the rotor at some fraction of rotor speed. The number of stall zones and the propagation rates vary considerably. Rotating stall can often lead to increased vibrational stresses and may produce resonance conditions. Typically, a rotating frequency of 30 to 70 percent of running speed is detected. If blades are excited by the regions of flow breakaway and the exciting frequency, or an integral multiple, coincides with the natural frequency of the blade, resonance can occur. In general, the lower the order of the harmonics, the higher the stress levels. According to the Allinaz Company [13], ratios of natural blade frequency to a maximum rotational frequency of 4.5:1 or higher will not normally cause blade breakage. Rotating stall can occur in *both* axial and centrifugal flow compressors. It is interesting to note that the phenomenon was first noted in centrifugal compressors. One of the best ways of preventing rotating stall would be careful operation when at low speeds. Often, part speed operation causes lower densities in the rear stages which results in choke. This then limits the flow in the inlet stages causing a stall condition.

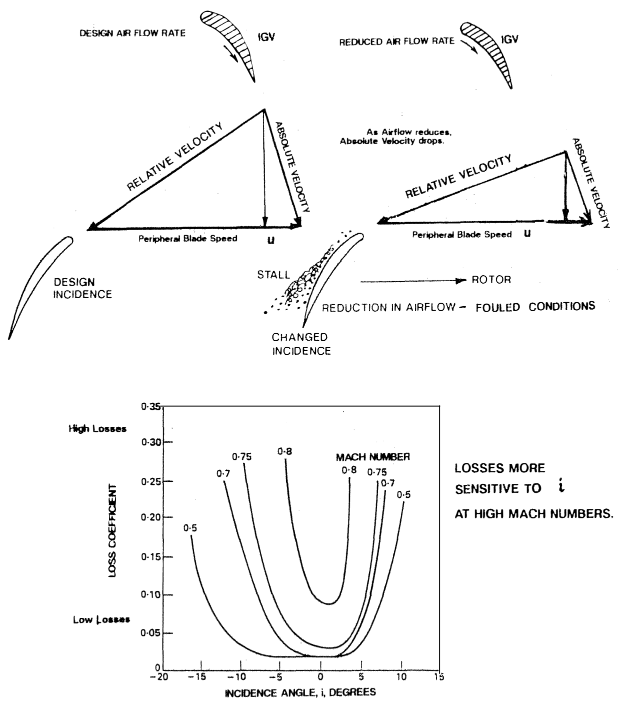


Figure 23. Velocity Triangles and Effect of Fouling on Incidence Angle.

Individual Blade Stall—This type of stall occurs when all the blades around the compressor annulus stall simultaneously without the occurrence of a stall propagation mechanism. The circumstances under which individual blade stall is established are unknown at present. It appears that the stalling of a blade row generally manifests itself in some type of propagating stall and that individual blade stall is an exception.

Stall Flutter—This phenomenon is caused by self-excitation of the blade and is an aeroelastic phenomenon. Stall flutter occurs due to the stalling of the flow around a blade. Blade stall causes Karman vortices in the airfoil wake. Whenever the frequency of these vortices coincides with the natural frequency of the airfoil, flutter will occur. At stall conditions, a sudden decrease in lift takes place. When this occurs, the aerodynamic damping reduces (negative damping). The blade movement then accentuates the instability which creates a self-excited oscillation. Stall flutter problems are relatively rare in industrial gas turbine compressor sections. Some studies indicate that it can cause blade excitation. Improper adjustment of inlet guide vanes (IGVs) can cause transgression into a flutter regime. Alternatively, a reduction in flow will cause a change in relative velocity (if IGVs are absent or do not work).

Surge control on most single shaft gas turbines is accomplished by the use of interstage bleed valves which open at part speed operation, say less than 95 percent of operating turbine speed. The control system is expected to keep the operating line sufficiently away from the surge line during full speed operation. There have, however, been cases of single shaft turbines exhibiting disastrous surge behavior due to the following reasons:

- Excessive fouling of compressor blading
- Deviations in stagger angle of blade tips
- Deviation in tip solidity (blade-chord/tangential distance between blades)
- Deviation in blade camber or curvature

Dundas [14] has conducted an excellent analytical investigation into the deterioration of turbine operation including drop in compressor efficiency, fouling, first stage nozzle distortion, internal bleed seal deterioration, drop in turbine efficiency, inlet filter fouling and low fuel heating value. These were examined to study the effect on the turbine operating line. His study concluded that:

- Compressor fouling had a pronounced effect on the operating line. This effect was more pronounced on cold days, but did not exist on hot days. Dundas [14] states that a possible reason for this could be the shape of the constant temperature lines which are steeper on cold days than hot days. The effect of fouling on the compressor operating line is shown in Figure 24.
- Turbine nozzle area also had a significant impact on the operating line as shown in Figure 25.

While these two effects caused movement of the operating line towards the surge line, there are other factors that can cause

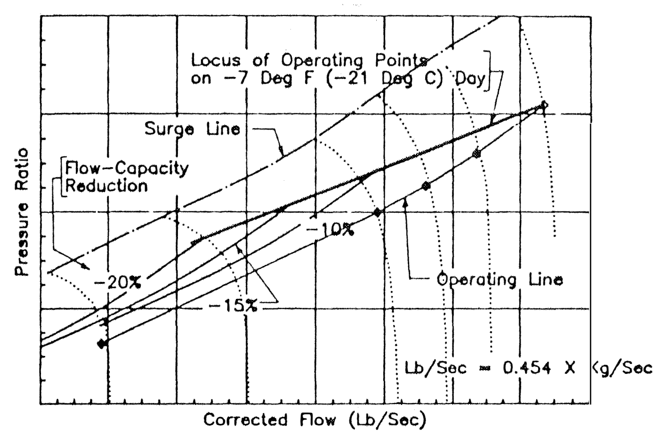


Figure 24. Fouling Effects on Compressor Operating Line [15].

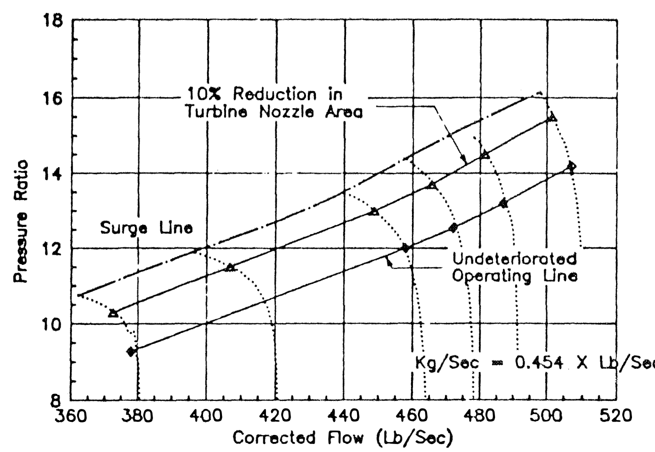


Figure 25. Effect of Turbine Nozzle Vane Distortion on Compressor Operating Line [15].

movement of the surge line itself. Erosion of compressor blading can effect boundary layer development and increase the tendency towards separation. Stall can, therefore, occur at lower incidences than with smooth compressor blading. Heavy erosion can also reduce blade tip chords, thereby reducing blade tip solidity, which would adversely effect stage stability.

The importance of considering surge as a serious problem becomes more important with the use of gas turbines in combined cycle cogeneration applications and with steam injection applications [15]. In combined cycle operations it is important to maintain superheat conditions for the steam turbine; this is attained by operating at reduced air flowrates (by the use of IGV control) at part load while maintaining the turbine inlet temperature. The variable inlet guide vanes throttle the flow, dropping the inlet pressure and maintaining the operating line. Part load operation with engine deterioration such as fouling can result in surge.

If prolonged surge occurs, the blades can be severely damaged because of the excitation caused by the flow reversals. Moreover, rapid speed changes (cycling) can occur as the unit is loaded and unloaded. It is possible that one or more stages pass through resonance conditions. An examined axial flow compressor (146 lb/sec, pressure ratio = 3.8) experienced surge for about five minutes. The casing was severely distorted, resulting in uneven blade tip clearances. Uneven tip clearances have been known to create blade vibration problems.

Steam Injected Gas Turbines—
Axial Compressor Surge Margin

Historically, water and steam injection into a gas turbine has been used for a long time. Steam injection or water injection is currently used routinely for NO_x control. Several turbine manufacturers are offering STIG cycles, and intercooled STIG (ISTIG) cycles are also being considered. STIG cycles are particularly useful in cogeneration applications when part load operation is required or where thermal requirements fluctuate a lot. In the STIG cycle, exhaust energy is converted into steam and then injected in total, or in part, into the compressor, combustor and turbine section (or a combination of these). In addition to increased mass flow, there is also a benefit from the specific heat effect as mixtures of air and steam have higher specific heat than air. The increase in specific heat can be as great as 25 percent which increases the output. For example on a 20 MW (ISO) aeroderivative gas turbine, the output can be augmented by about 30 percent. This could be accomplished by steam injection at about seven to eight percent of the air flow. Steam is injected into the compressor discharge port and combustor. In multispool engines with high pressure ratios, steam can also be introduced in the LP turbine section.

As aeroderivative engines are designed for emergency power situations, they are designed with wide surge margins to allow slam accelerations (sudden increase in turbine entry temperature). Because of this, aeroderivative engines are suited to steam injection while maintaining adequate surge margins.

Because of these considerations, gas turbines operating on steam injected cycles should have compressors with substantial surge margins that can accommodate fouling deterioration in addition to the increased back pressure.

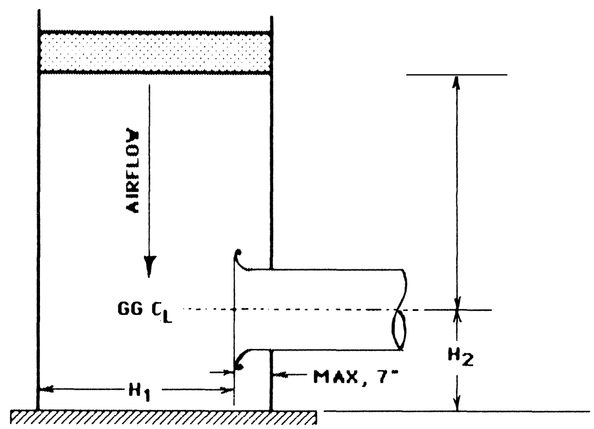
Inlet Distortion and Effects on Axial Compressors

There have been several cases where excessive distortion of the inlet air flow has tringered a surge event resulting in compressor damage. Icing, causing uneven inlet circumferential distortion, or uneven clogging of filters, possibly due to a bend in the inlet duct before the filter or improper inlet system design, can create distortion effects which could result in surge.

Studies as reported by Schweiger [16] have resulted in criteria to ensure distortion free air distribution to help avoid surge problems and excessive excitation of the first row of compressor blades or variable IGVs. Some guideline are shown in Figure 26 [15] for geometrical rules for intake design. The rules provided in this figure show approximate distances for the intake boundary walls from engine flare and centerline. If the walls are brought too close to the engine, strong vortices can be generated. It is important that intake design be carefully considered. Manfrida et al., [17] have conducted an experimental and analytical study of flow in inlet ducts.

Improperly designed inlets, excessively fouled IGVs or fouled struts can induce severe air flow distortion to an axial flow compressor. The results of distortion tests are shown in Figure 27 for an aircraft axial flow compressor [18]. The figure shows the effect of simulated distortions on the surge line. Distortion was induced by the use of screens. Dashed lines represent the compressor map in a clean, undistorted airflow condition. The solid lines are stall lines for distorted flow cases. As can be seen in Figure 27, the hub radial position distortion has little effect on the surge line, but tip radial distortions cause a more serious loss in stall compressor pressure ratio. The circumferential distortion pattern shown in the lower right causes the most loss in compressor pressure ratio at stall. The partial hub radial distortion pattern (Figure 27 upper-right) also has an effect on the surge line.

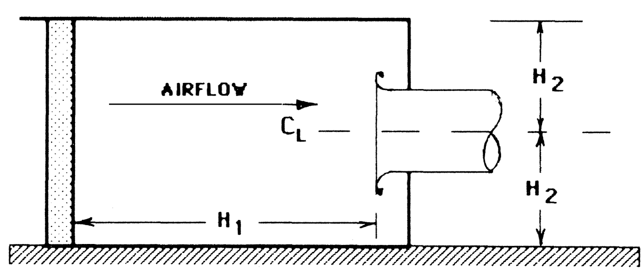
Air flow distortions that are severe enough to cause problems will usually affect performance of the gas turbine. Thus, by reg-



VERTICAL INTAKE

$$H_1 = \frac{\text{MASS FLOW}}{\left[1 - \left(\frac{\text{Hub Dia}}{\text{Tip Dia}}\right)^2\right]}$$

$$H_2 = \frac{H_1}{1.394}$$



HORIZONTAL INTAKE

Figure 26. Intake Plenum Geometry Recommended. (e. g., Avon Engine H_1 = 6 ft, H_2 = 4.26) [16].

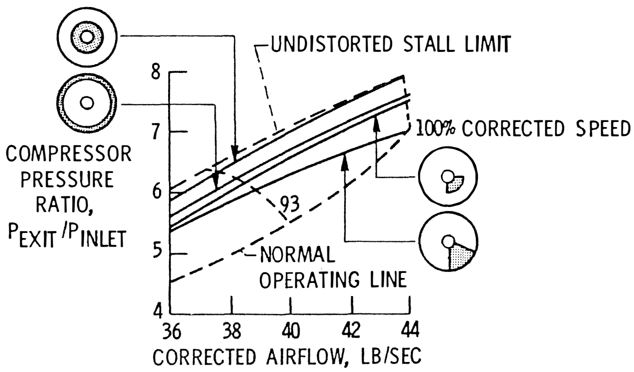


Figure 27. Effect of Distortion on an Axial Flow Compressor [18].

ular and *frequent* performance monitoring this deterioration can be detected.

Effects on Compressor Blading Integrity

Blading failure is a serious and frequently occurring problem in gas turbines. According to Allinaz [13], about 42 percent of the damage cases on gas turbines relate to blading. In the category of turbocompressors (axial and centrifugal compressors), about 18 percent of damage cases relate to blading and inlet guide vanes.

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

- By promoting surge or rotating stall which might have a dangerous effect on blades.
- In some unusual cases, blading natural frequencies can be affected by the increase in mass due to dirt buildup on the airfoil ($f_n = \sqrt{K/M}$). A well known case of this occurrence was on a very early Avon turbine where extensive fouling (⅛ in deposits) pulled the blade's first natural frequency down into the running range [19].
- Blading roughness, and, therefore, efficiency and performance, can be adversely affected by corrosion, erosion and fouling.
- Excessive dirt on the blades can cause unbalance, and a consequent increase in running speed vibration. In some cases, dirt can get between the bearing surfaces of the blade root, causing the blades to operate in a non-normal position, which would add to the stresses. If the root constraint is changed due to buildup in the fir tree region, a change in natural frequency could result (as the boundary condition changes).
- Foulant buildup on compressor blading can lead to a serious corrosion problem, especially when humidity is high (galvanic action can be set up). Pitting of the blading can lead to local stressing which reduces the blade's fatigue life. Some airborne salt may be one to three microns in size. This problem will occur if saltwater or salt particles are ingested in the compressor. The dry salt or brine will absorb moisture during high humidity operation or during water washing.
- Particles causing erosion are normally ten microns or greater. Five to ten microns represent the transition zone between fouling and erosion. (Note: 10 microns = 1/15 diameter of human hair).
- On a relatively small gas turbine, a 0.1 mm coating applied to the blading can cause a flow reduction of ten percent and a reduction of compressor efficiency of five percent [7].

Axial compressor blading should ideally have low density, high stiffness (this is more important on aircraft engines), a high strength/weight ratio, and good resistance to corrosion, impact, foreign object damage, fretting, moisture, erosion, and thermal fatigue. In the case of high pressure ratio compressor rear stages, resistance to creep is an important attribute. Compressor blades have to withstand conditions of surge, rotating stall, and unsteady flow created by inlet distortions. A discussion of blading problems has been made by Meher-Homji and Focke [20].

Common causes of vibration of axial flow compressor blading are:

- stator passing frequency wakes.
- inlet guide vane created wakes.
- rotating stall.
- surge.
- choking.
- inlet or diffuser supporting vanes or struts.
- inlet flow distortion.
- blade flutter.
- variable blade tip clearance.
- rotor vibration (mechanical origin).

Several of these can be accentuated by fouling. The influence of wakes from blades or stators can be felt at a considerable distance. Blades have been known to break in fatigue as far as three stages away from the excitation, upstream and downstream. The influence is still strong after passing through six rows of blades.

An excellent discussion of compressor airfoil reliability is made by Passey and Armstrong [21] and by Armstrong [22]. Experience has shown that safe alternating stress levels for compressor airfoils are relatively low—10,000 peak to peak lb/in^2 for aluminum alloys and about 25,000 peak to peak lb/in^2 for titanium alloy and steel [21]. There are several reasons why these levels are so low. They include:

- Surface imperfections—can be created during manufacturing, or be the result of foreign object damage. Compressor coatings can be of value in this situation.
- Notch sensitivity effects.
- Temperature effects on fatigue strength—for example, a typical 12 percent chrome steel will show a ten percent reduction in fatigue and tensile strength at a temperature of 752°F. This effect is more important on high pressure ratio compressors.

Valuable information on blading can be obtained by the use of carefully located, high frequency accelerometers. Generally, these accelerometers should be case mounted. The vibration signatures obtained from gas turbines are complex with several peaks occurring which correspond to the different number of blades and gearing present. Blade passing frequencies will be quite predominant. By knowing the number of blades present on a certain disk or stage, it is possible to detect problems by observing *relational changes*. Amplitudes are typically functions of pressure ratio (therefore speed), stator blade angles, and flowrate. This is an important point to be noted if any meaningful trending is to be accomplished. It is difficult to pass judgments on the absolute amplitudes of vibration that the actual blade is experiencing, but qualitative judgments can be made. In real life field troubleshooting problems, difficult value judgments have to be made with incomplete data. In such cases, data from accelerometer readings can prove invaluable.

Mitchell [23] describes work done to determine if changes in the blade passing frequency (BPF) can be used to detect blade problems. Relational changes in the blade passing frequency and its harmonics do provide useful information. Blade passing frequency amplitudes seem to increase at both low flow (near surge) and at high flow (approaching choke) conditions. A case of a large centrifugal turbocharger is shown in Figure 28 where

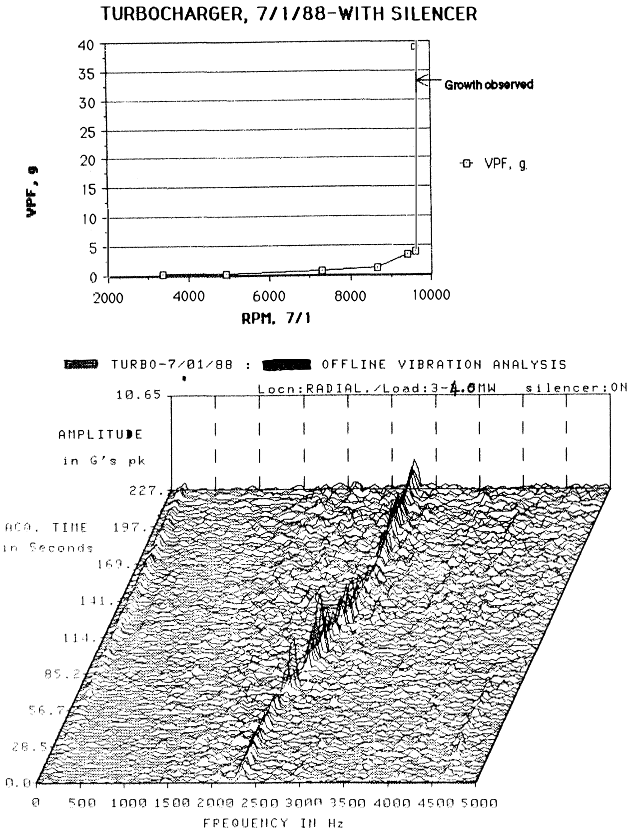


Figure 28. Growth in Vane Pass Frequency As Experienced on a Centrifugal Turbocharger Due to Stall. Effect was found to be very speed sensitive.

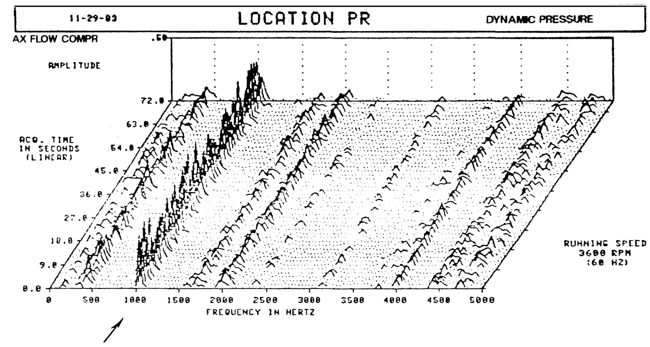


Figure 29. Dynamic Pressure Pulsations on an Axial Flow Compressor. This can be used as an aid in detecting stall.

the sudden growth in vane passing frequency was linked to aerodynamic stalling (surge). A contributing factor here was intake distortion caused by a distorted inlet-silencer. There has even been a case of a destructive surge event occurring in a syngas centrifugal compressor due to a hand-file left in the inlet section [24]. A cascade plot is shown in Figure 29 of dynamic discharge pressure taken on a large axial flow compressor that had a problem with varying tip clearance. The 1000 Hz component was felt to be related to a flow instability. Many more carefully designed experiments need to be conducted to correlate blade problems with signatures, but there is little doubt that useful information does exist in signature analysis.

Associated Problems

There are some problems that are often discussed along with fouling. These are:

Erosion is the abrasive removal of materials by hard particles suspended in the airstream, typically above ten microns in diameter. Erosion impairs aerodynamic performance and can affect the blade mechanical strength. Erosion first increases blading surface roughness, thus lowering efficiency slightly. As erosion progresses, airfoil contour changes occur at the leading and trailing edge and at the blade tip. Severe erosion has also been known to cause changes in blade natural frequency. Some factors that affect erosion rate are:

· angle of impingement.

· velocity of particle impact.

· characteristics of particles (size, density, hardness and shape).

· properties of blade material and coatings.

Erosion problems can occur along with corrosion. Tabakoff [25] has done extensive work in the area of erosion.

Corrosion is caused due to chemical reactions between the engine components and airborne contaminants. Salts, mineral acids, and aggressive gases (e. g., chlorine) along with water can cause wet corrosion and compressor blade pitting. This can lead to local stress raisers which can diminish blade fatigue life. Compressor coatings are of value in this instance. An important corrosion process in compressors is known as electrochemical or wet corrosion. This can occur in the following forms: a) *Uniform*—all areas corrode at the same rate. b) *Galvanic Corrosion*—occurs when dissimilar metals are in contact (e. g., brazed joints). c) *Pitting Corrosion*—is a localized attack that can seriously affect blade strength. Salts and moisture can easily collect in existing pits. d) *Intergranular Corrosion*—is another localized form of attack where grain boundaries are affected. Both (c) and (d) act as stress concentrators reducing blading fatigue life. The electrolytes that cause these problems develop when hygroscopic particles deposited on the airfoils from the intake air absorb airborne moisture during periods of high humidity. The absorption of air pollutants such as NO_x or SO_2 can cause acidification of salt droplets thus causing very corrosive conditions. Axial blading coatings can be of value in controlling corrosion.

Cooling air passage blockage—The blocking (or partial blockage) of cooling passages of hot section stators and blades can be caused by fine foulants (typically less than five 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.

Unbalance caused by foulants—Long term accumulation of foulants can at times cause unbalance problems (high 1 × rpm vibration). Severe and nonuniform fouling (or cleaning) could create this condition.

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

ECONOMIC EFFECTS OF COMPRESSOR FOULING

Fouling causes diverse concerns to different operators. For most petrochemical, pipeline and cogeneration plants, there are three economic areas of concern: 1) Loss of thermal efficiency (i. e., higher fuel costs), 2) loss in throughput/output (i. e., less gas pumped or lower MWs generated), and 3) loss of steam generation (in cogeneration plants).

The economic impact can best be judged by the following findings:

- The largest cost during the life cycle of a gas turbine is the fuel cost. An 80 MW_e gas turbine will consume $23 million in fuel in a year. Based on 1986 estimates in the gas distribution industry for a 12 MW aeroderivative gas turbine, the fuel cost is $870/fired hour, compared to an overhaul/maintenance cost of $20/fired hour (i. e., fuel costs are approximately 40 times overhaul cost) [26]. Because of this, and the fact that the gas pipeline is a major consumer of fuel itself, the incentive for maintaining high thermal efficiency is of paramount importance.
- Operating with dirty compressors can increase fuel consumption by up to five percent.
- A fouled compressor might easily result in an output drop of ten percent.

A performance deterioration economic analysis is shown in Table 3 for five gas turbines ranging from 10 MW to 116 MW operating in cogeneration service. An estimated drop in air flow due to fouling effects has been considered and performance decrements used are based on Zaba's research [9]. The significant penalty costs can be easily seen from this table.

DETECTION OF FOULING

Gas turbine manufacturers and operators typically develop guidelines as to when fouling deterioration calls for corrective action. This is usually based on a combination of load and exhaust gas temperatures (EGT). Users also monitor compressor discharge pressure and compressor efficiency. Graphs can be plotted to show expected (clean) *vs* measured parameters. It is the opinion of some operators that the only way to detect a fouled compressor is by visual inspection. With most turbine designs, however, this means shutting the unit down, removing the inlet plenum hatch and visually inspecting the compressor inlet, bellmouth, inlet guide vanes (IGVs) and visible early stage blading. The following factors can be used as indicators of fouling: (1) Drop in compressor mass flowrate on fixed geometry engines. (2) Drop in compressor efficiency and pressure ratio. The most sensitive parameter of the above factors is the mass flowrate. Meher-Homji and Boyce [27] cover a technique for performance computation of the air flowrate when it is not being measured. This technique utilizes thermodynamic heat balance methods.

Table 3. Analysis Showing Economic Effect of Fouling Degradation on Performance of Five Gas Turbines Ranging from 10 MW to 116 MW. This is provided to give users a feel for the magnitude of the numbers involved.

	A	B	C	D	E	F
1	ECONOMIC ANALYSIS FOR GAS TURBINE PERFORMANCE WITH FOULING DEGRADATION					
2	**					
3	NOTE: (1) This analysis is based on deterioration criteria of Zaba [9]					
4	(2) Deterioration is estimated on year average basis					
5	(3) ISO Conditions assumed					
6	(4) Unit assumed in Cogeneration Service.					
7	(5) Turbine section deterioration is assumed to be 0 %					
8	(6) Exhaust Cp = 0.26					
9	(7) Five Gas Turbines Considered ranging from 10-116 MW.					
10	GAS TURBINES------------------------------>	GT-A	GT-B	GT-C	GT-D	GT-D
11	Gas Turbine Rating, KW	10,450	26,300	37,400	79,750	116,400
12	Number of Gas Turbines	1	1	1	1	1
13	Design Air Flow Rate, lbs/sec	115	270	301	633	897
14	Design Heat Rate, BTU/KWhr	13,320	11,820	10,860	10,690	10,350
15	Exhaust Gas Temp, F	978	901	903	993	984
16	Estimated HRSG Stack Temp, F	300	300	300	300	300
17	Export Power Rate, $/KWhr	0.06	0.06	0.06	0.06	0.06
18	Supp Firing Efficiency	0.85	0.85	0.85	0.85	0.85
19	--					
20	Duty Cycle	0.95	0.95	0.95	0.95	0.95
21	Fuel Costs, $/MMBTU	3	3	3	3	3
22	Estimated drop in air flow rate, %	5	5	5	5	5
23	Estimated drop in Compr Eff, %	5	5	5	5	5
24	Output drop due to airflow decrement, %	4.6	4.6	4.6	4.6	4.6
25	Output drop due to Compr Eff decrement, %	8.5	8.5	8.5	8.5	8.5
26	Heat Rate incr. due to Air flow decrement, %	0.5	0.5	0.5	0.5	0.5
27	Heat Rate incr. due to compr eff decrement, %	5.83	5.83	5.83	5.83	5.83
28	Total Heat Rate increase, %	6.33	6.33	6.33	6.33	6.33
29	Total Drop in Output, %	13.1	13.1	13.1	13.1	13.1
30	Drop in Output, KW	1,369	3,445	4,899	10,447	15,248
31	Degraded Heat rate, BTU/KWhr	14,163	12,568	11,547	11,367	11,005
32	Heat rate increment, BTU/KWhr	843	748	687	677	655
33	--					
34	Undegraded Annual Fuel cost, $	3,475,117	7,761,081	10,140,290	21,284,202	30,077,539
35	Degraded Annual Fuel cost, $	3,211,035	7,171,298	9,369,706	19,666,766	27,791,877
36	Undegraded Electrical Revenue	5,217,894	13,132,116	18,674,568	39,820,770	58,120,848
37	Degraded Electrical Revenue, $	4,534,350	11,411,809	16,228,200	34,604,249	50,507,017
38	Annual Loss in Revenue, $	683,544	1,720,307	2,446,368	5,216,521	7,613,831
39	Drop in Exhaust Energy, MMBTU	15,183	31,600	35,345	85,424	119,479
40	Cost to Supplement Fire, $	53,589	111,528	124,747	301,497	421,691
41	--					
42	Total Annual Penalty, $	$473,051	$1,242,053	$1,800,531	$3,900,582	$5,749,861

The real problem is not how to detect fouling, but how to detect it at an *appropriate* time before a significant power drop has occurred and a fuel penalty cost has been paid. Several philosophies are in use. Some operators believe in periodic washing of the machine while others base washes on condition (i. e., some set of performance parameters). The philosophy utilized is a function of normally expected fouling level, its severity, washing effectiveness, and plant operation criteria. Periodic washing can maintain a reasonable state of cleanliness, but actual operating performance, seasonal based fouling and other factors are obviously not determined. With the advent of online monitoring systems [28], the capability now exists for obtaining a good picture of how turbine performance is being affected by fouling.

Some methods suggested by gas turbine OEMs of detecting fouling include:

- Use of bell-mouth instrumentation to measure flow—on several machines, the inlet section to the compressor is equipped with two tap positions. The first tap is located upstream of the inlet flange and the second tap is located downstream at the inlet to the axial compressor. Due to the acceleration of air, there is a pressure differential between the two tap locations. This pressure differential is a function of flow (flow being approximately proportional to the square root of the pressure differential). The procedure followed is:
  - When machine is new or overhauled, data pertaining to barometric pressure, inlet duct static drop, differential pressure drop, and compressor inlet temperature for a range of rpm are obtained.
  - From this data a curve of referred (corrected) rpm *vs* the referred pressure drop is constructed.
  - During operation, operating values of referred pressure drop are plotted to see if it falls on the line. If the test valve is significantly below the curve, then this indicates a fouled compressor.
- Another procedure suggested by a manufacturer is to run the unit at base load while recording the power output, exhaust gas temperature, and compressor discharge pressure and temperature. Fuel consumption can also be measured. Then pressure ratio *vs* corrected speed can be calculated along with compressor efficiency. Using performance curves current performance values are checked with baseline values.

Scott [29] has conducted extensive studies into fouling detection methods. His technique of measurement of air-intake depression is a practical and economical method. It involves measuring intake depression as an analog of air flowrate. In this approach, the gas turbine inlet bellmouth is utilized as a flowmeter. The layout is shown in Figure 30. This technique has been applied by Scott to several Avon engines with good results. A parametric analysis performed by Scott reported that as the compressor fouled, the largest effect was on intake depression

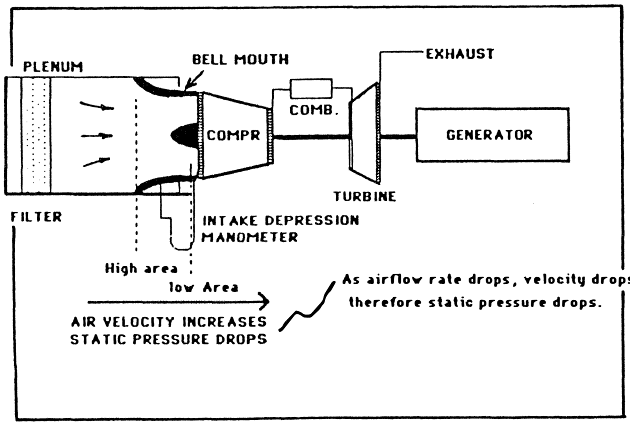


Figure 30. Intake Depression Measurement Method for Fouling Detection.

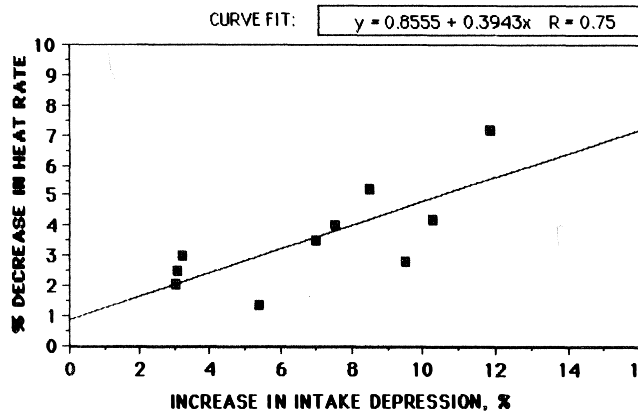


Figure 32. Intake Depression and Heat Rate Relation [29].

compared to the drop in air flow and change in compressor efficiency. This is shown in Figure 31 which was based on tests performed on ten Avon gas turbines. Ten parameters (both directly measured and calculated) were evaluated. The large percent change in intake depression (ΔP_1) clearly indicates its sensitivity to fouling (or performance recovery upon compressor washing). Scott points out the unreliability of relating axial compressor efficiency to heat rate. This is because, for accurate determination of axial efficiency, an accurate discharge temperature is required. Using one or two thermocouples may not be adequate to accurately depict average outlet conditions. The effect of cleaning engines is shown in Figure 32 in terms of intake depression change and heat rate change.

On an Avon engine, the inlet Mach number is approximately 0.5, resulting in an intake depression of over 60 inches of water. Newer high performance aeroderivatives would operate at higher Mach numbers and intake depressions would be correspondingly higher. Manometers using a fluid with specific gravity of 1.75 along with a thermometer can be used. To compare readings under varying speed conditions, ambient temperature and barometric pressure the following corrections to ISO conditions may be made.

$$(\Delta P_1)_{corr} = \Delta P_1 \times SG \times \frac{30}{(BP - \Delta P_f)} \tag{7}$$

$$(N_1)_{corr} = N_1 / \sqrt{\Theta}$$

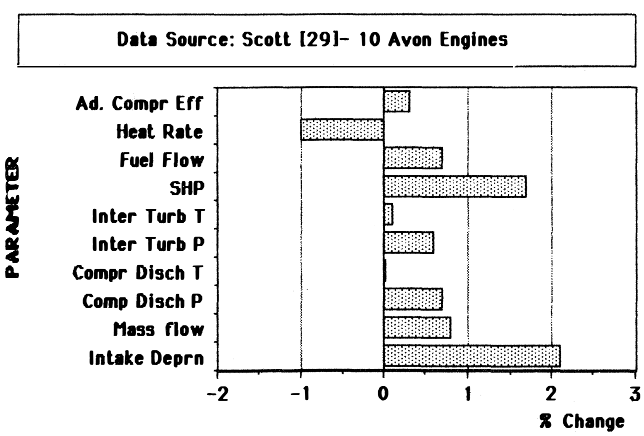


Figure 31. Sensitivity of Ten Parameters on an Avon Engine Upon Engine Cleaning. The high intake depression sensitivity can be noted [29].

where:

$(\Delta P_1)_{corr}$ = Corrected Intake Depression (in.-H_2O)
BP = Barometric Pressure (in.-Hg)
SG = Manometer Fluid Specific Gravity
T_{inlet} = Compressor Inlet Temperature (°F)
N_1 = Gas Generator RPM

$$\Theta = \frac{(T_{inlet} + 460)}{520}$$

A baseline plot of intake depression is shown in Figure 33 for a newly installed engine [29]. The "newly installed" line can be obtained by the manufacturer or determined by means of a field test. It is important to note that variable IGV operation (for part load EGT control) and use of variable stators makes this method more complicated to apply.

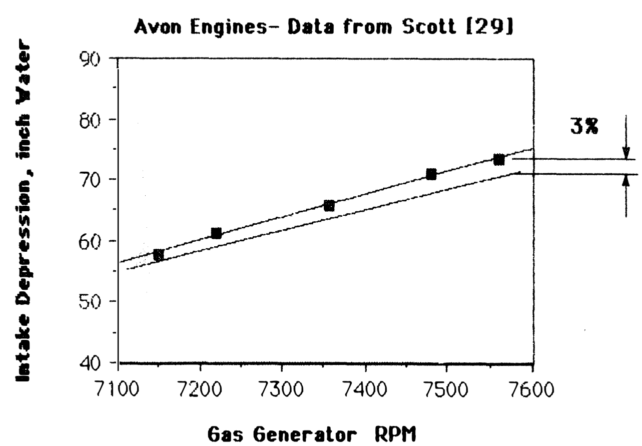


Figure 33. Baseline Showing Gas Generator rpm vs Intake Depression.

It is possible to apply classical optimization techniques to define optimal compressor wash intervals. It is also possible to trade off cost of inefficient operation with compressor cleaning costs. This tradeoff is best accomplished if online monitoring is used so that real time deterioration and fuel consumption can be considered.

CONTROL OF FOULING

Fouling is best controlled by a combination of two methods. The first line of defense is to employ a high quality air filtration system. If fouling occurs (and it usually will), then the compressor can be cleaned by either abrasive materials or by liquid wash systems.

Filtration

There are several types of filters that can be broken into the following groups:

Inertial filters—are used 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—are medium efficiency filters made of cotton fabric or spun fiberglass. They are relatively inexpensive and serve as protection for high efficiency filters.

Coalescers—are constructed by the use of wire mesh which acts as an agglomerator. The mist in the inlet air is agglomerated and the moisture is thus removed.

Louvers and vanes—are typically used in the first stages along with coalescer filters to remove water droplets.

High efficiency filters—remove smaller particles of dirt. They are typically barrier or bag type filters.

Self-cleaning filters—consist of a bank of high efficiency media filters. Air is drawn through the media at a low velocity. At a predetermined pressure drop (about two to three inch water gauge) a reverse blast of air is used to remove dust buildup. These filters are made by several manufacturers and are very successful. The efficiencies of filters on different particle sizes are shown in Figure 34.

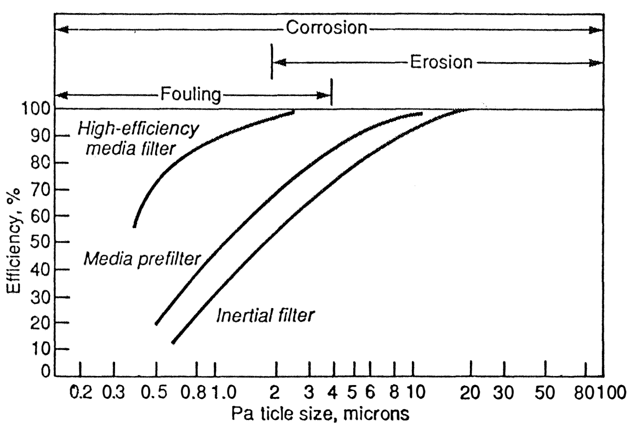


Figure 34. Inlet Air Filter Efficiency.

A very important point is that *air tightness is a must* for any gas turbine inlet system as even the most efficient filtration system will be useless if unfiltered air flow leaks in and enters the compressor. Some common causes of leakage are:

· bypass door leakage.

· poor gaskets and seals at flanged points.

· modifications made on the inlet ducting. Over the years, personnel may add structures or devices to the inlet system which might cause problems.

Corrosion in carbon steel inlet ducts has also been a source of problems. At times, the corrosion can be severe enough to cause a loss of integrity. Because of this, several users are now utilizing stainless steel for the filter houses and inlet ducts. The effectiveness of a filtration system is impacted by its design, installation, and maintenance.

Some important considerations in intake filter design are:

• Aerodynamic design should be such as to keep intake velocities uniform across the entire filter area.

• Filter housing should be of a bolted and welded design fabricated of steel no less than 3/16 in thick and reinforced by steel members. Filter house should withstand 12 in water gauge pressure. All seams and joints should be air tight. All nuts and bolts used inside the clean air plenum should be welded after assembly to prevent air leaks and foreign object damage to the turbine.

• Design should facilitate 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, loose retaining nuts, ungasketed washers, etc.

• Filter design should ensure that the inlet air is drawn at *least* ten feet above grade level. (In some locations, a greater height may be required).

• A stainless steel trash screen with one 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 air flow 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. Several times, flange distortion has allowed air ingress. Users should specify types of seals required and call for a filter house integrity test under specified depression to ensure airtightness.

• System design in the case of pulsed cleaning systems should be such as to minimize flow distortions and pressure pulsations due to pulse cleaning. More than five percent of the total filter elements should not be cleaned simultaneously.

• Filter system pressure drop is an important parameter affecting gas turbine performance. The following *rough rules of thumb* may be applied: For aeroderivative gas turbines, every four inch H_2O of inlet loss will result in a 0.7 percent increase in heat rate and a 1.6 percent reduction in power. For heavy duty industrial machines, a four inch H_2O inlet loss will result in a 0.6 percent increase in heat rate and a 1.6 percent drop in output power. It is important to consider both new filter pressure drop *and* the pressure drop increase over time.

Axial Compressor Blade Coatings

While hot section coatings have been popular for several years, an increasing number of users are now utilizing compressor coatings to help enhance durability and performance. There are also several users that utilize axial blading coatings in order to reduce blade skin friction (thus improving aerodynamics to increase efficiency and performance), and to reduce wear.

While coatings do not necessarily prevent fouling, they enable more efficient and durable operation of a compressor. There is some evidence that smooth coatings can help resist dirt buildup. Typically, compressor blading and stators are made of stainless steel, low alloy steels, titanium, or aluminum. All of these can be prone to corrosion and erosion. Some typical coatings are: nickel-cadmium, teflon®, and aluminum.

Coatings may be classed into two main groups—overlay (sprayed on) and diffused coatings. Sprayed on coatings are applied by spraying on aluminum suspension and then applying a mechanical or thermal process. It does not "diffuse" into the substrate. Diffused coatings are applied by means of a pack diffusion process, resulting in about half the thickness being below the blade surface and half above.

In applying coatings several factors should be considered:

- Type of blading and environmental conditions. For example, turbines operating in the Gulf Coast area of Texas certainly have benefited from coatings.
- Vendors should be consulted and a careful, balanced evaluation of claims should be made. Industrial users should consider coating behavior in terms of 20,000 to 100,000 hours.
- The environment conditions are particularly important in the choice of coatings. Ni-Cd coatings have performed well in mild environments, but have caused severe pitting problems in Gulf Coast plants producing chlorine.
- There is a school of thought that some coatings may *promote* fouling. There are also publications that show performance improvements with coatings stating numbers like two percent increase in compressor efficiency and a two percent decrease in heat rate. While performance improvements may certainly be possible, considerations need to be given to factors such as:
  - On what *statistical* basis were the performance changes evaluated? How long did they last? Were they merely random scatter effects?
  - Were there attendant changes in the turbine (overhaul, seal changes, blade tip clearances)?
  - Were corrected performance parameters utilized in evaluations?
- Coatings must be evaluated in context of the full system—air filtration, washing procedures, and environmental conditions.
- Even the best coatings will not be successful if the application is done poorly.

Becker and Bohn [30] have made a study of intake filtration, coatings and washing for large gas turbines.

COMPRESSOR CLEANING

Two approaches to compressor cleaning are abrasion and solvent cleaning. The choice as to which technique should be used is dependent on the nature of the deposits and the manufacturers' recommendations.

Abrasive Cleaning

Common abrasives used are nut shells, rice, or spent catalyst. The abrasives should have sufficient mass to achieve the momentum required to dislodge the dirt. As they are ingested, they are hit by the leading edge of the airfoils and, therefore, the trailing edges are not abraded. The abrasives must be sufficiently durable to resist breakage.

Abrasive cleaning seems to work well with hard dry deposits and may be carried out at operating load and speed. If the foulant is oil, salt, or a combination of both, then solvent washing is preferable. Some operators may elect to use both a dry and a wet wash in order to fully restore performance.

Some problems with abrasive cleaning are that it can erode airfoils, clog nozzles and, most important, remove protective coatings. In some cases, it can compact deposits in certain areas of the compressor. The use of abrasives can cause problems with flame scanners, which may have to be bypassed to avoid a trip.

In the case of regenerated gas turbines, certain abrasive mate rials used can cause problems in the regenerators. This occurs when an accumulation creates hot spot problems.

Water washing (with or without detergents) cleans by water impact and by removing the water soluble salts. It is most important that the manufacturers' recommendations be followed with respect to water wash quality, detergent/water ratio and other operating procedures. Typically, wheel space temperatures must be below 200°F to avoid thermal shock, and the water wash is done with the machine on crank. Water washing using a water soap mixture is an efficient method of cleaning. This cleaning is most effective when carried out in several steps which involve the application of a soap and water solution, followed by several rinse cycles. Each rinse cycle involves the acceleration of the machine to approximately 50 percent of the starting speed, after which the machine is allowed to coast to a stop. A soaking period follows during which the soapy water solution may work on dissolving the salt. If the machine is cleaned under full operating conditions using rice, walnut shells or some other solid material injected into the inlet, the foulants and corrosive elements deposited within the compressor are removed rapidly and flushed through the turbine.

If the machine is running at substantial load and the gas temperatures are in the region where the salt can be fused, it will have the opportunity of combining with the sulphur (which is present in the fuel) and form sodium sulphate. The sodium sulphate may then fuse, become sticky and deposit itself on the hot parts of the turbine causing severe and rapid attack, which is characteristic of hot corrosion (sulphidation).

The method recommended for determining whether or not the foulants have a substantial salt base is to soap wash the turbine and collect the water from all drainage ports available. Dissolved salts in the water can then be analyzed. This sort of analysis was conducted on some FT-4-1D gas turbines [31, 32]. Results of this research indicated the following:

- A fraction of the airborne salt always passes through the filter.
- Inertia type filters, during periods of high humidity, were found to dump contaminants into the machine, causing a drop in power. There is also evidence of phenomena where compressors can "clean" themselves. Some graphical results relating to this are shown in Figure 35. An analysis of the water used to wash the turbine for determination of where the salts were located was made. Results are shown in Figure 36. A typical plot (Figure 37) shows gallons of water used *vs* conductivity change (inlet-to-outlet).

Water washing a machine under power must be closely monitored to prevent the possibility of liquid water impinging upon hot turbine parts. Carefully controlled water flow using turbine operating parameters as control functions can produce efficient cleaning with minimal or no damage to the machine and with a very short period of reduced power output.

Solvent and Surfactant Washing

Washing can be accomplished by using petroleum based solvents, water based solvents or by the use of surfactants. The solvents work by dissolving the contaminants while surfactants work by *chemically* reacting with the foulants. Water based solvents are effective against salt, but fare poorly against oily de-

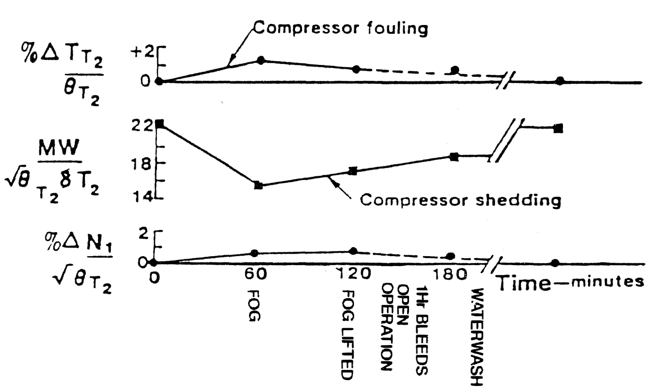


Figure 35. Effects of Fouling on Engine Parameters [32].

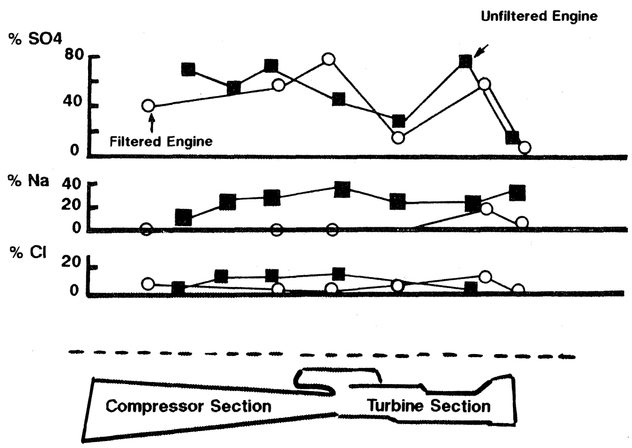


Figure 36. Composition of Residue vs *Engine Location [32].*

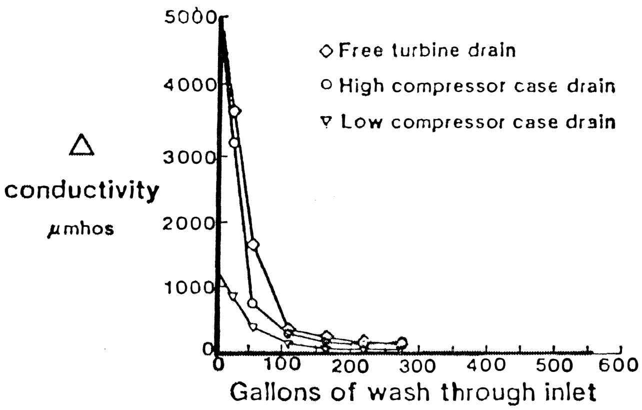


Figure 37. Water Wash Test Results on a Natural Gas Fired Engine [32].

posits. Petroleum based solvents do not effectively remove salty deposits. With solvents, there is a chance of foulants being redeposited in the latter compressor stages.

There are several popular surfactant cleansers that are being used increasingly. These surfactants are effective against both organic and inorganic contaminants. Problems of redeposition of foulants do not occur with surfactants. Surfactants are typically applied on a regular basis under load conditions.

Some important practical considerations for compressor washing:

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

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

• Examine the spray nozzle design to ensure that there is no chance of it coming loose and creating a FOD incident.

• Stainless steel for tanks, nozzles, and manifolds are recommended to reduce corrosion problems.

• If the commercial cleaner being used requires dilution, the *quality* of the dilution water is important. Strength and wash intervals should be adjusted based on performance improvements.

• Prior to washing a very dirty compressor, attempt to clean the inlet plenum and bellmouth.

• Carefully follow manufacturers' requirements relating to drains, valving, protection of piping, and flushing.

The effect of cleaning is shown in Figure 38 of power output of a gas turbine.

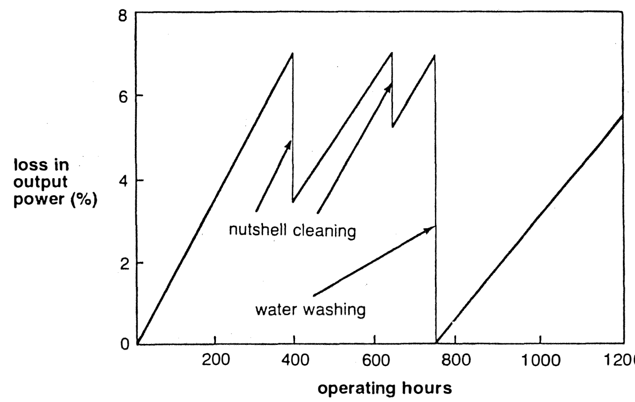


Figure 38. Effect of Cleaning on Gas Turbine Power Output.

TURBINE SECTION FOULING

Contaminants that cause turbine fouling can enter the gas turbine through:

- inlet air.
- fuel (may contain Na and V).
- fuel additives.
- NO_x control injection fluid.

In the hot turbine section, and in the presence of hot gases, low melting point ashes, metals, and unburned hydrocarbons can be deposited in the form of scale. As hot combustion products pass through the first stage nozzle, they experience a drop in static temperature and some ashes may be deposited on the nozzle blades. As the throat area of the nozzle controls the compressor-turbine matching, a reduction in throat area causes a movement away from the design match point [3]. This then causes a loss in performance. Deposits will also form on the rotating blades, causing a further loss in performance. Blade and disc cooling can also be impaired by foulants, causing a reduction in component life or even failure.

The problem of hot corrosion or sulfidation can occur with the nickel and cobalt based super alloys used in gas turbines. Sulfidation is the rapid and destructive reaction that occurs when extensive quantities of alkali salts (Na, K) and vanadium are present in the combustion products along with sulphur. These may be found in fuel, water or steam (used for NO_x control) or, more importantly, in inlet air. As the fuel flowrate is typically about two percent of the air mass flowrate, one ppm Na entering from the fuel would have the same effect as just 20 ppb airborne salt entering the air flow. This is a significant requirement considering that most manufacturers call for not more than one ppm of Na.

Normally, with liquid fuels (especially heavy fuels), fuel treatment calls for the removal of sodium by water washing. If vanadium (V) is present in the fuel, then its effect is inhibited by the addition of epsom salts ($MgSO_4$). This additive causes severe turbine section fouling. As the magnesium is added in a 3:1 weight ratio to the vanadium, fuels high in vanadium cause frequent fouling of the turbine. The frequency of turbine washes required depends on the ash content, turbine inlet temperature and operating cycle (i. e., base load *vs* cyclic operation). Peaking units operated on heavy fuels (high vanadium) do not require washing, since the thermal cycling removes the deposits.

Modern gas turbines operate at a turbine inlet temperature of between 1900°F and 2200°F, which makes them highly sensitive to corrosion problems. This calls for care to be taken with respect to air filtration and compressor washing. Even with good filtration, salt can collect in the compressor section. During the collection process of both salt and other foulants, an equilibrium condition is quickly reached, after which reingestion of large particles occurs. This reingestion has to be prevented by the removal of salt from the compressor prior to saturation. The rate at which saturation occurs is highly dependent on filter quality. In general, salts can safely pass through the turbine when gas and metal temperatures are less than 1000°F. Aggressive attacks will occur if the temperatures are much higher. During cleaning, the actual instantaneous rates of salt passage are very high together with greatly increased particle size.

Even with good air filtration, salt deposits will occur in the compressor. As the air moves through the compressor towards the combustion section, it is heated and compressed, causing removal of the remaining moisture from the airborne salt particles. These particles are deposited heavily in the first few stages, sometimes going back as far as half way through the compressor. In general, the condensation nuclei that pass through the compressor without being entrapped can also pass through the turbine without depositing themselves or adhering to the hot parts. A difficulty arises, however, when the salt that has been collected within the compressor stages becomes so thick that large flakes are reingested into the engine. When this occurs, the local concentration of salt in the air immediately surrounding these large flakes is extremely high. These salt flakes actually have sufficient mass to stick firmly to the turbine hot parts and are responsible for many gas turbines suffering from hot corrosion damage.

Regenerator Fouling

The principal problem in regenerator fouling is gas side fouling. Regenerator gas side fouling depends on factors such as: (a) exhaust emissions, and (b) type of fuel utilized. This problem is not prevalent on large industrial type natural gas fired turbines. There have been investigations in this area by Bowen, et al., [33] and Miller [34].

USER-SURVEY QUESTIONNAIRE RESULTS

A number of questionnaires were sent to gas turbine users at several locations on the North American continent and abroad. The questionnaire objective was to obtain user insight into fouling problems and to determine what fouling detection and control measures were taken. Due to the relatively small sample size, *no statistical conclusions should be drawn from the results*. The quantitative results presented must be viewed tentatively. It must be noted that fouling of axial flow compressors is a complex phenomenon involving intake systems, filters, environmental and climatic conditions and a host of other factors. Hence, fouling behavior is very site specific.

Some questionnaire results are shown in Figure 39. A population of about 25 gas turbines was considered, and included both heavy duty and aeroderivative units. Power recovery numbers after wash ranged from one to four percent with 1.5 to two percent being typical.

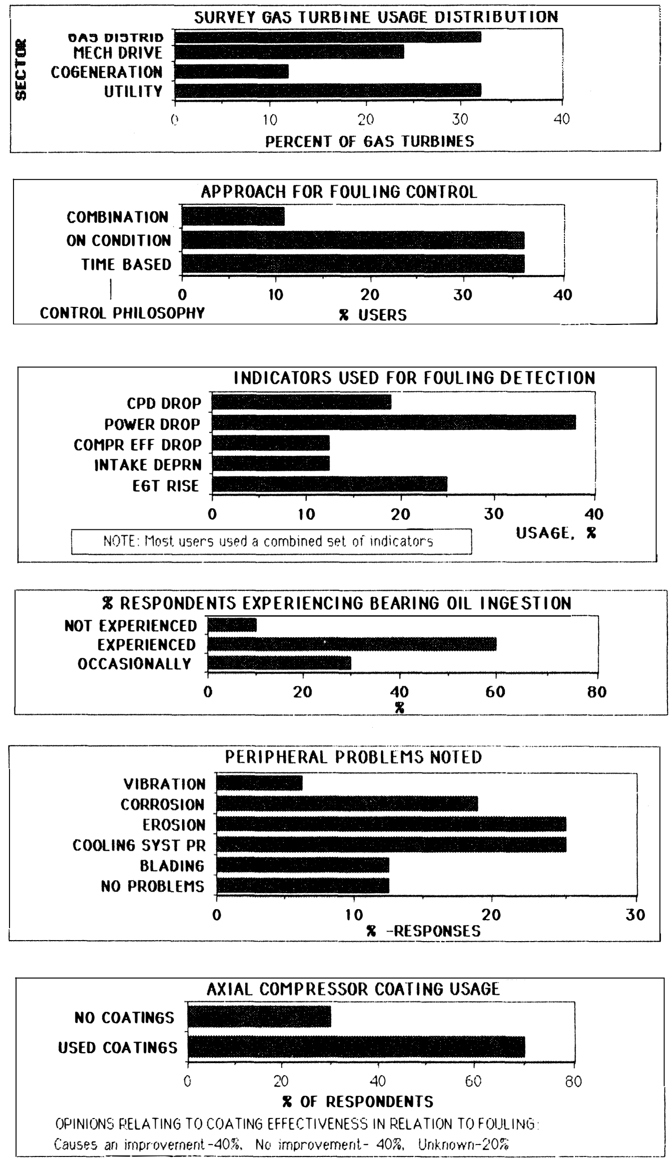


Figure 39. Questionnaire Results.

CONCLUSIONS

The fouling of axial flow compressors and turbines in gas turbine engines is a common operating problem. Understanding the causes and effects helps operators combat this problem. The bibliography provides additional sources of information on gas turbine fouling and related subjects. The influences of fouling can be detected by performance monitoring allowing corrective action to be taken. The first line of defense is, however, a good air filtration system and appropriate fuel preparation and treatment, in order to minimize the effects of compressor and turbine fouling.

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REFERENCES

1. Meher-Homji, C. B., "Compressor and Hot Section Fouling in Gas Turbines—Causes and Effects," Proceedings of the 9th Industrial Energy Technology Conference, Houston, Texas A&M University (1987).
2. Air Quality Data—1978 Annual Statistics, EPA - 450/4-79-037, U. S. Environmental Protection Agency (1978).
3. Upton, A. W. J., "Axial Flow Compressor and Turbine Blade Fouling, Some Causes, Effects and Cleaning Methods," First Gas Turbine Operation and Maintenance Symposium, National Research Council, Canada, (October 1974).
4. Kimm, M. H. P., and Langlands, D., "Gas Turbine Intake Filter Systems Related to Offshore Platform Installations," ASME Paper No. 85-GT-109, ASME Gas Turbine Conference, Houston, Texas (1985).
5. Tatge, R. B., Gordon, C. R., and Conkey, R. S., "Gas Turbine Inlet Filtration In Marine Environments," ASME Paper 80-GT-174, (1980).
6. Hsu, L. L., "Total Corrosion Control For Industrial Gas Turbines: Airborne Contaminants and Their Impact On Air/Fuel/Water Management," ASME Paper No. 88-GT-65, Gas Turbine & Aeroengine Congress, Amsterdam, (1988). (This paper covers equations for evaluating contaminant ingestion quantitatively).
7. Zaba, T., and Lombardi, P., "Experience In The Operation of Air Filters In Gas Turbine Installations," ASME Paper No. 84-GT-39, 1984 ASME Gas Turbine Conference (1984).
8. Bammert, K., and Woelk, G. U., "The Influence of Blading Surface Roughness on the Aerodynamic Behavior and Characteristic of an Axial Compressor," Trans. ASME, Journal of Engineering for Power, *102*, pp. 283-287 (April 1980).
9. Zaba, T., "Losses In Gas Turbines Due to Deposits on the Blading," Brown Boveri Review 12-80, pp. 715-722 (December 1980). (This is a classic paper on fouling effects).
10. Saravanamutto, H. I. H., Lakshminarasimha, A. N., "A Preliminary Assessment of Compressor Fouling," ASME Paper 85-GT-153 (1985). (The first of a series of excellent publications on fouling conducted at Carleton University.).
11. Lakshminarasimha, A. N., and Saravanamuttoo, H. I. H., "Prediction of Fouled Compressor Performance Using Stage Stacking Techniques," 4th ASME Fluid Mechanics Conference, Turbomachinery Performance Deterioration (May 1986).
12. Aker, G. F., and Saravanamutto, H. I. H., "Predicting Gas Turbine Performance Degradation Due to Compressor Fouling Using Computer Simulation Techniques," ASME Paper 88-GT206, ASME Gas Turbine and Aeroengine Congress, Amsterdam (1988).
13. Allianz Insurance Group, *Handbook of Loss Prevention*, New York: Springer Verlag (1978).
14. Dundas, R. E., "A Study of the Effect of Deterioration on Compressor Surge Margin in Constant Speed, Single Shaft Gas Turbine Engines," ASME Paper No. 86-GT-177, ASME Gas Turbine Conference (1986).
15. Dundas, R. E., "Review of Design Parameters in Gas Turbines for the Prospective User," ASME Paper No. 88-GT-23, ASME Gas Turbine and Aeroengine Conference, Amsterdam (1988).
16. Schweiger, T., "North Sea Operating Experience—Air Intake Design," Fifth NRCC Symposium On Gas Turbine Operations and Maintenance, National Research Council of Canada, Calgary (September 1983).
17. Manfrida, G., et al., "Experimental Study of Flow In Gas Turbine Inlet Duct Models," Quaderni Pignone 44, pp. 19-26 (June 1988).
18 "Aircraft Propulsion," NASA - SP-259, Chapter 10, "Effects of Engine Inlet Disturbances On Stall Performance" (November 1970).
19. "History Of The Gas Turbine—The Rolls Royce Avon," Turbomachinery International (March 1985).
20. Meher-Homji, C. B., and Focke, A. B., "Aerothermal and Vibration Monitoring for the Prevention of Gas Turbine Airfoil Failures," 6th ASME Failure Prevention and Reliability Conference, Cincinnati, Ohio, ASME Publication H-331 (1985).
21. Passey, R. D. C., "Reliability of Compressor Airfoils," Program Aerospace Science, *17*, pp. 67-92, Pergamon Press, U.K. (1976) (This presents a comprehensive overview of compressor blading problems).
22. Armstrong, E. K., and Stevenson, M. A., "Some Practical Aspects of Compressor Blade Vibration," Journal of the Royal Aeronautical Society, *64*, (591), pp. 117-130 (March 1960).
23. Mitchell, J. S., "Examination of Pump Cavitation, Gear Mesh and Blade Performance Using External Vibration Characteristics," *Proceedings of the Fourth Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1975).
24. Zech, W. A., "Discussion of Mechanical Failures of Centrifugal Compressors as Experienced in High Pressure Installations," presented at the Symposium on Los Prevention in the Process Industries, St. Louis, (February 1968).
25. Tabakoff, W., "Causes For Turbomachinery Performance Deterioration," ASME Paper No. 88-GT-294, ASME Gas Turbine and Aeroengine Congress, Amsterdam (June 1988). (Contains several references.)
26. Scott, J. H., "Reduced Turbomachinery Operating Costs with Regular Performance Testing," ASME Paper No. 86-GT-173 (1988).
27. Meher-Homji, C. B., and Boyce, M. P., "Aerothermodynamic Gas Path Analysis for Health Diagnostics of Combustion Gas Turbines," Proceedings of the 1982 Conference on Detection, Diagnosis and Prognosis, National Bureau of Standards, Cambridge University Press (1983). (Covers a computational procedure for gas turbine performance analysis to iteratively compute air flowrate and T.I.T.)
28. Meher-Homji, C. B., and Mani, G., "Online Condition Monitoring Systems for Offshore Turbomachinery—System Design and Operating Experience," Proceedings of the 43rd Meeting of the Mechanical Failure Prevention Group, Symposium on Advanced Technology in Failure Prevention, National Bureau of Standards (October 1988).
29. Scott, J. H., "Axial Compressor Monitoring by Measuring Intake Air Depression," Third Symposium on Gas Turbine Operations and Maintenance, National Research Council of Canada Symposium (September 1979). (This is a classic paper outlining the method and experiences with intake depression monitoring.)

30. Becker, B., and Bohn, D., "Operating Experience with Compressors of Large Heavy Duty Gas Turbines," ASME Paper No. 84-GT-133 (1984).

31. "Parameter Monitoring for Corrosion Control of Utility Gas Turbines," EPRI Report AF665, Project 643-1, Electric Power Research Institute, Palo Alto, California (March 1978).

32. Krasij, M, et al., "Parameter Monitoring for Corrosion Control in Gas Turbines," ASME Paper 79-JPGC-6, ASME Power Generation Conference, Charlotte, North Carolina (October 1979).

33. Bowen, T. L., Guimond, D. P., and Muench, R. K., "Experimental Investigation of Gas Turbine Recuperator Fouling," ASME Journal of Engineering for Gas Turbines and Power, *109*, pp. 249-256 (July 1987).

34. Miller, J. A., "Mechanisms of Gas Turbine Regenerator Fouling," ASME Paper 67-GT-26 (March 1967).

BIBLIOGRAPHY

Felix, P. C. and Strittmatter, W., "Analysis of Pollution on the Erection Site of a Brown Boveri Gas Turbine," Brown Boveri Review, *66*, pp. 97-103 (February 1979).

Boyce, M. P., *Gas Turbine Engineering Handbook*, Houston, Texas: Gulf Publishing (1982).

Robbins, W. H., and Dugan, J. F. Jr., "Prediction of Off-Design Performance of Multistage Compressors," *in* "Aerodynamic Design of Axial Flow Compressors," NASA SP-36, ch X, pp. 297-310, (1965).

Stone, A., "Effects of Stage Characteristics and Matching on Axial-Flow-Compressor Performance, Trans ASME, pp. 1273-1293 (August 1958).

Scheper, G. W., Mayoral, A. J., and Hipp, E. J., "Maintaining Gas Turbine Compressors for High Efficiency," Power Engineering (8) pp. 54-57 (1978).

Bagshaw, K. W., "Maintaining Cleanliness in Axial Flow Compressors," 1st Gas Turbine Operations and Maintenance Symposium, National Research Council, Canada (October 1974).

Turner, B., "On-Stream Cleaning of Turbomachinery," *Proceedings of the Second Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1973).

Allen, H. L., "Gas Turbine Operation in the Real World," ASME Paper No. 84-GT-1 (1984).

Matthews, T., "Field Performance Testing to Improve Compressor Reliability," *Proceedings of the 10th Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1981).

Lages, F. P., "Air Filtration and Sound Control Systems for Gas Turbines - The State of the Art," *Proceedings of the 8th Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1979).

Pulimood, M. K., "Field Experience with Gas Turbine Inlet Air Filtration," ASME Paper No. 81-GT-193 (1981).

Ostrand, Gary G., "Gas Turbine Inlet Air Filtration," *Sawyer's Turbomachinery Maintenance Handbook*, First Edition, *3*, Chapter 10 (1980).

Nordberg, Dennis A., "Air Filters—Types, Selection and Application," *Sawyer's Gas Turbine Engineering Handbook*, Second Edition, 2, Chapter 5 (1972).

"Causes and Cures for Sulfidation Corrosion, Erosion and Fouling," Gas Turbine World, (March 1975).

Peterson, R. C., "Design Features for Performance Retention in the CFM56 Engine," *in* Turbomachinery Performance Deterioration, 4th ASME Fluid Mechanics Conference (1986).

Brown, G. J., "Engine Design to Minimize Foreign Object Damage," Proceedings of the Second Symposium on Gas Turbine Operation and Maintenance, National Research Council of Canada, Calgary (June 1977).

Boucher, G. R., "Current Gas Turbine Blade Coating Technology," Proceedings of the Fifth Symposium on Gas Turbine Operations and Maintenance, National Research Council of Canada (September 1983).

Ewins, D. J., "Studies to Gain Insight into the Complexities of Blade Vibration Phenomena," Paper No. C.184/76, Institution of Mechanical Engineers, U.K. (1976).

Rao, J. S., "Natural Frequencies of Turbine Blading: A Survey," Shock and Vibration Digest (October 1973).

Strub, R.A., "Field Measurements of Blade Stresses on Industrial Turbomachines," *Proceedings of the Third Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, College Station, Texas (1974).