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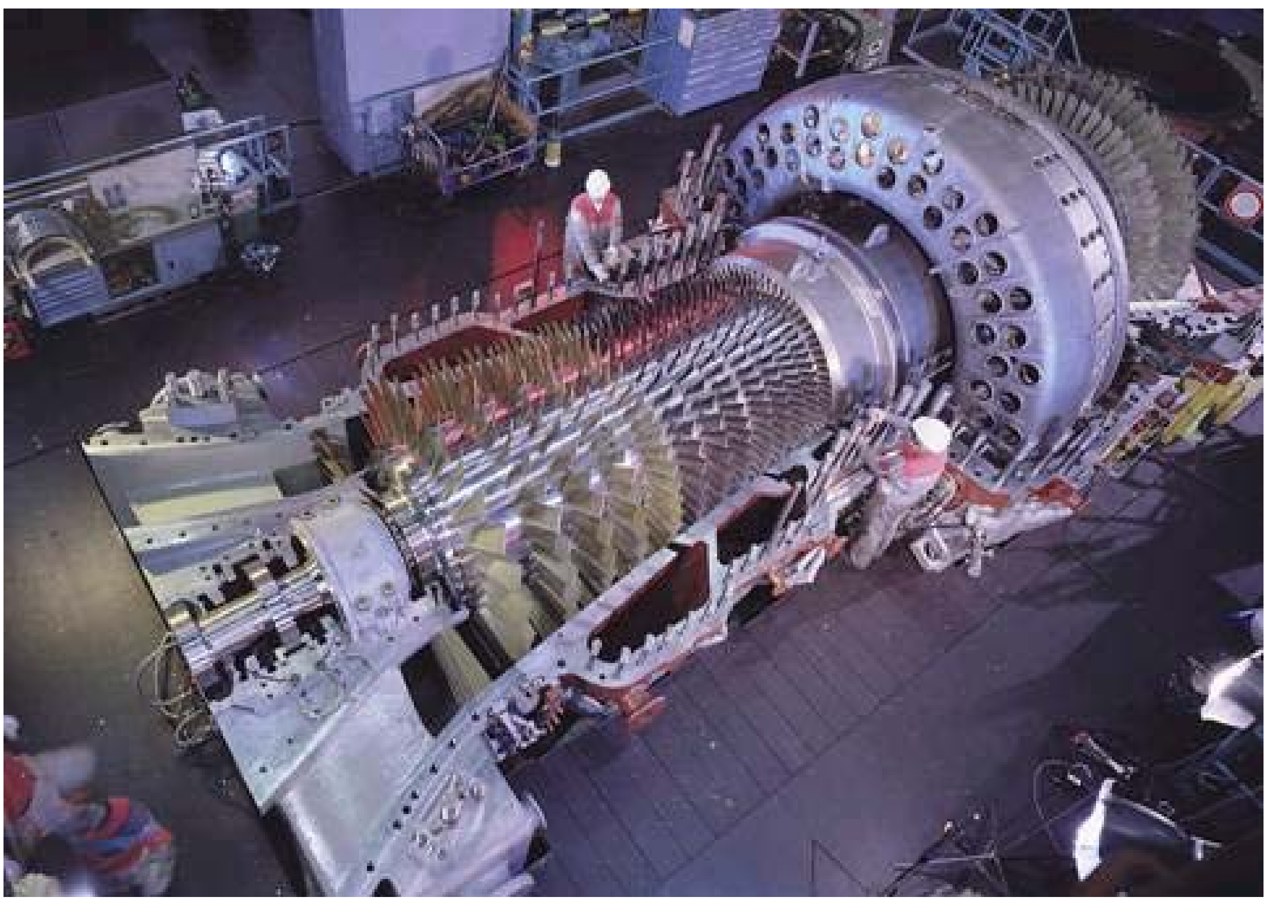
Axial Compressor Performance Maintenance Guide Update

1008325



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Axial Compressor Performance Maintenance Guide Update

1008325

Technical Update, February 2005

EPRI Project Manager

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PRODUCT DESCRIPTION

To deal with volatile fuel prices and growing pressures to limit green house gas (GHG) emissions, CT operators are striving for maximum fuel efficiency. The axial compressor is a leading cause of short term and long term CT efficiency losses due to fouling, corrosion and erosion. This report reviews the technology being advanced for compressor maintenance to achieve improved compressor and the CT efficiencies.

Results & Findings

Customized coatings to deal with site specific or engine specific issues are described in the report; hard surface coatings for erosion resistance, smooth anti-corrosion coatings for high aerodynamic efficiency and anti-fouling coatings that work together with water washing to maintain high efficiency. On-line wash systems are also described that use less water and are more effective.

Challenges & Objectives

The object of this report is to provide an easy-to-understand explanation of compressor operating principals and the corresponding O&M practices. An unprecedented growth in the use of combustion-turbine-based generating equipment, along with intensifying competition, has hastened the need for practical knowledge about the operation and maintenance (O&M) of combustion turbine equipment. Often people with little combustion turbine experience find themselves responsible for combustion-turbine-based units. Compressor thermodynamic and operating principals are complex, making it difficult for many CT managers to understand the relevance of several compressor O&M practices. As a result, several units have experienced damage that could have been avoided. Also, CT output can drop 5% to 10% within weeks without proper compressor maintenance. Hence, knowledge of axial compressor principals and practices is necessary to maximize profit from a CT power plant.

Applications, Values & Use

Operators, performance engineers and maintenance personnel can benefit from this report by implementing the new technology to improve efficiency resulting in increased profitability. The periodic restoration of airfoil surfaces as a result of over spray operation is shown to be cost effective when selectively done for periods of high spark spreads.

As operating data becomes available on the new water wash systems and coatings it is expected that further improvements will be made. It is anticipated that interest in compressor airfoil restoration will increase and the use of commercial aviation techniques will be employed to reduce the costs of maintenance.

EPRI Perspective

CT managers interested in improving profit and reducing risk of compressor damage should become familiar with the principals and practices outlined in this guide. The guide explains the principals of CT axial flow compressor operation in terms of blade aerodynamics and the "compressor map". Various performance loss mechanisms are explained. O&M practices are described including condition monitoring, damage mitigation, corrosion control, surge prevention, cleaning techniques, and coatings. In addition, EPRI's Combustion Turbine Performance and Fault Diagnostic Module or CTPFDM software (Product ID 1004971) provides combustion turbine operators with a low-cost, easy-to-install, easy-to-use program for monitoring axial compressor and overall turbine performance. Combustion turbine operators can use this update report and the CTPFDM software to diagnose the condition of axial compressors and to determine the benefits of maintenance actions.

Approach

Experts knowledgeable in CT design and operating principles surveyed published literature on issues related to axial flow compressors. These issues included performance loss mechanisms, corrosion, coatings, stall and surge prevention, cleaning techniques, and coatings. The experts then selected and edited the information for relevance to CT operators.

The goals of the report were to update CT operators regarding new technology that can improve the efficiency of their units and reduce operating and maintenance costs. Implementation of the new technology reaching the market place can help meet the goals.

Keywords

Combustion Turbines
Axial Compressors
Water Wash Systems
Compressor Coatings

ABSTRACT

This report contains an updated review of the state-of-the art in axial compressor maintenance to sustain high capacity and efficiency for gas turbines in power generation applications. The fundamentals of operation and causes of performance losses and surge margin loss due to degradation of the condition of the compressor are reviewed. The causative factors of performance degradation and loss in surge margin that are considered in the report are fouling, erosion, corrosion, changes in clearances due to creep and operational malfunction of the controls. Alternative direct and indirect techniques for monitoring the condition of the compressor and software to supplement this report are described. Instrumentation and analytical requirements for different condition monitoring techniques are described. Techniques to minimize operational and maintenance costs associated with washing a compressor are described.

Water wash system technology is reviewed for on-line or off-line operation and liquid or solid compound cleaners. A summary of the liquid and solid compound cleaners currently used is presented. A method to devise an optimum cleaning plan and interval for a particular site is also described. A summary of the state-of-the art of compressor coatings to retard corrosion and erosion and to provide smooth airfoils that yield more efficient operation is given. The impacts of non-recoverable losses and additional compressor maintenance on the operating profitability are estimated.

CONTENTS

FIGURES

TABLES

1 INTRODUCTION

Scope of the Guide

This guide reviews the fundamental aspects of axial compressor performance maintenance. The scope of this axial compressor maintenance guide includes:

- Performance Loss Mechanisms
- Compressor Damage Mitigation
- Condition Monitoring
- Cleaning Methods
- Coatings

Axial-Flow Compressor Design and Operation Fundamentals

The principal type of compressor used in gas turbines for power generation is the axial-flow compressor. Axial-flow compressor designs meet the needs of power generation gas turbines, namely:

- High Efficiency
- High Flow Capacity within a Relatively Small Structure
- High Pressure Ratio
- Reasonable Manufacturing Costs
- Robust and Reliable

The efficient design of an axial compressor requires a major aerodynamic design effort. First attempts (before the year 1935) to design axial compressors for use with gas turbines resulted in compressor efficiencies so low (on the order of 50%) that the engine would not run at all or had a thermal efficiency of only a few percent. Real progress in the design of axial-flow compressors was not realized until the potential of the turbojet engine was recognized, beginning in the mid-1930s. Since that time, funding for basic research and large scale test facilities, sponsored by the US and the UK, has led to the necessary basic design data and methods that are used today to achieve high efficiencies between 80% and 90%. The long historical development needed to achieve good designs is indicative of the sensitivity of the axial compressor performance to the fine details of aerodynamic design and construction (Refs. 1, 2, and 3). Once the gas turbine is placed in operation, it is found that the engine performance (efficiency and power) is sensitive to preserving the aerodynamic design against changes caused by fouling, corrosion, erosion, or damage of the blade and vane surfaces (Ref. 4).

The term "axial-flow" indicates that the air flow enters and leaves the compressor in a direction parallel to the axis of rotation. The air flow path is an annulus between the outer fixed stator casing and the rotating hub, with alternating rows of rotating rotor blades and fixed or variable stator vanes (VSV) as shown in Figure 1-1. The air flow turning angles, which play a key role in the affects of roughness and fouling on compressor performance degradation, are defined in Figure 1-1. The terms used to describe the air flow path in the blades and vanes at a particular radius are:

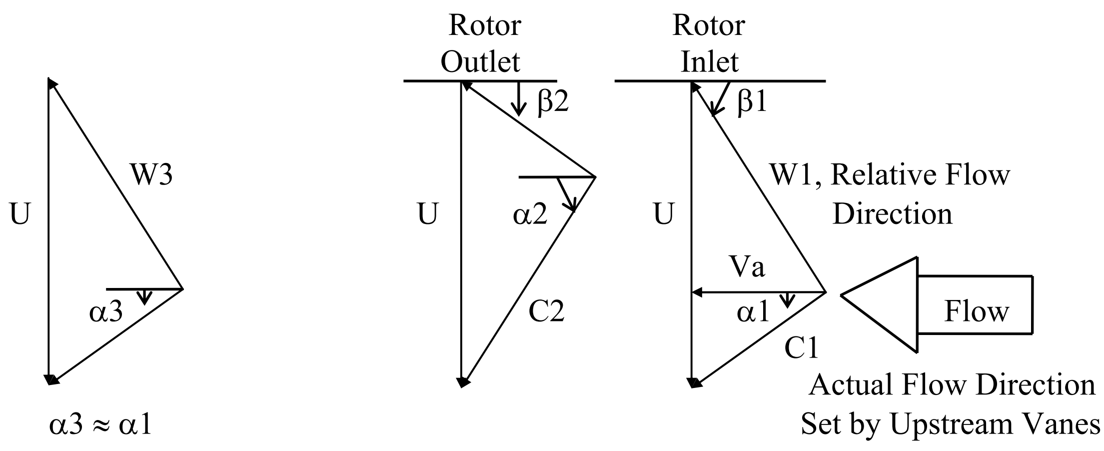
- C1 = actual inlet air flow velocity into a rotor blade row, fps
- C2 = actual outlet air flow velocity leaving a rotor blade row, fps
- U = actual blade speed, fps
- W1 = relative air flow velocity into a rotor blade row, fps
- W2 = relative air velocity leaving a rotor blade row, fps

The angles alpha (α) and beta (β) are defined by the flow vectors as shown in Figure 1-1.

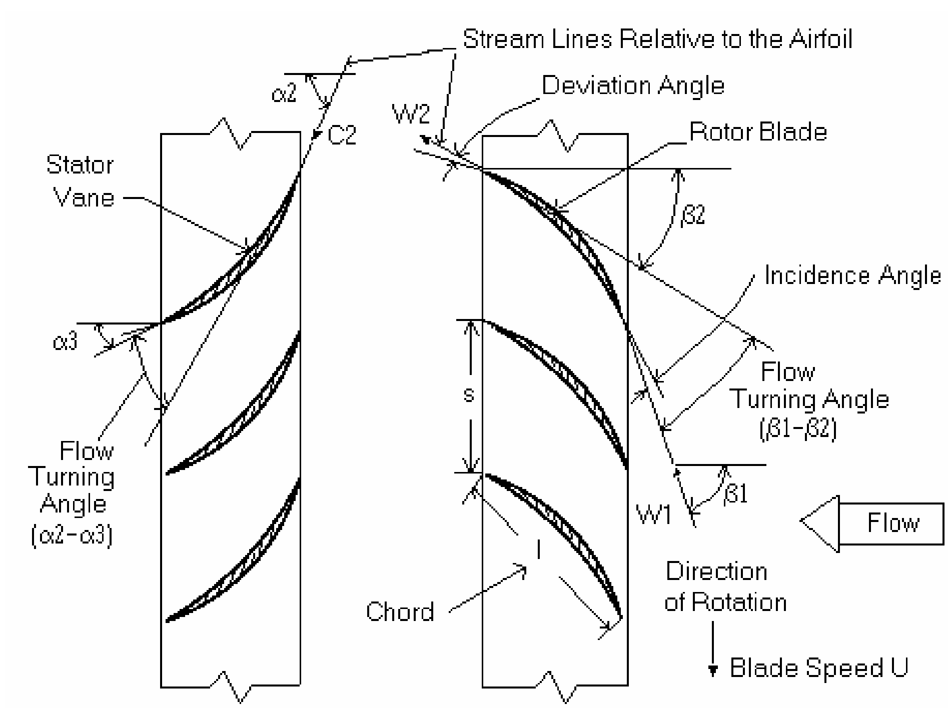
The shaft power imparted to the rotor increases the angular momentum and kinetic energy of the air flow by turning the air flow direction. The air flow turning angle in a rotor blade row is defined as $\beta 1 - \beta 2$ and in a vane row as $\alpha 2 - \alpha 3$. The turning angle is indicative of the amount of shaft power imparted to air. The reference dimension of a row of blades or vanes is the chord (Ref. l) and the spacing(s) of the vanes or blades.

Each consecutive pair of rotor blades and stator vanes forms a compressor stage. After the air is turned by the rotor blade, the stator vanes restore the flow direction. The kinetic energy imparted by the rotor blades to the air is efficiently diffused in the rotor blades and stator vanes to achieve the maximum pressure rise for a given power input. The air temperature rise across each stage corresponds to the amount of power added. Efficient turning of the air is best done in small steps, so the compressor has a number of stages, each raising the pressure of the air a small amount. To achieve high efficiencies of the overall multistage compressor, each stage has an even higher efficiency, typically in the 90 - 95% range.

The highest blade speeds are typically in the first blade row of a compressor and can exceed 1,000 fps (305 mps). The inlet air velocity (C1) to the first blade row is typically between 400 - 500 fps (122 - 152 mps), so the relative velocity to the rotor blade may be as high as 1200 fps (366 mps). The high relative flow velocities to the first stage rotor blades make them more likely to be subject to erosion and fouling as discussed in Chapter 2.



(a) Flow Velocity Triangle



(b) Flow Turning Angles

Figure 1-1
Axial-Flow Compressor Aerodynamic Conventions (NASA Aerodynamic Design of Axial-Flow Compressor, 1965, Ref. 1)

The operating conditions of the axial compressor in a gas turbine are constrained to match the flow resistance imposed by the combustor and turbine sections. The hot gases leaving the combustor expand through the turbine section that drives both the compressor and the generator. In order to maximize the turbine power, the turbine inlet flow area is reduced through the first row of turbine inlet vanes to a limiting "throat" area causing a back pressure on the compressor.

The compressor discharge pressure is sufficient to overcome the pressure drop caused by the combustor and to pass the gas flow through the throat area. The ratio of the discharge total pressure to the inlet total pressure is the compressor pressure ratio. The addition of fuel, water injection, or steam injection into the combustor causes the pressure ratio to increase.

The required pressure ratio that matches the combustor and turbine flow resistance forces each compressor stage to produce the amount of air turning and pressure rise needed. There are aerodynamic limitations on the pressure rise of each stage and if these are exceeded, the flow through the compressor can stall and become unsteady with a potential for reversing flow direction (pulsation and surging) that can cause severe damage.

Theoretical considerations show that compressor performance over the complete range of operating conditions can be described in terms of a compressor map of pressure ratio, corrected air flow, and corrected speed (Ref. 1) as shown in Figure 1-2. The map shows the corrected air flow, pressure ratio, surge margin, and efficiency for an expected baseline operating line that satisfies all the matching conditions described above. (This is the dashed line in Figure 1-2.)

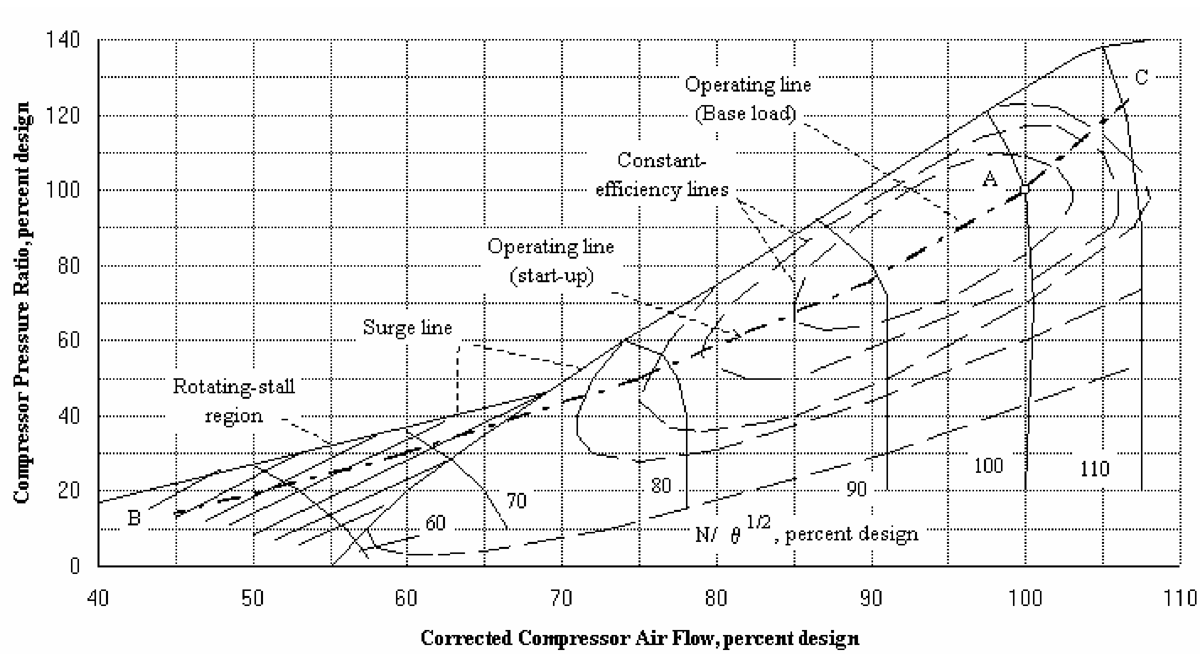


Figure 1-2
Typical Multi-Stage Axial Compressor Operating Map Features (NASA Aerodynamic Design of Axial-Flow Compressor, 1965, Ref. 1)

The efficiency referred to in the compressor map is the isentropic or adiabatic efficiency. Loci of constant efficiency tend to form ellipses, where the smaller ellipse has the highest efficiency. The efficiency relates the ideal power required to raise the air pressure to the actual power required. Humidity effects are usually very small except in climates where the ambient temperature and relative humidity are both high. The term $N / \theta^{1/2}$ is the corrected speed of the compressor where $\theta = (T_{in} / Mw) / (T_{in} / Mw)_{ref}$ and N = rpm. Aerodynamically, the compressor performance (efficiency) at any operating condition is dependent on the corrected speed and the pressure ratio. The corrected speed term accounts for the effects of rpm, inlet temperature (T), and specific humidity (molecular weight) of the air on the efficiency.

The lines of constant corrected speed are called "characteristic lines". The characteristic lines are vertical at high flow, indicative of a maximum flow condition at the inlet (also known as choking), and curve to lower flows with negative slope, whereas the operating line has a positive slope. The intercept of the operating line and a characteristic indicates a stable operating point. Note that the surge line is indicated at the maximum pressure ratio and minimum flow conditions for stable operation. As shown in Figure 1-2, the compressor is able to operate in a rotating stall condition without surge. The degree of stability of the compressor is related to the slope of the characteristic lines. As the slope tends to the horizontal, the stability diminishes and the disposition to surge increases.

The base load operating line will pass through the 100 percent design point (point A) where the compressor design is optimized for high efficiency at high pressure ratio and flow. At 100% corrected speed, pressure ratio and air flow is governed by the 100% characteristic line passing through the design point (higher pressure ratio caused by water or steam injection will reduce air flow or could increase the potential for surge). The distance between the operating line and surge line is indicative of the "surge margin". In cold weather, the lower inlet temperature can cause the base load operating point to move to a higher corrected speed (point C), with a potential for significantly lower efficiency as shown in Figure 1-2.

During startup, point B to point A, the operating line, may pass through regions of potential stall and low surge margin. The manufacturer usually incorporates flow control during startup by using variable inlet guide vanes (IGVs) or variable stator vanes (VSVs) and/or bleed valves. The reduced flow allows the compressor to operate safely without stall or surge. As compressor speeds approach the design point, the IGVs are opened and the bleed valves are closed. For more details on the theory of axial compressors, the reader is referred to Ref. 5.

The use of water injection into the compressor is being exploited to reduce the compressor power and enhance the CT power output due to the inter-stage cooling benefit (Refs.6 and 7). As the water evaporates in the compressor, the lower air temperature results in changes of the velocity triangles such that the airfoil incidence angles change as shown in Figure 1-3 and Figure 1-4 (Ref. 8). These results are based on a simulation of a 17-stage axial compressor using a classical pitch-line stage stacking calculation that defines the velocity triangles at the design point and then calculates the changes in the velocity at off-design with inter-stage cooling. The changes caused by the water injection limit the amount of water injection to less than 2% of the air flow to avoid excessive incidence angles that could lead to stall. Ideally, if all the water evaporates so as to saturate each stage, then all the water would evaporate by about the 5th stage (Ref. 8), but there may be some relatively large droplets present in the spray that do not evaporate until later. The gains in CT power output with inter-stage cooling may be as much as 14% due to inter-stage, but there is some loss in stall margin and there is increased maintenance due to airfoil erosion.

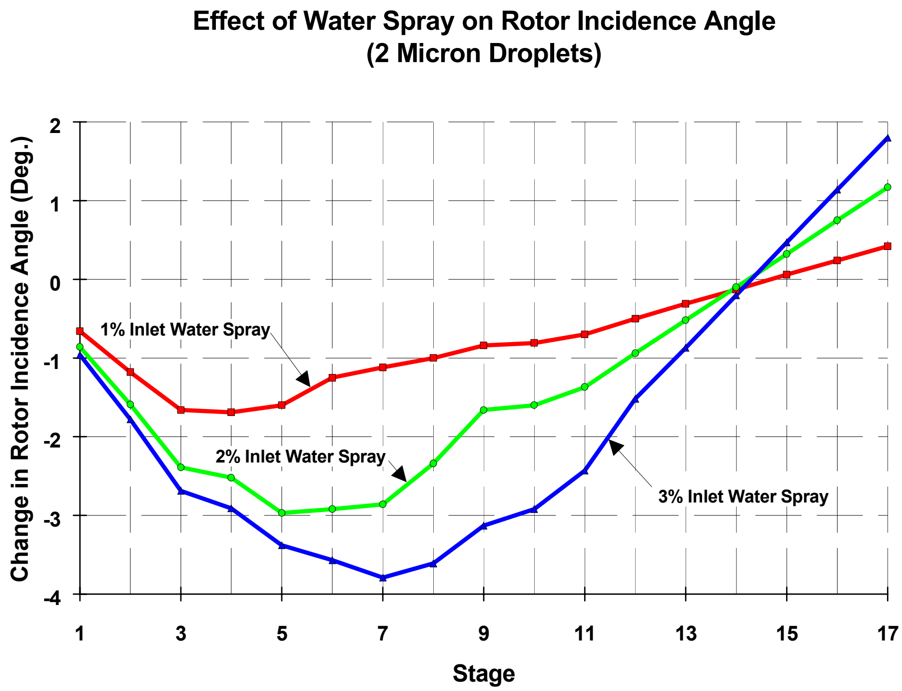


Figure 1-3
Effect of Inter-Stage Cooling Spray on Rotor Incidence Angles (Ref. 8)

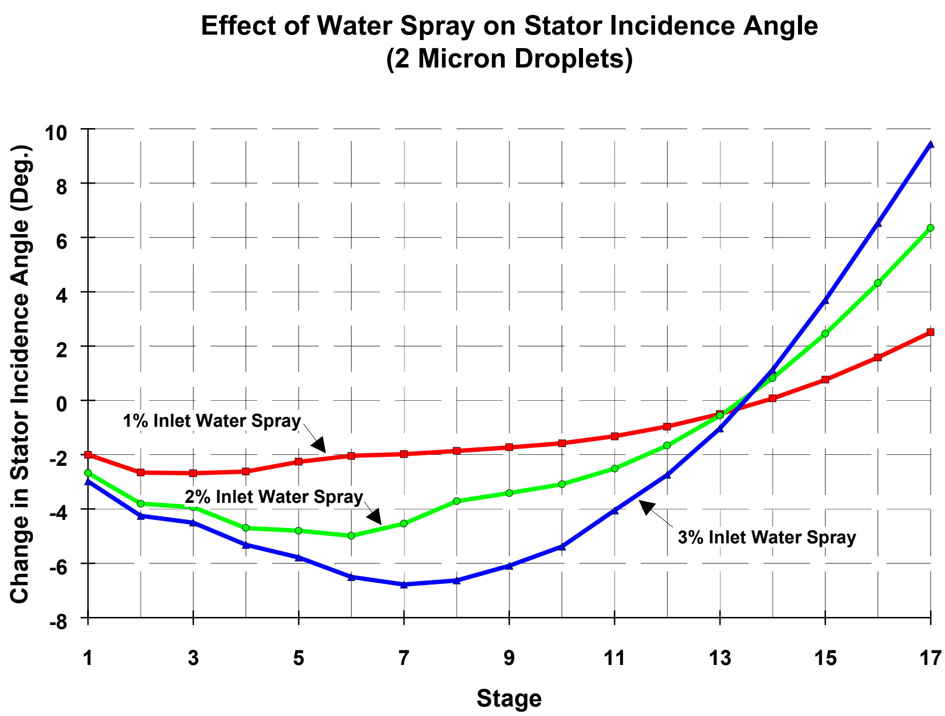


Figure 1-4
Effect of Inter-Stage Cooling on Stator Incidence Angles (Ref. 8)

2
PERFORMANCE LOSS MECHANISMS

Causes of Axial Compressor Performance Losses

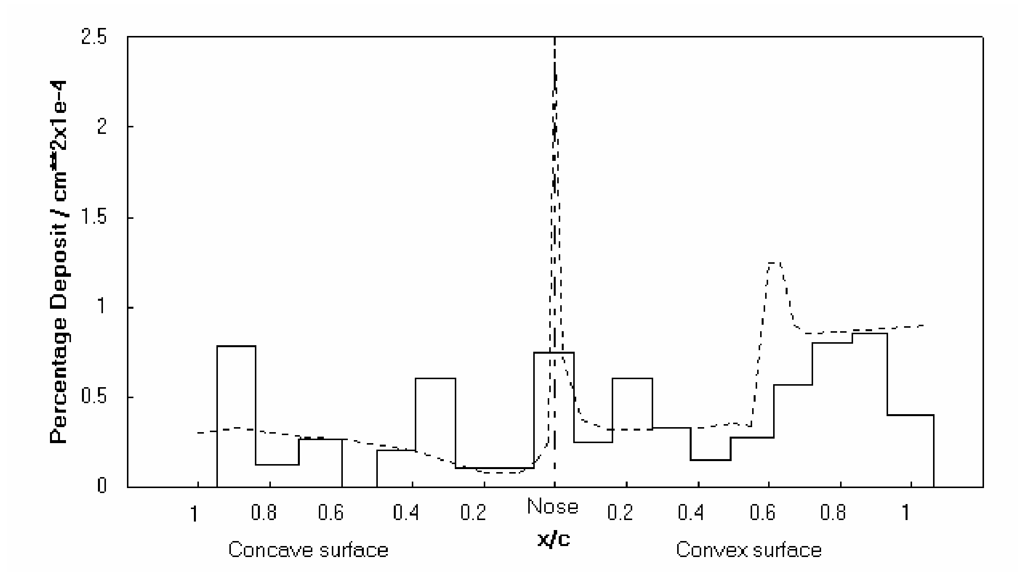
The performance loss mechanisms include:

- Fouling of Airfoil Surfaces
- Corrosion of Airfoil Surfaces
- Erosion of Airfoil Surfaces
- Peripheral Equipment Degradation
- Distortion and Wear of the Casing and Rotor Assembly
- Foreign or Domestic Object Damage

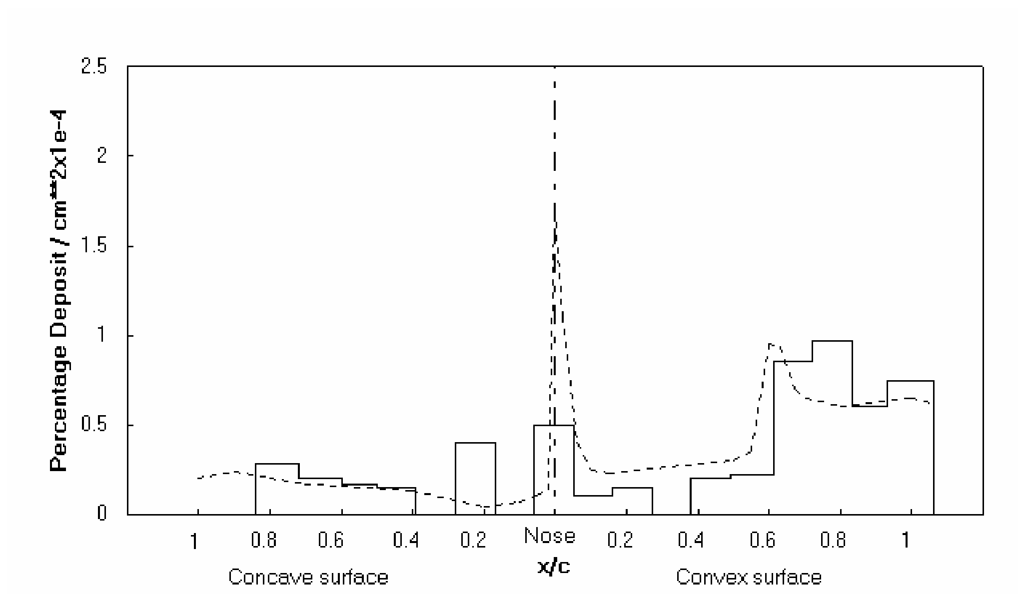
Airfoil Fouling Phenomena

Compressor fouling is due to the size, amount, and chemical nature of the aerosols in the inlet air flow, dust, insects, organic matter such as seeds from trees, rust or scale from the inlet ductwork, carryover from a media type evaporative cooler, deposits from dissolved solids in a water spray inlet cooling system, oil from leaky compressor bearing seals, ingestion of the stack gas or plumes from nearby cooling towers.

Studies of the flow phenomena that cause compressor fouling phenomena are described in Refs. 9 and 10. Fine particles adhere to rotating blades due to their stickiness that is sufficient to hold the particles even on the rotating blades where the centrifugal forces are high. Particles less than 1 micron in diameter diffuse to the surface in proportion to their original concentration and in accord with random Brownian motion. Transport of larger particles to the surface is enhanced by turbulent mixing and diffusion. Sample results of the estimated deposition rates for different regions of the blade surface are shown in Figure 2-1. Relatively low deposition rates are found for the leading edge region. The highest deposition rates are on the suction surface near the trailing edge where the boundary layer is thick and turbulent. As some diffusion (adverse pressure gradient) effects may be present on the suction side near the trailing edge, there is the potential for some reverse flow that could drag particles forward from the wake.



(a) Particle Mass Median Diameter 0.13 µm



(b) Particle Mass Median Diameter 0.19 µm

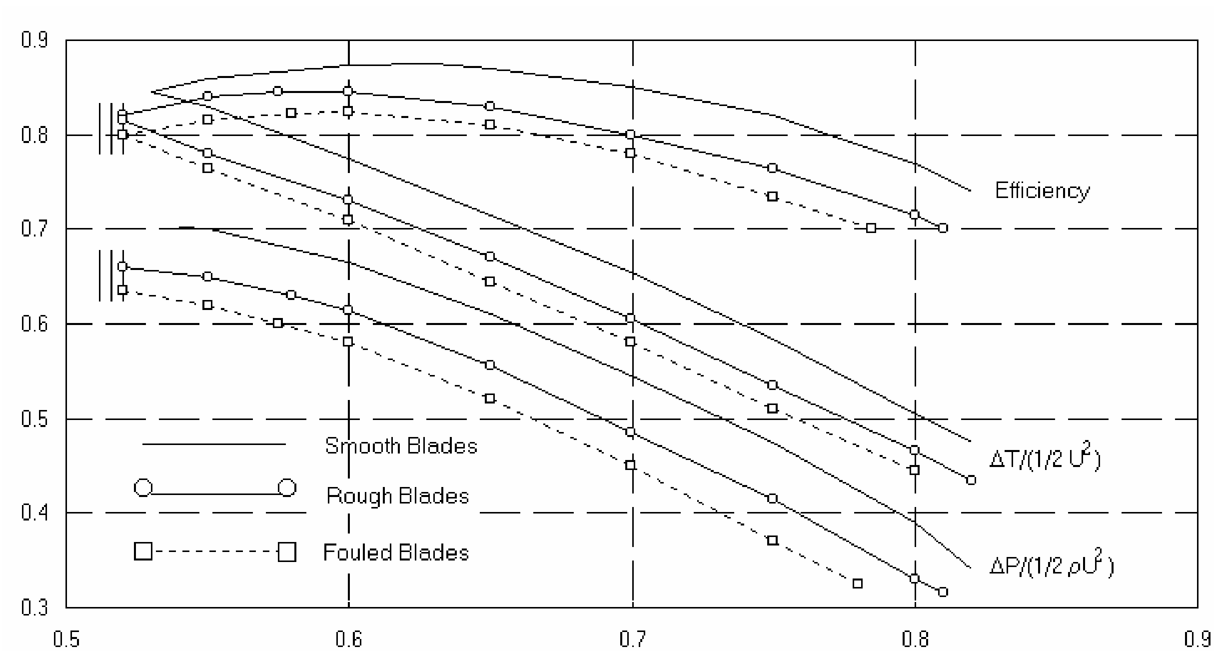
——————— Experimental Results

- - - - - - - - - Theoretical Predictions

Figure 2-1
Fouling Deposition Rates on Axial Compressor Airfoil (Parker & Lee, 1972, Ref. 10)

Effects of Fouling on Compressor Performance

The effects of fouling and adding roughness to compressor blades were carefully tested by using a special laboratory model and instrumentation (Ref. 11). The test results shown in Figure 2-2 indicate that either fouling or roughness decreases the efficiency, reduces the pressure rise (as measured by $\Delta P / (\frac{1}{2}\rho U^2)$), and reduces the power added (as measured by $\Delta T / (\frac{1}{2}U^2)$). Careful flow measurements of the boundary layer indicate that the roughness caused it to thicken much more on the suction side than on the pressure side (Ref. 12). This boundary layer phenomenon causes the blade to take on the aerodynamic character of a different shape, one where the blade is thicker, but also has less curvature (less camber) that reduces the flow turning angle, reducing the amount of power the blade can transmit to the air. The effects of varying the severity of roughness on airfoil frictional losses and flow turning are shown in Figure 2-3 and Figure 2-4.



Ratio of Axial Air Velocity to Rotor Blade Velocity

Figure 2-2
Effect of Fouling and Surface Roughness on Axial Compressor Performance (Turner & Hughes, 1956, Ref. 11)

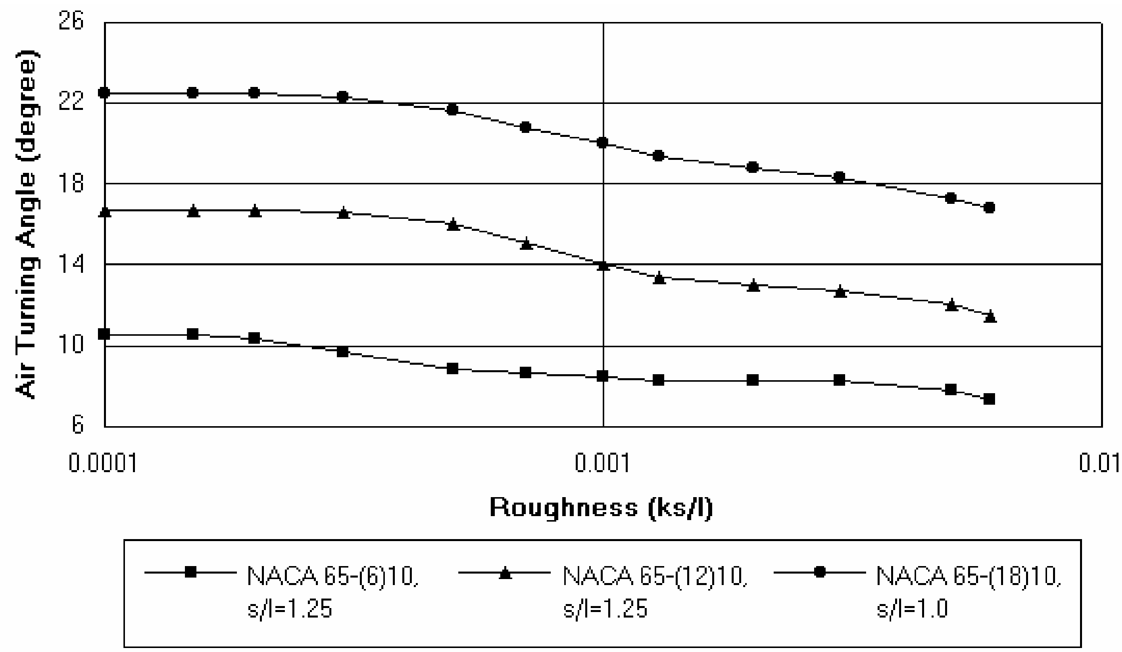
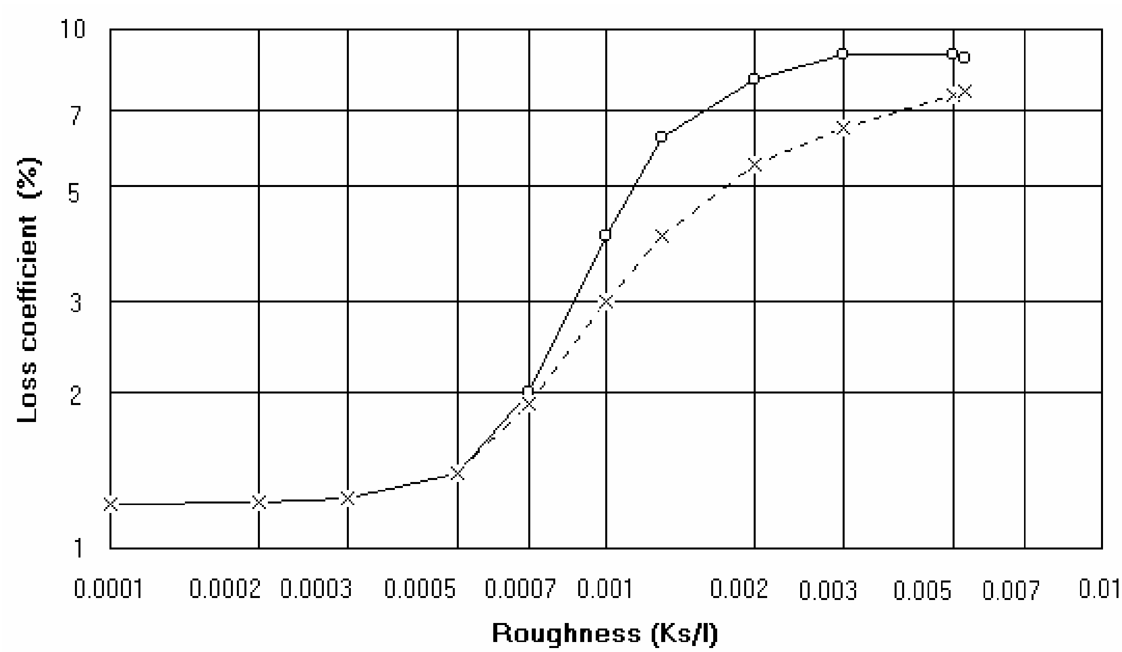


Figure 2-3
Effect of Surface Roughness on Air Flow Turning Losses, $R_e = 4.3 \times 10^5$ (Bammert & Milsch, ASME 72-GT-48, Ref. 12)



o - From wake survey
× - From boundary layer measurement

Figure 2-4
Effect of Surface Roughness on Flow Friction Losses for s/l = 1.25, $R_e = 4.3 \times 10^5$ (Bammert & Milsch, ASME 72-GT-48, Ref. 12)

The application of these results to an axial compressor implies a loss in efficiency due to the increased losses and reduced air flow and surge margin due to the reduced flow turning.

The conclusion that roughness causes reduced flow and reduced surge margin requires further explanation. An axial compressor running at constant speed has the characteristic behavior that reduced air flow always accompanies an increase in pressure ratio. Consider that a clean compressor is operating at a certain pressure ratio and air flow. As roughness is added, the flow turning capability is reduced, so the pressure rise is not adequate to match the back pressure. This condition forces a change to a lower flow that helps to match the back pressure condition in two ways: 1) reduced flow reduces the matching back pressure, and 2) reduced flow increases the turning angle capability as the incidence angle increases. A new equilibrium condition is found that satisfies all the matching requirements, but at a reduced flow. The reduced flow and higher incidence angle causes an increase in frictional flow losses and decreases the airfoil stall margin as shown in Figure 2-5 (Ref. 13). If the design of the compressor has a small stall range for certain blade rows, this may lead to a rotating stall and/or surge condition when fouling occurs.

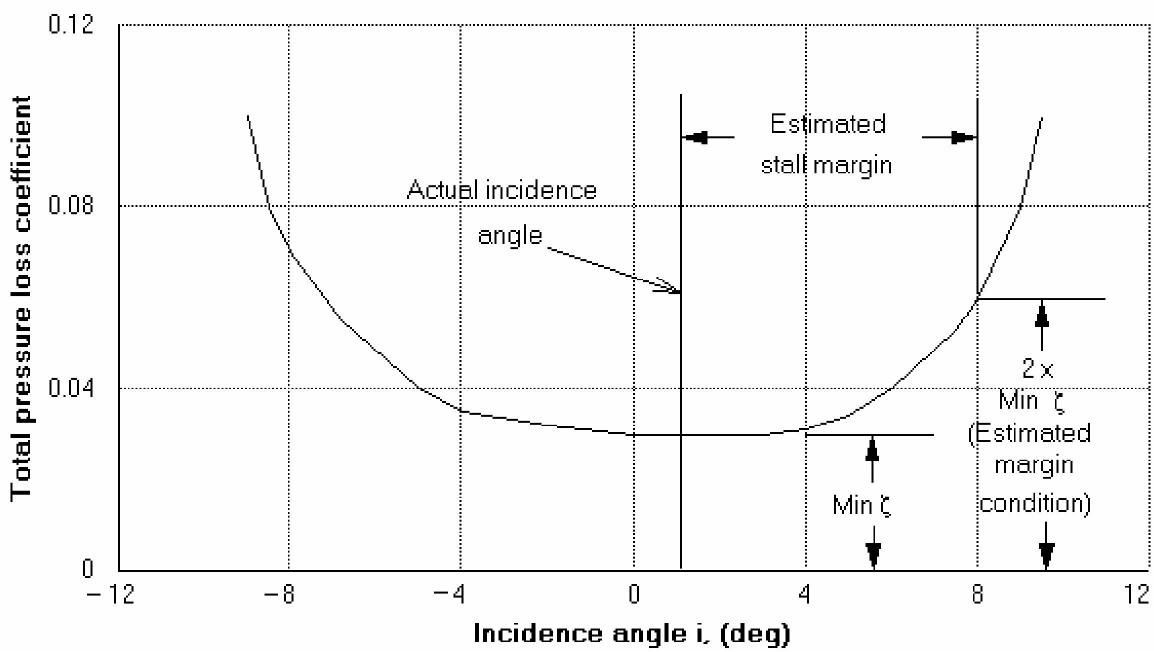


Figure 2-5
Effect of Incidence Angle on Airfoil Friction Losses and Stall Margin (Lieblein, 1960, Ref. 13)

The concept that roughness is akin to fouling raises the possibility of directly monitoring the condition of the blade. Inspection of Figure 2-3 and Figure 2-4 shows that the blade losses change very little until a critical amount of roughness is added, corresponding to a Reynolds number of 90 based on roughness height (Ref. 14). This result suggests the possibility of monitoring roughness and fouling due to deposits on vanes and blades and correlating this with a loss criterion to optimally control the cleaning interval. The results of roughness and fouling testing also suggest that the effects of roughness and fouling may be more critical on the latter stages of a compressor where the Reynolds number is higher due to the increased density.

Methods to mitigate the effects of fouling and surface degradation are covered in Chapters 5 and 6.

Corrosion Phenomena

Corrosion in axial compressors may be due to chemical or electrochemical actions. Chemical actions can occur in an industrial environment where sulfur or chlorine may be present in the air. Electrochemical action is more common, where a film of moisture coats the airfoils, forming a galvanic cell between sites of different electromotive force. Dissimilar metals such as zinc and copper form a galvanic cell where zinc, having the higher electromotive force (emf), becomes the anode and copper the cathode. The reaction proceeds where the anodic material dissolves in the form of positively charged ions which plate out as hydrogen ions on the cathodic metal surface. In the case of a single material, small potential differences arise due to variations in composition, surface finish, stress, deposits, or inclusions. Penetration proceeds at the anodic sites resulting in a wide variety of pitting, roughening, and material loss (Ref. 15).

Corrosion test results given in Ref. 16 illustrate that the corrosion phenomena are sensitive to the type of material, coating, atmospheric concentrations of moisture, salt, sulfur compounds, and maintenance techniques. See Chapter 6, Compressor Coatings, for more details.

Corrosion of the compressor airfoils can result in a pitted and roughened surface (Ref. 16) sufficient to lower compressor efficiency, lower air flow, and reduce surge margin. Corrosion proceeds, even if the gas turbine is not running, in climates where moisture condenses on the surfaces and where aerosols contain salt, sulfur, or chlorine. Pits are dangerous, as they cause stress concentrations and act as fatigue initiation sites. Once pits are created, they tend to grow as salts and moisture collect in them.

Coatings to reduce corrosion are covered in detail in Chapter 6.

Erosion Phenomena

Erosion can cause a change in the airfoil leading edge and trailing edge dimensions and shape, the airfoil thickness distribution, roughness, and tip clearance (Ref. 17). All of these effects may have a serious effect on performance and the surge margin.

Analytical and experimental studies of the erosion mechanisms are reported in Refs. 18 to 20. These studies indicate that the erosion rate depends on the following:

- Liquid or Solid Particles
- Impact Velocity
- Impact Incidence Angle
- Concentration of Particles
- Size of Particles
- Surface Material

The damage mechanism is fundamentally different for solid and liquid particles. The maximum damage due to solid particles occurs at an impact incidence angle of 35 degrees relative to the surface, resulting in damage that is primarily scoring of the surface (Ref. 18 and 19). The erosion tends to be higher near the leading and trailing edges of the vanes and blades.

There is a potential for water droplet erosion in compressors when using a spray or overspray evaporative cooling system. Droplets may coalesce, impact, and create a surface film on vanes that is carried to a trailing edge where large droplets may be shed onto downstream blades. Droplets larger than 30 microns in diameter are likely to impact the surfaces of the compressor before they have enough time to evaporate (Ref. 6). Further discussion of airfoil erosion is presented in Chapter 5 with regard to the use of on-line water washing.

The results of research on erosion effects of water droplets in steam turbines (Refs. 21 and 22) are assumed to be applicable to understanding the potential for erosion in axial compressors. Estimates of the droplet sizes shed from the trailing edge of an airfoil with a liquid film are on the order of 100 - 200 microns. Since all industrial gas turbines have inlet guide vanes, the accretion of droplets on these vanes is likely unless the droplets are very small, on the order of 10 microns or less. The droplets that are shed from the trailing edge have a low velocity and will likely impact the rotor blades, as there is not enough time to accelerate the droplets. Impacts on the first stage rotor are of primary concern, as the tip velocity and relative flow velocity are high. Because of the low velocity of droplets that are shed from the vanes, the impacts on the blade will tend to occur on the concave (pressure) side. The maximum damage from liquid droplets occurs at a 90 degree incidence angle. For soft and ductile materials, the resulting damage is plastic indentation of the material with subsequent failure of the material by low cycle fatigue. For hard materials, the damage is in the form of spalling and pits caused by shear stresses induced below the surface. There appears to be a velocity threshold for damage of about 1000 fps for stainless steels (Ref. 22), which suggests that pitting may occur in the rotor tip region, which in turn would cause losses due to roughness. Materials with high Brinell hardness and good low cycle fatigue strength have the best resistance to erosion. Titanium and stainless steels are good material choices.

Use of Inlet Filters

The effects of erosion on compressor performance and engine life may be nullified by the use of inertial and mechanical air filters. There appears to be a threshold such that erosion caused by particles smaller than 10 microns in diameter is negligible (Ref. 19). Efficient air filters can remove dust particles above two to three microns in diameter. Air sampling tests are needed to assess the concentration of particles and size distribution and form the basis of a suitable filter specification.

Advances in the design of inlet filters have resulted in high efficiency (as shown in Figure 2-6), low pressure drop, and ease of maintenance. The use of inertial filters with a pressure drop of approximately 1 inch w.c. (2.5 mbar) can remove essentially all the particles over 10 microns in diameter and hence virtually eliminate concern of erosion. Manufacturers of engines with dry low emissions (DLE) combustors may also have an inlet air specification requiring that the particle content size be limited, as these combustors are often designed with very fine holes to mix the fuel and air. Using a high efficiency filter can increase the pressure drop to 2 - 3 inches w.c. (5 - 7.5 mbar) and introduce added maintenance to avoid excessive pressure drops as the filter becomes loaded with particles.

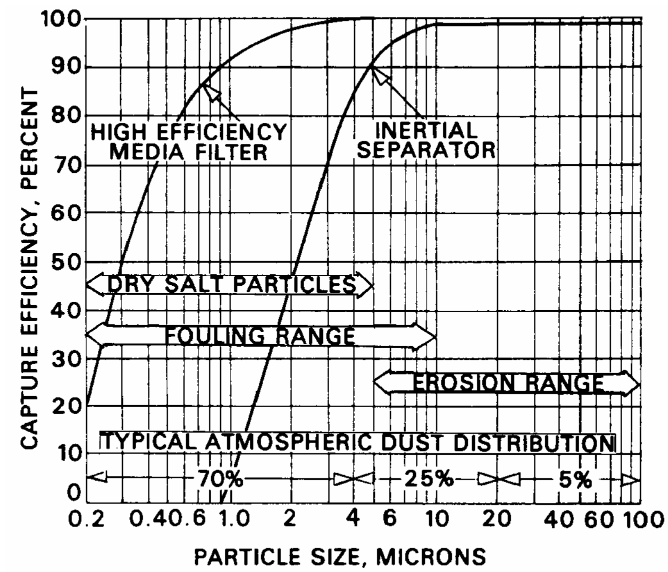


Figure 2-6
Effective Filter Range (Sawyer, Sawyer's Turbomachinery Maintenance Handbook, 1980)

To overcome the maintenance requirements, self-cleaning filter systems are offered where air is periodically pulsed in the counter-flow direction to blow off the accumulated particles and collect them at the bottom of the filter. Operation in very humid conditions may also increase the pressure drop if the filter media is hydroscopic. To guard against excessive pressure drops, bypass inlet doors are installed. While these doors limit the pressure drop, they also are susceptible to leakage of air around the filter and must be periodically inspected to ensure that leakage is minor. Considerations for the use of inertial filters include:

- Reduced erosion of thin corrosion resistant coatings for the compressor and hot section
- Reduced erosion of coatings used to enhance the smoothness of the compressor airfoil surfaces
- Reduced erosion or plugging of fine cooling air holes in the hot section
- Reduced erosion of thermal barrier coatings
- Reduced erosion of blade tips

Considerations for the use of high efficiency filters include:

- Reduced fouling
- Reduced salt particle ingestion

Some selected features of various filter suppliers are given below:

American Air Filter (AAF)

Contact
AAF
10300 Ormsby Park Pl., Suite 600
Louisville, KY 40223
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Peripheral Equipment Degradation

The maintenance of the following peripheral equipment is important (Ref. 23):

- Inlet Guide Vane (IGV) or Variable Stator Vane (VSV) Actuation Systems
- Compressor Air Bleed Valves
- Inlet Filter
- Evaporative Cooler
- Silencers
- Seals

The IGV or VSV actuation systems should be inspected regularly for proper setting and looseness. Improper settings may reduce the operating limits on surge and reduce air flow and power. Looseness in the control linkage could result in flutter (vibration) of the vanes causing high cycle fatigue and flow pulsations.

If the compressor is equipped with bleed valves that bypass flow during start-up to prevent stall or surge, then periodic maintenance of the valve should be considered to prevent it from becoming stuck in an open position. Performance losses due to an open valve emulate those due to a fouled compressor.

Monitoring the compressor performance should include monitoring the inlet filter pressure drop to help identify the source of the performance losses. The filter should also be inspected periodically to determine if it has degraded and is allowing some inlet air to bypass it or if there is a potential for loose or corroded parts being ingested into the compressor.

The use of media-type evaporative coolers poses the risk of untreated water being carried over to the compressor that can cause deposits and corrosion. Over time, the media may become loaded with deposits left by the evaporation of the cooling water, causing blockage and an increase in the cooler pressure drop and reduced efficiency of the cooler. Spray-type coolers using de-mineralized water should be periodically inspected to ensure that the water quality is being maintained to avoid build up of deposits on the airfoils.

Periodic inspection of the silencers is needed to monitor their condition. Deterioration of the silencers can result in ingestion of rust, metal fragments, and sound absorption material.

The failure of bearing seals may result in oil vapors escaping and entering the compressor inlet. Clean seal air is supplied to the bearings and if the seal air pressure is inadequate or the seal is defective, then oil vapors may escape. Since the air pressure in a compressor inlet bellmouth drops below atmospheric pressure, it is possible for oil vapors to travel through structural joints to enter the inlet. Oil vapors that escape from bearing seals can cause rapid and serious fouling as oil deposited on an airfoil surface creates a sticky surface for deposits to form.

The wear of labyrinth seals at the rear of the compressor may lead to leakage, causing a significant power loss. Performance losses due to leakage emulate those due to a fouled compressor.

Distortion and Wear of the Compressor Casing and Rotor Assembly

Compressor casings are subject to distortion due to the added stiffness in the flange areas so that ovaling can result after a long period of operation. This distortion can lead to blade rubs and increased clearances and degradation of performance.

The rotor assembly may incur rubs due to bowing of the compressor shafting, or wear and looseness of the blade attachments. The blade rubs lead to increased clearances and degradation of performance.

Foreign or Domestic Object Damage

Damage to the blades due to ingestion of small metal parts can lead to increased vibration and performance degradation. A sudden change in the vibration level or performance of the compressor is indicative of object damage and visual and/or borescope inspections are needed to resolve the cause of the change.

3
COMPRESSOR DAMAGE MITIGATION

Background

Severe compressor damage may occur in spite of our knowledge of the potential causative factors such as:

- IGVs and bleed valves that do not operate properly causing excessive rotating stall loads or surge during startup
- High cycle fatigue caused by accelerating through regions where resonant torsional and/or lateral bending modes occur during a start
- Pitting due to corrosion introduces a stress concentration that significantly reduces the high cycle fatigue life of a blade
- Excessive vibration caused by excitation of a vibration mode during rotating stall
- Surge due to the deterioration of the compressor condition such as fouling, erosion and roughness
- Surge due to exceeding the recommended operating limits on fuel rate (acceleration), water or steam injection
- Coatings and airfoil erosion due to the use of excessive cleaning compounds, particularly solid cleaners
- Excessive erosion and/or fouling resulting from fogging or media type evaporative coolers

At the rated operating condition, the rear stages of the compressor experience an increase in density that results in specific aerodynamic design conditions for the blades and vanes to operate efficiently. During a start, the density is low in the rear stages, resulting in a mismatch of the aerodynamic design needed to avoid stall at all speeds. To overcome this mismatch, a designer may use IGVs to reduce the air flow at low speeds and bleed valves to bypass some of the air flow and thereby reduce the flow velocities in the rear stages to avoid stall and surge.

Improper settings and operation of the IGVs and bleed valves can lead to rotating stall and surge. As shown in Figure 1-2, rotating stall may occur and persist during starts up to and beyond 70% of the rated rpm. The rotating stall pattern may have many cells, say 5 for example, resulting in pulsation frequencies that can range from approximately 1/3 to twice the shaft rpm, potentially exciting a resonant vibration mode of a blade.

Determination of the causes of vibration-related blade failure can be very complicated depending on the involved vibration modes, excitation frequency, magnitude of the loads, high cycle fatigue strength of the material, and the effects of corrosion pitting that results in stress concentrations. During a start, resonant blade vibrations excited by blade passing frequencies may also occur. If excessive time is spent in the resonant condition during a start, then the life of a blade may be seriously reduced.

Preventative maintenance actions are:

- Monitoring changes in the vibration signatures of the fore and aft compressor bearings during a start
- Periodic inspection of the air foil surfaces for pitting and erosion
- Inspection of the corrosion coating
- Check of the operation and settings of the IGVs and bleed valves
- Periodic compressor washing to maintain clean compressor conditions

Stall and Surge

Audible flow fluctuations or rumbling noise emanating from an axial-flow compressor are commonly referred to as surge. Certain flow fluctuations are not due to surge, but are a result of a phenomenon called rotating stall. The term surge refers to flow fluctuations different from rotating stall. Surge involves fluctuations in the net flow through the compressor, whereas rotating stall is characterized by low flow zones around the flow annulus of the compressor, but with a constant net average flow through the compressor (Ref. 24).

Stall may be classified as either progressive or abrupt stall. Progressive stall is characterized by a gradual reduction in stage pressure ratio as the flow is reduced and generally results in a rotating stall pattern consisting of more than one stall zone rotating around the flow annulus of the compressor. Abrupt stall results in a discontinuous drop in compressor pressure ratio and efficiency and has a single zone pattern. When an abrupt stall occurs, there is a sudden drop in pressure ratio coincident with a sudden drop in flow, followed by an increase in flow as the pressure drops and a low frequency limit cycle flow pulsation (surge) may occur.

As shown in Figure 1-2, rotating stall can occur over the entire range of flow for low compressor speeds, and the compressor will operate without surge, but there will be pulsating loads on the blades due to the stall. Surge can only occur when the compressor operating condition is unstable. Stable operation is dependent on the slope of the speed line and the aerodynamic damping. The flow through the compressor decreases as the pressure ratio increases at constant speed, so the slope of the speed lines for stable operation, as shown in Figure 1-2 (pressure ratio vs. flow), are all negative. The slope of the speed lines becomes less negative and can approach zero as the flow is reduced. Therefore, the compressor operating surge limit line occurs at the operating condition of minimum flow and maximum pressure ratio. The slope of the speed line at the surge line may still be negative, so as to leave a surge margin of safety. The surge margin will increase if the load is reduced (reduced fuel and firing temperature) or if some of the compressor discharge air is by-passed (dumped) to the exhaust, thereby reducing the pressure ratio.

Surge Prevention

Preventative maintenance on the gas turbine is important to ensure that the surge margin is not reduced by the degradation of the compressor, control system, and turbine section. As noted in Chapter 2, compressor fouling, corrosion, and erosion of the airfoils reduce their design flow turning ability and increase the flow friction losses, causing a reduction in air flow and in the stable (stall and surge free) operating range. While the OEM provides a surge margin to accommodate some performance degradation, maintenance is needed to guard against an excessive loss in margin. Monitoring the compressor performance of air flow by trending measured vs. expected air flow is useful to define a cleaning schedule that minimizes the flow losses. Inspection of the vane and blade surface, IGV settings, and clearances should be done periodically to guard against a performance degradation that may reduce the surge margin. If rotor tip clearances increase due to erosion, damage, or creep, then more air is by-passed around the rotors, so the blades need to turn the flow more to compensate for the added clearance losses and achieve the required operating pressure ratios. As noted in Chapter 2, increasing the turning of the air flow causes a reduction in the stall margin.

Periodic checking of the fuel flow control operation is important to avoid the possibility of an excessive fuel flow rate that can cause an overshoot in the pressure ratio. Tests reported in Ref. 25 illustrate the danger of an excessive rate of fuel flow on the initiation of surge. Engines with dry low NOx (DLN) controls may experience combustor pressure pulsations at lean fuel mixture operation. Combustor pressure pulsations at low frequencies might reduce the surge margin by causing perturbations on the compressor flow (negative damping) that would further reduce the stable operation range of the compressor. Preventative compressor maintenance is likely to be more important with engines that use DLN systems. Engines that employ steam injection for control of NOx or for added power are susceptible to excessive pressure ratio and surge in the event that excessive steam flows are permitted.

Deposits on or damage to the first row of turbine vanes in the hot section of the turbine may restrict gas flow and cause the compressor to operate at a higher pressure ratio, reducing the surge margin. Significant deposits may arise as a result of the use of additives for treating heavy fuel oils. Some additives form deposits on the vanes, reducing the first stage vane throat area and increasing the pressure ratio needed to pass the gas flow. Monitoring the pressure ratio can guard against an excessive increase that would reduce the surge margin.

Instrumentation for Surge Protection

Tests on nine different axial compressors (Ref. 26) indicated that stall always preceded surge, confirming what many other researchers have found. Stall warning signals were found in the form of low amplitude traveling waves (pressure pulses). A stall warning time of approximately 100 revolutions may be possible, but in order to achieve this stall warning capability, sophisticated instrumentation and software are required.

Experience with turbomachinery indicates that the pulsation frequency due to a single rotating stall cell is about 1/3 the rotor speed. A multiple stall cell condition can arise so the pulsation frequencies may be higher than the shaft speed. An indication of rotating stall can be detected by the spectral analysis of accelerometer or velocity transducer measurements (Ref. 27). These measurements cannot detect which stage goes into a rotating stall, but are sufficient to confirm that the operating conditions can lead to fatigue and fretting of the airfoil attachments. As rotating stall is more likely at low flow conditions, a practice of monitoring spectral signatures (velocity or amplitude) during startup using a waterfall plot could indicate a stall condition caused by a degradation in the compressor condition and the need for cleaning or repair.

Incipient surge detection was successfully demonstrated (Refs. 28 to 30) using dynamic pressure transducers. The transducers are installed between stages at several stage locations. The rotor and bearing housing vibration measurements do not give as much warning time for incipient surge as did the pressure transducers. Surge protection against excessive acceleration is described in Ref. 30 based on a measured level of the disturbance pressure that exceeds the allowable limit causing a bleed valve to open that by-passes some compressor discharge air to the exhaust. All the required transducers and signal conditioning components are commercially available. However, regular functional and calibration tests need to be done to ensure a reliable system. Dynamic pressure transducers with a high temperature operating ability of up to 1200°F (650°C) and a reliability of up to 500,000 hours mean time between failures (MTBF) are described in Ref. 31.

The use of audible noise measurements to detect flow pulsations prior to surge is suggested in Ref. 32. Pulsation frequencies due to single or multiple rotating stall cells are typically on the order of 1/3 to twice the rotational speed and monitoring for frequencies in this range could provide a warning of a potential stall and surge condition. If a warning occurs while at full load, an action to reduce load could alleviate this condition.

Water Ingestion Stall Warning

Water ingestion into the compressor may be intentional, with the object of increasing power by the evaporation of water internal to the compressor resulting in cooling and less compressor work (Ref. 8). The "overspray" effect has the potential for a large change in incidence angle due to internal cooling without significant change of the compressor pressure ratio or air flow. The test results obtained during the tests reported in Ref. 33 showed that the incidence angles for the last stage changed by as much as 2 degrees due to internal cooling. An increase in the incidence angle beyond the normal operating limits could lead to stall and possibly surge. The monitoring of the incidence angle may be done using the available information on the blade, vane and casing geometry of the last stage, and the discharge temperature and pressure. This type of stall monitoring may also be useful if water ingestion occurs due to a heavy rainfall.

Monitor Erosion & Pitting with Foggers, Evaporative Coolers, and On-Line Washing

The intent of an evaporative cooling system is to evaporate any water droplets prior to entry into the compressor, but evidence suggests that not all the water evaporates, resulting in carry-over and impingement of droplets on the airfoils. Droplets larger than 10 microns may cause erosion in highly stressed blade sections such as the root of the 1st rotor. If carry-over is suspected as a result of measurements and/or visual inspections that show streaks on the inlet surfaces, then the airfoils should be inspected in accord with OEM recommendations which may be as often as every 100 hours of combined fogging, evaporative cooler, and on-line water wash operation. Leading edge erosion also causes a loss in compressor efficiency due to sharp corners resulting from increased bluntness. When water droplets entering the compressor are anticipated due to evaporative cooling, fogging, or on-line washing, a corrosion and erosion resistant coating should be considered, at least to the first 5 stages depending on experience with similar CTs and the OEM's recommendations.

4
CONDITION MONITORING

Direct Detection of Condition

The detection of the condition of the compressor can be by direct or indirect methods. The direct measurement monitoring method can be a stand-alone method or combined with any of the other methods and provides more information on the condition of the compressor. This method is finding use in evolving limits on water washing and/or carry-over that will limit erosion and roughness. Direct measurements include:

- Visual Inspection of the Condition of the Airfoils
- Surface Measurements of Roughness
- Dimensional Measurements, Thickness & Chord
- Clearance Measurements
- Leading Edge Erosion

Visual inspection involves shutting the unit down, opening the inlet plenum hatch, and visually inspecting the compressor inlet, bellmouth, inlet guide vanes, and the first several rows of vanes and blades. The use of a borescope and a remote visual inspection (RVI) system is useful to track changes in the surface condition (Ref. 34).

As noted in Chapter 2, there is a critical level of roughness that triggers a rapid increase in the losses. A correlation of the on-set of losses with increased roughness (fouling) is given in Ref. 35, where the roughness Reynolds number is less than 90:

$$V \times Ks \times \text{density} / \text{viscosity} <= 90$$

The sand roughness (Ks) may be related to the center line average (Ra) approximately as:

$$Ks = 6.2 \times Ra$$

Surface roughness measurement methods (Ref. 36) are summarized in Table 4-1. Roughness is characterized by the average (Ra) or center line average as defined in Figure 4-1. Contact surface roughness testers use a stylus that is dragged over the surface that generates an electrical signal which is then amplified and processed to yield the Ra value (Ref. 37). Non-contact test equipment, such as glossmeters, uses optical techniques (Ref. 38) to measure and analyze surface qualities. Optical reflection sensors (Ref. 39) work best with specular reflecting surfaces, but can be calibrated for the random scattering found with diffuse reflectors.

Table 4-1
Characteristics of Various Surface Examination Methods (ANSI B46.1-1978)

	Stylus Instruments			**Optical Reflection Measurement**	
		Profile			
	Averaging	**Regular**	**Ultra High**	**Gloss-Meter**	**Light Scattering Specular**
Practical Peak to Valley Range (nm)	30 to 125,000	10 to 125,000	2 to 125,000	100 to 2500	2 to 30
Practical Measurement Area (mm^2)	1000	1000	10	(1)	(1)
Contacting (C) or Non-Contacting (NC)	C	C	C	NC	NC
Profile (P) or Non-Profile Quantitative (NPN)	NPN(2)	P	P	NPN	NPN
Approximate Vertical Resolution (nm)	2.5	2.5	0.5	100	Does Not Apply
Approximate Horizontal Resolution (mm)	2500	2500	100	Does Not Apply	1000
Output Data Form	Meter	Meter & Graph	Meter & Graph	Meter	Graph or Meter

1 nm (nanometer) = 0.03937 μ in. (micro-inch)

(1) Techniques are relatively insensitive to specimen displacement, thus specimen may continuously pass through measurement area.

(2) Roughness output here is not in the form of a profile. However, the output is based on electrically processed profile information.

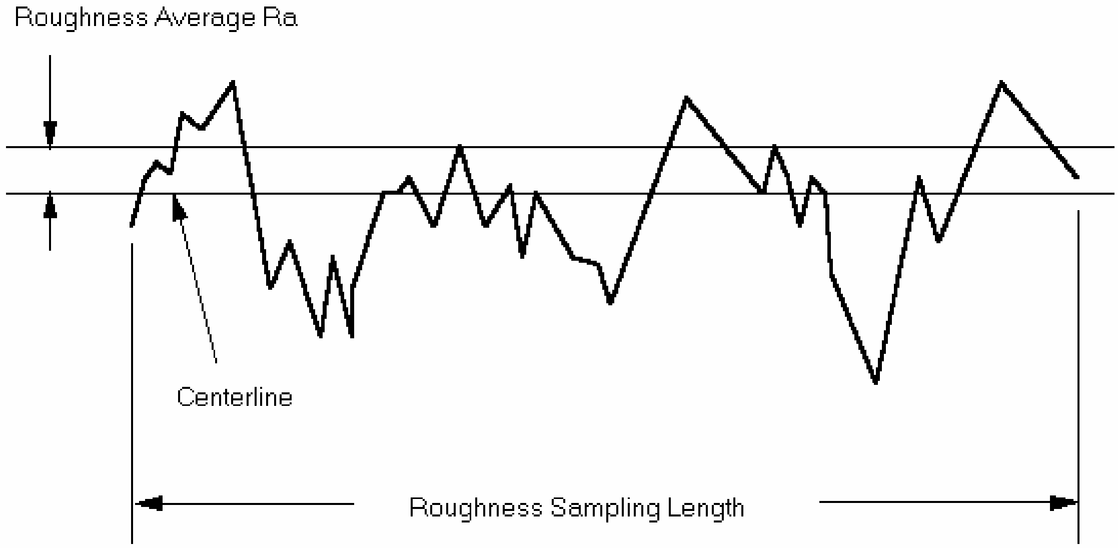


Figure 4-1
Definition of Average Surface Roughness (ANSI B46.1.1, 1978, Ref. 36)

On a first stage vane, the critical value of Ra is typically 70 micro-inches (1.778 microns), so clean surface roughness should be much less than 70 micro-inches (1.778 microns). Because of the high relative speed on the first stage rotor, the critical roughness could be as low as 25 micro-inches (0.635 microns). Continuously viewing the surface and getting an indication of the roughness could serve to automate the detection of fouling and the control of an on-line wash system. Such a system has not been demonstrated to date. Any CT operators interested in pursuing this can contact EPRI.

Long term erosion or distortion effects that result in non-recoverable performance due to increased running tip clearances can be monitored with a proximity gauge (Ref. 29). The long-term effects of erosion that cause a reduction in blade thickness can be monitored with an ultrasonic micrometer (Ref. 37). Erosion monitoring may be done by regularly taking dental molds on the leading edge of the 1st stage compressor rotor blades. The OEM may have strict limits on the amount of allowable erosion to avoid loss of structural integrity (Ref.40).

Indirect Detection of Condition

Indirect methods that rely on measuring the aerodynamic and thermodynamic performance of the compressor and/or the whole gas turbine are a common approach to condition monitoring. These methods rely on quality instrumentation with highly repeatable data and an accurate estimation of what the new and clean or baseline performance should be in terms of power, thermal efficiency, air flow, pressure ratio, or compressor efficiency.

There is a diversity of indirect methods that can be used to monitor the condition of the compressor. Therefore, software that offers the user the choice of a number of different methods is desirable. In general, a database is needed on the expected baseline performance. Databases already exist for some gas turbines such as that in EPRI's SCAAD program (WO3401-01). The database contains the baseline performance for a number of gas turbines widely used by utilities, including power, heat rate, and air flow. To cover different methods, added data should include pressure ratio and compressor efficiency.

Power Output Monitoring Method No. 1

The degradation of the axial compressor performance causes an observable decrease in power. Experience has shown that although other factors may cause a decrease in the power output, the degradation of the compressor condition is the most likely cause of the power loss. Accurate power measurements are required (+/-0.25%) and using the techniques of ASME PT-22 or special instrumentation is important to trend the small changes that can occur on a daily basis. This method is described in Ref. 41, where it was applied to derive an optimum on-line water wash schedule to maintain high capacity with minimal down time for crank washing. The measured power is transposed to a standard reference condition, say dry ISO operation, so that the deviations in power are all with respect to a standard condition. Ancillary measurements of inlet air temperature, barometric pressure and specific humidity are needed to correct the power to standard conditions. Trending should be restricted to base load operation as correcting the power output to standard conditions at part load is less reliable. To correct the power, measurements of inlet and exhaust pressure drops and water and steam injection flows are also required.

Heat Rate & Power Monitoring Method No. 2

In addition to monitoring the output power, fuel flow monitoring is useful to perform economic optimization of the compressor maintenance interval (Ref. 42). The fuel flow measurement must be accompanied by regular determinations of the fuel heating value. A fuel flow meter with good repeatability (±0.25% or better) is needed to trend the small differences in fuel use and heat rate that can occur daily. For natural gas fuels, an on-line gas chromatograph facilitates the monitoring of the heat rate.

Air Flow Monitoring Method No. 3

A further refinement to monitoring the compressor condition is to monitor the reduction in air flow caused by fouling of the compressor and/or the inlet filter (Ref. 43). Air flow can be monitored by a Pitot static probe in the compressor inlet or by using static pressure taps in the wall of the inlet scroll just forward of the inlet guide vanes. As noted previously, detection of a reduction in air flow could also serve as a stall warning indicator. The measured air flow would be compared to a predetermined correlation of the expected air flow variation with inlet temperature and barometric pressure. Monitoring the air flow is useful when monitoring combined cycle performance and when using inlet fogging.

Compressor Pressure Ratio Monitoring Method No. 4

As the compressor pressure ratio is directly influenced by the air flow and is relatively easy to measure, it is often used in lieu of the air flow measurement noted above. Typical results of monitoring the pressure ratio are shown in Figure 4-2 (Ref. 21). Also shown in Figure 4-2 is the effect of a compressor bleed valve that is stuck open. However, to distinguish whether a decrease in the compressor discharge pressure is due to fouling or seal (or recuperator) leakage, monitoring both the inlet air flow and the pressure ratio is recommended. Compressor leakage from valves and seals changes the flow ratio of air flow/discharge pressure when the gas turbine is being controlled at constant firing temperature (base load), as the discharge pressure is in actuality dependent on the gas flow through the 1st stage vane throat area.

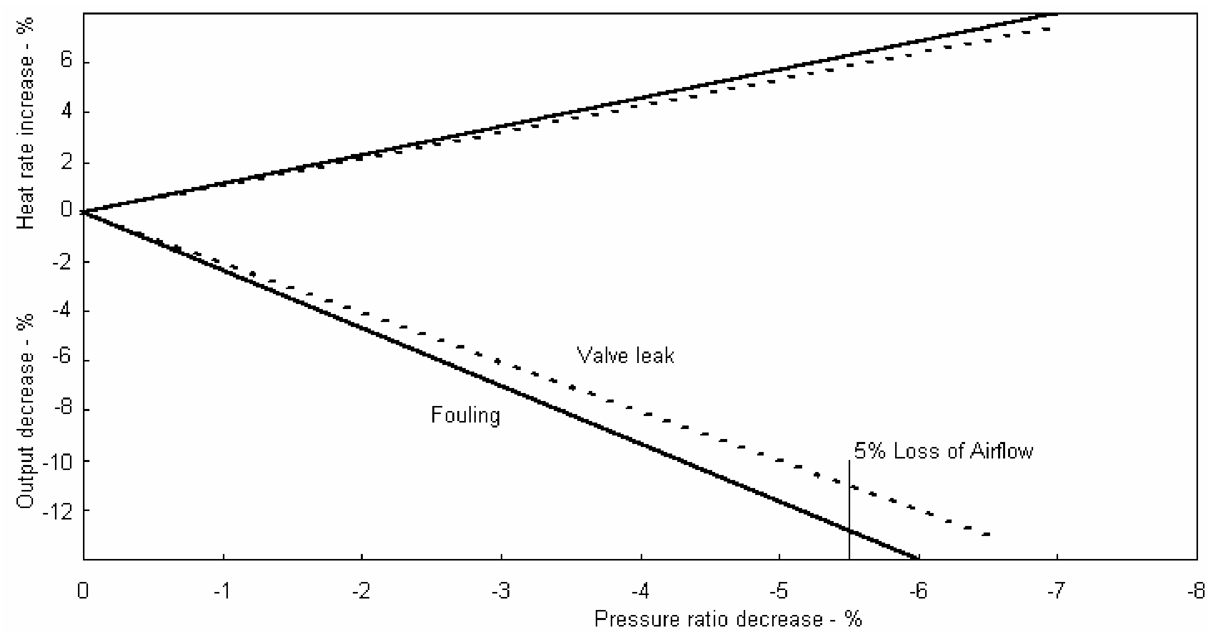


Figure 4-2
Effect of Pressure Ratio on Gas Turbine Performance (Scheper, et al, 1978, Ref. 21)

Compressor Efficiency Monitoring Method No. 5

Inaccuracies arise in the compressor efficiency measurement as the calculation involves four measurements: inlet pressure, discharge pressure, inlet temperature, and discharge temperature. Accurate monitoring the discharge temperature is hampered by the temperature stratification so that a temperature rake measurement is needed for reliable data. There are also effects of humidity on the discharge temperature which must be accounted for. Whereas the compressor efficiency measurement is useful in diagnosing the cause of the performance loss, the air flow loss is generally the largest contributor to the loss in output power. Compressor efficiency degradation has a strong effect on heat rate as shown in CT performance given in Table 4-2.

Table 4-2
Gas Turbine Performance Losses Due to Compressor Degradation

	Loss	Power Loss	Heat Rate Increase
Inlet Air Flow	1%	1.1%	0.2%
Compressor Efficiency	1 pt.	1 - 1.5%	1 - 1.5%

By monitoring the degradation in compressor efficiency, the operator can establish whether the compressor performance is fully recovered after a wash. If the efficiency is not recovered, repair of damage or use of other cleaning technique(s) is indicated. If the efficiency is recovered but the power is low, then leakage or low air flow could be causing the performance degradation.

Instrumentation

A summary of the instrumentation required for each indirect monitoring method is given in Table 4-3. The measurements marked with an asterisk are needed to correct the power and heat rate to standard conditions.

Table 4-3
Instrumentation for Condition Monitoring Methods

Measurement	Symbol	Options, Flags	Units, English	Units, SI	Method(s)
Barometric Pressure*	PBAR	Measure or Default	Psia	Bara	All
Inlet Temperature*	CIT	Measure	Degrees F	Degrees C	All
Inlet Pressure Drop*	IPD	Measure	In. of Water	Mbar	All
Exhaust Pressure Drop*	EPD	Measure	In. of Water	Mbar	All
Relative Humidity*	RH	Measure or Default	%	%	All
Compressor Speed* (Aero-Derivatives)	RPM	Measure	RPM	RPM	All
CT Fired Hours	CTFH	CT Counter	Hours	Hours	All
Power	KW	Measure	KW	KW	All
Fuel Flow	FFLOW	Measure	Lbs/sec	Kg/sec	2,5
Fuel Heating Value	LHV	Measure	Btu/Lb	KJ/Kg	2,5
Air Flow	WAIR	Measure	Lbs/sec	Kg/sec	3
Compressor Discharge Pressure	CDP	Measure	Psia	Bara	4, 5
Compressor Discharge Temperature	CDT	Measure	Degrees F	Degrees C	5

Expected Baseline Performance Database

A database of baseline performance includes:

- KW Output vs. Inlet Temperature
- Heat Rate vs. Inlet Temperature
- Air Flow vs. Inlet Temperature
- Pressure Ratio vs. Inlet Temperature (only for Methods No. 4 & 5)
- Compressor Efficiency vs. Inlet Temperature (only for Method No. 5)
- Performance Correction Sensitivity Coefficients: C1, C2, C3, C4, C5, C6, C7, C8
  - C1, Effect of Inlet Pressure Drop on Power Output
  - C2, Effect of Inlet Pressure Drop on Heat Rate
  - C3, Effect of Water Flow/Fuel Flow Ratio on Power Output
  - C4, Effect of Water Injection Flow/Fuel Flow Ratio on Heat Rate

- C5, Effect of Steam Injection Flow/Fuel Flow Ratio on Power Output
- C6, Effect of Steam Injection Flow/Fuel Flow Ratio on Heat Rate
- C7, Effect of Exhaust Pressure Drop on Power
- C8, Effect of Exhaust Pressure Drop on Heat Rate

- Reference ISO Performance, Dry Operation
  - Power
  - Heat Rate
  - Air Flow
  - Inlet Pressure Drop
  - Exhaust Pressure Drop

Trending

Trending the degradation of the compressor performance with time is essential to successful compressor maintenance. The measured performance is transposed to the reference ISO conditions. The deviations of power, heat rate, pressure ratio, air flow, and efficiency between the transposed and reference ISO performance are stored in a historical database for trending. The historical file should also include a log of the maintenance actions such as off-line and on-line washes, coating application date, wash compound used, and dosage. These records enable the operator to develop an efficient maintenance technique to maintain the compressor in good condition. The on-condition maintenance assessment is made on the increases in the differences in fuel use and capacity loss. The frequency of measurements can be as low as once-per-day.

Non-Recoverable Compressor Performance Loss Monitoring Method

Non-recoverable losses are defined as the decreased output that cannot be recovered without removing the compressor shell and manually cleaning the airfoils and/or refurbishing the airfoils and other compressor elements. Results of monitoring non-recoverable compressor performance losses are presented in Ref. 44.

Causes of the non-recoverable losses include:

- Deposits that are not removed by water wash
- Increased rotor tip clearances as a result of creep and/or rubs
- Leading edge erosion and chord reduction
- Increased airfoil roughness and/or loss of coatings
- Foreign or internal object damage

An OEM typically supplies a non-recoverable performance curve as shown in Figure 4-3 which includes the turbine section losses as well as the compressor losses. However, the turbine rate of degradation is much slower than that for the compressor, so the curve may be used as a conservative estimate for the short term non-recoverable compressor losses.

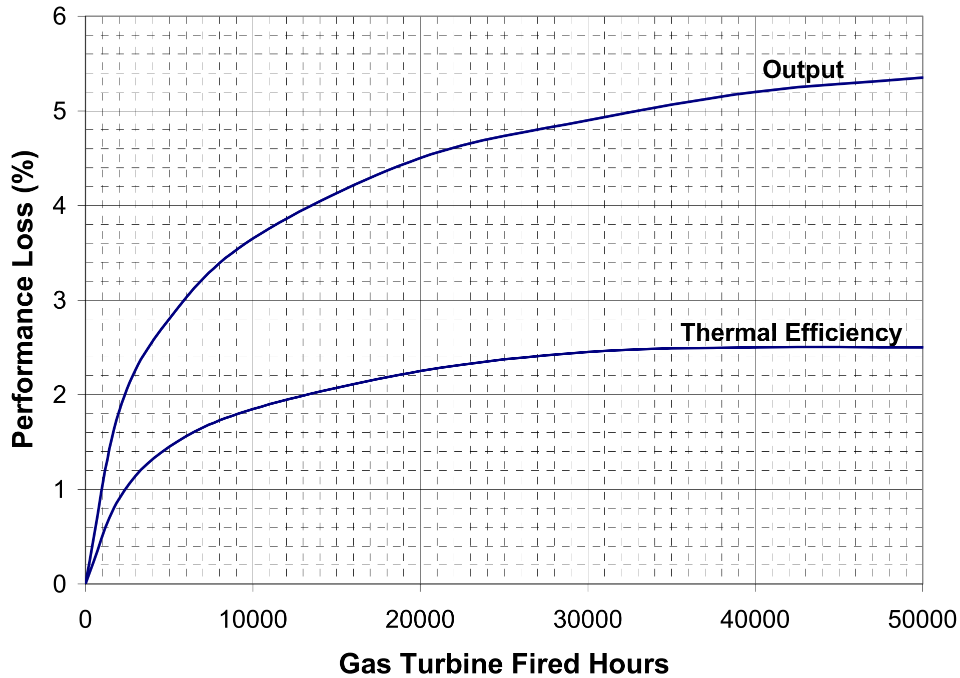


Figure 4-3
Estimated Non-recoverable Performance Losses Based on Normal Maintenance Procedures

As the testing for non-recoverable performance can span the whole period between an engine overhaul, say 5 - 6 years, then the instrumentation must be carefully maintained to ensure that the results are not compromised by measurement errors. The test results of Ref. 44 show that there is about a 2 - 3% loss in capacity in the first year of service, followed by about another 3% loss more over the next five years.

Effect of Compressor Fouling on Gas Turbine Maintenance

A loss in compressor performance not only influences CT performance, but also the CT maintenance. Higher compressor discharge temperatures resulting from a decrease in compressor efficiency can increase the metal temperature of the cooled components in the hot section as the cooling air temperature may increase also. The higher discharge temperature may also result in a higher flame temperature in diffusion flame combustors resulting in increased NO_x emissions. The effects of compressor fouling are to some extent dependent on the type of gas turbine and the control system. A single shaft heavy duty gas turbine operating at base load may have a control algorithm that uses the compressor discharge pressure and the exhaust temperature to maintain a constant firing temperature. If there is a reduced air flow resulting in lower compressor discharge pressure (as noted previously), then the control algorithm will cause the turbine to operate at a higher exhaust temperature. Circumferential variations in compressor discharge temperature may lead to an increased spread in the exhaust gas temperature.

Economic Optimization of Compressor Maintenance

The optimal economic interval between crank washes occurs when the average O&M cost per MW over the interval is minimal. The O&M costs include the reduced revenue caused by taking a base-load unit off-line, as well as the labor, material, and other shut-down and start-up costs plus the added cost of fuel as the performance degrades with time. Historical test results must be gathered on pre-wash performance vs. post-wash performance to quantify the degradation in performance and the benefits of a crank wash. An optimum appears as frequent off-line washes result in high O&M costs, while over a very long interval without cleaning, the O&M costs may also increase due to excessive degradation. An off-line wash is typically justifiable in anticipation of a period of high demand where the spark spreads are likely to be favorable. The cost of on-line washing is relatively low and operators tend to wash daily to defer off-line washes and maintain high efficiency. Continued developments are resulting in more effective and lower operating cost on-line water wash systems. A review of on-line water wash technology is given in Chapter 5.

In the case of non-recoverable losses, the approach is to determine an outage maintenance interval that in-effect recovers the cost of the outage (lost revenue) and reduces losses as a result of compressor blade repair such as coating or restoration. At issue is whether the costs and benefits of additional compressor inspections and maintenance are economically justified. The trend of power loss and heat rate increase with operating time is approximated by the curve in Figure 4-3. This is a typical OEM curve used to correct test data for the number of fired hours and, hence, may be conservative.

The decision to repair or replace blades depends on an inspection of the condition of the surface, dimensional and twist measurements, and a determination of whether the blades are repairable and at what cost. As shown in Figure 4-4, re-contouring a blunted leading edge may lead to added aerodynamic losses unless the resulting profile results in smooth airflow around the airfoil (Ref. 45). A blade reaches the end of its life when it becomes too thin for restoration and is unable to maintain structural integrity. The cost of compressor blade restoration and/or replacement is dependent on the ease of maintenance built in by the OEM, the availability of the material, the blend limits of the OEM, and the availability of specific tooling for welding, profiling and surface finish. The use of custom materials such as 450 SS improves erosion and corrosion resistance, but results in added cost for replacement.

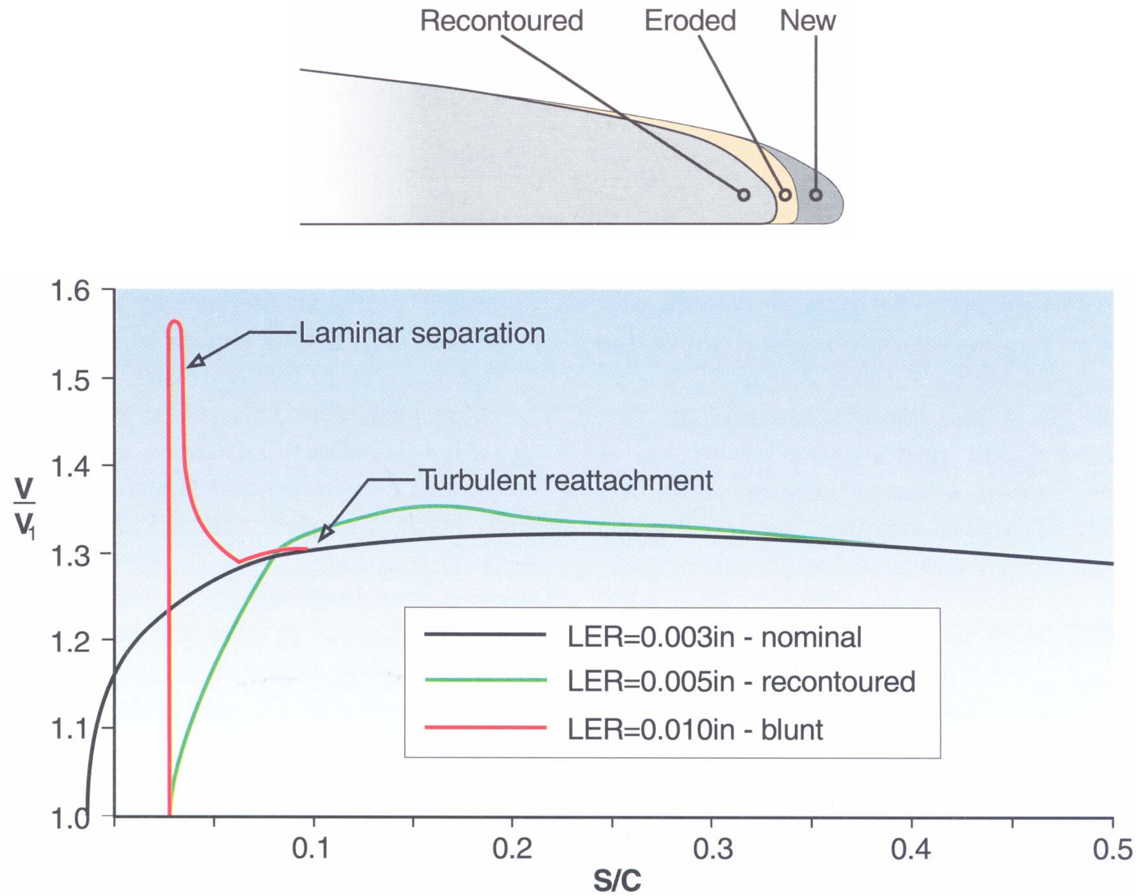


Figure 4-4
Leading Edge Suction Surface Velocity Distributions for a Compressor Blade Section with Various Leading Edge Radii (Ref. 45)

The first 5 stages of a compressor have the greatest impact on the loss in performance due to fouling, roughness, airfoil erosion, and increased clearances as shown in Figure 4-5. The results shown in Figure 4-5 were calculated from a multi-stage compressor simulation by implanting faults at various blade rows and then determining the change in compressor efficiency and the impact on the CT cycle performance (Ref. 46).

The mechanical design of the CT influences the cost of compressor repairs such as the ease of compressor rotor removal and coating the stacked rotor. CTs may be designed to allow all the compressor blades and stationary diaphragms to be removed and replaced without rotor removal. Other maintenance actions may include: corrosion coating of the rotor disks and coatings to reduce diaphragm seal and angel wing clearances.

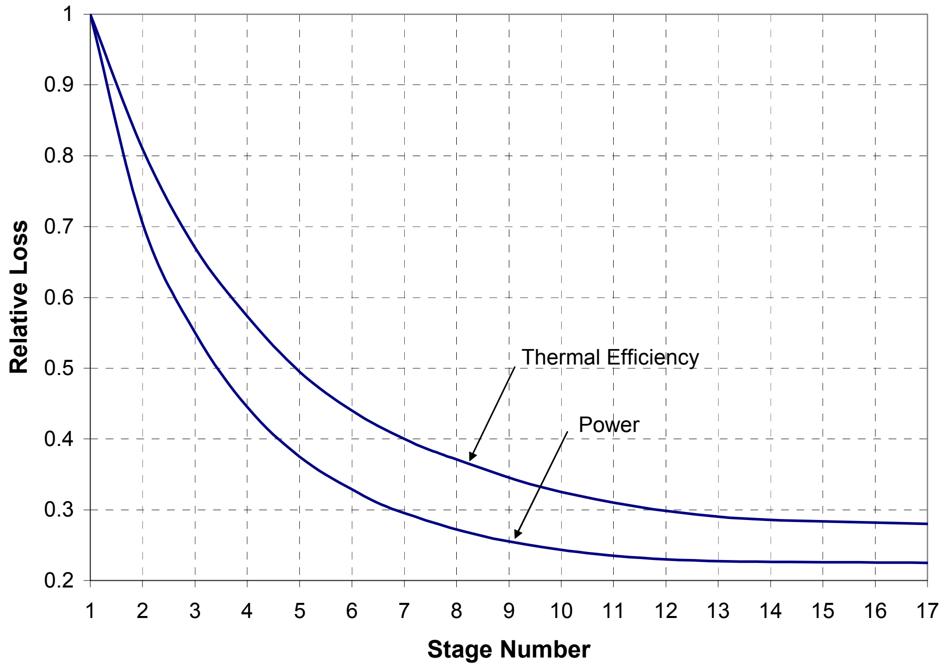


Figure 4-5
Estimated Effects of Compressor Stage Loss Coefficient on GE 7EA CT Power and Thermal Efficiency Performance (Ref. 46)

An economic model based on the concept of spark spread is used to determine the cost/benefits of additional compressor inspections, trading off the added costs vs. the improved performance benefits. The spark spread is the margin between the power sales price and the fuel cost and in the real world can have marked peaks during a year, so a detailed site-related strategic analysis is desirable. For the present, the potential of this maintenance is evaluated based on average spark spreads of \$5, \$10 and \$20/MW-hr. The calculations can include the effects of ambient temperature as well, but for the present, an average ambient temperature is used. The following example is only to illustrate how this may be done and the potential benefits.

Referring to Figure 4-3, the increment in MW-hrs lost during each major overhaul interval is given by:

$$\Delta \text{MW-hrs} = \int (P0 - P(t))\, dt$$

where:

$$P0 - P(t) = P0 \times SF \times pplf(t) / 100$$

SF = loss scale factor due to repairs that lower the rate of the loss
pplf(t) = percent power loss from Figure 4-3.

Using the approach where the spark spread is the net difference in the power sales price and the cost of fuel, then the loss in net revenue (NR) during each major overhaul interval (MOI) is calculated as:

$$\Delta NR = ES \times \Delta \text{MW-hrs} - (OC + RC) \times (MOI / MCI - 1)$$

where ES is the spark spread in dollars/MW-hrs, OC is the outage cost, RC is the repair cost at each outage, MCI is the compressor maintenance interval.

The calculation is done for various intervals of compressor maintenance (MCI) – 1000, 2000, 4000, 8000, 16000, 24000 hours – to search for an optimum. The overall net loss without any added maintenance inspections is also calculated based on 48,000 hours between major overhauls. Say, for example, there are inspections at 8000 hours, or 5 added inspections between overhauls as shown in Figure 4-6. The total net revenue loss over a maintenance overhaul interval would be 5 times that calculated for each compressor inspection interval. The benefit is then the difference between the net revenue lost due to lower power without any compressor inspections to that amount lost with added compressor inspections. The cost of the 5 added inspections must be subtracted from the revenue to get the net benefit.

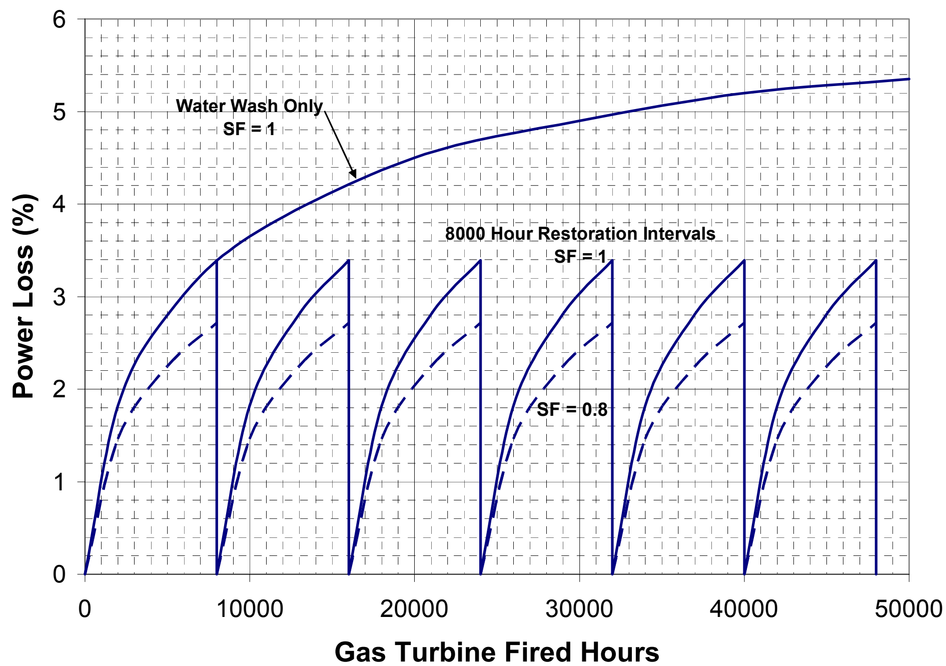


Figure 4-6
Loss Model with 8000 Hour Maintenance Interval

When the compressor is inspected, the measurements to assess the condition of the blades are used to determine the extent of repair. Some sample parametric results are calculated for the following specific inputs:

P0 - 150 MW, new & clean average power

H0 - 10,665 Btu/kW-hr, new & clean average heat rate

ES - 5, 10, 20, average spark spreads, \$/MW-hrs

OC = 48 × ES × PO, outage cost

RC = \$ 0.1, millions, cost of repairs to inspect, restore, smooth and/or coat airfoil surfaces

SF = 1, 0.8, 2

The scale factor adjusts loss trends in Figure 4-3 on the basis that repairs such as coatings may lower the loss rate and/or indicates sensitivity to loss estimates.

Calculated results of the differences in net operating revenues for different maintenance intervals are given in Table 4-4 for SF = 1. The results show that benefits are relatively small when the spark spread is small.

Table 4-4
Benefits Analysis Results, SF = 1, RC = 0.1

Spark Spread $/MW-hr	Optimum Maintenance Interval, Hours	Total Net Benefit $ Millions
5	8,000	0.2
10	8,000	0.6
20	8,000	1.7

When the rate of degradation is slowed such that SF = 0.8, the results in Table 4-5 show increased benefits of additional maintenance. The determination of the optimum interval is shown in Figure 4-7 which shows the effect of maintenance interval on profit.

Table 4-5
Benefits Analysis Results, SF = 0.8, RC = 0.1

Spark Spread $/MW-hr	Optimum Maintenance Interval, Hours	Total Net Benefit $ Millions
5	16,000	0.4
10	16,000	1.0
20	8,000	2.4

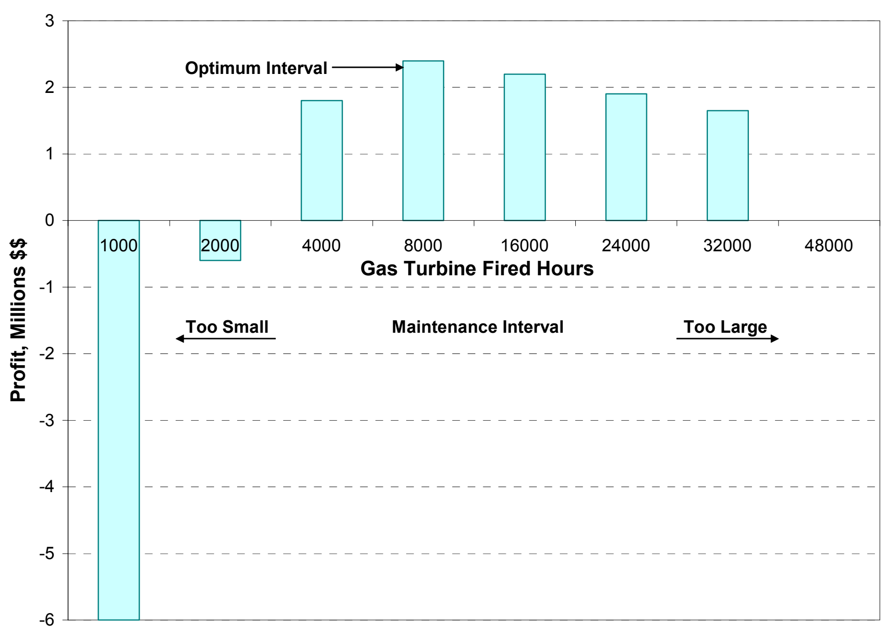


Figure 4-7
Effect of Maintenance Interval on Profit Over a 48,000 Hour Major Overhaul Interval (SF = 0.8, RC = 0.1, ES = 20)

As there may be planned outages for other inspections, there would be no outage cost penalty for the compressor. The results shown in Table 4-6 are more favorable.

Table 4-6
Benefits Analysis Results, SF = 0.8, OC = 0, RC = 0.1

Spark Spread $/MW-hr	Optimum Maintenance Interval, Hours	Total Net Benefit $ Millions
5	16,000	0.5
10	8,000	1.3
20	4,000	3.3

At a cost of $5/million Btu, the average fuel costs over the 48,000 hour major overhaul interval varied between $53.43 and $54.25 per MW-hr for all the cases shown. The lowest fuel costs occur for the most frequent maintenance interval. The attempt to determine an optimal compressor maintenance procedure does not show any dramatic benefits. The outage and repair costs appear to offset the benefits in lower fuel costs.

The benefit of added maintenance was also investigated for an increased scale factor of 2 to simulate the impact of compressor erosion for an inter-stage cooling system. The simulation increased the repair cost to $1 million and increased the power by 7%. The results shown in Table 4-7 indicate the potential for increased profitability. Experience with inter-stage cooling indicates the benefits may not always be favorable (Ref. 46), but as the results in Table 4-7 show, the benefits are most likely to be favorable in periods where there is a premium on the spark spread and if the maintenance interval is optimized. Restoration of blades as a result of erosion caused by inter-stage cooling may require replacement of several rows of blades, but the use of inter-stage cooling may be justified in any case for the increased capacity need in hot weather. The use of a hard, smooth, erosion-resistant coating may be very attractive for use with inter-stage cooling as discussed in Chapter 6. The results of an example calculation where the added coating limits erosion are given in Table 4-8, which show a much larger potential benefit for inter-stage cooling.

Table 4-7
Benefits Analysis Results, SF = 2, RC = 1, PO Increased 7%

Spark Spread $/MW-hr	Optimum Maintenance Interval, Hours	Total Net Benefit $ Millions
5	24,000	0.3
10	24,000	1.6
20	16,000	4.5

Table 4-8
Benefits Analysis Results, SF = 0.8, RC = 0.1, PO Increased 7%

Spark Spread $/MW-hr	Optimum Maintenance Interval, Hours	Total Net Benefit $ Millions
5	8,000	2.9
10	8,000	6.0
20	8,000	12.5

Summarizing, it appears that compressor maintenance may result in a significant increase in profit if it is done during a planned outage and if the repair work results in a lower rate of degradation. An example of this type of maintenance may be coating refurbishment and airfoil restoration that keeps the clearances low and the airfoils very smooth. Services for airfoil restoration of tips, leading edges, and trailing edges using automated welding and re-contouring techniques are available from vendors listed in Chapter 6.

5
COMPRESSOR CLEANING TECHNIQUES

Off-Line Compressor Washing Technologies

An OEM typically provides an off-line wash system and the washing procedures (Ref. 48). For CTs in base load service, a dispatcher must allow the CT to be shut down for six hours or more (Ref. 41) resulting in a loss of revenue. The off-line wash typically uses a hot water and detergent mix and must be done at ambient temperatures above 40°F (4.4°C) to avoid icing. A recent CFD study on the flow field during an off-line wash concludes that the benefits of heating the water are lost before the spray impacts the airfoils due to droplet cooling and evaporation (Ref. 49). However, if sufficient water is sprayed to saturate the air, then the evaporation is negated. The hot water may also be needed to mix the detergent. The off-line wash is done at crank speed so the air flow is lower (see Figure 1-2) and the water needed to saturate the air is reduced. The procedure involves preparing the CT for the wash, washing, rinsing and restoring the CT to service. Several washes and rinses may be needed before the rinse water is clean. The benefits of on-line washing at full power are to increase the off-line maintenance interval and retain a high average thermal efficiency of the CT. The importance of compressor maintenance is brought into focus when testing for performance guarantees where the compressor must be washed off-line before a test, with only a small allowance for fired operation after washing before the actual test is performed. Environmental issues with crank washes include the disposal of the drain water containing solvents that are used and the deposits removed from the compressor.

On-Line Compressor Washing Technologies

The on-line water wash scenario is one where water is sprayed into the inlet to create a wet film covering the entire blade surfaces in a blade row while the CT is either at full load or a slightly reduced load. The droplet impacts may dislodge the deposits where they strike and the resulting water film dissolves the deposits and/or simply washes them away as particulates. Solvents may be needed to mix with the water to enhance removal of organic deposits while, on the other hand, a non-stick surface coating such as Teflon may not require dissolving the deposit. Successive downstream blade rows receive the remaining wash water that has not evaporated. Because of evaporation, the amount of water left for washing is reduced for each succeeding blade row and finally all the water disappears. When all the water is gone, all the dissolved deposits appear as particulates and may be deposited on succeeding blade rows. If the deposits are organic, then they will likely vaporize if the penetration is far enough into the compressor.

Typical benefits of on-line washing are shown in Figure 5-1 (Ref. 41). By daily on-line washes, the rate of capacity loss is slowed and economic considerations justify a shut-down and an off-line wash in 66 days vs. off-line washes only every 22 days.

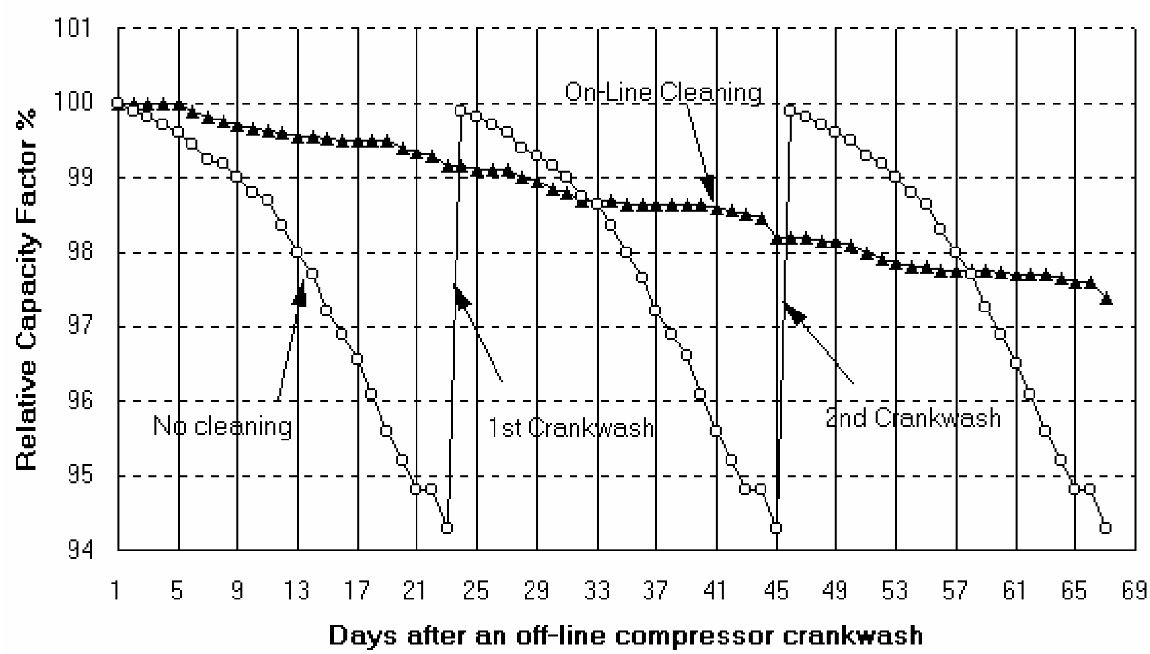


Figure 5-1
Benefits of On-Line Washing (Haub, ASME 90-GT-107, Ref. 41)

There is a substantial increase of capacity of about 2% due to the better performance and lower off-line time when using on-line washing. The results shown in Figure 5-1 are site- and wash technique-specific. Other field testing experience (Ref. 42) varying the washing procedures confirm the potential benefits of on-line washing.

An on-line water wash system consists of an array of water nozzles intended to provide full coverage at the inlet guide vanes (IGVs). A review of on-line wash systems categorizes them in terms of pressure and flow (Ref. 50). A discussion of the nozzle spray parameters needed to wet the surface of the airfoils completely indicates the importance of nozzle orientation and installation design (Ref. 51). A nozzle array designed for full coverage with a minimum number of nozzles is shown in Figure 5-2 for an aero-derivative CT. Note that the spray nozzles are located close to the IGVs and they are elevated and angled to target the annulus completely. In heavy duty CTs, the compressor inlet scroll is designed for a side inlet so that the scroll must turn the air 90 degrees as depicted in Figure 5-3. Note that in this particular design there is a double array of nozzles located in the surface and penetration into the flow must be achieved to have full coverage at the IGVs. As shown in Figure 5-3, the nozzles body is a ball-joint so the nozzle angle can be tuned for optimum penetration. The nozzles have flexible connections to annular manifolds. To enhance the penetration of the spray further for the larger CTs, a nozzle was developed combining an air curtain to shield the water spray from deflection by the air flow. The resulting flush-mounted nozzle design is shown in Figure 5-4. Alternatively, nozzles customized for each model of CT could also be elevated and angled to achieve good penetration. When the nozzle is close to the IGVs, the air flow speed is relatively high, so a high-pressure nozzle is favorable to achieve a nozzle exit velocity close to the air flow velocity.

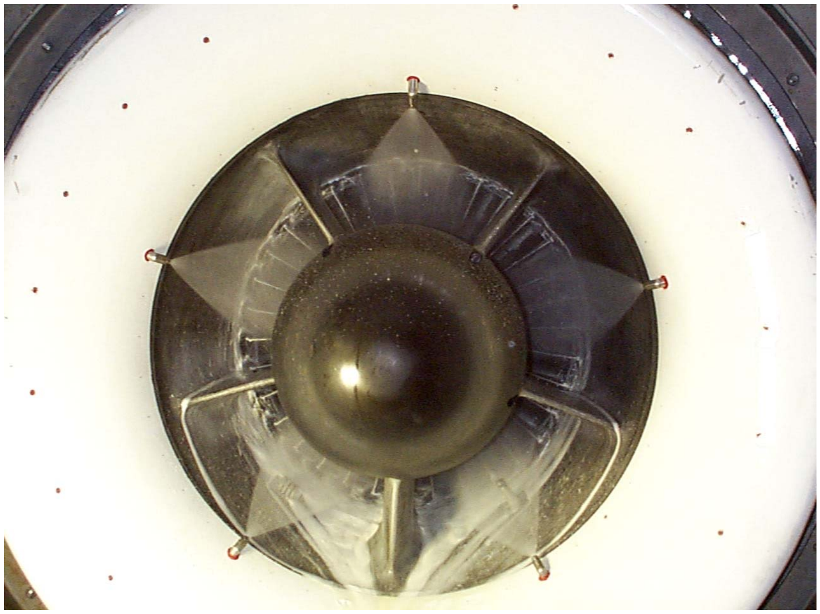


Figure 5-2
GTE Spray Nozzle Plume Coverage (Courtesy of GTE)

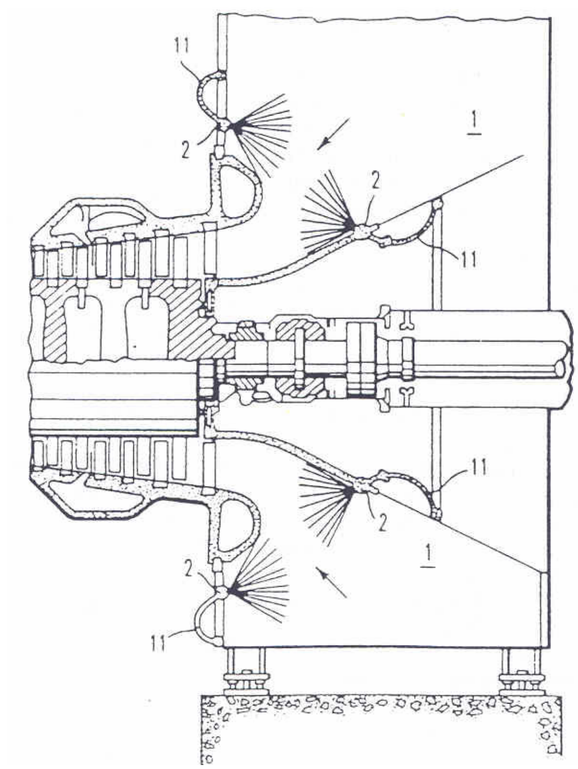


Figure 5-3
Typical Heavy Duty Inlet Scroll with Typical Nozzle Locations (Courtesy of Turbotect, US Patent 5,193,976, March 16, 1993)

The nozzle designs shown above use less water as there is less wasteful surface wetting, also known as streaking, before the spray impacts the IGVs. The droplet size is also increased to enhance penetration and coverage which results in reduced droplet evaporation before the spray enters the IGVs. Water heating is advocated by some vendors, but as noted above (Ref. 48), the perceived benefits may not be important as there is rapid droplet cooling before they impact the airfoil surface. To enhance this benefit of heated water, the use of added water via foggers to saturate the air is beneficial as evaporation of the heated water is negated. While large size droplets, 50 to 250 microns, offer the prospects of better penetration into succeeding stages due to reduced evaporation and better washing, the downside is the effect on erosion. Erosion has become a critical issue in advanced CTs where the airfoils are optimized for maximum efficiency while meeting the stress requirements. The airfoil erosion may be at the leading and trailing edges. The leading edge erosion is due to the direct impact of the droplets while the trailing edge erosion is due to high surface water film velocities leaving the trailing edge.

Figure 5-4
Turbotect MK3 Nozzle with Air Curtains to Enhance Penetration (Courtesy of Turbotect)

Water flow rates are typically suggested by vendors of wash systems, but may be tuned to achieve maximum benefits. Some of the water droplets initially impact the airfoil surfaces and form a film as shown in Figure 5-5, which in-turn flows off the trailing edge breaking up into large droplets to reform a film again on a downstream surface.

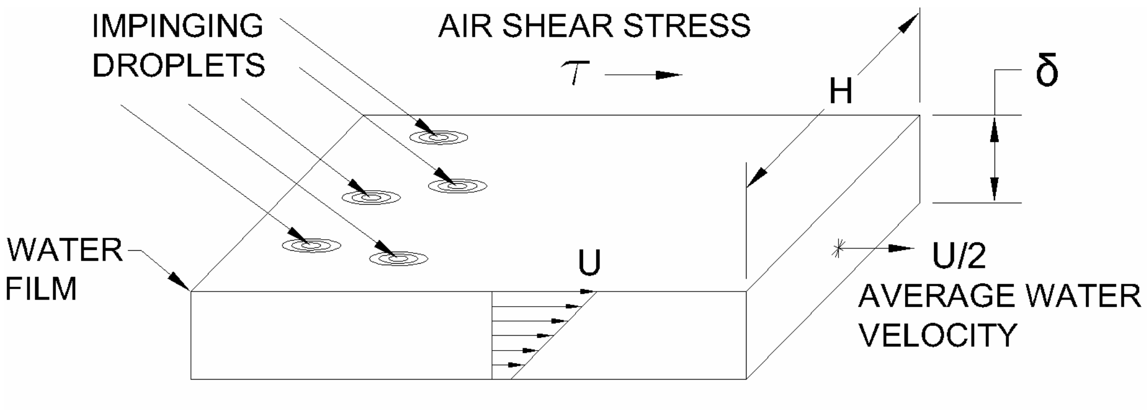


Figure 5-5
Water Film Development on the Airfoil Surfaces

The basic expression for a film of water flow over a surface of span (H) and an average film thickness of (δ) on both sides is:

$$W = \text{water flow} = 2\sigma_w u_w\, \delta\, H$$

Where σ_w is the water density and u_w is the average axial water velocity in the film. A relationship between the film velocity and the film thickness is developed in Appendix A so that the water flow can be estimated for a specified film thickness. The thickness of the film (on both sides of each blade) should probably be larger than the anticipated heights of the deposits which may be on the order of 100 micro-inches (2.54 microns). The estimated water flow given in Appendix A for a film thickness of 1000 micro-inches (25 microns) and the 1st rotor on a GE 7E is 8.6 gpm (33 lpm). The estimated water flow is dependent on the square of the film thickness, so a lower water flow reduces the film thickness. Besides the water flow needed to sustain a water film, some of the water intended for washing is ineffective due to evaporation and casing wetting. As a nozzle emits a spray with a droplet size variation, the finer droplets will tend to evaporate and/or follow the streamlines without benefiting the water wash. There is also evaporation from the water film surface due to convection. The maximum amount evaporated in the first stage depends on the climatic conditions and whether a fogger is used. If a fogger is used, then the air entering the 1st rotor is nearly saturated, but further evaporation occurs as the temperature rises in the stage. The maximum droplet and surface film evaporation in a stage to saturate is approximately a water/air ratio of 0.005. This result is found by adding water during the compression process so as to keep the air saturated. The water flow to keep the air saturated in the 1st stage and maintain the water film thickness of 1 mil (25 microns) is 30 gpm (115 lpm), bringing the potential total water flow needed to as much as 0.7 % of the air flow. Large gains in the reduction of the water flow rate may be possible using larger droplet sizes that slow down evaporation so that saturation does not occur. To make the most efficient use of the de-mineralized water, there is a need to balance the water flow rate and the duration of the wash. The benefits of a large water flow rate include: 1) a thicker water-washing film and 2) increased penetration into the compressor. A study on the sensitivity of the stage location to the loss in performance due to fouling shows that penetration into at least the first three stages is desirable (See Chapter 6). For a given amount of water use per wash, a higher rate of flow means a shorter wash period and a shorter soak period. The limit on water rate is also dependent on the impacts on the CT. For the example above, a water rate of 0..7% introduces important inter-stage cooling effects that: 1) boost the power by ~5%, 2) reduce the compressor discharge air temperature

~30°F (10°C), and 3) increases fuel ~5%. Note that as the water flow is increased to yield a thicker film, the film velocity increases which can lead to trailing edge erosion.

A survey (50) of on-line water wash systems correlated the water/air ratio with CT output power and the results are shown in Figure 5-6. The systems are categorized by high or low pressure (HP, LP) and high, medium and low flow (HF, MF, LF). Using the model relationships developed in Appendix A, the air/water ratio scales as the square root of the air flow for similar aerodynamic parameters such as aspect ratio, solidity and hub/tip ratio. The results shown in Figure 5-7 for a range of heavy duty CTs exhibit a similar trend with power as found in the survey.

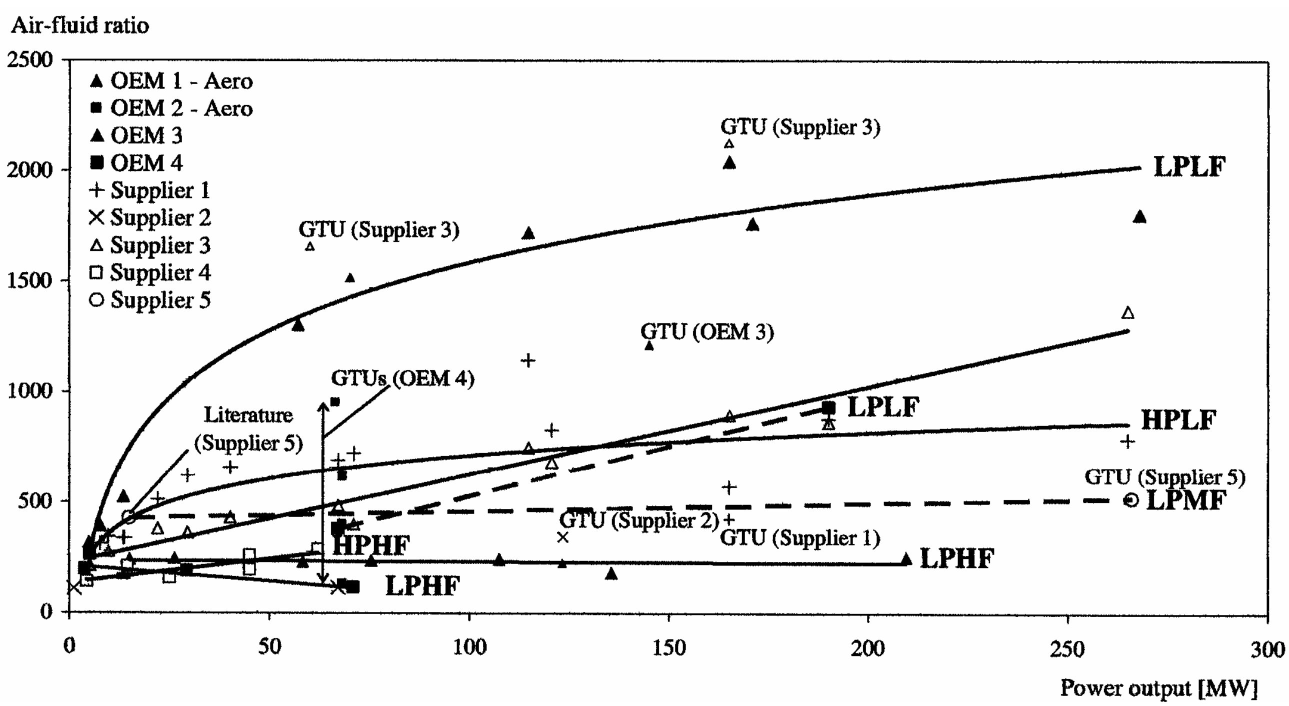


Figure 5-6
Air-Fluid Ratio for Different Compressor Washing Systems

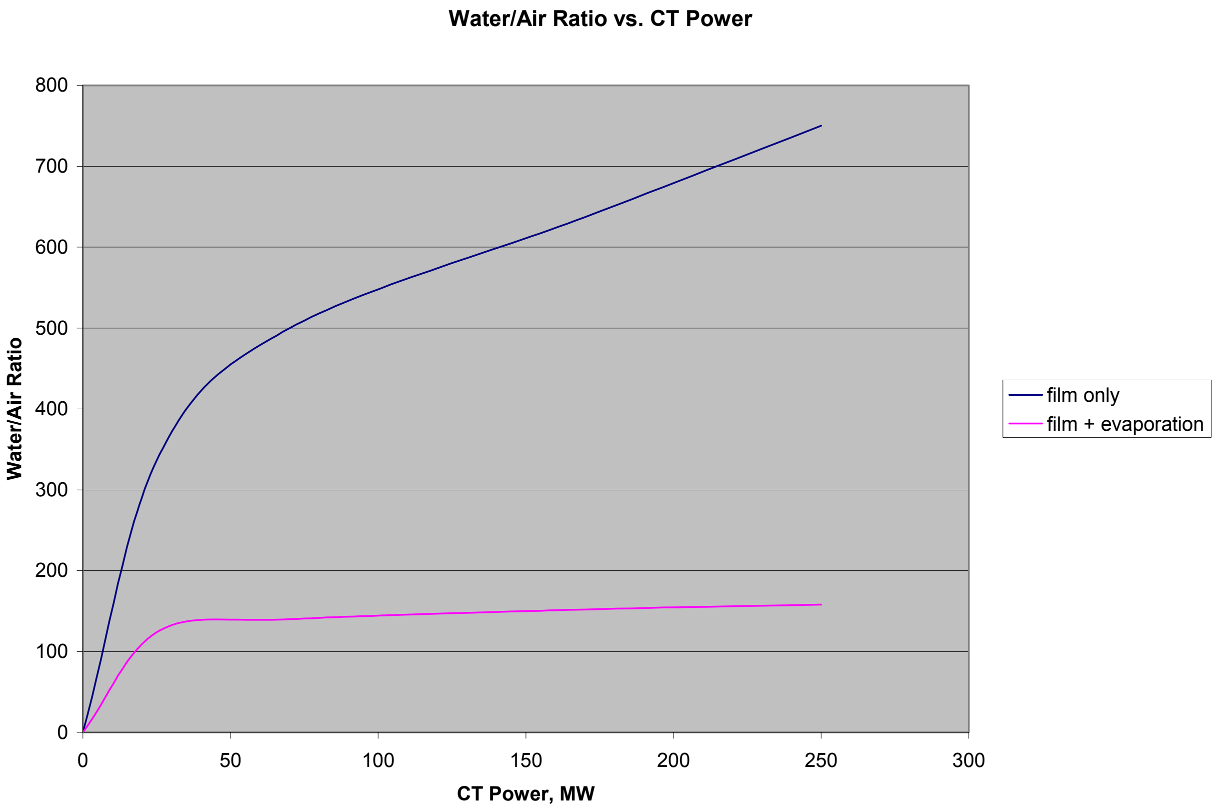


Figure 5-7
Air/Water Ratio Scaling with CT Output Power

Erosion is sufficiently serious for OEMs to limit the amount of on-line washing and/or resize the systems for lower water flow. While a long period of injection is desirable to soak the deposits, a short pulse is preferable to minimize water use and erosion. When the water is injected, the fuel flow increases to maintain the firing temperature, so staging of the water flow injection rate may be advisable for high injection rates to alleviate potential thermal cycling damage to hot section components.

A comparison of OEM system designs and vendor designs is given in Table 5-1 as influenced by the concern of erosion. The original OEM system has an air/water ratio of 250. The revised design increased the water/air ratio to 725. The advanced vendor design has an air/water ratio of 1570. Information on the amount of water injected is limited (Ref. 51) and the spread in the injection interval varies from less than 1 minute to as much as 30 minutes. If a pulse of 1 minute is sufficient, the total water usage in the case described above is 30 gallons (115 liters). The use of anti-fouling coatings (see Section 6) is likely to reduce the amount of water usage needed to recover the performance loss. The water flow rate, pulse interval, and frequency of the wash are parameters that can be established by performance monitoring. The mechanical and control elements of the wash system must have flexibility to conveniently adjust these parameters.

Table 5-1
GE – Turbotect Comparison of OLWW System Design Data (Courtesy of Turbotect)

GE Frame 9FA (256 MW)	GE Original OLWW	GE New/Modified OLWW (Nov. 2002)	Turbotect Mk1 Nozzle Eems (Design: 1993)	Turbotect Mk3 Nozzle for Large GTs (2003)
Number of Nozzles	18	9	40 Erosion 8,9 mils After 51,936 hrs	28
Total Water Mass Flow	38 gpm 144 l/min	App. 13 gpm App. 48 l/min (Approx. 1/3 of GE Orig. Mass Flow)	8 gpm 31 l/min (21% of GE Orig. Mass Flow)	6 gpm 22 l/min (15% of GE Orig. Mass Flow)
Water Pressure	100 psi 6.9 bar	40 psi 2.8 bar	58 psi 4 bar	58 psi 4 bar

Summarizing the key points of the results above and in Appendix A:

- A method is derived to relate the water flow requirements for a complete water wash film to the airfoil geometry and its condition.
- A smooth airfoil requires less water than a rough airfoil to maintain a film for a complete water wash film.
- The air/water ratio that is required to maintain a film is proportional to the square root of the air flow.
- Evaporation can reduce the film thickness and added water is needed to compensate for the losses.
- A water wash system should have the flexibility to conveniently adjust the water flow rate and the injection time to maximize the benefit of the water used.
- The results suggest increased water flow rate to enhance washing effectiveness and reduced injection time to limit the amount of water and the erosion.

Liquid Cleaning Options

For on-line systems, the compressor may be cleaned while operating at full speed or the load may be reduced to minimize the effects of added vibration or unsteady loads (stresses) that may arise due to the ingestion of liquids. On-line washing systems (Refs. 45 to 55) use de-mineralized water or mixtures of de-mineralized water and various chemical agents depending on the type of fouling. If the fouling is of an oily hydrocarbon nature, then a solvent based cleaning compound or surfactant is justified. A rinse may be helpful if a sticky residue results after the cleaner is applied. A biodegradable detergent compound is useful for lightly fouled surfaces. Water alone may be sufficient to remove dry deposits. Commercially available liquid cleaners are noted in Table 5-2.

Table 5-2
Liquid Cleaning Compounds

Product	Source	Type	Application
Fyrewash	Rochem	Solvent or Water Based Concentrate	On-Line, Off-Line
Krankwash	Rochem	Solvent or Water Based Concentrate	Off-Line
ECT Inc.	R-MC	Water Based Surfactant	On-line
Turbotect 950	Turbotect.	Water Based Concentrate	On-Line, Off-Line
Turbotect 927	Turbotect.	Solvent Based Concentrate	On-Line, Off-Line
Turbotect 2020	Turbotect.	Water Based Concentrate	On-Line, Off-Line
ZOK 27	ZOK	Water Based Concentrate	On-Line, Off-Line

The selection of the appropriate cleaning compound is a result of testing and evaluation. A test program described in Ref. 53 evaluated different products, water/chemical mix ratios and wash intervals to determine the most economically advantageous approach. The product selection was found to be site-dependent due to the different nature of the fouling deposits. A stronger mix of solvents is more effective where there are heavy hydrocarbon deposits. Surface wipes and laboratory analysis are performed to identify the nature of the deposits and the requirements of a washing compound. The use of detergents may require one or more rinses to clear the nozzles to prevent clogging. The off-line and on-line cleaning compounds generally are described in the operating permits indicating whether they meet biodegradable and non-hazardous material criteria as given by EPA, DOT, and IATA.

On-Line Compressor Wash Products & Services

Conntect, Inc.

Contact
Brookfield, CT 06804
(203) 775-8445

Products & Services
Conntect offers on-line and off-line compressor water wash systems, including the skid and nozzle assemblies and cleaning fluids.

Engine Cleaning Technology, Inc (ECT)

Contact
ECT, Inc.
401 E 4th, Bldg #20
Bridgeport, PA 19405
(610) 239 5120

Products & Services
ECT manufactures R-MC compressor cleaner, and offers nozzle/manifold assemblies for on-line or crank wash applications. ECT also offers wash skids.

Gas Turbine Efficiency (GTE)

Contact
Hugh Sales
12 Greenway Plaza, Suite 1100
Houston, TX 77019

Products & Services
GTE offers a second generation water wash system including the skids and nozzle assemblies for both off-line and on-line water washing that utilizes a minimum number of nozzle and high pressure heated water designed for low water consumption, good coverage and penetration and minimal erosion.

The nozzle pressures are 750 - 1400 psi (51.7 - 96.5 bar) resulting in the bulk of the droplets to be between 60 - 90 microns. A water skid/console is provided for water storage and water heating to 160°F (71.1°C). GTE recommends de-mineralized water without solvents for cleaning, but their system can use any OEM approved consumables.

GTE products are used in large and small heavy duty and aero-derivative CTs.

Rochem Technical Services (USA) Ltd.

Contact
1308 SE 20th St.
Cape Coral, FL 33990
(936) 609-7184

1532 Hartwick St.
Houston, TX 77093
(281) 227-1447

4711 SW Huber St., Suite 7E
Portland, OR 97219

Products & Services
Rochem offers the following Fyrewash family of concentrated (to be mixed with de-mineralized water) compressor cleaning systems:

- Fyrewash F1 is a petroleum based solvent for on-line and off-line heavy duty cleaning.
- Fyrewash F2 is a non-petroleum-based biodegradable solvent cleaner for oily/carbonaceous fouling.
- Fyrewash F3 is a water-based detergent for on-line and off-line cleaning.
- Krankwash KSB solvent is formulated to be environmentally safe off-line wash cleaner (20 minute soak).
- Krankwash KWB is a non-solvent, highly biodegradable, detergent off-line cleaner (20 minute soak).

Rochem offers complete custom design and supply of water wash off-line or on-line systems and performance monitoring software. Their water wash system controls allow for adjustment of numerous wash parameters.

Turbotect Ltd.

Contact
Turbotect (USA), Inc.
9203 Hwy 6 South
Suite 124, PMB #422
Houston, TX 77083

Products & Services
Turbotect offers compressor wash skids, injection nozzles and cleaning fluids. Both mobile and stationary wash skids are offered with local or remote control. The injection nozzles are a patented ball-type nozzle that can be oriented to optimize the spray for a specific flow situation. Ready-to-use water-based (Turbotect 9500) or concentrated solvent-based (Turbotect 927) cleaning fluids are offered.

A second generation nozzle design, the MK3, uses a nozzle that has compressed air jets that shield the water spray jet from flow deflection resulting in improved spray penetration and coverage, especially for large CTs. The new nozzle design uses less water, reducing the potential for erosion.

Zok International Group Ltd.

Contact
Zokman Products
1220 East Gump Road
Ft. Wayne, IN 46845
(260) 637-4038

Product & Services
- Zok offers ZOK mix, a high-strength, environmentally-friendly, concentrated solvent cleaner.
- The ZOK mix RTU (ready to use) is premixed with de-mineralized water, so no on-site water supply is needed.
- Zok 27 is their standard compressor cleaner which also comes premixed as ZOK 27 RTU.

On-Line Water Wash System Application Guidelines

Gas turbine operators interested in applying on-line water wash techniques to enhance capacity and reduce operating and maintenance costs should consider the following steps:

1. Gather Data and Formulate Objectives
    A. Gas Turbine Model Data
        a. Compressor Inlet Scroll Cross-Section and Dimensions
        b. Site Conditions
        c. New and Clean Performance

   B. Review Maintenance Experience
      a. Frequency of Crank Wash
      b. Maintenance Cost, Lost Revenue, Added Fuel Cost Estimates
      c. Type of Deposits and Removal Issues
      d. Aerosol Measurements
      e. Erosion or Corrosion Issues
      f. Peripheral Equipment Issues
   C. Specify Capacity Requirements
      a. Critical Peaking Conditions
      b. Critical Hot Weather Conditions
   D. Specify Economic Requirements
      a. Optimize Revenue
      b. Minimize O&M Costs

2. System Specification
   A. Dosage Control, Amount and Concentration
   B. De-ionized (DI) Water Mix and Rinse Rate Control
   C. Capability to Vary Cleaner Type and Quantity
   D. Manual Control or Automated
   E. Nozzle Manifold Features - Safety, Adjustable
   F. Meets OEM-Recommended Limits on Chemical Content, Dosage, and Quality

3. System Selection Criteria
   A. Meets Specifications
   B. Documented Experience
   C. Installed Cost
   D. Operating Cost

4. Test Matrix of Dosage/Interval/Cleaner Type
   A. Benchmark dry test for 30 days
   B. Test supplier-recommended injection rate, duration, interval, and dosage
   C. Repeat test with changed injection rate
   D. Repeat test with changed injection duration
   E. Repeat test with changed injection interval
   F. Repeat test with changed dosage
   G. Analyze trend data, refine the operating parameters, and evaluate
   H. Repeat other tests as needed

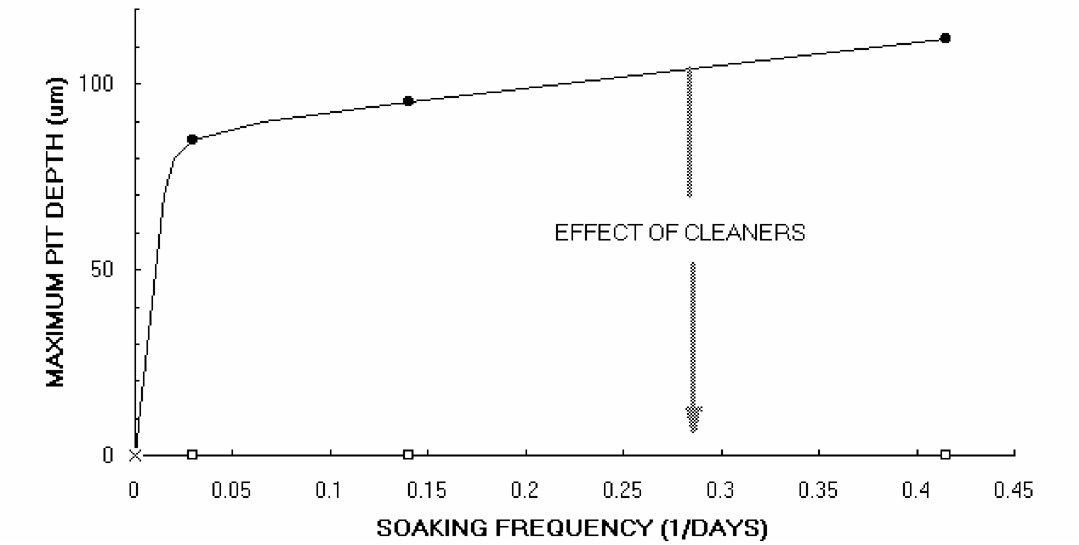
5. Monitoring to Support Test Matrix
   A. Instrumentation
      a. Power
      b. Inlet Temperature
      c. Barometric Pressure
      d. As Required to Achieve Objectives
   B. Software
      a. Power Deviations
      b. Trending
      c. Economic Analysis

Off-Line Water Wash Guidelines

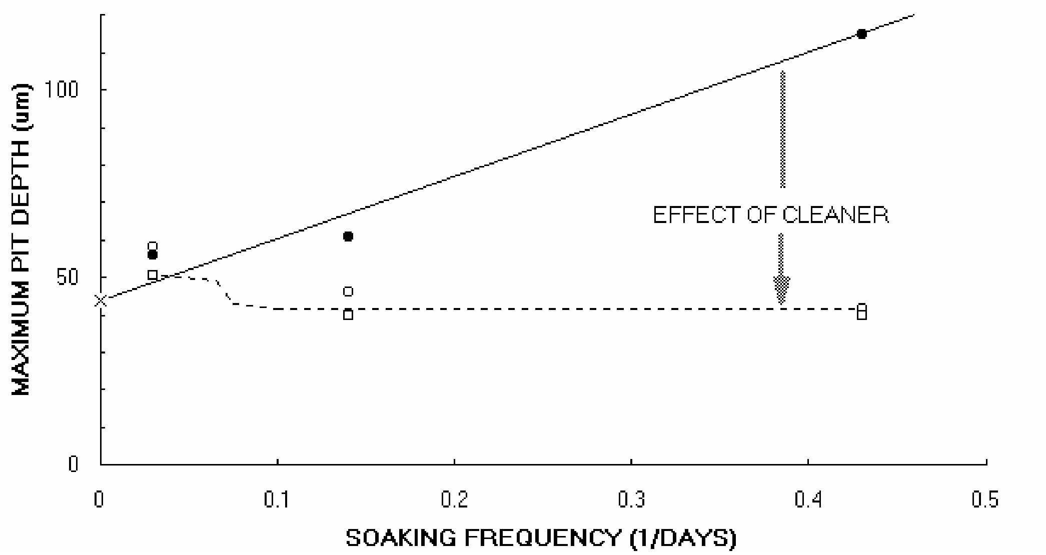
The off-line crank wash procedures may be fully automated by the OEM, with spray nozzles and an injection ring for a solid compound cleaner, and with valves located in the air line piping to prevent water entrainment. If the system is not automated, then:

1. Preparations
   A. Block air lines from water/cleaner entry
   B. Equip air lines with traps if necessary
   C. Open drains
   D. Close flame detector ports
   E. Guide vanes in a full open position
2. Cleaning Cycle
   A. Rinse 5 - 10 minutes with compressor at low speed
   B. Inject a water/detergent mix while decelerating rotor
   C. Inject at a rate sufficient to thoroughly wet airfoil and wall surfaces
   D. Stop injection when rotor speed is too low to carry water throughout compressor
   E. Allow a period of soaking to loosen deposits
   F. Repeat wash cycle until drain water is clear, check conductivity values
   G. Rinse and run engine for at least 20 minutes to dry out
   H. During the first few fired hours after the wash, apply a solid compound cleaner

The use of corrosion inhibitors can limit additional corrosion that may be induced by leaving the blades wet (no drying) when in a humid, industrial atmosphere. Corrosion tests on metal coupons simulating wash with 200 ppm of salt are reported in Ref. 54. The test results in Figure 5-8 show the benefits of rinsing and drying after each wash.



(a) 17-4 PH Coupons



(b) 410 Coupons

Symbol	Soaking In	Simulates
×	Not soaked	Washing with rinsing and drying
•	200 ppm NaCl	Washing without rinsing and drying
o	200 ppm NaCl + Cleaner	Washing with cleaner without rinsing and drying
□	Cleaner	Less severe circumstances

Figure 5-8
Effects of Washing Processes on Corrosion (Kolkman, 1993, Ref. 55)

Solid Cleaning Options

There are two types of solid compounds used: organic (nutshells and rice) and inert (catalyst support, spent catalyst, polishing powders). Combustible compounds are preferred to inert compounds. Inert compounds are only used where nutshells may build up and cause a danger of fire. When the deposits are solid and dry, solid cleaners are often sufficient.

Solid compound cleaning is done at full speed and reduced load to reduce the potential for hot corrosion in the turbine section by the cleaning compound. Particle sizes are normally between 40 and 200 microns to ensure cleaning action, keep the particles airborne, and minimize blade erosion and the possibility of plugging cooling holes.

The dry solid compounds listed in Table 5-3 have been used for many years to clean the compressor of dry deposits. Experience has shown that the solid particles are detrimental to coatings and blade finish and can block cooling air holes. Hence, the trend is towards water and detergent cleaners. However, if performance cannot be recovered by a crank wash and if fouling of the latter stages is suspected, an application of a solid cleaner may help recover the lost performance.

Table 5-3
Solid Cleaning Compounds

Product	Source	Type
Jet Blast	Jet Blast Co.	Nutshells
Shelblast	Pangborn Corp.	Walnut Shells
Carboblast	Turco Products, Inc.	Nutshells
Pecan Shells	Composition Materials Co.	Pecan Shells
Rice	Common White, Uncooked	Medium or Long Grain

The gas turbine manufacturer-recommended procedures for compressor cleaning with liquid or solid compounds should be followed to prevent voiding the warranty agreement. The selected product should have the approval of the OEM for the intended use regarding:

- Injection Rate
- Chemical Content
- Size
- Quality

Solid compound cleaners are not recommended for use on aero-derivative gas turbines, coated compressors, or current heavy duty gas turbines that rely extensively on air-cooled parts in the hot section. The use of organic solids in regenerative gas turbines may result in a fire hazard if particles accumulate in the flow passages. When solids are used, caution should be taken in regard to the rate of injection to avoid clogging of instrumentation lines or cooling passages.

6
COMPRESSOR COATINGS

Compressor Coatings Selection Criteria

When considering retrofit compressor coating applications, the CT fleet operator typically has critical operating and maintenance issues. The issues may involve loss in performance, lower profit and/or accumulated corrosion and erosion effects that threaten parts failures. The type and location of the surface to be coated within a compressor impacts the optimal choice of coating. As shown in Figure 6-1, coatings on the first 5 stages of a compressor are likely to experience surface temperatures lower than 400°F (204.4°C), so that coatings pigmented with PTFE to make them slippery are appropriate. The high pressure stages of aero-derivative CTs impose the most severe temperature environment.

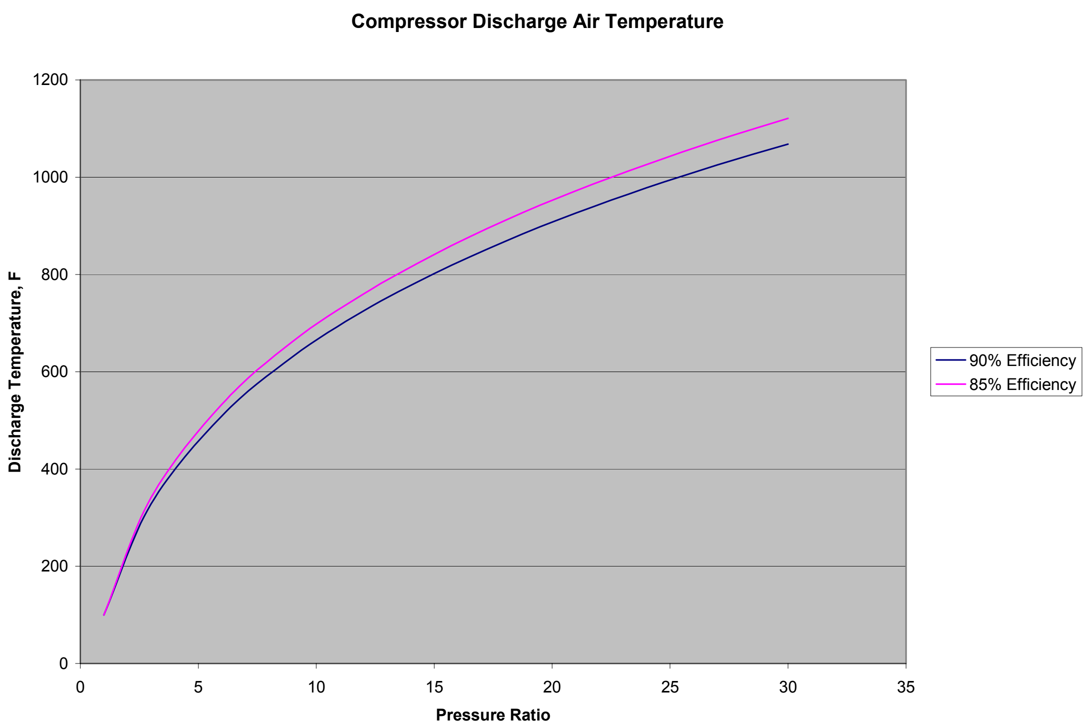


Figure 6-1
Compressor Airfoil Coating Temperature Environment

The benefit of airfoil coatings in improving the compressor efficiency was demonstrated through side-by-side testing of CTs with coated and uncoated compressor airfoils (Ref. 56). The test results showed that improvements in power and heat rate would result in a payback period of less than one year. As seen in Figure 6-1, a reduction in compressor efficiency raises the discharge air temperature, so that by measuring the temperature change one can infer the loss in efficiency. The difference in the compressor power and the net CT output power is proportional to the difference in discharge temperature. Whereas all companies engaged in coating compressor airfoils have documented success stories, a decreased benefit of an airfoil coating could arise if: 1) the initial roughness of the airfoil is too large (greater than 100 micro-inches (2.54 microns)), 2) there is a poor coating thickness distribution that changes the air foil contours, 3) there is excessive thickness, roughness or waviness due to the application (Ref. 36), or 4) the surface was not prepared well resulting in a weak bond.

The performance improvements resulting from coating the airfoils are a result of recovering losses by improved smoothness that cannot otherwise be recovered by water washes, as well as improving the benefits of water washes by using non-stick finishes such as those containing PTFE (Teflon). Maintaining the coating in good health and periodically stripping and re-coating may be justifiable as fuel costs rise (Ref. 46). When the surface roughness is reduced to a point where the surface is hydraulically smooth, then further reduction in roughness yields no added aerodynamic benefit. The typical roughness (Ra) level for this limiting condition is estimated to range from 30 - 50 micro-inches (0.762 - 1.27 microns). Further reductions in roughness are still beneficial as they delay the effects of fouling or increased roughness until the limiting condition is achieved. Commercial coatings for compressor blades are available with a surface smoothness of 25 micro-inches (0.635 microns) or better (Ref. 57).

Site conditions play a key role in selecting the optimum compressor coating. Coastal, fog, smog, and industrial environments places greater emphasis on corrosion resistance for all the compressor components, airfoils, wheel disks, etc. (Ref. 58). The corrosion protection of different coatings can be compared based on the ASTM B117 standard. This rating method is based on the use of a salt spray chamber to test how long a coating will survive without substrate corrosion. The service life of the coating is dependent on its adhesion strength. The ASTM test D 3359 is useful to rate the adhesion strength of different coatings. In this test, the coating is cut and a sticky tape is applied. The extent of flaking is assessed when the tape is removed. A rating of 5B indicates no flaking; a lower rating indicates flaking.

Erosion and increased clearances are exacerbated if there is little, if any, inlet filtration (as for older peaking CTs) so that a hard surface coating is a good candidate. Similarly, a hard erosion resistant coating may be favored for installations using inter-stage cooling that subjects the airfoil surfaces to a high loading of large droplets. A novel compressor airfoil coating is being evaluated for corrosion and erosion protection at a site using inter-stage cooling (Ref. 59). A chrome carbide coating protects the leading edges of the first five compressor blade rows. The total surface of the blade is then coated with Sermatel 5380 for corrosion protection and smoothness. Erosion (or abrasion) resistance is rated in accord with ASTM D 968 and defined as the amount of abrasive (liters/mil, L/mil) required to wear through a unit film thickness of the coating when tested in accord with the method in the ASTM standard. This standard provides a means for comparing the abrasion resistance of different coatings. Another aspect of the coating performance is the resistance to impact. A standard test for impact resistance is given in ASTM 2794. The standard test involves adjusting the height of a weight so that the energy of impact causes an indent into the coating. The rating is then in terms of (kg-meters or inch-lbs). In this way, the resistance to impact of different coatings can be compared.

Abradable coatings may be applied to the casing to reduce rotor tip clearance losses. A choice of an abradable coating applied to the casing depends on the material of the blade and the tip geometry of the blade (Ref. 60, 61). There are countering criteria such as durability, yet easily abradable. The goal is to produce a seal that does not cause blade wear, which maintains a smooth shroud surface and which remains intact for thousands of operating hours. Thermally-sprayed abradables release fine wear debris when machined by high velocity blade tips. The released material must be able to readily escape, which requires thin blade tip cross-sections or rough sandpaper-like surfaces on the blade tips. Experience has shown that when the blade tip width is larger than 0.7mm, the likelihood of blade wear and shroud rupture increase strongly.

Environmental, health, and safety concerns have resulted in strict control on the use of cadmium in coatings and presently there is concern about the presence of hexavalent chromium. As a result of environmental concerns, the EPA has targeted compounds of cadmium and chromium for reduced use. As a result of these environmental concerns, coatings that have no hazardous compounds are being developed (Ref. 62, 63). In particular, Sermatech has developed a chromium-free aluminum ceramic coating that meets the same specifications as the Sermatech 5380 DP coating.

Compressor Coating Application Technologies

There are a number of different coating application technologies used on compressors (Ref. 64).

- Air Spraying
- Thermal Spraying
- Physical & Chemical Vapor Deposition
- Electroplating
- Pack Aluminide
- Non-Drying Oil Coatings

Air Spraying

Air spray coatings are used in lower temperature applications which require corrosion resistance, high aerodynamic efficiency and a fine surface finish. Air spray coatings are applicable to any ferrous alloy substrate and thorough surface preparation is required. Air spray guns are used to apply an aluminum rich base coat for galvanic corrosion protection and a ceramic topcoat sealer for a barrier to corrosive attack. An optional anti-fouling topcoat of PTFE (Teflon) may be beneficial with on-line water wash systems. Coatings are applied to a uniform thickness and smoothness.

Thermal Spraying

Thermal spray coatings are well suited for retrofit applications for abradable coatings, hard surface coatings, and high temperature coatings. Many material compositions may be sprayed, provided there is a stable liquid phase, such as metals, ceramics, carbides and plastics or any combination. A thermal spray coating bond is chemical or physical, so stripping and repair is possible. The application of thermal coatings is by spray guns and the gun technology is continually improving. Spray guns of interest include:

Wire Flame Spray

An oxyfuel fuel gas wire spray gun feeds the coating in the form of a wire into the center of a multi-jet flame, melting the tip of the wire and spraying the molten material on to the substrate. This method is used for low melting point materials as aluminum, tin, zinc, and Babbitt. The fuel gases are oxyacetylene and air.

Powder Flame Spray

An oxyfuel gas powder spray gun feeds the coating material in the form of a powder into the spray flame. The molten mix of coating material is sprayed onto the surface. This approach gives a continuous supply of coating material, but the relatively low energy of the spray stream tends to result in low density deposits and low adhesion strength.

High Velocity Oxygen Fuel (HVOF) Flame Spray

The HVOF gun burning oxygen and fuel gas yields a higher coating density and improved adhesion to the substrate. Specially designed spray guns give a higher velocity to the coating material.

High Velocity Fuel Detonation Spray

The spray gun (D gun) has a barrel into which the powdered coating material and an oxygen/fuel mixture are metered and then ignited by a spark discharge. The mixture detonates resulting in an extremely high spray velocity. The operating cycle is repeated to yield a continuous deposit of the highest density and best adhesion. The noise of this process is very high and is usually carried out in an automated chamber.

Electric Arc Wire Spray

Electric arc wire spraying uses twin wires that are fed together into a spray head in which an arc is struck. Molten material is projected from the arc by means of a high pressure gas jet. Different materials may be used for each wire to form a composite. While high spray rates are possible, the resulting density of the coating is lower than plasma or D gun applications.

Plasma Spray

The high temperature plasma produced by a high current electric arc enables almost all materials to be sprayed. Deposits are of high density and strongly bonded to the substrate. The spraying may be done in air (APS) or in a vacuum, but the vacuum application eliminates oxidation of the spray material resulting in superior coating properties. Plasma spraying is costly and requires bulky equipment.

Physical vapor deposition (PVD) techniques are used to make very thin films utilizing various techniques such as vacuum evaporation, gas scattering deposition and sputter coating. Materials that can be used for (PVD) coatings include the wear resistant nitrides and carbides. Chemical vapor deposition (CVD) is used to apply thin or relatively thick coatings. The process is especially applicable to depositing pure 100% dense metals such as refractory metals, which cannot otherwise be electroplated.

Electroplating a NiCd coating on the forward stages of the compressor was used extensively by OEMs to substantially reduce corrosion. Existing NiCd coatings are subject to deterioration at an increased rate in the presence of humid and industrial environments that contain aerosols of sulfur compounds. Diffused pack aluminide coatings provide corrosion and particulate erosion protection, but their higher cost and time required for application are disadvantages for use on compressors (Ref. 65).

A coating for increased protection against solid particle erosion (SPE), as found in most steam turbines, has been developed under EPRI sponsorship. The patented SPE-resistant coating consists of a chromium carbide-FeCrAlY mix. This coating can be applied by a thermal spray (Ref. 66). This coating may be applicable to the airfoils in the forward section of a compressor where severe erosion may occur.

In the case of peaking units located in a corrosive environment, the use of an easily strippable coating may be desirable as a low cost approach to prevention of corrosion. An acid-free, non-drying oil coating that can be removed by a detergent spray wash would protect the surface against chemical or electrochemical attacks.

Compressor Restoration and Coating Products & Services

AAR

Contact
148 Industrial Park Drive
Frankfort, NY 13340
(315) 731-3700

Products & Services

- AAR offers aluminum ceramic coatings for compressor components corrosion control with a capability for stacked rotor coating application.
- Tungsten and chromium carbide coatings are offered for erosion protection.
- Services include inspection and restoration of compressor blades and vanes.

Ambeon
(Division of Sulzer)

Contact
Fort Saskatchewan
Alberta, Canada
(780) 992-5283

Products & Services
Ambeon offers Durablade, a thermal spray-abradable seal material, and Neomet honeycomb, also used for gas path sealing and clearance control in gas turbines.

BayState Surface Technologies
(Subsidiary of Aimtek, Inc.)

Contact
201 Washington St.
Auburn, MA 01501
(800) 772-0104

Products & Services
BayState Surface Technologies offers a complete line of thermal spray materials; pure metals, alloys, ceramics, carbides and the PG Series PlasmaGun. The hand-held PlasmaGun has evolved over the past 30 years.

Hickham Industries
(Division of Sulzer)

Contact
11518 Old LaPorte Rd.
LaPorte, TX 77571
(713) 567-2700

Products & Services
Hickham offers:

- Air spray coatings that provides corrosion resistance and a fine surface finish, HICoat A08, A21 and A24 coatings.
- HVOF thermal spray chrome carbide coatings and abradable coatings.
- Plasma coating for abradable coating applications
- Full service repair and restoration of compressor components.

Liburdi Engineering

Contact
404 Armour St.
Davidson, NC 28036
(704) 892-8872

Products & Services
Liburdi Engineering offers physical vapor deposition (PVD) wear resistant coatings, including titanium nitride, chromium carbide, and tungsten carbide coatings on compressor airfoils.

Natole Turbine Enterprises (NTE)

Contact
P.O. Box 1167
LaPorte, TX 77572
(281) 470-9226

Products & Services
NTE offers:

- Gas turbine parts
- Repairability inspections
- Vendor verification services
- Cost/benefit analysis of restoration vs. replacement

Pratt & Whitney Power Systems

Contact
1165 Northchase Pkwy
Marietta, GA 30067
(770) 859-1999

Products & Services
Pratt & Whitney offers advanced technology coatings and repairs for gas turbine compressors.

Praxair

Contact
39 Old Ridgebury Rd.
Danbury, CT 06810
(800) 772-9985

Products & Services
Praxair offers a product line that includes metallic, carbides and ceramic powders and wires. Application equipment products include plasma, HVOF, D-gun, and arc spray equipment.

Sermatech

Contact
Windsor, CT
(203) 683-0711

Limerick, PA
(215) 948-5100

Lantana, FL
(305) 582-6080

Sugar Land, TX
(713) 240-1444

Compton, CA
(213) 604-0018

Bruce McMordie
Manager R&D Lab
(610) 948-2840

Products & Services
Sermatech offers the Process 5380 Dense Pack which is a dense corrosion/erosion resistant coating and has an aerodynamically smooth surface finish. The coating provides protection to ferrous alloys at temperatures up to 1200°F (649°C). An air sprayed and cured two-coat metal-ceramic system, the base coat is an inorganic polymer mix containing aluminum powder particles. A top coat is added for improved corrosion resistance. The coating is burnished with glass beads for improved coating structure. The metallic-ceramic coatings result in an electrically conductive coating with base metal corrosion resistance for ferrous alloys as a result of the higher galvanic potential of the aluminum.

Sermatech also offers complete restoration services.

Sulzer Metco

Contact
1101 Prospect Avenue
Westbury, NY 11590
(866) 238-5933

Products & Services

- Sulzer Metco is a global supplier of a broad line of thermal spray surface technology equipment and materials.
- The proprietary SUME brand coatings are currently available only in Europe and Japan.
- Sulzer Metco offers a complete portfolio of plasma spray guns.

The Nanosteel Company

Contact
485 North Keller Road
Suite 100
Maitland, FL 32751
(407) 838-1427

Products & Services
Nanosteel has patented the Super Hard Steel™ coating material which can be thermally-sprayed for wear resistance and is more ductile than redractory materials.

Triumph Gas Turbine Services Group

Contact
Triumph Gas Turbine Services Group
6519 W. Allison Road
Chandler, AZ 85226
(520) 796 6300

Products & Services
Triumph provides compressor blades and diaphragms and airfoil restorations.

Turbine Resources Unlimited (TRU)

Contact
William Howard, President
P.O. Box 430
West Winfield, NY 13491

Products & Services
TRU offers Flow 2000, a two layer coating with a corrosion-protective sacrificial aluminum base coat and a smooth Teflon seal coat. The maximum operating temperature is limited to 500°F (260°C). Applications include compressor rotating and stationary airfoils and inlet bell housings.

Seal 3000 is a two-layer coating with a corrosion protective aluminum base coat and a smooth hardened seal coat with a maximum temperature capability of 1150°F (621°C). Applications include rotating and stationary airfoils.

TRU applies abradable coatings that wear upon incidental contact to control clearances in compressors.

TurboCare

Contact
2140 Westover Rd.
Chicopee, MA 01022
(800) 887-2622

Products & Services
TurboCare offers repair, reconditioning and coating of compressor blades and vanes. TurboCare can apply APS and HVOC, diffused aluminides, and wear resistant coatings. Blade inspection and surface restoration services are also offered.

Uniquecoat Technologies, LLC

Contact
1070 Merchants Lane
Oilville, VA 23129
(804) 784-0997

Products & Services
Uniquecoat products include a patented thermal spray technology based on an Activated Combustion High Velocity Air-Fuel (AC-HVAF) gun. The spray is done at a temperature lower than the melting temperature of the material and so avoids oxidation and degradation of the coating material. The limitations of the Cold Spray are avoided so that hard alloys and cemented carbides for hard facing can be sprayed. Thermal coatings applied with the AC-HVAC have high density. The company offers proprietary spray equipment and coating services.

White Engineering Services

Contact
One Pheasant Run
Newtown Industrial Commons
Newtown, PA 18940
(215) 968-5021

Products & Services
White Engineering Services offers the WESTCOAT thermal coating services for metals, ceramics (aluminum oxide, chromium oxide), carbides, cermets, Teflon, dry films, metal alloys, diffused coatings. Application technologies include wire arc spray, plasma spray, HVOF, and flame spray.

Other Resources

AZoM

Contact
USA & Pacific Rim Sales
139 Hudson Parade
Clareville, Sydney
NSW 2107, Australia
+61 (0)2-9918-7475

Products & Services
The AZoM (A to Z of Metals) web site AZOM.com offers a database on materials and industries that covers thermal spray coatings and abradable materials.

ASM Thermal Spray Society

Contact
Materials Park, OH

International Thermal Spray Association (ITSA)

Contact
Materials Park, OH

Metal Powder Industries Federation

Contact
105 College Road East
Princeton, NJ 08640

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A
Appendix

Estimate of Water Flow Rate Requirements for On-Line Water Wash

A flow model is hypothesized to estimate the water flow rate needed to sustain a water wash film over the airfoil surface of an entire blade row. The model assumes that some of the water droplets initially impact the airfoil surfaces and although there may be splashing, they form a film. The use of liquid films for cooling is well known and in this case, the film is evaporating due to convective heat transfer and cooling the air flow. The water film is likely to be at the local wet bulb temperature. The film concept is as depicted in Figure A-1.

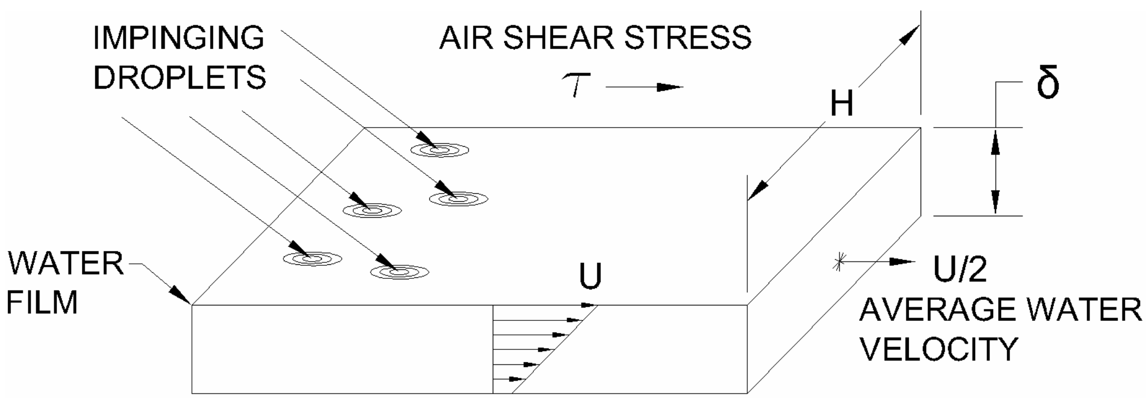

Figure A-1
Water Film Formed on an Airfoil by Impinging Droplets

The film model assumes that water flows off the trailing edge of an airfoil breaking up into droplets to reform a film again on a downstream surface. The droplet impacts around an airfoil are not uniform, as the inertial effects that cause the impacts are more strongly felt on the leading edge and the pressure side of an airfoil. Nevertheless, a simplified model serves to identify parameters that can lead to a better understanding of the water wash phenomena.

An estimate of the water film thickness formed by the droplet impacts and the film flow velocity can be made based on two conditions: 1) the water film must cover the entire blade surface to effectively clean the surface and 2) there is a viscous shear stress in the water film that transfers the aerodynamic drag to the surface of the blade. At the interface between the air and the water film the water velocity is much lower than the air velocity, so the drag by the air is approximately the same as it would be on a solid surface under the condition of no slip. The fluid shear stresses in the air and the water at the interface are the same. The shear stress at the surface is also the same as that at the interface, so the drag of the air is transferred to the surface.

The description above translates into the following expressions for the water flow rate for a surface of width (H) and a film thickness of (δ) on both sides:

$$W = \text{water flow} = 2\ \sigma_w u_w\ \delta\ H$$

Where σ_w is the water density and (u_W) is the average axial water velocity in the film. The effects of body forces on the fluid film are neglected. In order to calculate the water rate flow, a relationship is needed between the average water velocity and the film thickness. The approach relates the boundary condition that the shear in the water film transfers the aerodynamic drag to the airfoil surface. The shear in the water film is simply estimated as:

$$\tau = \text{shear stress} = \mu / g\ (du / dy)$$

Where (μ) is the viscosity of water and du/dy is the gradient of the water velocity normal to the surface. Note that u = 0 at the surface under the condition of no slip, and since the shear is transferred to the surface, it is assumed constant so that du/dy = U/δ where U is the velocity at the edge of the film and δ is the film thickness. The shear stress in the water film becomes:

$$\tau = \mu U / (g\delta)$$

The shear stresses due to the air flow vary around an airfoil due to the initial laminar flow, transition to turbulent flow, roughness, airfoil shape and velocity distribution. Airfoil losses are commonly presented in terms of a total pressure loss that is measured across a cascade of airfoils. When the compressor airfoil is operating close to the optimum design condition, the predominant contribution to the airfoil drag is due to viscous shear stresses. The relationship between the drag force and the total pressure loss coefficient is shown in Figure A-2.

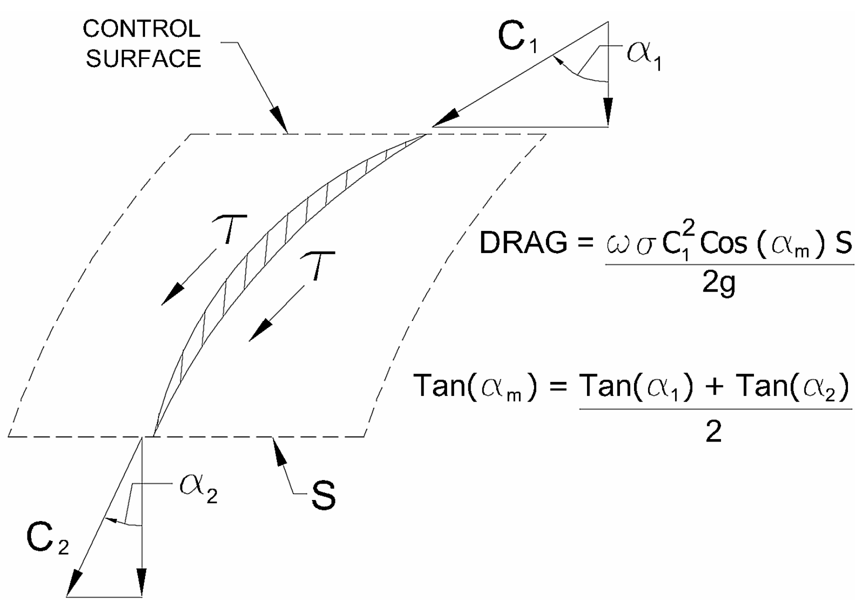

Figure A-2
Airfoil Drag and the Total Pressure Loss Coefficient (2)

An average surface shear stress is estimated by balancing the drag the airfoil with the total pressure losses such that, approximately (Ref. 2):

$$\tau = \omega\ \sigma_a\ V^2(S / C)\cos(\alpha_m) / (4g)$$

Where σ_a is the air density, V is the relative air velocity to the airfoil, ω is the total pressure loss coefficient, S is the pitch, C is the chord and α_m is mean flow angle relative to the axial direction. Estimates of ω are given in Figure A-3 (Ref. 67). This approximation is a conservative estimate for the average shear stress (high side) as there may be other significant losses due to the incidence angle, trailing edge loss and separation on the suction surface.

Equating the air-drag shear stress and the water film shear stress:

$$U = \tau\ \delta g / \mu$$

The thickness of the film (on both sides of each blade) should be larger than the anticipated heights of the deposits, but limited so as to minimize water consumption say for example, 10 times 100 micro-inches or 1 mil (0.001 inch). For a 1st stage example, ω =.01 , σ_a = 0.07 lbs/ft^3 (1.2 kg/m^3), V = 800 fps (243.8 mps), δ = 0.001/12 ft (0.025 mm), μ = 0.0006 lb/ft-sec (0.0009 kg/m-sec), S/C = 0.7, α_m = 40 deg. resulting in τ = 1.7 lbs/ft^2 (8.3 kg/m^2) and U =7.7.fps (2.4 mps).

To estimate the water flow required for the above film parameters on rotor with 30 blades, wet on both sides and the blade height is H = 1 ft (0.305 m) and N = 30 blades in a row:

$$W = \sigma_w\ U\delta\ H\ N = \sigma_w\ \tau\ \delta^2 g/\mu\ H\ N = 62.4 \times 1.77 \times 32.2 \times (0.001/12)^2 \times 1 \times 30 / 0.0006$$

$$W = 1.2 \text{ lbs/sec (8.6 gpm (33 lpm), or 0.2 \% of the air flow)}$$

The estimates above correspond to a CT such as a GE 7E.

Besides the water flow needed to sustain a water film, some of the water is evaporated. As a nozzle emits a spray with a droplet size variation, the finer droplets will tend to evaporate and/or follow the streamlines without benefiting the water wash. There is also evaporation from the water film surface due to convection. The maximum amount evaporated in the first stage depends on the climatic conditions and whether a fogger is used. If a fogger is used, then the air entering the 1st rotor is nearly saturated, but further evaporation occurs as the temperature rises in the stage. The maximum droplet and surface film evaporation in a stage to saturate is approximately a water/air ratio of 0.005. This result is found by adding water during the compression of the air so as to keep the air saturated. The water flow to keep the air saturated in the 1st stage and maintain the water film thickness of 1 mil (25 microns) is 30 gpm (115 lpm), bringing the potential total water flow needed to as much as 0.7 % of the air flow. Utilizing a spray with large droplet sizes slows the droplet evaporation and reduces the amount of water required, but the impact effects are higher.

A scaling relationship for the water/air ratio for aerodynamically similar compressors but of different size may be derived as the product of HN may be written as:

$$HN = H / C \times C \times \pi D / S$$

where:

H/C = aspect ratio
S/C = solidity
D = compressor diameter

Since the air flow (Wa) may be expressed as:

$$Wa \sim D^2 \times (1 - HTR^2)$$

where HTR is the hub/tip ratio.

Therefore, for aerodynamically similar compressors, the air/water ratio scales as:

$$Wa / W \sim (Wa)^{1/2}$$

It can also be shown that the water film velocity U scales as:

$$U \sim (Wa)^{1/4}$$

Summarizing the key points of the results above:

- A method is derived to relate the water flow requirements for a complete water wash film to the airfoil geometry and its condition.
- A smooth airfoil requires less water than a rough airfoil to maintain a film for a complete water wash film.
- The air/water ratio required to maintain a water film is proportional to the square root of the CT air flow.
- The air water ratio required is lower as the air density increases in later stages
- Evaporation may reduce the film thickness and added water is needed to compensate for the losses.

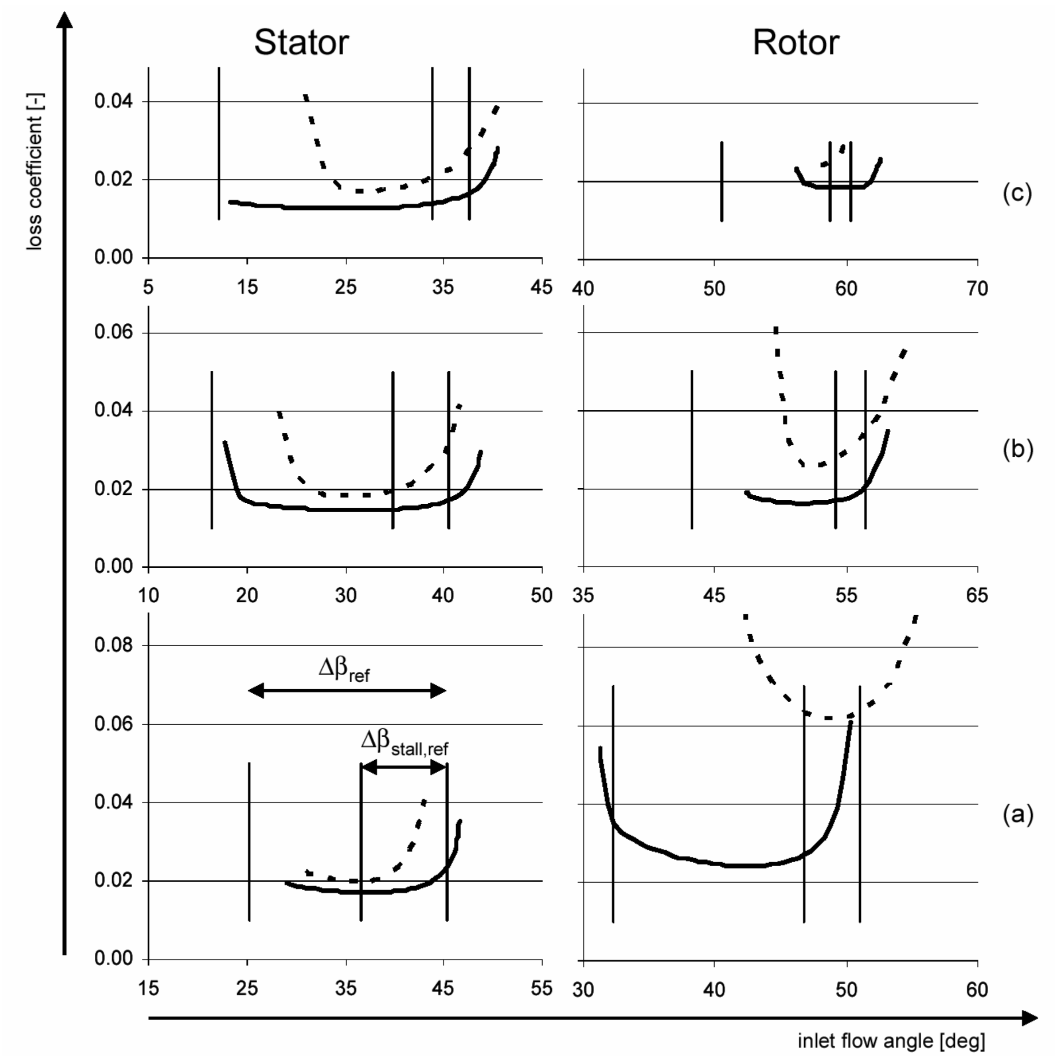

Figure A-3
Loss Coefficient for (a) Hub, (b) Midspan, and (c) Casing Section of Stator and Rotor for Existing (Dotted) and Optimized Profile

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