ABSTRACT

Power enhancement technologies for gas turbines such as inlet fogging, interstage water injection, saturation cooling, inlet chillers, and combustor injection are being used by end users without evaluating the potentially negative effects these devices may have on the operational integrity of the gas turbine. Specifically, the effect of these add-on devices, off-design operating conditions, nonstandard fuels, and compressor degradation/fouling on the gas turbine’s axial compressor surge margin and aerodynamic stability is often overlooked. However, aerodynamic axial compressor instabilities caused by these factors can be directly linked to blade high-cycle fatigue and subsequent catastrophic gas turbine failure. A careful analysis should always precede the application of power enhancement devices, especially if the gas turbine is operated at extreme conditions, uses older internal parts that are degraded and weakened, or uses nonstandard fuels.

This paper discusses a simple method to assess the major factors that affect the aerodynamic stability of a single shaft gas turbine’s axial compressor. As an example, the method is applied to a frame type gas turbine, and the results are presented. These results show that inlet cooling alone will not cause gas turbine aerodynamic instabilities, but it can be a contributing factor if, for other reasons, the machine’s surge margin is already slim. The approach described herein can be employed to identify high-risk applications and bound the gas turbine operating regions to limit the risk of blade life reducing aerodynamic instability and potential catastrophic failure.

INTRODUCTION

As the widespread introduction of inlet cooling and water injection technologies for gas turbines is a relatively recent development, only limited historical field operating data and information on the effect on parts life is available. Nonetheless, because of the commercial advantage these technologies may bring to the operator, a large number of the devices have been installed over the past five years. Since the introduction of inlet cooling and water injection technologies for ground-based gas turbines, over 800 gas turbine installations worldwide have been built or retrofitted with these technologies. However, inlet air-cooling, in combination with off-design operating conditions, nonstandard fuels, combustion dilution, and compressor blade degradation, can lead to compressor aerodynamic instabilities and subsequent gas turbine high-cycle fatigue blade failures. Namely, if the axial compressor’s surge margin is significantly reduced by aberrant gas turbine application and operation, the machine may well be in an operating region of localized rotating stall and blade flutter. Flutter has a very strong potential to cause blade damage due to high-cycle metal fatigue.

A number of different gas turbine power augmentation technologies are currently available. They can be generally classified into two categories:
• Evaporative cooling—These include wetted media, fogging, wet compression, overspray, and interstage injection
• Chillers—Mechanical and absorption chillers with or without thermal energy storage

In general, chillers only affect the gas turbine by directly cooling the inlet air temperature. This increases the gas turbine’s output power but has otherwise minimal effects on the gas turbine’s internal aerodynamics, i.e., the gas turbine is behaving as if it is running on a cold weather day. Other than some cases where ice formation on the turbine inlet has caused problems with casing distortion, rubbing, and foreign object damage, few gas turbine failures have been directly linked to inlet chilling. However, inlet chillers are generally very large, expensive to operate, and economically often only marginally feasible. Thus, they are not widely employed.

On the other hand, evaporative inlet cooling relies on injecting water droplets or vapor into the gas turbine’s compressor (either upstream or interstage). Within the context of this paper, the authors will treat fogging and conventional evaporative cooling as having the same thermodynamic effect on the gas turbine. However, two different principles are employed: evaporative coolers utilize a wetted media that is exposed to the inlet air flow, while inlet fogging sprays water mist into the gas turbine inlet system. Either system is normally designed to avoid liquid water carryover into the engine inlet, but fogging systems can sometimes (unintentionally or intentionally) “overspray,” i.e., a significant amount of water does not evaporate before entering the gas turbine inlet. Overspraying has a similar effect as interstage water injection in that water droplets will be present inside the first stages of the axial compressor.

Evaporative cooling has been demonstrated to provide between five to 10 percent power augmentation on hot/dry days and is, thus, commercially attractive. The principal difference between the various commercially available evaporative cooling technologies is the quantity of water (percent air saturation), the water droplet size, and the location of the water injection ports into the gas turbine. However, the basic functional principle of all inlet water augmentation technologies is that they effectively reduce the gas turbine’s inlet air temperature from the air’s dry bulb temperature to the wet bulb temperature of the ambient air. This effective temperature difference depends on the ambient air’s relative humidity and temperature, as well as the evaporative cooler’s efficiency. The efficiency of evaporative devices is typically measured as the percent of the difference between dry and wet bulb temperature achieved; efficiencies of 80 to 95 percent have been reported.

When discussing gas turbine compressor aerodynamic instability, one must also address the issues of combustor diluent injection and low heating value fuels. Although these technologies are primarily concerned with NOx reduction and fuel flexibility, gas turbine power output is increased, and the gas turbine’s aerodynamic stability can be severely affected. Most combustor diluent injection applications employ steam or water as the medium, but, in some process power plants, CO2 or nitrogen has successfully been utilized for NOx reduction. Similarly, the use of low heating value fuels in a gas turbine produces the same net aerodynamic effect as combustor diluent injection. Thus, the influence of diluent injection and low heating value fuel combustion on gas turbine operational stability is also briefly discussed in this paper.

Previous Work on Evaporative Cooling and Wet Compression

Because of the recent popularity of water injection, a number of researchers have studied the effects of the various technologies available on the market. For example, Horlock (2001), Zheng, et al. (2002), Chaker, et al. (2002a, 2002b), Bhargava, et al. (2003a, 2003b), Hartel and Pfeiffer (2003), White and Meacock (2003), and Bhargava and Meher-Homji (2002) all studied various aspects of wet compression in gas turbines. Most of this analysis focused on thermodynamic effects and did not evaluate the long-term effects on the gas turbine operation and life, or the effects of water injection in combination with other nonstandard gas turbine conditions.

The study of wet compression technology in gas turbines dates back to the 1940s and 1950s, as is evident from the work on jet engine water injection by Kleinschmidt (1947) and Wilcox and Trout (1951). However, detailed thermodynamic analysis of wet compression in ground-based gas turbines is more recent. For example, Utamura, et al. (1999), studied the impact of droplet size in an industrial gas turbine. Later, Zheng, et al. (2002), and Hartel and Pfeiffer (2003) performed an analysis of the thermodynamic effects of wet compression inside a gas turbine. In addition, interstage water injection methods for gas turbine compressors were discussed by Ingistov (2001, 2002).

Operational Effects of Evaporative Cooling

On standard gas turbine applications, such as natural gas power generation, few operational problems have been reported in the public domain that were directly attributed to inlet evaporative cooling. Ingistov (2002) even documented that water injection has reduced compressor fouling and improved maintenance intervals. However, in a number of cases where excessive inlet water injection was combined with other factors, such as low heating value fuels, combustor steam injection, highly degraded blades, and off-design operating conditions (load, speed, and ambient temperatures), performance issues and failures have been reported (Bagnoli, et al., 2004; Behnken, 1997; Behnken and Muray, 1997; Greitzer, 1976; Hill, 1963; Jolly, 2003; Moore and Greitzer, 1986; Zheng, et al., 1997, 2003). Thus, prior to installing an evaporative cooling system on an existing gas turbine plant or determining the feasibility of water augmentation for a greenfield plant, a proper design review should be performed to prevent any damaging gas turbine operation.

The two most critical effects of gas turbine inlet evaporative are the influence on the aerodynamic stability of the gas turbine’s axial compressor and the increased heat transfer of the air-water vapor-fuel mixture on the gas turbine hot-section vanes and buckets. These two effects are separately treated herein for single shaft gas turbines. Namely, detailed discussion is provided on:

• The onset of aerodynamic compressor instabilities, such as rotating stall and surge that can result from the usage of evaporative cooling power augmentation technologies in gas turbines is analyzed. A simple method is discussed to determine the compressors’ operational safety margin. This method can be employed as a design tool for new installations or as a safety check on existing facilities to ensure that the gas turbines operate with a proper margin from potentially detrimental conditions.

• The increased heat transfer due to the higher water vapor content of the exhaust mix is analyzed. A life fraction curve that can be used to estimate the hot-section parts life reduction is presented. This curve can aid the operator in evaluating increased operating costs and maintenance intervals of gas turbines with water inlet injection.

COMPRESSOR AERODYNAMIC INSTABILITY

A typical performance map for an axial compressor shows pressure ratio as a function of inlet volumetric flow (or mass flow) for a range of compressor speeds; this compressor map indicates that there are limits on the operating range of such a compressor (Figure 1). The limit for low-flow (or high-pressure ratio) operation is set by a flow instability known as surge. The exact location on the compressor map at which surge occurs can range widely depending on operating condition and, as a result, a surge control line is established with a significant margin above the flow at which surge is expected to occur. A typical surge margin is 10 to
15 percent or more of the design flow. While in driven centrifugal compressors, surge is actively avoided using a recycle loop and valve; most axial compressors in a gas turbine have no actively controlled surge devices other than startup compressor discharge bleed.

Axial compressors will surge when forward flow through the compressor can no longer be maintained due to an increase in pressure across the compressor, and a momentary flow reversal occurs. Once surge occurs, the reversal of flow reduces the discharge pressure or increases the suction pressure, thus allowing forward flow to resume until the pressure rise again reaches the surge point. This surge cycle continues at a low frequency until some change is made in the process or compressor conditions.

Thus, surge is a global instability in a compressor’s flow that results in a complete breakdown and reversal of flow through the compressor. Surge occurs just below the minimum flow that the compressor can sustain against the existing suction to discharge pressure rise (head). When surge occurs, both flow rate and head decrease rapidly and air flows backward within the compressor. Full surge is a source of large axial dynamic forces applied to the gas turbine’s elements and, hence, a flow phenomena that must be avoided.

Prior to reaching a surge event, a number of other aerodynamic flow instabilities are likely to occur within an axial compressor. These phenomena include rotating stall and blade flutter. Rotating stall is a localized flow separation (and/or reversal) cell that rotates at about 30 to 70 percent rotor speed in the direction of the compressor blades. As the compressor’s operation condition moves close to the surge line, rotating stall typically initiates in the stator first and then, once fully developed, enters the rotating blades.

Compressor rotating stall is a well-understood instability phenomenon in axial compressor aerodynamics. Rotating stall occurs when aerodynamic stability margins are exceeded, and results in a periodic excitation of blade loading. Emmons, et al. (1955), described rotating stall as a momentary aerodynamic overloading of an already highly loaded blade, which causes this particular blade to exhibit separation and blockage. This blockage restricts the flow through a blade passage, and consequently diverts the incoming streamlines, causing an increase in incidence on one side of the blade passage and reduced incidence on the other. The increased incidence on the adjacent blade row will cause separation and blockage there, and a stall cell will propagate from blade to blade at a speed different from the compressor running speed.

Thus, rotating stall occurs when the flow’s kinetic and pressure (potential) energy cannot overcome the required differential pressure across a blade passage to maintain forward flow and the flow locally “stalls out.” As the surge line is passed, multiple rotating stall cells from both the rotor and stator elements combine to generate a full surge event with fully reversed and oscillating flows throughout the entire compressor. Rotating stall has to be distinguished from violent surge, where the flow pattern through the entire compressor breaks down. In many machines, rotating stall may manifest itself only by elevated vibration levels, but the dynamic flow instability leads to blade forces at frequencies other than multiples of running speed or blade pass frequency. The frequencies of rotating stall forces are typically not identified in a Campbell diagram.

Rotating stall is, thus, a precursor to full surge in which the flow locally and significantly deviates from its design magnitudes and angles. Rather than cleanly following the rotor and stator blades, the flow tends to separate on both the leading and trailing edges of the vanes, and periodic vortex shedding is initiated at a frequency determined by the flow’s Strouhal number. (Strouhal number is a vortex shedding frequency multiplied by a characteristic length divided by the free stream flow velocity.) If this frequency coincides with any natural frequency of the blade, the blade will oscillate in harmony with the vortex shedding and begin to “flutter.” Clearly, flutter imposes significant aerodynamic lateral and torsional forces on the blades at subsynchronous frequencies that can have a very detrimental effect on the life of the blades. For example, Figure 2 shows a gas turbine compressor blade that failed due to flutter-induced high-cycle fatigue. Both rotating stall and flutter are often experienced when the compressor’s operating surge margin is less than five percent. Standard gas turbine instrumentation is usually not adequate to measure the occurrence of either of these precursors to surge.

Analysis Method

To investigate whether a compressor is operating in a region where rotating stall or flutter may occur, an aerodynamic surge analysis of the compressor must be performed. A basic axial compressor surge analysis consists of a one-dimensional (1-D) thermodynamic and two-dimensional (2-D) mean streamline flow analysis for each stage of the multistage compressor. The 2-D mean streamline analysis yields power absorbed by each stage, total compressor power, and blade-to-blade flow angles. Using the results from the 2-D analysis, the 1-D thermodynamic analysis results for a particular stage’s operating point can be mapped on the stage’s characteristic performance curve to determine the local surge margin. Surge margin (SM) for axial compressors is typically defined as:

\[
SM = \frac{Q_{\text{design}} - Q_{\text{surge}}}{Q_{\text{design}}} \tag{1}
\]

where \(Q\) is the volumetric flow rate.

For evaporative cooling, the model assumes that all water instantly vaporizes unless the flow is already saturated, in which case the model assumes that the water is carried by the flow axially through the compressor until the flow can absorb higher moisture content. Evaporative cooling primarily affects the gas turbine’s inlet air temperature. Other thermodynamic parameters, such as flow density and specific heat of the air, are also affected by water injection, but the dominant effect is cooling. Nonetheless, although these physical properties play a minimal role in the overall...
thermodynamic performance of the gas turbine, they do affect the aerodynamic stability of the compressor and must be included in a surge analysis. Thus, the model locally recalculates the mixture’s physical properties based on the moisture content in the air.

Characteristic compressor stage performance curves are typically not available from the gas turbine manufacturer for a particular gas turbine model, so generic single-stage axial compressor curves can be adopted for the application by matching global performance parameters (power, efficiency, and mass flow) with given geometry and operating condition (speed and pressure ratio) information. Namely, using the general thermodynamic equation for stage efficiency:

$$\eta_b = \frac{T_d - T_i}{T_i \left(\frac{P_d}{P_i}\right)^\frac{1}{\gamma} - 1} \quad (2)$$

The compressor’s stage power requirements can be determined from:

$$W = \Delta h \cdot \frac{dm}{dt} = \frac{dm}{dt} \left(\frac{T_d - T_i}{C_P}\right) \quad (3)$$

and

$$W = \Delta \tau \cdot \omega = \omega \frac{dm}{dt} \left(V_d^0 - V_i^0\right) \quad (4)$$

Unless specific information on the compressor stages is available, a 15 percent original design surge margin is a reasonable and conservative assumption for all stages. Also, mean streamline individual stage compressor blade design angles can be estimated using Euler’s turbomachinery equation and flow vector analysis:

$$\frac{P_d}{P_i} = \left(1 + \frac{\varphi \omega}{C_P T_i} (c_2 \sin \alpha_2 - c_1 \sin \alpha_1)^{\frac{1}{\gamma}}\right)^{-1} \quad (5)$$

The same equation is subsequently used to determine actual blade incidence angles in the surge/flutter analysis. As the aim of the method is to determine changes in surge margin, rather than the total surge margin for a given operating condition, small inaccuracies of the characteristic compressor stage maps do not significantly affect the individual decreases in surge margin calculation.

**Aerodynamic Stability Criteria**

The results from the characteristic stage curve operating point mapping should assure that the lowest stage surge margin of any stage is always above five percent to conservatively avoid operating the compressor in a region of local aerodynamic instability. In addition, from the 2-D analysis, the highest local stator and rotor blade incidence angles should not exceed 8 and 12 degrees, respectively; high blade incidence angles are an indicator for the potential of rotating stall development.

To avoid high-flow compressor stall, modulated inlet/stator guide vanes are employed on most gas turbines to reduce (throttle) the gas turbine’s airflow at low load and startup/shutdown conditions (refer to Figure 1). The gas turbine’s axial compressor’s surge margin is decreased when operating at these conditions due to high incidence flow on the modulated stage blades. Hence, as compressor aerodynamic instabilities are more likely to occur during off-design operation, the above stability criteria must be met at all gas turbine operating points, including startup and shutdown.

Factors that can have a significant effect on surge margin in a gas turbine and that should be parametrically evaluated in the surge analysis are:

- Inlet water injection (evaporative cooling).
- Interstage compressor water injection.
- Combustor diluent (steam, water, nitrogen, etc.) injection.
- Compressor blade degradation and fouling.
- Low or medium equivalent heating value fuels.

As each of these factors can individually affect a gas turbine compressor’s surge margin, their total impact must be determined for a given gas turbine operating condition. One should note that effects on surge margin are net additive, i.e., if, for example, five percent of surge margin is lost due to degradation and six percent of surge margin is lost due to fogging, the total surge margin decreases by 11 percent. Namely:

$$\Delta SM = \sum_{i=1}^{n} \frac{\partial M}{\partial \Phi_i} \cdot \Delta \Phi_i \quad (6)$$

where $\Phi$ is the individually assessed factor influencing the stage surge margin. By parametrically mapping the individual factor effects on surge margin, a global operation limit can be established.

**INSTABILITY FACTOR INFLUENCE ANALYSIS**

Although the above described surge analysis is based on the sequential application of 1-D thermodynamic and 2-D mean streamline equations that are readily available in the public domain (refer to Zheng, et al., 2002, for a comprehensive review), few end users or operators perform this safety check prior to installing water inlet and stage injection on their gas turbine. This has led to some gas turbine operational problems and even blade failures. Thus, each of the principal factors that can affect the surge margin (evaporative cooling, interstage injection, combustor injection, blade fouling, and low heating value fuels) is individually discussed below. A parametric study of the relative influence of each factor on the axial compressor surge margin is provided for a typical large gas turbine power generation application. Although the analysis results presented herein are based on single shaft gas turbines, a similar methodology can be developed and applied to two-shaft gas turbines.

**Evaporative Cooling**

Evaporative cooling of the inlet air into a gas turbine has become a popular method to increase a gas turbine’s power. This technology was originally designed to only provide a power boost during peak demand times, but some plants now use fogging even for base load power. Long-term impacts on the gas turbine due to evaporative cooling are not well understood and documented at this time. The primary effect of evaporative cooling is that it reduces the inlet air’s temperature to its wet bulb temperature and changes its specific heat constants. Typically, a power increase between five to 10 percent can be achieved on hot, dry days using evaporative cooling.

To demonstrate the reduction of surge margin due to evaporative cooling, example results of a surge analysis (as described above) are included herein for a large (>100 MW) industrial frame type gas turbine. For this case, Figure 3 shows analysis results for surge margin versus percent inlet fogging. Clearly, inlet evaporative cooling moves the compressor operating point closer to the surge line. At 100 percent saturation fogging, the surge margin was decreased by approximately four percent—if the ambient relative humidity is about 20 to 30 percent. Thus, evaporative cooling techniques, such as inlet fogging alone, will not decrease the surge margin sufficiently to cause aerodynamic instabilities. However, they can be a contributing factor if, for other reasons, the machine’s surge margin is already slim.

The impact of fogging on the gas turbine’s behavior, and in particular the surge margin of the compressor, cannot be understood without a simulation of the stage-by-stage behavior of the compressor and the interaction of the compressor with the remaining engine. Changing the operating point of one compressor stage, for
example, by introducing a different working fluid (such as changing the content of water vapor in the air) will have a cascading effect on the operating point of the subsequent stages (Kurz and Brun, 2000).

Evaporating water will reduce the inlet temperature into the engine compressor. This will increase the machine Mach number of a single shaft, constant speed engine. On the other hand, water vapor is lighter than air and has different thermodynamic properties. The change in properties will lead to a mismatch between the compressor stages, which, in turn, means that for the same operating conditions, the overall surge margin of the compressor is reduced.

The relationship between flow and pressure ratio (because the engine at full load always operates at maximum firing temperature) is determined by the flow capacity of the turbine. If the compressor operates at unchanged mechanical speed, adding water, without changing the inlet temperature, would move the operating point to a lower flow, and a lower pressure ratio, thus lowering the surge margin considerably. However, the evaporation of water lowers the inlet temperature, which counteracts this effect to some extent. The flow is still lower, but the pressure ratio is increased. This, again, will reduce the compressor surge margin.

Increasing the water content of the air increases the speed of sound under otherwise identical conditions, which means that the Mach number becomes lower. The reduction of temperature has the opposite effect and would increase the Mach number. For evaporative cooling, the effects would tend to partially compensate each other.

Sample calculations for a gas turbine at full load, with the model described above, show that with the amount of water to lower the inlet temperature from 104°F (40°C) to 80°F (26°C), the surge margin is reduced by 1.8 percent, the engine pressure ratio is increased by 4.5 percent, and the engine mass flow is increased by 4.5 percent. The engine power will be increased by 9.5 percent.

**Interstage Water Injection**

Beyond simple inlet fogging, some users have recently tested injecting water directly into the axial compressor through ports in the first stages of the gas turbine’s compressor. Injecting water downstream from the inlet of a gas turbine has a similar effect as over spraying; it locally reduces the airflow temperature to that of the air’s wet bulb temperature. This increases gas turbine output power but also decreases the compressor’s surge margin. Figure 4 shows a parametric study that evaluates the effect of stage water injection on the frame type example gas turbine. The cases that are analyzed are:

- Compressor Stage 1 water injection (spray).
- Compressor Stage 2 water injection (spray).
- Both Stages 1 and 2 water injection (spray).

Compressor interstage water injection is seen to have a very strong effect on a compressor’s surge margin. Also, study results showed that the further downstream in the compressor water is injected, the larger the loss in compressor surge margin. This is primarily due to the fact that a stage’s aerodynamics sensitivity to physical mixture property increases at higher pressures and temperatures. Specifically, a temperature reduction from dry bulb to wet bulb temperature in a downstream stage has a more significant effect on the blade flow incidence angles (and possible flow separation) than in an upstream stage. Similarly, Figure 5 shows a parametric study of 100 percent saturation inlet evaporative cooling in combination with interstage water injection. Here, saturation inlet evaporative cooling was assumed for all cases.

Results in Figure 5 demonstrate that evaporative cooling combined with interstage water injection can move a gas turbine’s compressor into aerodynamic instability. Again, the further downstream the injection occurs, the more detrimental the effect on surge margin. The net effect of evaporative cooling and interstage injection is clearly additive.

As previously mentioned, if the compressor’s remaining surge margin is less than five percent, the machine may well be in an operating region of localized rotating stall that can subject the compressor to blade flutter. Flutter has a very strong potential to cause blade damage.

**Equivalent Lower Heating Value**

When operating a gas turbine with a fuel that has a heating value lower than that of its original design fuel, compressor aerodynamic stability can be affected. Most gas turbine fuel systems are designed for natural gas fuel, which has a heating value of approximately $5 \times 10^7$ Btu/kg, and can handle a Wobbe Index variation of $\pm 10$ percent. Steam or water injection into the combustor to reduce NOx emissions have a similar effect as using lower heating value fuels, since it decreases the equivalent heating...
value of the fuel mixture in the combustor. If the fuel mixture’s heating value is significantly lower than that of its design fuel, more fuel must be injected into the combustor to achieve the required equivalent gas turbine heat input and firing temperature. However, the increased volumetric input into the combustor also increases the compressor backpressure (when the gas turbine nozzle is flow choked) and, thus, decreases the compressor’s surge margin. Therefore, care must be taken when operating a gas turbine with very low heating value fuels, such as synthesis gas, blast furnace gas, or refinery off-gas. Figure 6 shows parametric study results for axial compressor surge margin versus fuel equivalent lower heating value. Equivalent heating value is defined as the resulting net heating value of a fuel/steam/water mixture in the combustor. Results from this analysis show that for a large industrial turbine, the compressor’s surge margin can be reduced by about half when burning extremely low heating value fuels. Clearly, other physical effects related to burning low heating value fuels could cause aerodynamic instabilities in the combustor; however, a full discussion on combustion stability is beyond the scope of this paper.

Compressor Blade Degradation

While all effects discussed above cause the gas turbine compressor operating point to move closer to its surge line, compressor blade degradation due to fouling, erosion, corrosion, and particle fusing, actually moves the surge line toward the operating point. Nonetheless, the net effect is the same in that the gas turbine operates closer to a range of possible aerodynamic instability.

Compressor degradation is principally caused by loss of material on the leading and trailing edges of the blades (corrosion, erosion, and “sandblasting”) and surface fouling (build-up of material and roughening) on the concave and convex sides of the airfoils. Particle fusing and pitting generally does not affect an airfoils’ aerodynamic performance and is more related to crack formation and other material life limiting issues. Degradation affects a blade’s ability to guide the flow. Namely, if the airfoil chord is shortened, the blade profile changed, or the surfaces roughened, the airflow will not follow the blades the same way it would on a new blade; the flow may slip and even separate (refer to Kurz and Brun, 2000). Slip is an indicator of how effectively a rotating blade imparts tangential kinetic energy onto the airflow. Slip is, thus, a direct function of blade degradation. Figure 7 shows a severely degraded compressor blade.

Compressor flow analysis results for surge margin versus degradation are shown in Figure 8 for an industrial gas turbine. Here, fouling and corrosion are presented as an equivalent chord loss parameter (ECLP = equivalent chord loss/total chord length). The ECLP is often employed to account for all types of degradation in one convenient parameter. ECLP assumes that aerodynamic degradation due to fouling can be equated to an equivalent decrease of blade chord. Typical values for this parameter range from zero percent to 0.3 percent in frame type gas turbines, and zero percent to one percent in aero-derivative gas turbines. Surge margin results in Figure 8 show that even if blades are significantly degraded, a normal industrial gas turbine axial compressor would have had adequate surge margin to be operating well away from rotating stall and flutter. However, if inlet evaporative cooling is installed on a gas turbine with a severely degraded compressor, the gas turbine could be operating in an area of compressor aerodynamic instability, such as rotating stall and flutter.
assessing the overall life reduction was included herein. The combustor) will reduce the hot-section turbine parts life; a curve any water injection into a gas turbine (inlet, interstage, or fatigue and possible catastrophic gas turbine failure. Furthermore, aerodynamic instabilities can be directly linked to blade high-cycle problems, such as rotating stall and flutter, will likely occur. These degraded blades, gas turbine compressor aerodynamic stability as low heating value fuels, combustor steam injection, and highly and interstage water injection is combined with other factors, such turbine, and results were presented. Results showed that when inlet simplified method to evaluate the principal factors that affect the national and life-limiting problems. This paper discussed a stability review should be performed to prevent any potential oper- evaporative cooling systems, a proper gas turbine aerodynamic discharge and, as this air also contains a higher water vapor content, it provides increased heat carrying capacity of the cooling flow. Nonetheless, a detailed analysis shows that this offset does not fully overcome the increased heat transfer into the blades, because of the high ratio of forced external versus mostly convective/conductive internal blade heat transfer. Furthermore, for cases of steam/water injection directly into the combustor, the effect is exasperated, as the external flow will clearly have a higher water vapor content and increased mass flow. An analysis of the effect of inlet evaporative cooling on a large industrial gas turbine shows that the design life of blades and vanes will be severely reduced (Figure 9). Analysis results in Figure 9 are based on a total heat transfer to blade life model. For example, if saturation evaporative cooling is employed in the industrial gas turbine, the water volume fraction in the exhaust can be as high as 15 percent. In this case, the hot gas path parts life would be reduced to about 88 percent of its original design life. Clearly, this is a significant increase of the gas turbine’s maintenance costs and a reduced availability.

Figure 9. Hot-Section Parts Life Fraction.

CONCLUSION

Only limited historical field operating data and information on the long-term effect of gas turbine inlet evaporative cooling is available. However, over the past 15 years, more than 800 gas turbine installations worldwide have been built or retrofitted with these technologies. The basic functional principle of all evaporative cooling technologies is that they effectively reduce the gas turbine’s inlet air temperature from the air’s dry bulb temperature to the wet bulb temperature of the ambient air. Prior to utilizing evaporative cooling systems, a proper gas turbine aerodynamic stability review should be performed to prevent any potential operational and life-limiting problems. This paper discussed a simplified method to evaluate the principal factors that affect the aerodynamic stability of a single shaft gas turbine’s axial compressor. As an example, the method was applied to a frame type gas turbine, and results were presented. Results showed that when inlet and interstage water injection is combined with other factors, such as low heating value fuels, combustor steam injection, and highly degraded blades, gas turbine compressor aerodynamic stability problems, such as rotating stall and flutter, will likely occur. These aerodynamic instabilities can be directly linked to blade high-cycle fatigue and possible catastrophic gas turbine failure. Furthermore, any water injection into a gas turbine (inlet, interstage, or combustor) will reduce the hot-section turbine parts life; a curve assessing the overall life reduction was included herein. The method described in this paper can be employed by end users and operators to identify high-risk applications, and bound the gas turbine operating range to limit the potential risk of blade life reducing aerodynamic instability and subsequent catastrophic failure. Because the problems described are often due to the coincidence of several factors, proper and timely engine maintenance can avoid many of the severe problems described herein.

NOMENCLATURE

\[ c_1, c_2 = \text{Local velocities} \]
\[ c_p = \text{Specific heat} \]
\[ \frac{dm}{dt} = \text{Mass flow} \]
\[ h = \text{Enthalpy} \]
\[ k = \text{Specific heat ratio} \]
\[ P = \text{Pressure} \]
\[ Q = \text{Volumetric flow} \]
\[ r = \text{Radius} \]
\[ SM = \text{Surge margin} \]
\[ T = \text{Temperature} \]
\[ V = \text{Velocity} \]
\[ W = \text{Power} \]
\[ \omega = \text{Angular speed} \]
\[ \alpha = \text{Flow incidence angles} \]
\[ \eta = \text{Stage efficiency} \]
\[ \tau = \text{Torque} \]
\[ \theta = \text{Tangential direction} \]

Subscripts

\[ d = \text{Discharge} \]
\[ s = \text{Suction} \]

REFERENCES


