



# INNOVATIVE GAS TURBINE PERFORMANCE DIAGNOSTICS AND HOT PARTS LIFE ASSESSMENT TECHNIQUES

by

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## ABSTRACT

Traditional service and maintenance contracts are evolving to new forms of relationship between providers and operators requiring deep changes in service offerings, with strong emphasis on high-tech contents. In gas turbine applications, remote monitoring associated with advanced condition and performance diagnostics and hot parts remnant life assessment are requirements being increasingly set by operators as a must for global, long-term service contracts. The work described herein is aimed at providing effective answers to these requirements through development of advanced tools for accurate gas-path analysis with custom tuning for each individual machine based on monitored data. Generated gas-path information is used to diagnose internal component degradation and retune the control system accordingly, to reduce deterioration effects on performance and consequent needs for preventive or unscheduled maintenance. Hot part metal temperatures and stresses are continuously evaluated from gas-path information and used to both assess part damage evolution related to actual operation and to predict remnant life as a function of expected operating requirements.

## INTRODUCTION

Turbomachine plants and particularly those utilizing gas turbines as prime drivers are undergoing deep changes in maintenance and servicing strategies. These are more and more driven by user requirements to progress from traditional maintenance based on scheduled parts replacement and refurbishment toward “global service” strategies where providers are directly involved in keeping equipment condition and performance at best levels with outages reduced to a minimum. This new vision is leading to nontraditional

types of contracts denominated long-term service agreements (LTSA), established between plant owners and service providers to share responsibilities in maximizing plant operability, reliability, and profitability. In this new scenario, remote monitoring is an essential tool for service providers to constantly keep the equipment under surveillance. However, monitoring alone would not be sufficient to provide the kind of service expected in the LTSA without prompt diagnosis of equipment conditions and of problem sources, immediately followed by guidelines to continuously optimize performance and reliability consistent with diagnosed states and actual operator requirements.

To respond to requisites of the new service contracts, Nuovo Pignone, a part of the recently established GE Power Systems Oil & Gas Division, three years ago started development of a system to provide comprehensive remote monitoring services for gas turbines, with the main focus on mechanical drive applications. Integration of monitoring with high-level diagnostic support was considered a must to provide complete and high added value packages incorporating top knowledge and technology available from the company’s gas turbine and compressor development and design background. This approach led to setting up an integrated diagnostic system that utilizes engine data measured with standard plant instrumentation and continuously transmitted to the diagnostic center. Monitored data are used to perform detailed thermal cycle analyses, determine component efficiencies consistent with field measurements, and generate internal gas-path information normally not available during commercial operation from standard machine instrumentation. Turbine nozzle and rotating blade metal temperatures and stresses are then evaluated from hot gas-path data by means of special functions built by using complex aerodynamic and structural analysis tools normally employed for product development and design. Finally, metal stresses and temperatures evaluated from monitored data in real time are used to assess damage progression on critical hot parts and related contribution to attainable residual life. As a result, life history of each critical component can be assessed as a function of actual operating conditions and without need for metallurgical analyses requiring destructive tests. The methodology can also be used proactively, by specifying scheduled operating conditions to predict related damage progress and finally assess achievable residual life of critical parts. Effects of different possible operating criteria on predicted component remnant life can also be assessed with this methodology and the results used to select plant management strategies most suitable to meet life extension targets.

The system is presently fully operative for cycle analysis and performance optimization, while life assessment modules are under advanced development and will start pilot application on a selected number of currently monitored gas turbine units by end of the present year.

## GAS TURBINE CYCLE ANALYSIS AND THERMODYNAMIC DIAGNOSTICS

One of the main problems in performing gas turbine thermodynamic and performance diagnostics is the relatively scarce amount of data available from commercial instrumentation

installed on machines in industrial operation. Other issues are typically relatively low sensor precision, failure during operation, and tendency to lose calibration in time.

In developing a new diagnostic system, a basic assumption was to avoid needs for substantial implementation of new instrumentation, particularly of sophisticated types or those not strictly necessary for normal operation and control requirements. A major goal was therefore to replace missing measurements with advanced cycle analysis tools suitable to assess individual component performance and gas-path parameters with precision suitable for diagnostic purposes.

The scheme in Figure 1 shows the flow diagram of a two-shaft gas turbine with indication of locations along the gas-path where normally standard instrumentation is provided (framed symbols indicate parameters for which measurements are usually available). The most important missing data are, usually, combustion chamber exit and turbine inlet temperatures and conditions between high-pressure and low-pressure turbines; also compressor inlet air mass flow measurement is not usually available. On older installations being retrofitted with a new diagnostic system, ambient pressure and humidity measurements are often not available, together with inlet and exhaust pressure drops. Fuel heating value, particularly for gaseous fuels, is updated periodically through direct analysis, or is derived from specifications provided by the supplier mainly for billing purposes.

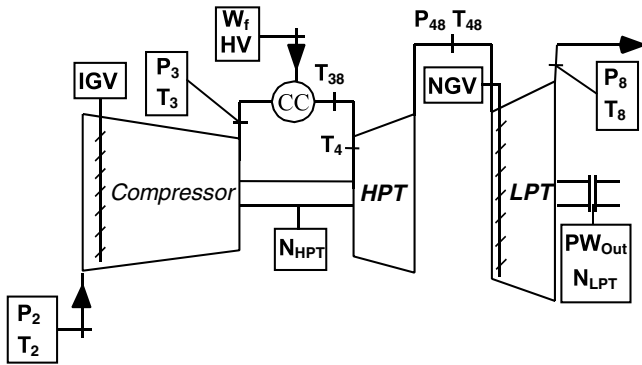


Figure 1. Standard Flow-Path Measurements on a Two-Shaft Gas Turbine.

To evaluate thermodynamic and gas-path data shown in Figure 1, a cycle model is routinely used, consisting of a set of equations correlating boundary conditions (e.g., inlet and exhaust conditions, fuel characteristics, geometry, etc.), component efficiencies, internal cooling, and leakage flows to satisfy energy and mass balances. If evaluation has to be made for operating conditions different from design, the component characteristics are specified as maps of parameters instead of single values; the resultant cycle analysis tool is a computer program commonly designated as “cycle-deck.” Figure 2 schematically shows the main blocks of the cycle program and correlation among boundary conditions, operating data, and component characteristics. A standard cycle-deck version is typically utilized to evaluate performance of new, clean engines and is calibrated in order to model average characteristics of production machines when the manufacturing process is steady and repetitive.

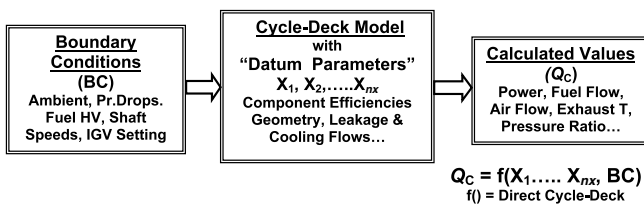


Figure 2. Direct Cycle Model Program (“Cycle-Deck”) Scheme.

When dealing with diagnostics, the problem needs to be posed in completely reversed terms, since performance is measured but internal component states consistent with actual performance are not known, particularly when the machine is deteriorated and significantly far from new and clean conditions. Ideally, it would be necessary to evaluate component characteristics for each monitored condition with a process exactly opposed to that used in the traditional cycle analysis, i.e., component efficiencies and internal gas data should be evaluated as a function of the measured performance and boundary conditions. Schematically, the so-called “inverse cycle” analysis process is outlined in Figure 3.

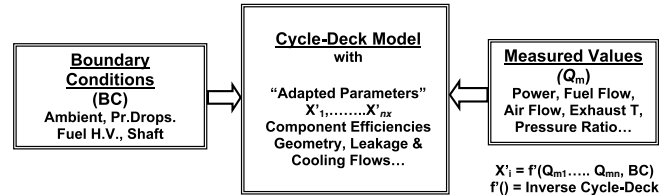


Figure 3. “Inverse” or “Adaptive” Cycle-Deck Model Scheme.

For several years the author and others (Benvenuti, et al., 1993 and 1994; Bettocchi, et al., 2000) have been working to develop an “inverse cycle” methodology simple and reliable enough to acceptably work with unavoidable monitored data inconsistencies and discrepancies, but suitable enough to provide acceptable precision for diagnostic purposes.

The “inverse cycle” method is based on the principle that all thermodynamic and performance parameters computed by the cycle-deck are a function of two sets of data used for cycle calculations:

- “Fixed parameters”—represented by performance and geometric data specific to gas turbine components: compressor and turbine performance maps (curves of flow and flow coefficients, pressures and enthalpies, speeds, variable inlet guide vanes, and nozzle angles), leakage and cooling flow discharge coefficients, combustion chamber pressure drop and efficiency, turbine nozzle throat areas
- “Variable parameters”—defining the machine working configuration. They are, respectively, boundary conditions (ambient pressure and temperature, relative humidity, pressure drops at compressor inlet and turbine exhaust, fuel composition, and heating value) and operating parameters (for a two-shaft turbine corresponding to Figure 1, these are high-pressure (HP) and low-pressure (LP) shaft speeds, compressor inlet guide vanes, and LP turbine variable nozzle settings).

Therefore, calculated estimates “ $Q_c$ ” of measurable parameters provided in the cycle-deck output are also a function of the machine characteristic parameters “ $X$ ” (fixed parameters) and gas turbine working configurations (variable parameters):

$$Q_c = f(X_1, \dots, X_{nx}, \text{Variable Parameters}) \quad (1)$$

where “ $f$ ” is a complex “function” represented by the gas turbine real cycle model linking fixed and variable data with gas-path and engine performance parameters. This function cannot be expressed explicitly as a single ordinary mathematical function, but is an assembly of equations that perform specific tasks inside the thermal cycle model, the principal being:

- Energy equations linking pressures, temperatures, enthalpies, and efficiencies at stations along the gas-path. Efficiencies are obtained by interpolation on the built-in compressor and turbine maps.
- Functions evaluating internal secondary, leakage and cooling mass flows, and mixing with the main stream to determine total mass flow and after-mixing conditions at each station

- Combustion chamber energy balance linking fuel and air flows, heating value, compressor delivery temperature, injected steam flow, and enthalpy (when present) with turbine inlet temperature
- Flow discharge through the turbine first stage nozzle throat, linking turbine inlet pressure, temperature, and mass flow with throat passage area according to the well-known relationship:

$$W = C_{D,N} \frac{P_3 - \Delta P_{Comb}}{\sqrt{TIT}} A_{TH} \quad (2)$$

$C_{D,N}$  is the nozzle discharge coefficient provided in the turbine maps as a function of pressure ratio and speed, and  $A_{TH}$  is the first stage nozzle geometrical throat area. This equation is of great importance in the inverse cycle algorithm to determine air mass flow when measurement is not available, as is usually the case, from measured compressor delivery pressure and turbine inlet temperature determined in the combustion chamber energy balance.

Since the relationships corresponding to the component maps are nonlinear, determination of  $Q_c$  through the cycle model "function" is typically iterative.

When designing a new gas turbine, the characteristic parameters " $X_i$ " are set as inputs and determined from calculations and previous experience on similar engines. In diagnostics, with deteriorated engines differing from design, these parameters are not known *a priori*, but must be determined from measured performance.

The method for calculating the characteristic parameters " $X_i$ " uses a special version of the gas turbine cycle-deck where the original parameters describing the new and clean engine are "adapted" by an iterative process until measured performance is matched with sufficient accuracy. The mathematical process consists of simultaneous solution of an equation system, obtained by setting the residuals between computed and measured values to zero:

$$\left\{ \begin{array}{l} (e_{Q_i})_1 = \frac{[Q_c(X_1, \dots, X_{nx}, \text{Variable Parameters})]_1 - (Q_m)_1}{(Q_m)_1} = 0 \\ \dots \\ (e_{Q_{ne}})_{ne} = \frac{[Q_c(X_1, \dots, X_{nx}, \text{Variable Parameters})]_{ne} - (Q_m)_{ne}}{(Q_m)_{ne}} = 0 \end{array} \right. \quad (3)$$

This system of equations is nonlinear due to the above mentioned nonlinear cycle model "function" structure. The solution is therefore iterative, and is achieved by using a constrained minimization algorithm that repeatedly calls the cycle-deck routines and modifies the characteristic parameters " $X_i$ " in order to minimize the "objective" function:

$$F_{ob}(X_1, \dots, X_{nx}) = \sum_{i=1}^{ne} [(e_{Q_i})_i]^2 \quad (4)$$

where residuals  $e_Q$  are subjected to the constraint Equation (4). A simplified flow diagram of the complete inverse cycle algorithm is shown in Figure 4.

The method was initially checked and calibrated by using data from rig tests of extensively instrumented development engines, with availability of measurements of most of the gas-path data to be evaluated through the inverse cycle algorithm. These checks made it possible to identify and eliminate one major issue of the algorithm when used with test data subjected to some degree of error and uncertainty, i.e., establish ranges in which the " $X_i$ " parameters are allowed to vary during the iterative solution. In fact,

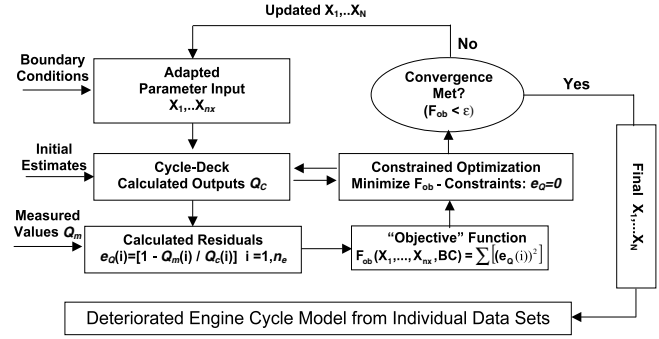


Figure 4. Inverse Cycle-Deck Simplified Conceptual Block Diagram.

too narrow ranges may make iterations not converge, while too large bands tend to divert solution toward unrealistic results. The issue was resolved thanks to the mentioned availability of good and detailed flowpath measurements from rig tests, which led to establishing rules to assign suitable ranges to parameter variability as needed for actual use in routine diagnostic analyses.

Application of the system in pilot form to a limited number of remotely monitored 5 and 10 MW gas turbines in mechanical and generator drive applications has shown sufficient robustness to run without major convergence or instability problems, even with relatively poor monitored data quality. A distinctive merit of this method when applied to existing engines retrofitted with a remote monitoring system was that of identifying major instrumentation calibration and control system tuning defects that could be quickly corrected with immediate benefits on performance and reliability.

However, even if the algorithm is in general capable of providing a solution from virtually any set of data, irregularities in the input stream may result in substantial scatter in evaluated engine parameters, thus making it difficult to clearly identify trends in the short-medium term. An important issue concerning results provided by the algorithm is steadiness of supplied data. Although the monitoring system provides flags to identify major operating mode changes (acceleration, deceleration, load change, startup, shutdown, etc.), smaller fluctuations that are not automatically identified as state changes can produce substantial scatters in the analysis results.

Efforts were therefore made to identify a "steady-state index" (SSI) suitable to quickly identify "windows" in data streams with true steady-state operation characteristics. A first approach was to select relatively narrow data windows (e.g., encompassing one-minute readings at one-second frequency). In each window, normalized standard deviations (NSTDD) of five key cycle parameters: compressor variable inlet guide vane (IGV) position, compressor discharge pressure, low pressure turbine variable nozzle angle, exhaust temperature and shaft power were evaluated. The SSI was then defined as the arithmetic average of individual normalized standard deviations over each window. Figures 5 and 6 show two data windows extracted from same day readings at one-hour intervals. The SSI in the two windows is 4.08 percent and 0.25 percent, respectively, while the arithmetic parameter average values are nearly identical. Figures 7 and 8 show results obtained by running the inverse cycle analysis on each of the 25 measured data sets in the two windows. Plotted output parameters are compressor flow and efficiency, high- and low-pressure turbine efficiencies, and overall turbine expansion efficiency. For comparison purposes, values of each parameter have been referred to the respective simple arithmetic averages. The difference in the quality and scatter of the calculated parameters is quite evident and dramatic. Finally, Figure 9 shows the effect of the SSI index on the normalized standard deviations of the same parameters for a set of data windows with SSI ranging from 0.25 percent to 4 percent. The diagram shows that, if effects of imperfect steady-

state conditions on calculated cycle parameters have to be kept within 0.1 percent, SSI should not exceed 0.5 percent. This is a criterion set to achieve a certain degree of automation in the monitored data filtering before submission to the inverse cycle analysis, so that only data windows complying with the 0.1 percent maximum SSI are selected. Although this is not yet considered as the ultimate approach to assure submittal of consistent and reliable data to analysis, practical application to currently monitored units shows remarkable consistency in assessing trends of critical parameters, thus helping to provide reliable recommendations to operators.

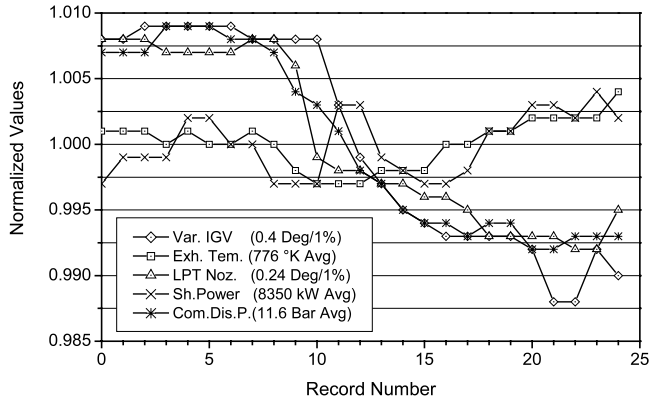


Figure 5. Sample of Monitored Data Window with High Steady-State Index.

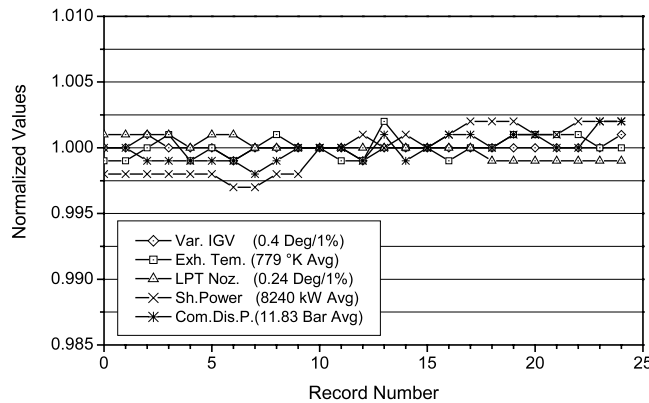


Figure 6. Sample of Monitored Data Window with Very Low Steady-State Index.

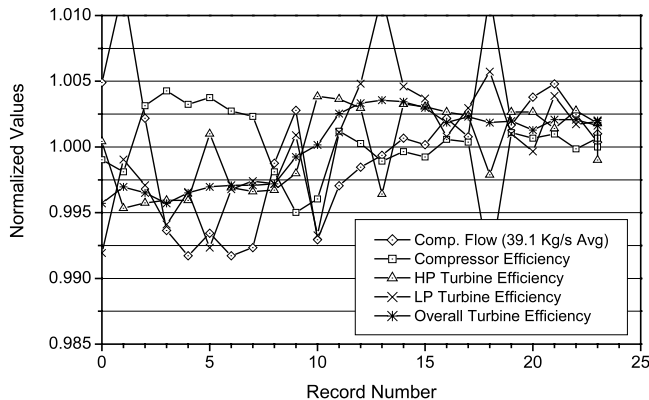


Figure 7. Inverse Cycle-Deck Output Scatter over a High Steady-State Index Data Window.

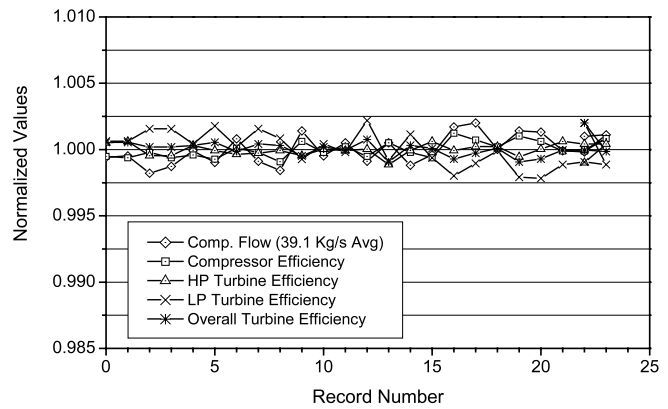


Figure 8. Inverse Cycle-Deck Output Scatter over a Low Steady-State Index Data Window.

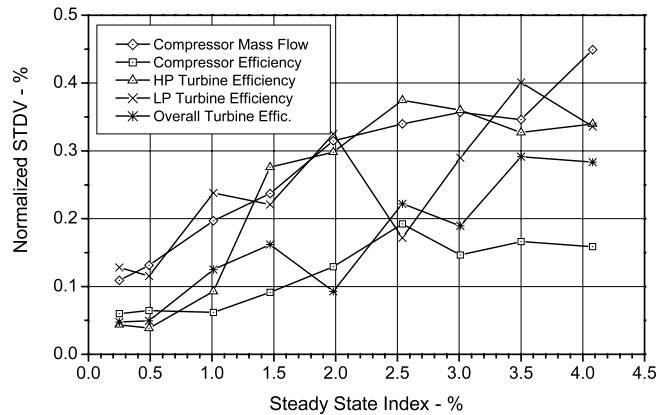


Figure 9. Steady-State Index Effects on Inverse Cycle-Deck Output Scatter.

One major advantage of the inverse cycle method approach is that, for any set of monitored data, cycle parameters calibrated on the corresponding engine condition are available. This feature leads to availability of a continuously updated direct cycle model. It can be used for a variety of purposes, like accurately evaluating performance of deteriorated engines at different operating conditions, and updating control laws to compensate effects of component mismatches produced by degradation. The second and third capabilities are particularly important in providing advanced diagnostic services oriented to optimize maintenance strategies and maximize actual performance, according to LTSA requirements mentioned earlier.

In terms of maintenance strategies, capability of accurately forecasting engine performance in the near-mid term through dynamic calibration of the cycle model allows operators to reliably evaluate costs of reduced outputs and compare them with benefits from reducing performance driven maintenance. A typical example is a request for peak power on hot summer days. By accurately predicting achievable output from data monitored in advance during the colder season, operators have a tool to assess if cost of preventive engine outage for servicing and refurbishing compares favorably with cost of lost production during hot months.

An important effect of engine deterioration corresponding to decrease in turbine expansion efficiency during life is exhaust temperature increase for a given turbine inlet temperature and pressure ratio. Typically, a gas turbine is controlled at base load by setting the control exhaust temperature curve as a function of compressor discharge pressure (CDP) to control TIT via pressure ratio and expansion efficiency. The corresponding CDP- $T_X$  curve

is set according to cycle-deck calculations for an average new engine. This relationship is valid until turbine expansion efficiency remains nearly unchanged. If, as is often the case, turbine efficiency decreases in time, exhaust temperature increases at constant pressure ratio and TIT. If no action is taken, the control system reduces TIT in accordance with the specified CDP- $T_X$  control curve. This process is schematically shown in Figure 10. T-s cycle diagram: to match actual exhaust temperature with the control curve when expansion efficiency decrease requires that TIT is reduced from the new engine  $T_4$  to  $T_{4d}$  deteriorated state value. As a result, a double output power loss occurs, one due to turbine efficiency degradation, that cannot of course be recovered, and the second due to control system mismatch leading to TIT decrease. This latter can be fully recovered through appropriate retuning of the CDP- $T_X$  control curve with support of the recalibrated cycle-deck model taking the true turbine efficiency into account. This capability is now being used to regularly check and retune monitored engine control curves, and in several cases led to performance recoveries that helped customers to delay maintenance shutdowns with substantial cost savings.

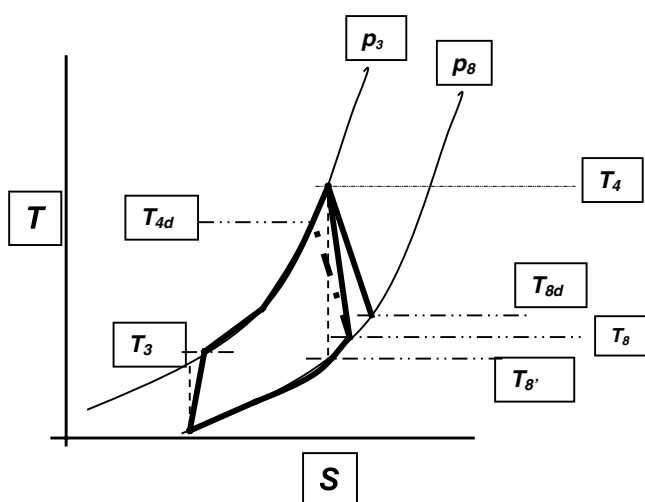


Figure 10. Effect of Turbine Efficiency Degradation on Turbine Inlet Temperature.

A case history with application of this methodology to a 10.7 MW gas turbine numerically shows sensitivity in detecting internal anomalies and effectiveness in changing the control law to partially offset deterioration effects. The machine considered is one of the oldest in the fleet, and has cumulated over 80,000 base-load operating hours in paper mill cogeneration service. The machine is equipped with fuel nozzle steam injection for NOX emission control, with a nominal steam/fuel ratio of 1/1. One and a half years ago, an appreciable decay in performance was detected, together with lower than normal pressure ratio, that could not be explained by airflow decrease associated to compressor efficiency deterioration nor by incorrect variable inlet guide vane settings. The pressure ratio decrease and corresponding exhaust temperature increase, associated with its limitation to 520°C (968°F) for heat recovery boiler requirements, produced remarkable output decrease at ambient temperatures just over 20°C (68°F), consistent with the compressor CDP- $T_X$  standard control curve built in the control system.

A number of conventional analyses made by using the design cycle model were unable to match measured data satisfactorily. The inverse cycle algorithm, undergoing pilot testing at that time, was therefore applied to measurements taken at a previous time on a moderately hot day requiring exhaust temperature limitation. First trials showed that to match the measured low

compressor delivery pressure, a substantial increase in turbine first stage nozzle throat area had to be assumed. This hypothesis was excluded since a previous borescope inspection had shown no major damage in the nozzle throat and airfoil trailing edge areas. In addition, this scheme did not allow for satisfactory simultaneous matching of measured output power and exhaust temperature, unless the range of turbine expansion efficiency variability was unrealistically increased. The next hypothesis was then to simulate a flow bypass over the first stage nozzle due to seal damage, a situation sometimes observed on other machines of similar age not yet retrofitted with new seals. By assuming a nozzle bypass flow corresponding to 5 percent of the inlet airflow in place of the nozzle throat area increase, measured data could be matched satisfactorily by the inverse cycle algorithm. Results of calculations showing individual component performance shifts with reference to datum values can be seen in Table 1. As can be observed, compressor flow and efficiency deviations from average new engine data are minor, denoting the healthy state of this component as a consequence of good air filtering and regular washing. Instead, overall turbine efficiency shows a 3 percent decrease that, associated with energy loss due to first stage nozzle bypass, shows to be greatly responsible for performance degradation. Additional power loss is generated by decrease in firing temperature associated to exhaust temperature limitation prematurely triggered by the lower than normal pressure ratio. Moreover, measured fuel flow rate shows to be around 10 percent lower than the value required in cycle calculations to satisfy thermal balance. As a consequence, actual injected steam mass flow results are proportionally lower, thus producing a further output decrease.

Table 1. Example of Inverse Cycle Analysis Results.

Ambient Temperature	deg C/F	27.5/81.5
Exhaust Temperature	deg C/F	520.2/968.4
$\Delta$ Compressor Flow (1)	%	-0.96
$\Delta$ Compressor Efficiency (1)	%	-0.2
$\Delta$ Pressure Ratio (2)	%	-5.3
$\Delta$ Firing Temperature (3)	deg C/F	-17.8/-32.0
$\Delta$ Expansion Efficiency (1)	%	-3.05
$\Delta$ Fuel Flow (4)	%	-10.4

#### NOTES

- (1) Difference between values needed for test data match and corresponding values referred to new and clean engine
- (2) Difference referred to standard engine with same flow and firing temperature
- (3) Difference due to lower expansion efficiency at same pressure ratio
- (4) Difference between measured value and thermal balance requirement

The standard cycle program was then calibrated on the actual engine condition by using component performance shifts determined in the inverse cycle calculation, and was used to investigate possible ways of recovering performance acceptably and avoid rescheduling of a major engine overhaul foreseen one year later. Two possible strategies were identified:

- To partially offset exhaust temperature limitation effects at ambient temperatures over 20 to 25°C (48 to 77°F)—raise pressure ratio by increasing steam injection. The steam/fuel ratio was therefore set to the allowable limit of 1.4 by taking advantage of surplus plant steam availability, after recalibration of the fuel

flow measurement to assure corrected steam/fuel injection ratios. A substantial power increase resulted from both the larger steam mass flow and the firing temperature gain made possible by the rise in pressure ratio associated with the larger steam flow.

- *In the lower ambient temperature range*—raise the exhaust temperature control curve to bring the firing temperature back to nominal value consistently with the lower turbine efficiency and the air bypass through the first stage nozzle seals. A new CDP- $T_X$  control curve was evaluated with the recalibrated engine cycle model. In this operating range, the steam/fuel ratio increase was maintained, with maximum steam flow limited to 3.3 tons/hr at low ambient temperatures as required by the existing steam supply system.

The revised CDP- $T_X$  control curve can be seen in Figure 11, while Figure 12 shows the power gains over the entire ambient temperature range from zero to 35°C (32 to 95°F), in which this machine typically operates. A power increase of 6 to 7 percent could be achieved over most of the temperature range. This made it possible to profitably operate the machine for the rest of the year, with particular benefits in the hot season, before carrying out a scheduled major overhaul at the beginning of the present year, during which the diagnosis of the internal states was substantially confirmed.

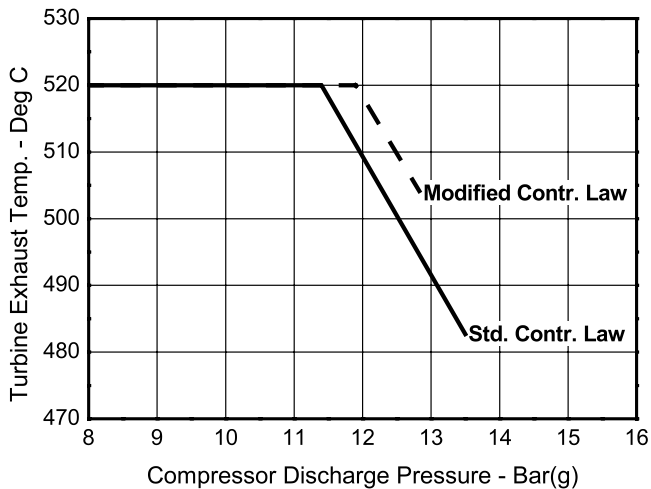


Figure 11. Example of Exhaust Temperature Control Law Modification.

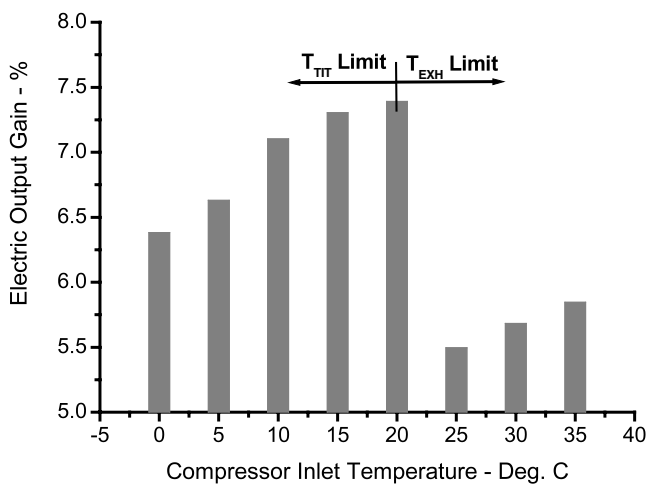


Figure 12. Effects of Control Law Modifications on Deteriorated Gas Turbine Output.

The analysis just described was one of the first applications of the inverse cycle methodology to modify control laws according to actual internal states and achieve significant performance recovery without surpassing the component operating limits. This analysis is presently performed routinely during contractual monitoring and diagnostic activities, with adjustments made to control laws when suitable to optimize performance following changes and deterioration of internal components.

## HOT PARTS LIFE ASSESSMENT

Improvement of reliability and availability of gas turbine plants heavily relies on assessment of component deterioration and damage in relation to previous operation, and on ability of predicting remnant life potential as a function of future operational and plant production requirements.

Assessment of cumulated damage is typically made either through metallurgical analyses on samples withdrawn from engines or through comparison with other machines in the fleet if a significant and consistent database exists. In the first case, interruption of service and replacement of parts is required, while in the second case significant under or overestimates of residual life capabilities may occur because of remarkable differences in operating criteria and site conditions. For example, even if all machines considered in the analysis are constantly operated at base load, actual metal temperatures of cooled turbine nozzles and blades can significantly differ if installations are in geographic areas with very different climates. In fact, turbine cooling air extracted from compressor delivery can have significantly different temperatures depending on site conditions at which engines operate, and therefore implies significant differences in cooled nozzle and blade metal temperatures even with the same TIT on all units. This will be confirmed by an example shown in the following. In mechanical drive applications, variability is further increased by extended operation at part loads and speeds with reduced TIT, thus making simple statistical life correlation from average fleet data even more uncertain than for generator drive units, typically operated at base load for most of their life.

Therefore, the capability of reliably assessing hot gas-path conditions by means of the inverse cycle approach opens a way to improve parts life assessment by taking environmental and operational variables into proper consideration, thanks to the monitoring database support.

The idea for developing this technique is based on a long-term goal for transferring design methodologies and tools into the diagnostics area. When designing a new gas turbine, a reference design thermal cycle is established for base load and standard operating conditions (normally ISO). The cycle gas-path data (pressures and temperatures) are used as boundary conditions for aerodynamic analyses that determine detailed, three-dimensional flow fields over the turbine nozzle and blade airfoils. This process is iterated until the most efficient airfoils are designed; aerodynamic analysis is finally used to evaluate heat transfer coefficients between hot stream and airfoil surfaces. Parallel calculations are made to determine flows and heat transfer coefficients in the internal airfoil cooling cavities traversed by air extracted from compressor delivery. Finally, heat transfer coefficients and airfoil geometry are used as inputs to finite-element mechanical and thermal analyses to assess metal temperatures and stresses and locate areas of highest creep and crack damage potential. The entire process is iterated until the optimum tradeoff between aerodynamic efficiency and parts life design target is achieved.

In diagnostics, engine geometry is frozen, but operating conditions are no longer constant, and change accordingly to environmental variables and operating requirements. Although the inverse cycle tool is capable of providing gas-path data in all situations, it is evidently impossible to directly use complex design

tools to evaluate links between gas-path data and metal temperatures and stresses routinely during monitoring and diagnostic activities. To find a solution, an approach based on dimensional analysis and turbine fluid-dynamic similarity laws was developed, in order to set up relatively simple semiempirical laws for real-time use in association with thermal cycle analyses. Parameters chosen to correlate turbine gas-path data are pressure ratio from inlet to exhaust and the so-called “aerodynamic speed” or corrected speed  $N_C$  expressed as the ratio between the physical shaft speed and the square root of TIT ( $T_4$ ):

$$N_C = N / \sqrt{T_4} \quad (5)$$

Aerodynamic similarity laws show that gas velocities and heat transfer coefficients over the airfoils can also be correlated through these two simple parameters. Therefore, if aerodynamic and heat transfer analyses are made for a suitable number of turbine map points within a pressure ratio and  $N_C$  range of interest for diagnostic purposes, heat transfer coefficients can be evaluated for any other condition simply as a function of pressure ratio and  $N_C$ . Similarly, metal temperatures can be correlated to hot gas and cooling air temperatures through a simple nondimensional relationship called “cooling effectiveness” and expressed as:

$$\varepsilon_c = \frac{T_{gas} - T_{metal}}{T_{gas} - T_3} = f\left(N / \sqrt{T_4}, P_3 / P_8\right) \quad (6)$$

In conclusion, metal temperatures in the operating range of interest for diagnostics can be easily correlated to the main cycle parameters, once aerodynamic and thermomechanical analyses have been performed on a suitable number of points at different pressure ratios and  $N_C$  over the turbine map. The entire process is synoptically shown in Figures 13 and 14. Determination of stresses associated with metal temperatures is straightforward, since they are simple functions of pressure drops across rows and rotational speeds. For each airfoil, gas-path data and metal temperature/stress correlation needs to be established for all locations having the highest damage potentials.

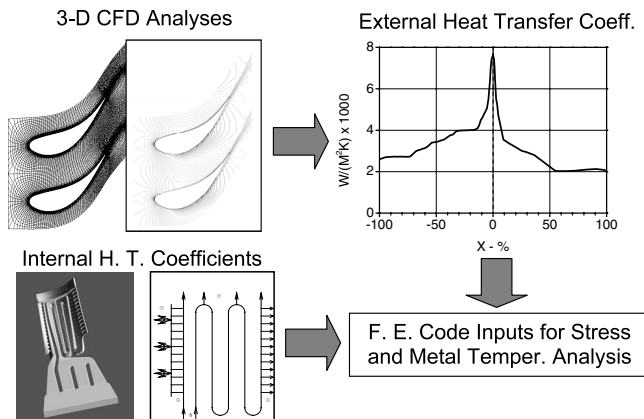


Figure 13. Remaining Life Assessment System—CFD and Heat Transfer Analyses.

The final step in life assessment is evaluation of critical parts damage in relation to monitored operational data analyzed according to the procedure described above. Overall damage is then evaluated as the sum of contributions from three operating states:

- *Steady or nearly steady operation*—Damage is evaluated in consecutive steps, each one corresponding to monitoring intervals with computed steady or nearly steady metal temperatures and stresses.

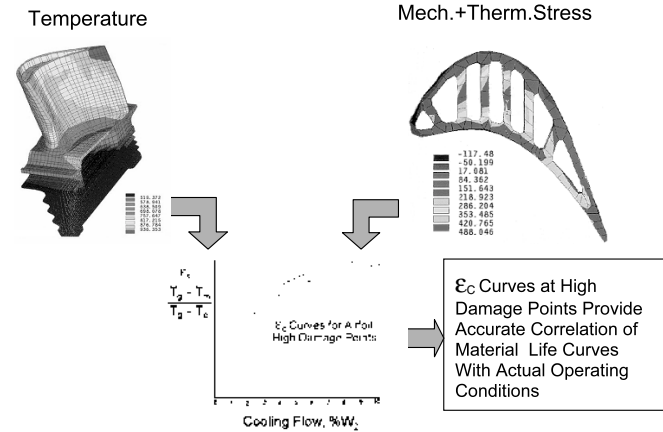


Figure 14. Remaining Life Assessment System—Thermomechanical Analyses.

- *Controlled state changes with slow transients* (e.g., controlled load changes)—Damage is evaluated as for steady-state, by reducing the width of time steps considered for damage evaluation. Low cycle fatigue effects are negligible in these cases.

- *Fast transients*, like rapid load changes, startups, trips, and fast shutdowns—In these cases, LCF effects are predominant and damage contribution must be evaluated with separate transient analyses and by making reference to existing experience.

To calculate total damage produced in the operating states mentioned above, several approaches are possible. While others are being considered, current applications are referring to a methodology already used in the author’s company, the continuum damage model (Mezzedimi, et al., 1998), where damage is incrementally evaluated as a combined contribution of creep and LCF. Up to now, damage has been evaluated *a posteriori* through metallurgical analyses (e.g., gamma prime phase distribution (Mezzedimi, et al., 1998)) on samples withdrawn from operation, and by making reference to design data when insufficient operating information was available.

The new methodology has the potential advantage of overcoming the need for assessing cumulated damage by analyzing samples, thus avoiding the need for machine shutdown and parts replacements if not actually needed. Another important feature is the capability of closely associating actual operation to damage through direct correlation of monitored data to metal temperatures and stresses. This makes it possible to properly account for effects of operating environment differences (e.g., ambient conditions) and of multiple operating modes (e.g., part load operation or possible overloads if needed by critical or emergency operating requirements).

From evaluation of hot parts damage incurred in past operation, the system can be easily turned into prediction of remnant parts life once the initial accumulated damage is known and expected operating conditions and modes can be anticipated. In absence of a monitoring database, initial component states need be assessed from available recorded data, comparison with similar machines in the fleet, and metallurgical analyses on samples.

Remnant life prediction is made through use of the direct cycle program, after calibration on recent operational data, provided either by continuous monitoring or by *ad hoc* measurements. To account for progressive deterioration during scheduled operation, degradation models can be added to the direct cycle model to account for compressor fouling, turbine erosion, clearance increase, etc. These models are presently being set up and refined through systematic analysis of a large number of monitored data. By using the calibrated cycle-deck incorporating degradation models, metal temperatures and stresses can first be calculated as a function of expected operating conditions and output requirements, and then

used to predict corresponding damage accumulation through the same approach used in monitored data analysis.

Two alternatives for remnant life estimates can be provided to operators:

- Predict life achievable with maximum output in all conditions; with this option the cycle model is run for base-load output and variable parameters are ambient conditions and degradation effects only.
- Assess achievable output capability to reach a preset remnant life target, to reliably schedule plant shutdowns and machine overhauls in advance. In this case, maximum engine outputs are estimated for specified operating conditions as a function of limiting levels in the turbine inlet temperature necessary to achieve life extension targets.

Depending on particular requirements, the above schemes can be mixed and integrated with economical analyses to assess the best tradeoff between maintenance frequency and plant production. For example, life reductions associated with limited periods of overfiring can be compared with benefits of keeping plant production at the highest levels in peak demand periods or adverse climatic conditions like very hot days.

A simple example taken from model sensitivity analyses made before launching this project can be useful to show the importance of adequately taking actual operating parameters into consideration if reliable assessment of critical component life has to be performed. The example referred to is a first stage turbine rotor blade of a state-of-the-art industrial engine with design turbine rotor inlet temperature in the 1200 to 1300°C (2200 to 2370°F) range. The blade has a high efficiency cooling system with multipass internal serpentine passages and “showerhead” leading edge film cooling. Cooling passage geometry and other heat exchange characteristics correspond to the configuration shown in Figures 13 and 14, synoptically illustrating the remaining life evaluation process. The example shows effects of cooling air temperatures associated with different average ambient temperatures at which one individual engine or different engines in the fleet can operate for periods of time. Reference data (cooling air temperature, blade stress, and life) are referred to operation at ISO conditions, and life comparisons were made for operation at average ambient temperatures of 5 and 25°C (41 and 77°F), respectively. For study purposes, a commercial high temperature alloy (Inconel 738) was considered, with creep and rupture data derived from tests made in the company’s metallurgical laboratories. Life trends of this material as a function of temperature are fully comparable to those of the alloy used on the actual engine.

Base-load operation with the same turbine rotor inlet temperature was considered for all cases. For these conditions, flow fields over the first stage rotor airfoil resulted close enough to assume equal cooling effectiveness in all calculations for the sensitivity analysis purposes. The external heat transfer coefficient distribution resulting from the viscous Navier-Stokes analysis is shown in Figure 15, and is an enlarged view of that visible in the synoptic diagram in Figure 13. The overall cooling effectiveness based on this heat transfer coefficient diagram and heat exchange characteristics of the internal serpentine cooling passages resulted equal to 48 percent. The ratio between relative total temperature at rotor inlet and turbine inlet temperature derived from the aero-analysis was 0.89. A summary of data used in the analysis and estimated rotor airfoil average metal temperature differences at the airfoil reference section for 5, 15, and 25°C (41, 59, and 77°F) ambient temperatures, respectively, are shown in Table 2.

Figure 16 shows creep life curves (rupture and 1 percent creep) for the selected material derived from lab test data by using the Larson-Miller parameter set for the average stress on the airfoil section considered. A 100,000 hour life for 1 percent creep was set at the reference airfoil metal temperature corresponding to 15°C (59°F) ISO ambient.

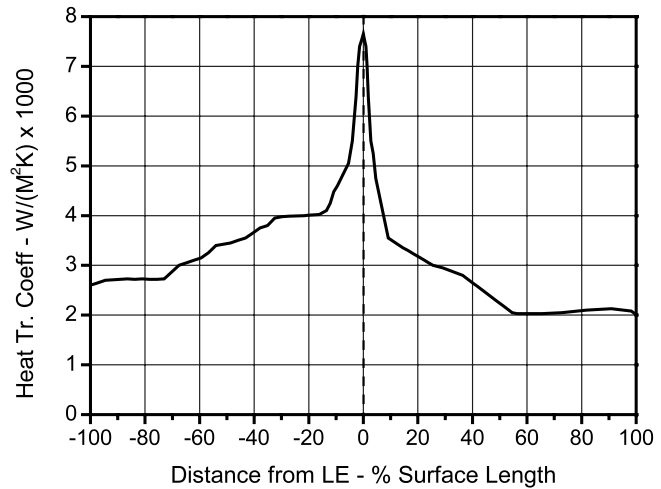


Figure 15. Heat Transfer Coefficient Distribution on a Turbine Rotor Blade Airfoil.

Table 2. Effects of Ambient Temperature on Turbine Blade Metal Temperature.

Ambient Temperature	deg C/F	4/39	15/59	25/77
Compressor Delivery Temp.	deg C/F	406/763	417/783	428/802
Cooling Effectiveness	%	48	48	48
Metal Temp. Difference	deg C/F	-5.7/-10.3	0/0	5.6/10.

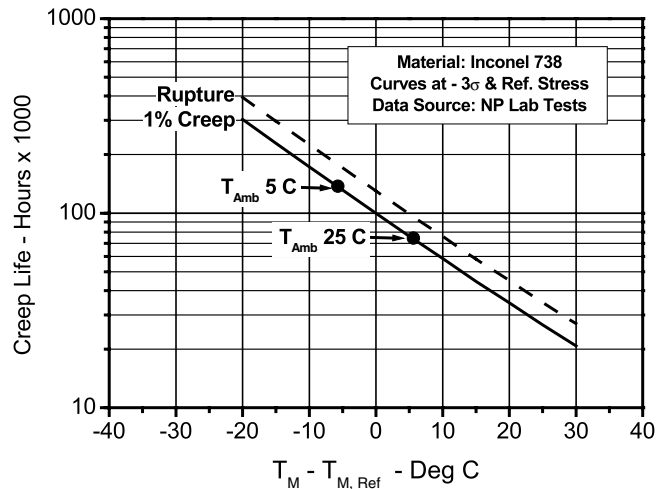


Figure 16. Operating Ambient Temperature Effects on Turbine Blade Creep Life.

As can be seen, life differences corresponding to operation at relatively close average ambient temperatures are quite remarkable and show the importance of taking all parameters that affect operating metal temperatures into proper consideration if accurate life estimates need to be made, or to properly compare fleet machines operating in different climatic environments. In actuality, situations are considerably more complex, if other factors like load changes, control system deviations, different fuel types, emergency, or unpredictable situations, etc., are taken into consideration. The adopted approach however, thanks to provision of rigorous cycle analysis and accurate gas-path data/thermomechanical correlations, can fully handle a wide variety of operating situations and reliably determine component metal temperatures and stresses in all cases, i.e., the real factors finally affecting hot parts life.



At the time of writing this paper, the remnant life model is under development for the 10.7 MW (14,350 hp) two-shaft gas turbine in the company's product line. Several of these units are under LTSA contracts with ongoing remote monitoring and collected data routinely analyzed with the inverse cycle program. The models for correlating cycle gas-path data with metal temperatures and stresses are under completion, and a major work is under way to analyze existing fleet data to complete the models with information related to LCF effects produced by transient operation, startups, scheduled, and unscheduled shutdowns.

## CONCLUSIONS

The work summarized in this paper represents a major effort being carried out to provide the highest added value to services offered under long-term service agreements. The architecture of the integrated gas turbine cycle analysis and hot parts life assessment presented herein has three essential requisites: knowledge of detailed machine design information, use of high-level proprietary analysis and design tools, and availability of a comprehensive operational background on the machine fleet under consideration. This wide knowledge and resource background is typically available to OEMs that also provide global service support to customers. However, a system of this kind can be exploited in its full potentiality only with a strict synergy between service providers and plant owners/operators to achieve full advantage from two major capabilities built in the tool:

- Dynamic modeling of the gas turbine thermal cycle to reliably assess actual deteriorated engine performance. This makes it possible to retune control parameters to match time varying component deterioration, and keep performance at levels suitable to achieve operator production plans and minimize preventive maintenance needs.
- Reliable assessment of cumulated damage on critical hot path parts without need for destructive analyses, and remnant life prediction as a function of planned operating strategies for best tradeoff between plant economic return and maintenance costs.

Application of the diagnostic cycle modeling tool to a restricted but significant number of gas turbines has demonstrated potentials of this method to achieve performance recoveries on degraded engines capable of satisfying immediate operator needs and in helping to avoid premature or unscheduled maintenance shutdowns. Extension of the system to hot parts remnant life assessment is under way and is expected to substantially increase accuracy of present methodologies through direct and continuous monitored database utilization and collaboration with operators in using the tool to optimize their operating strategies.

## NOMENCLATURE

$A_{TH}$  = Turbine first stage nozzle throat area  
 $CDP$  = Compressor discharge pressure used in the gas turbine control curve

$C_{D,N}$  = Turbine first stage nozzle flow discharge coefficient  
 $e_Q$  = Residuals in the inverse cycle solution equations  
 $f$  = Global function representing the gas turbine thermal cycle model  
 $F_{ob}$  = Objective function in the inverse cycle solution algorithm  
 IGV = Inlet guide vanes  
 LCF = Low-cycle fatigue  
 LTSA = Long-term service agreements  
 $N$  = Turbine shaft rotational speed  
 $N_C$  = Turbine corrected speed  
 $n_e$  = Number of constraints in the inverse cycle solution equations  
 $P_3$  = Compressor delivery pressure  
 $P_8$  = Gas turbine exhaust pressure  
 $Q_c$  = Values calculated by the inverse cycle program corresponding to measured  $Q_m$  data  
 $Q_m$  = Measured operating and performance parameters used in the inverse cycle method  
 $T_3$  = Compressor delivery temperature  
 $T_4$  = Turbine (rotor) inlet gas temperature  
 $T_{gas}$  = Hot gas temperature on turbine airfoil  
 $TIT$  = Turbine (rotor) gas inlet temperature  
 $T_{metal}$  = Turbine blade airfoil metal temperature  
 $T_x$  = Gas turbine exhaust temperature used in control curve  
 $X$  = Fixed parameters (component characteristics) in the inverse cycle algorithm  
 $\epsilon_c$  = Turbine airfoil cooling effectiveness

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