2000-GT-312

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Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Munich, Germany – May 8-11, 2000
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until August 30, 2000

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ABSTRACT

This paper describes the results of real-time, on-line performance monitoring of two gas turbines over a period of five months in 1997. A commercially available software system is installed to monitor, analyze and store measurements obtained from the plant's distributed control system. The software is installed in a combined-cycle, cogeneration power plant, located in Mass., USA, with two Frame 7EA gas turbines in April 1997.

Vendor's information such as correction and part load performance curves are utilized to calculate expected engine performance and compare it with measurements. In addition to monitoring the general condition and performance of the gas turbines, user-specified financial data is used to determine schedules for compressor washing and inlet filter replacement by balancing the associated costs with lost revenue. All measurements and calculated information are stored in databases for real-time and historical trending and tabulating. The data is analyzed *ex post facto* to identify salient performance and maintenance issues.

1. INTRODUCTION

Even under the best possible operating conditions, the performance of a gas turbine is subjected to deterioration due to compressor fouling and corrosion, inlet filter clogging, thermal fatigue and oxidization of hot-gas path components such as combustion liners and turbine blades.

The performance degradation attributed to compressor fouling is mainly due to deposits formed on the compressor blades by particles carried in by the air that are not large enough (typically a few microns diameter) to be blocked by the inlet filter. Depending on the environment, these particles may range from dust and soot particles to water droplets or even insects. These deposits result in a reduction of compressor mass flow rate, efficiency and pressure ratio which in turn causes a drop in gas turbine's power output while increasing its heat rate. This type of degradation is by far the dominant mode. Indeed, several studies of heavy-duty industrial gas turbines suggest that the decrease in output can easily reach 5% after a month's operation (e.g. Diakunchak, 1991).

Fortunately, compressor fouling is a "recoverable" degradation in that it can be alleviated by periodic on-line and/or off-line compressor washes. In an on-line wash, distilled water is injected into the compressor while the gas turbine is running such that water droplets impact the blades at high speeds to loosen and partially remove deposits. However, complete performance recovery can only be achieved by an off-line wash where distilled water (sometimes mixed with a special detergent) is sprayed into the gas turbine while being rotated by the starter at the crank speed.

Inlet filter clogging reduces gas turbine air flow and compressor inlet pressure and thus adversely affects gas turbine performance. Replacing the old filter with a new one can recover the lost performance.

The performance degradation associated with hot gas path components is influenced by myriad factors such as fuel quality, number of starts, amount of water/steam injection etc. and it is commonly referred to as "non-recoverable." Typically, for a base load machine this type of degradation can be 0.2-0.3% of the nominal (when new and clean) rating after a month's operation (Flashberg and Haub, 1992). The only remedy for non-recoverable degradation is an engine overhaul.

Although the benefits of compressor washing and inlet filter replacement are undoubted, the frequency of on-line and off-line washes and the type of detergent solvents to be used (if at all) and when to replace the filter are widely debated issues. It is essential to develop maintenance schedules based on the characteristics of the engine and its operating environment and/or cycle in order to balance the maintenance costs with lost revenue and extra fuel costs (Tarabrin et al., 1998).

In this paper, results obtained from an on-line monitoring system for two industrial gas turbines will be presented in a way to clarify the general issues enumerated in the preceding paragraph. The monitoring system itself, results and financial benefits thereof specific to the installation site were the subject of an earlier paper by Gülen et al. (1999).

2. PLANT DESCRIPTION

The 245-MW, gas-fired, combined-cycle cogeneration power plant consists of two GE PG7121EA gas turbines with evaporative inlet coolers, one 72-MW condensing, dual admission/dual extraction, axial exhaust steam turbine, and two heat recovery steam generators (HRSGs). The plant is designed to deliver up to 150,000 lb/hr of steam to a steam host at 675 psia and 750 °F and injection steam to the gas turbines at 310 psia and 550 °F for NOx control. Plant construction started in October 1991, and it was commercially operating by July 1993.

Both gas turbines are primarily controlled by GE Mark IV Speedtronic control systems which are connected to the overall plant distributed control system (DCS). The on-line performance monitoring system was installed in April 1997 and interfaces with the DCS to obtain gas turbine data tags.

3. SYSTEM DESCRIPTION

The software system used for performance monitoring and subsequent data analysis is a complete unattended, on-line, real-time performance monitoring system that is designed to track the current and historic technical and financial performance of a gas turbine. The operating principle of the software is shown in Fig. 1.

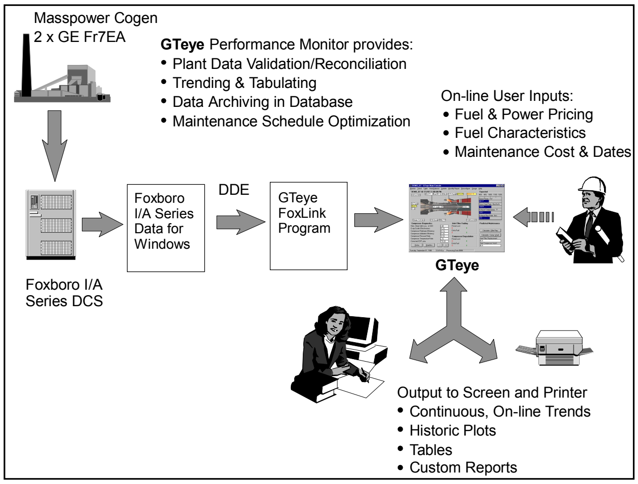

Figure 1 How the on-line, real-time performance monitoring system works.

The performance monitoring suite consists of separate modules for gas turbine model configuration, on-line performance monitoring, data trending and tabulating.

All software modules are 32-bit Windows programs designed to run on IBM-compatible personal computers (PC) with Windows 95 or NT operating systems (OS). The on-line monitoring module interacts with the plant's DCS via a special software module. The interface mechanism is the Dynamic Data Exchange (DDE) protocol. A proprietary Windows program provided by the DCS manufacturer is the data source in the DDE conversation and the data link module is the data destination.

At a user-specified interval, the data link module requests updated values for each of the gas turbine data tags identified during the gas turbine model configuration phase. The updated values are stored in an ASCII data file that is continuously queried by the main monitoring module. When the module finds that its input file contains new data, it reads and processes the new data. In the application described herein the monitoring module reads a new data every 10 seconds and averages data in intervals of one minute, one hour and one day. For more details of the software description, the reader is referred to Gülen et al. (1999).

4. GAS TURBINE PERFORMANCE

The performance expected from the gas turbine at its undeteriorated condition at prevailing site and loading conditions can be found as follows:

$$Y_{j,x} = Y_{j,b} \prod_{i=1}^{7} \mathrm{X}_{ij} \tag{1.a}$$

where Y_j is a performance parameter of the gas turbine guaranteed by the manufacturer when the gas turbine was (i) in its new and clean condition at the time of the acceptance test, or (ii) in its condition at the time of any performance test which is going to be used as a benchmark. The subscripts $_x$ and $_b$ denote expected and baseline values, respectively. The four cardinal gas turbine performance parameters are listed in Table 1. X_{ij} are the correction factors at a given value of a specific turbine operation parameter, ***i***, which, when multiplied with the baseline value of a turbine performance parameter $\boldsymbol{Y_j}$, reflect the change in $\boldsymbol{Y_j}$ due to a change in ***i*** (see Table 2.)

TABLE 1. Cardinal gas turbine performance parameters, Y_j.

j	GT Parameter Y_j
1	Gross output at generator terminals
2	Gross heat rate at generator terminals
3	Exhaust gas temperature
4	Exhaust gas mass flow rate

TABLE 2. Correction factor to Y_j, X_{ij}.

i	Correction in Y_j for
1	Ambient Temperature (or Compressor Inlet Temperature if there are inlet coolers and/or heaters)
2	Ambient Pressure
3	Ambient Humidity (or Compressor Inlet Humidity if there are inlet coolers)
4, 5	Inlet & Exhaust Pressure Drops
6, 7	Water/Steam Injection

When the gas turbine is running at part load, the performance of the gas turbine is calculated as

$$Y_{j,x} = \Pi_j \left(Y_{j,b} \prod_{i=1}^{7} \mathrm{X}_{ij} \right) \tag{1.b}$$

where Π_j is the part load correction factor to the gas turbine performance parameter Y_j. For the gas turbine power output, i.e. $j=1$, Π_j is the gas turbine's loading factor expressed as a fraction. For the others, Π_j is determined from manufacturer's part load correction curves.

4.1. Compressor performance

For a given speed, there is a characteristic line that describes the flow versus pressure ratio relationship for any axial or centrifugal compressor. This line has an upper limit that is referred to as the surge point and a lower limit that is commonly called as the choke or stonewall point. At the given speed, a compressor can operate in a stable fashion only between these two points. Above the surge point, airflow within the compressor will reverse directions rapidly, thereby leading to severe vibration and equipment damage. Below the choking point, flow will not increase and compressor operation will become unpredictable. The collection of characteristic lines over a certain range of rotational speeds comprises the performance map of a given compressor. The performance map usually also includes lines of constant adiabatic efficiency which are referred to as efficiency contours.

The compressor performance map is a very useful diagnostic tool that lends itself to engine performance monitoring with the most readily available measurements. The manufacturer's performance map for the compressors of their gas turbine can be specified during model configuration phase. Unfortunately, few turbine manufacturers publish such data. Although compressor maps for certain engines can be found in the literature, they are few and far between and not easy to locate. A representative compilation is available in a commercial software, GasTurb 8.0 (Kurzke, 1998). One can certainly construct a compressor map from the operational data. However this would require a reliable set of historic DCS data that is usually not the case.

When flow, rotational speed and compressor delivery pressure are expressed in non-dimensional terms, effects of any changes in the compressor air inlet conditions and physical properties are eliminated. Furthermore, such normalized performance maps of a variety of compressors are found to be nearly identical (e.g. see Saravanamuttoo & MacIsaac, 1983). The relevant flow, speed, and pressure rise parameters, respectively, are defined as follows:

$$\Phi = \frac{\dot{m}\sqrt{RT_0}}{p_0} \Bigg/ \left(\frac{\dot{m}\sqrt{RT_0}}{p_0}\right)_{design} \tag{7.a}$$

$$\mathrm{N} = \frac{N}{\sqrt{RT_0}} \Bigg/ \left(\frac{N}{\sqrt{RT_0}}\right)_{design} \tag{7.b}$$

$$\Pi = \frac{p_2}{p_0} \Bigg/ \left(\frac{p_2}{p_0}\right)_{design} \tag{7.c}$$

where $\dot{m}$ is the compressor air mass flow rate, p is the pressure, T is the temperature, N is the rotational speed, and R is the gas constant. Subscripts $_0$ and $_2$ refer to compressor inlet and exit, respectively.

5. GAS TURBINE PERFORMANCE DEGRADATION

5.1. Compressor fouling

The difference between expected and measured powers gives the lost kilowatts due to gas turbine degradation. However, this information does not enable one to separate the effects of compressor fouling (recoverable) from those of hot gas path degradation (non-recoverable).

The monitoring software algorithm utilizes the fact that non-recoverable performance degradation proceeds at a rate that is approximately one order of magnitude slower than that for recoverable degradation by attributing the lost power entirely to compressor fouling. The effect of non-recoverable degradation is introduced by using a user-specified *recovery factor*, that is

$$P_{lost} = \phi^n P_x - P_a \tag{2}$$

where ϕ is the recovery factor for power output, P_a is the actual measured power output, and n is the number of the off-line compressor wash since on-line monitoring started. Thus, we account for the fact that each off-line compressor wash restores only a fraction (equal to ϕ) of the power output previously restored by the preceding crank wash, and n^{th} off-line compressor wash restores only ϕ^n fraction of the original (baseline) power output.

A similar consideration applies to the heat rate; i.e. each off-line wash restores the heat rate to a level slightly higher than that restored by the preceding wash. In that case, extra fuel burned can be expressed as

$$F_{xtra} = f_a - \phi^n \phi'^n f_x \tag{3}$$

where ϕ' is the recovery factor for the heat rate, f_a and f_x are actual and expected fuel mass flow rates, respectively. The values of ϕ and ϕ' used herein are 0.998 and 1.005, respectively.

The average linear rate of change of power lost due to compressor fouling at a given time t can be found from

$$\Pi = \frac{2}{t^2}\int_0^t P_{lost}(t)\,dt\ . \tag{4}$$

Similary, for the average linear rate of change of the extra fuel burned due to compressor fouling

$$\Phi = \frac{2}{t^2}\int_0^t F_{xtra}(t)\,dt\ . \tag{5}$$

It can then be shown that the optimum time to do an off-line compressor wash is given by

$$\tau^* = \sqrt{\frac{2C_m}{\Pi C_p + \Phi C_f}} \tag{6}$$

where C_m, C_p and C_f are cost of maintenance, power sale price and fuel purchase price, respectively, and τ^* is the time from the last compressor off-line wash. The on-line monitoring software uses these formulas in its algorithm to continuously update the estimated off-line compressor wash dates as new data is received and processed.

Note that, although compressor fouling is detrimental to gas turbine output and heat rate, it actually decreases the fuel consumption at full load. For a typical Frame 7 machine, a 2.5% decrease in compressor efficiency and 5% reduction in air mass flow results in a 7.5% decrease in output and a 2.9% increase in heat rate whereas the fuel consumption drops by 4.8%. The net financial impact, however, is almost always negative unless the fuel purchase price is extremely high and/or power sale price is too low. The beneficial effect of an engine degradation on fuel expenditure is somewhat counterintuitive and deserves a closer look. This point and the condition under which the net financial impact can even be positive are expounded on in the Appendix.

5.2. Inlet Filter Fouling

For a gas turbine, neglecting leakage and assuming no blow-off during normal operation, the mass flow rate of air entering the compressor is equal to

$$\dot{m}_{air} = \dot{m}_{exh} - \dot{m}_{fuel} - \dot{m}_{inj} \tag{8}$$

From the baseline values of exhaust gas, fuel, and steam/water injection mass flow rates we determine a flow coefficient C for the inlet filter such that

$$\dot{m}_{air} = C \frac{\left(\rho \, \Delta p_{in}^{nc}\right)^n}{\mu^m} \tag{9}$$

Where ρ is the density of inlet air, μ is the viscosity and Δp_{in}^{nc} is the new and clean inlet pressure loss across the inlet filter. Exponents *m* and *n* have different values for laminar and turbulent flow regimes. Using *C* and the measured values of fuel and steam/water mass flow rates along with the exhaust mass flow rate determined from the engine part load curves in the above equations, a pressure drop across the filter is calculated. This pressure drop would only result with a filter that was new and clean. Lost generating capacity due to inlet filter fouling is found as follows:

$$P_{lost} = P_x \left(\frac{X_{41}^{nc}}{X_{41}} - 1 \right) \tag{10}$$

The superscript ^{nc} denotes that the correction factor X_{41} (see Table 2) is calculated using the new and clean inlet pressure drop obtained from Eq. (9). The resulting value is then used to evaluate the impact of filter clogging on gas turbine's performance as described in §6.2.

The filter replacement dates are estimated using the same logic and formulas presented above for compressor wash schedule by replacing the lost power and extra fuel terms by their corresponding definitions. Similar considerations described at the end of §5.1 apply to effects of inlet filter fouling as well.

6. ON-LINE MONITORING RESULTS

At the time the monitoring system was installed in the plant, both gas turbines were nearly four years old. Thus, nominal base-load performance data was specified using the average of several full-load measurements prior to monitoring system installation. Inlet filters for both gas turbines were installed in November 1995, seventeen months prior to the installation. Off-line compressor washes were performed on gas turbines A and B on April 28 and 19 1997, respectively. Also in April, gas turbine B was taken off line for a hot gas path inspection and compressor discharge packing replacement. The firing temperature was upgraded by 15 degrees.

Two on-line monitor modules run separately for each gas turbine in unattended mode on a PC with a Microsoft® Windows NT OS in the plant's administration building. The data acquired by the software and calculated performance parameters are written into Microsoft® Access databases that are stored in a dedicated folder on the plant owner's wide area network. For the analysis presented herein, the data is downloaded via modem to a remote PC and the monitoring module is run in the off-line (calculator) mode. The figures in the paper are generated using the Trend module and the daily-average data in the databases.

6.1. Compressor Performance

The monitoring system is used to track the compressor degradation and optimize the maintenance schedule. One specific goal was to assess the effectiveness of on-line washes and the benefit of using detergents. An on-line compressor wash program was developed in order to achieve this objective. Between May 5 and June 24 1997, both gas turbines had on-line washes with distilled water only. From June 24 on, gas turbine A compressor was subjected to on-line washes with detergent three times a week while gas turbine B was kept on water-only washing program.

Trends of the capacity factor[1] for both units shown in Fig. 2 indicated several facts:

(a) For both units engine degradation progressed at a rate of roughly 1% a month.

(b) After about two months engine degradation stabilized at a the level of 2%.

(c) There was no discernible benefit associated with using detergent in on-line washes as opposed to using distilled water only.

(d) There was a constant 1% difference between units A and B although both engines were identical.

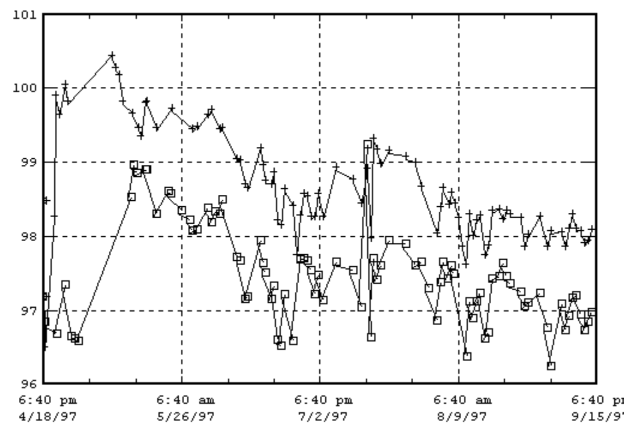

Figure 2 Capacity factor for units A (symbol) and B (symbol +).

During the same period, the heat rates for both gas turbines increased roughly by 1%. These findings indicate that the compressor fouling observed in both gas turbines is consistent with the lower end of the spectrum of observations for heavy industrial gas turbines as reported in the literature. The results supported the use of on-line water washes to limit compressor degradation and the previously reported exponential law behavior of compressor fouling due to the stabilization of the deposits on the blades (e.g. see Tarabrin et al., 1998).

The average linear rate of deterioration of output for both gas turbines as observed in capacity factor trends was calculated to be approximately 0.7 kW per hour. The average linear rate of decrease in the fuel mass flow rate, on the other hand, was found to be 0.2 lb/h per hour. Substituting these rates into Eq. (6) along with the financial information, the next crank wash was estimated to be 2,400 hours from the last one.

Compressor discharge measurements for unit A corrected for the ambient conditions are shown in Fig. 3. Same trends are observed for unit B as well. There is a 4-5% decrease in the corrected discharge pressure that stabilizes after two months similar to the trend observed

[1] Capacity factor, expressed as a percentage, is defined as the ratio of the measured power output to the expected power output.

in output degradation. The corrected discharge temperature (CDT) shows an unexpected trend in that it first decreases and then increases. This in turn results in decreasing and increasing compressor adiabatic efficiency trend. This result, at first puzzling, can be explained by examining the hourly trends of the corrected CDT and the adiabatic efficiency in Fig. 4.

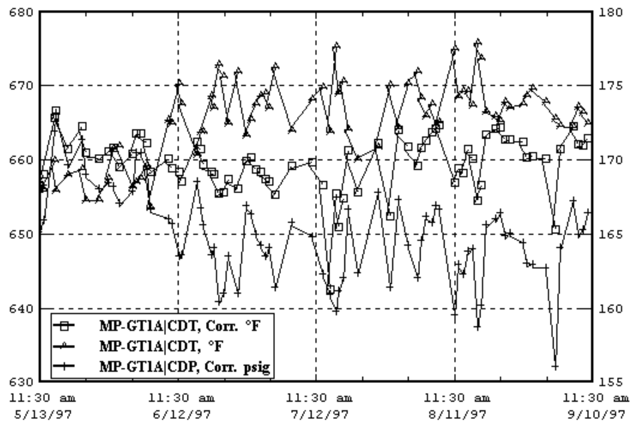

Figure 3 Compressor discharge data (corrected) for unit A. (Left axis = ºF and right axis = psig.)

The four on-line water wash periods can be clearly identified by the pronounced spikes in the hourly-averaged trends of Fig. 4.

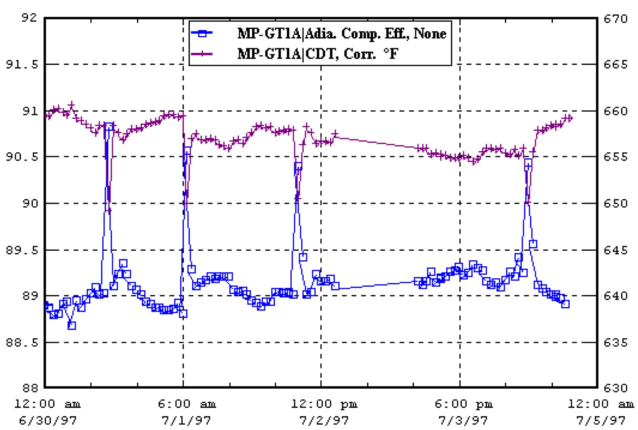

Figure 4 Hourly trends of corrected CDT and adiabatic efficiency for unit A.

These spikes are due to the inter-cooling effect of the evaporating water droplets as they are compressed and heated within the compressor. The decrease in efficiency and increase in CDT between the on-line washes can also be clearly seen. The spikes are (about 1.5% in efficiency and 5-10 degrees in CDT) major enough to cause a strong bias in daily averages. Thus, in the period following the off-line wash when the compressor was thoroughly clean the water washes bias the daily averages to the extent that they show the compressor getting progressively "cleaner." The inter-cooling effect slowly decreases over time as deposits build up and the daily-averaged plots start revealing the expected deterioration trends.

For a quick evaluation of the compressor degradation, pressure and flow parameters can be plotted on the compressor performance map as shown in Fig. 5. Note that the horizontal and vertical axes are normalized and corrected compressor air mass flow parameter (Eq. 7.a) and pressure rise parameter (Eq. 7.c), respectively.

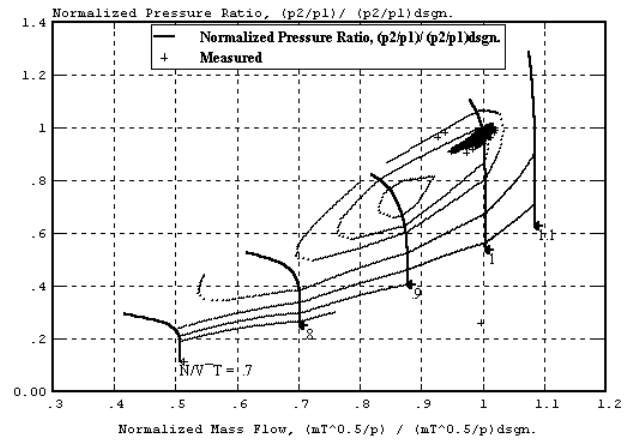

Figure 5 Normalized and corrected compressor performance parameters for unit A.

The performance map in the figure is the default map used by the monitoring software that is based on the generic map presented in Saravanamuttoo & MacIsaac (1983) and it should be a reasonably close approximation of a Frame 7 gas turbine's axial compressor. In Fig. 5, it can be seen that for the same period as in Fig. 2 there is a 5% drop in compressor air mass flow parameter and a 8% drop in compressor pressure rise parameter. The trend displayed on the map in Fig. 5 represents a shift in the operating point to the left and downward. Clearly, the shift in the speed line and partially the trends in the mass flow and pressure rise parameters are due to the increase in ambient temperature and humidity. Typically, for a 7EA the decrease in mass flow and pressure rise parameters due to a 4% increase in ambient temperature (absolute) and 40% increase in relative humidity would be 3.6% and 5.6%, respectively. The significant difference in measured values is clearly attributable to compressor fouling.

6.2. Inlet Filter Performance

Each gas turbine has a single-stage, self-cleaning intake air filtration system with an evaporative cooler downstream. The inlet filter utilizes cylindrical filter elements that are sequentially cleaned by reverse flow pulses of compressed air. The operational pressure drop is between 2 and 3 in. H_2O and the cleaning cycle starts when the pressure drop across the elements reaches the upper limit. The evaporative cooler consists of a direct-contact, irrigated media utilizing cross-fluted cellulose pads impregnated with anti-rot salts and rigidifying saturants with an operational pressure drop of 0.25 in. H_2O.

For both gas turbines, between 5/1/97 and 9/28/97, the trends shown in Fig. 6 indicated a performance degradation of roughly 25 kW per month due to inlet filter clogging. During this period the inlet pressure drop (inlet filter plus evaporative cooler) increased from 2.9 to 3.1 in. H_2O for unit A. Unit B displayed the same trend, but the measured drops were on the average 0.05 in H_2O higher.

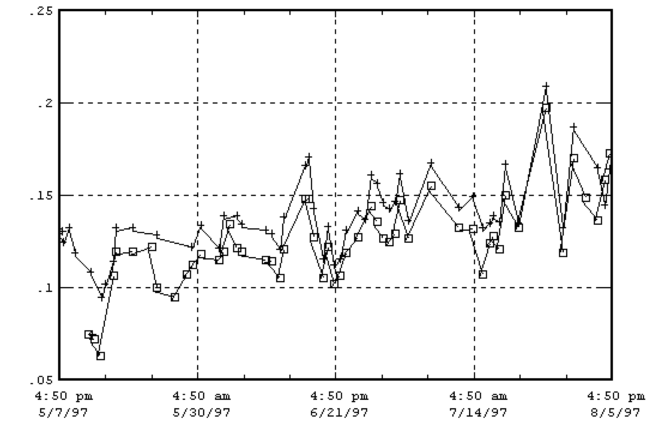

Figure 6 Lost power generating capacity [MW] due to inlet filter fouling for units A (symbol) and B (symbol +).

For both gas turbines A and B, the trends shown in Fig. 7 indicated less fuel burned due to inlet filter clogging at a rate of roughly 9 lb/h per month. Balancing the effects of both performance changes (converted into monetary terms using user-defined power prices and fuel costs) with the cost of filter replacement, a filter replacement date of 2.7 years from the last replacement date (end of 1995 for both units) has been predicted.

However, samples of filter units sent to its manufacturer and an independent consultant for testing revealed that they were deteriorated to the point of performing below minimum filtering specifications. These findings prompted the decision to replace the inlet filters in the summer of 1999. This experience clearly demonstrates that the pressure drop by itself is not a sufficient criterion to determine inlet filter performance degradation.

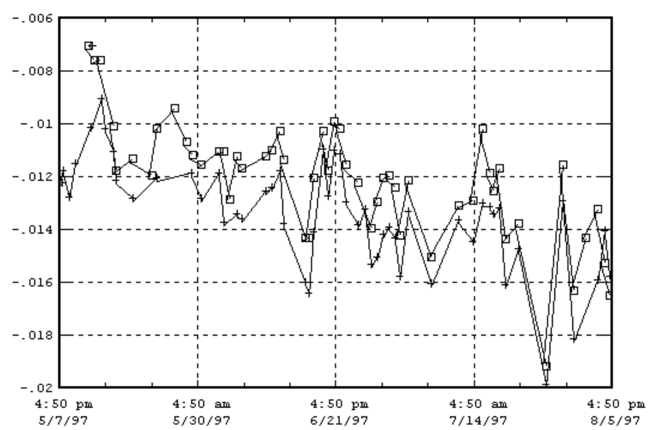

Figure 7 Extra fuel burned [lb/s] due to inlet filter clogging for units A (symbol □) and B (symbol +).

6.3. <u>Gas Turbine Performance</u>

During monitoring it was observed that gas turbine B had a consistently higher output than gas turbine A. On the average, the difference was roughly 1% of the nominal baseline output as shown in Fig. 2. Over the time period considered, i.e. approximately 2,000 hours of operating between 5/1/97 and 9/28/97, this translated to a lost power production of nearly 1,800 MWh. For an annual service factor[2] of 80% at base load this is equivalent to an annual loss of 6,300 MWh.

The trend plot shown in Fig. 8 implied that, although both gas turbines had identical temperature control curves, gas turbine A exhaust temperature as calculated by the monitoring software was roughly 3 degrees below the control value.

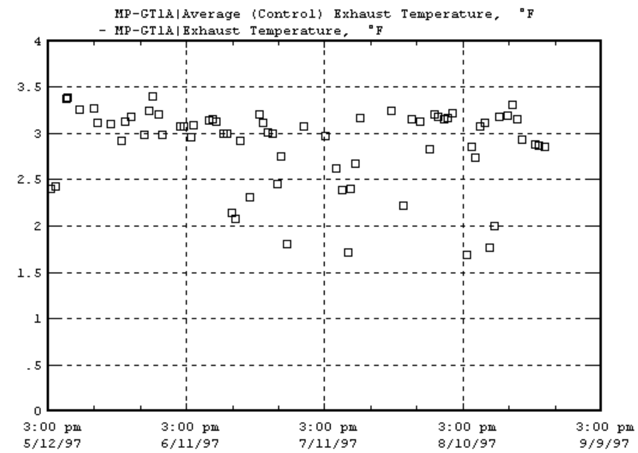

Figure 8 Deviation of average exhaust temperature from DCS control temperature signal [°F] for unit A.

To investigate this trend further, a quick visual diagnosis has been obtained from the exhaust temperature map utility of the Trend module as shown in Fig. 9. These plots are intended to be used in determining faulty combustors or fuel injectors using the known swirl angle for a particular engine. Indeed, it can be seen that unit A has two "hot" spots, i.e. thermocouples 1 and 12.

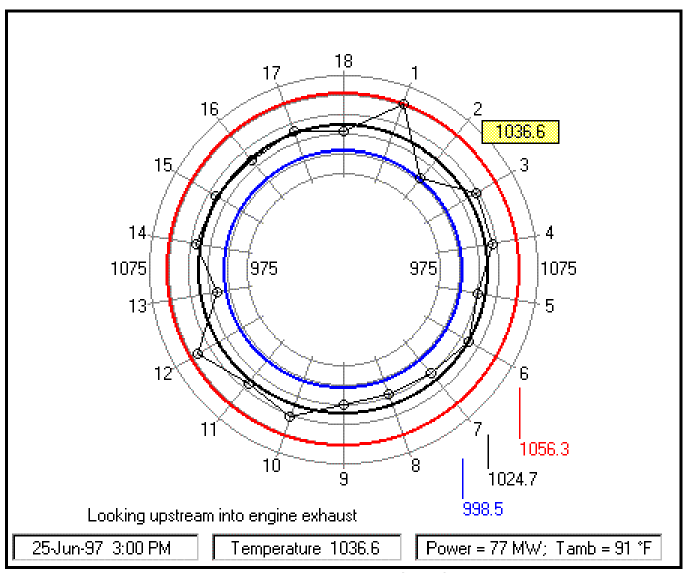

Figure 9 Exhaust temperature profile for turbine A.

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[2] Service factor is the operational use expressed as a percentage on an annual basis (8760 hours.)

From Figs. 8-9, it is quite clear that the “hot” thermocouples at the exhaust duct of gas turbine A cause a bias in the control exhaust temperature signal of gas turbine A (which is an average of eighteen thermocouples). In other words, the exhaust temperature control signal is higher than the actual exhaust temperature and thus results in under-firing. The monitoring system’s sensor validation detected the two outliers, thermocouples 1 and 12, and re-calculated an average exhaust temperature that is a better representation of the actual value and pointed to the source of the problem.

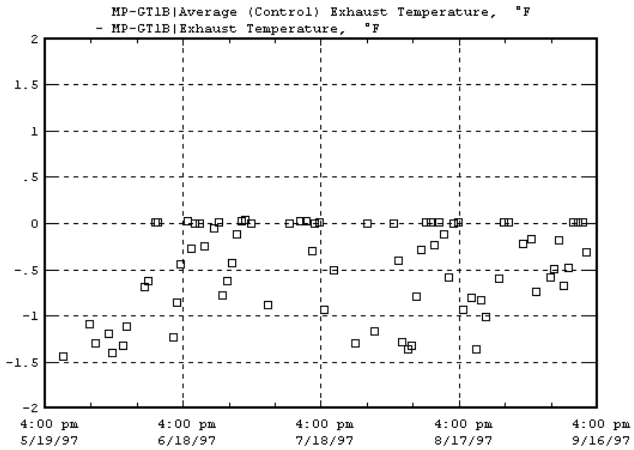

Figure 10 Deviation of average exhaust temperature from DCS control temperature signal [ºF] for unit B.

Exhaust temperature of gas turbine B, on the other hand, either corresponded to the control temperature or was one degree higher (Fig. 10). The exhaust temperature plot in Fig. 11 indicated a “cold” spot for unit B. The “cold” thermocouple 5 at the exhaust duct of unit B has the opposite effect of the “hot” ones in unit A; i.e. the control signal is lower than the actual exhaust temperature resulting in over-firing although the bias was not as severe in unit B as in unit A. Consequently, everything else being the same, an exhaust temperature difference of 4 degrees corresponds to a 7 degrees difference in firing temperature and roughly 0.6% difference in the power output.

On the other hand, when we factor in the 0.35% lower polytropic compressor efficiency for unit B and a 0.35% increase in turbine efficiency[3], this results in power output deficiency of roughly 1% for unit A and an decrease of 12 degrees in turbine inlet temperature[4]. Performance tests conducted in April and May of 1997 qualitatively agree with these observations based on the analysis of the stored data.

Unit A turbine section was overhauled in October 1998 during a major scheduled outage. This resulted in an increase in turbine efficiency as confirmed by a performance test conducted in November 1998. Furthermore, firing temperature of unit A was also increased by 11 degrees. Test results confirmed that unit A output increased substantially to the point that it exceeded unit B output by roughly 0.5 MW.

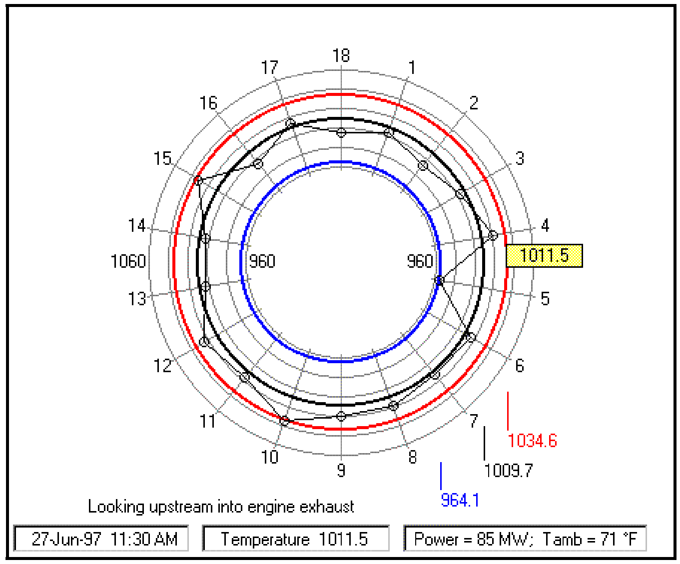

Figure 11 Exhaust temperature profile for turbine B.

6.4. Combustor Performance

An important diagnostic parameter that is calculated by the monitoring software is the shape factor. Also known as the profile factor, this parameter is the ratio of the maximum exhaust thermocouple reading to the average of all exhaust temperature thermocouples. It is a measure of the exhaust temperature spread that is the difference between highest and lowest thermocouple readings. Excessive and sudden changes or consistent upward trends in the shape factor may indicate combustor or fuel distribution problems. As can be seen in Fig. 12, unit A combustor shape factor displayed such an upward trend.

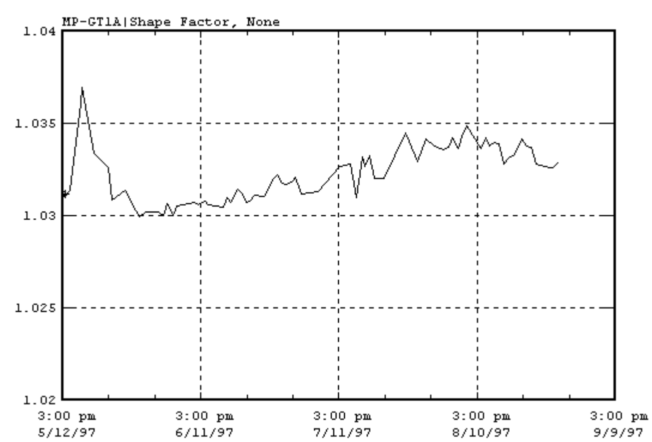

Figure 12 Combustor shape factor for unit A.

From the exhaust temperature map of Fig. 9, thermocouples 1 and 2 are identified as the highest and lowest reading sensors, respectively. The plot of the difference between the two thermocouples, i.e. the spread, shows a clear upward trend in Fig. 13. A combustor inspection

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[3] Note that turbine efficiency calculations require proprietary information and thus not included in the analysis algorithm. The assumption was based on the fact that a hot gas path inspection and maintenance was performed on unit B in April 1997.

[4] This analysis was carried out by using the GE7121EA gas turbine model of a commercially available heat balance software, GT PRO.

is required when this spread reaches a value of 70 degrees. The trend in Fig. 13 indicated that this particular level of deterioration would be reached in about 5 to 6 months.

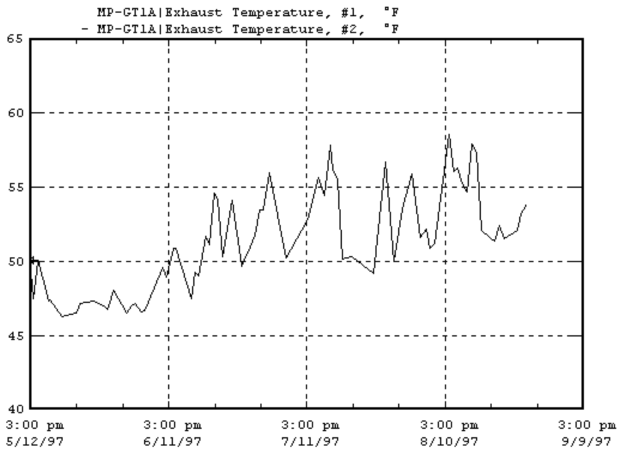

Figure 13 Exhaust temperature spread for unit A.

7. SUMMARY AND CONCLUSIONS

A real-time, on-line performance monitoring system was installed in a combined cycle cogeneration power plant with two Frame 7 gas turbines. The program runs continuously and unattended in a real-time, on-line mode to monitor two gas turbines. Collected data is stored in database files on a network and is available to local and remote users for historical plots, real-time monitoring, and other data analysis purposes. The software analyzes data for predicting off-line compressor wash and filter replacement scheduling. The software also calculates gas turbine parameters such as shape factor and compressor efficiency that can be used for on-line or off-line diagnostic as well as prognostic purposes.

The results highlighted several important aspects of gas turbine operation and maintenance:

- Regular on-line washing performed three times a week with demineralized water keeps compressor fouling at a minimum. No benefits were observed for using detergent in on-line washes. This is to be expected because the primary cleaning mechanism in the on-line wash is the water droplet impact on the first 5-6 row blades that loosens the deposits. There is ample evidence in the literature that for typical frame engines the degradation in output can reach 10% or more in 2,000 hours of base load operation. In the plant discussed herein the degradation in the same period was about 2%.
- The previously observed exponential law trend in output degradation was confirmed by the trends of this paper. In the absence of on-line washing, the mechanism was thought to be the stabilization of the thickness and shape of the blade deposits. In machines with regular on-line water washes, the mechanism can be the re-deposition of the dirt loosened from the front rows of blades on the back rows or a combination of the two.
- Over short term, performance degradation due to filter pressure drop is negligible as compared to compressor fouling. In this study, over 2000 hours of base load operation the lost generating capacity due to excess inlet filter pressure drop was measured roughly to be 70 MWh (at a rate of 25 kW per month). This translates to roughly 5% of the lost generation capacity due to compressor fouling in the same period (at a rate of 0.7 kW/h or 500 kW per month.)
- Inlet filter fouling is a long-term maintenance consideration. Most filters have a manufacturer recommended life of 2 years. In this project it was discovered that excess inlet pressure loss is one consideration in deciding whether to replace the inlet filter not. It is also important to ensure that filter elements are healthy enough to perform their job of blocking microscopic particles contributing to compressor blade deposits.
- An important performance degradation source can be the combustor problems. This was demonstrated by the under-firing of unit A. This problem, if not remedied, could lead to an annual lost generating capacity equivalent to 7800 MWh at base load. Most gas turbine controllers such as GE's Speedtronic Mark IV are able to detect and discard bad thermocouple readings to generate a reliable control signal. However, if the individual deviations (due to possible combustor problems) are not excessive, the resulting bias of a few degrees can still result in significant loss of output. Thus identifying and continuously trending exhaust thermocouple spread can pinpoint problems resulting from presently mild deviations that can lead to potentially serious engine problems or result in lost revenue.

In order to put these results into dollars and cents, consider an 80 MW base load unit with a service factor of 80% and power sale price of 3¢/kWh. The fuel cost is assumed to be $2/MMBtu and the total cost of an off-line wash is estimated at $10,000 including labor, materials and down-time. Regular on-line washing would recover $200,000 worth of power generation capacity. Compressor fouling, if not remedied by off-line washes, would result in $300,000 worth of lost power generating capacity annually assuming a 2% capacity factor loss with an exponential law trend. Performing an off-line compressor wash at intervals of 1,500 hours would save the operator $20,000 annually.

The savings will increase with increasing engine size, service factor and fouling rate. The savings will be directly proportional to power sale price and inversely proportional to fuel costs and off-line wash expenses. For the example numbers cited above, a software package that would estimate the optimum off-line wash interval would pay for itself in six months at a licensing cost of $10,000. This pay-back period would be proportional to the savings if only compressor maintenance is taken into consideration and could be even much shorter if the software helps to shed light on situations similar to that described in §6.3.

For a discussion of the actual benefits of the system in the site of installation the reader is referred to Gülen et al. (1999).

8. REFERENCES

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APPENDIX

The following relationship exists between the efficiency and power output of a gas turbine:

$$\frac{1}{\eta} \propto \frac{f\,LHV}{P} \tag{A.1}$$

where η is the engine's thermal efficiency and LHV is the lower heating value of the fuel. With some algebra, one can show that

$$\frac{\Delta f}{f_0} = \frac{\left(1+\dfrac{\Delta P}{P_0}\right)}{\left(1+\dfrac{\Delta \eta}{\eta_0}\right)} - 1 \tag{A.2}$$

where subscript $_0$ denotes the new and clean state of the engine and Δ denotes a change in the particular quantity. Compressor degradation results in a decrease in power output ($\Delta P < 0$) and an increase in heat rate. Since the heat rate is equivalent to $3412/\eta$, this translates into a decrease in the engine's thermal efficiency ($\Delta\eta < 0$). Thus, both terms in parentheses on the RHS of Eq. (A.2), as well as their ratio, are always less than unity because $|\Delta P|$ is always greater than $|\Delta\eta|$. Equation (A.2) then implies that Δf is always less than zero, i.e. fuel mass flow rate at full load decreases with increasing compressor fouling.

This conclusion can also be reached qualitatively starting from the fact that, since a fouled compressor has a lower efficiency and higher discharge temperature, a smaller amount of fuel is required to reach a set firing temperature.

In monetary terms, the total cost of engine degradation due to compressor fouling over a time period of τ is

$$\Delta C_{tot} = \left(C_f \Delta f - C_P \Delta P\right)\tau \tag{A.3}$$

Equation (A.3) suggests that the total cost can be negative, i.e. one could actually even save money due to compressor fouling. This would happen in the possible –albeit unlikely- event that

$$\frac{C_f}{C_P} > \frac{\Delta P}{\Delta f} \tag{A.4}$$

In order to give a numerical example, consider a typical frame 7EA with 84,920 kW output and 10,212 Btu/kWh heat rate. When compressor fouling leads to a pressure ratio decrease of 2%, engine output will drop by 5% (i.e. about 4.25 MW) whereas heat rate will increase by 2% (e.g. see figure 23 in Hoeft, 1993.) This implies a 3.1% decrease in fuel mass flow rate, i.e. about 27 MMBtu/h. Thus if C_f in [$/MMBtu] is greater than 158 times C_p in [$/kWh], there will be a saving due to this degradation. In other words, if the power sale price is 3 cents per kWh, the owner will save money because of compressor fouling if the fuel purchase price is higher than $4.74 per MMBtu. The same will hold true if the power sale price is less than 1.6 cents per kWh if the fuel purchase price is 2.5 $/MMBtu.

Note that these statements hold only for a gas turbine operating at full load. Same kilowatts as in a new and clean engine can always be produced at the expense of extra fuel consumption in a degraded engine running at part load. In that case $\Delta P = 0$ in Eq. (A.2) such that

$$\frac{\Delta f}{f_0} = \frac{1}{1+\dfrac{\Delta \eta}{\eta_0}} - 1 \tag{A.5}$$

Since $\Delta\eta/\eta$ is always negative and less than unity in magnitude, Δf is always greater than zero.