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GAS TURBINE AND DRIVEN MACHINERY MANAGEMENT AND DIAGNOSTICS

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ABSTRACT

The size and complexity of gas turbines has evolved tremendously over the years and the controls, instrumentation and diagnostics tools have kept pace with the advances. This paper discusses the progress of the tools to keep these complex machines running for continued reliability, efficiency, emissions compliance and power output. The technology to enable the user to manage their machinery on site and remotely will be discussed in this paper along with the benefits added by the technologies.

INTRODUCTION

As gas turbine development has continued to deliver ever increasing, efficiency, emissions compliance, reliability and power output, the oversight and expertise required to maintain these systems has grown in complexity. The changing operating environments for the gas turbines and their driven equipment are forcing users to adopt advanced techniques to remain competitive by allowing turbines and the plants in which they operate extend outages, inspections and overhauls.

One outcome of this complexity is the margin for error or deviation has become smaller to maintain compliance or competitiveness. The tools used to manage and monitor the equipment have needed to keep pace. Often these tools begin to take a plant level of monitoring instead of being based at the machine level.

Users have been exposed to the following tools over the years and using them wisely and in combination with one another can lead to positive results.

- Environmental Protection
  - Emissions Monitoring Systems
  - Dynamic Pressure (Combustor Acoustics)
  - Hazardous Gas Detection
- Economic-Performance
  - Thermodynamic Performance Monitoring
  - Temperature
  - Remaining Lifetime Calculations
- Machinery Monitoring and Protection
  - Shaft Relative Radial Vibration
  - Casing Radial Vibration
  - Shaft Absolute Radial Vibration
  - Thrust Position-Axial Vibration
  - Shaft Eccentricity (Shaft Bow)
  - Zero Speed
  - Over Speed Detection
  - Remote Monitoring Connectivity

Sometimes these tools can detect and identify anomalies on their own. More often it requires a combination to properly identify and develop corrective actions. It is therefore imperative that the organization must have access to expertise in these technologies to assist in their use and integration. For most operating units, an economical and reliable source of expertise that can interpret and correlate the differing technologies with one another may require remote access. In the current industrial environment, expertise is becoming limited in the expanding global sites. Remote monitoring and diagnostics systems are often implemented to allow for Original Equipment Manufacturers (OEM) and other third party service providers to access the operational and behavioral data over high-speed links to assist the local teams on necessary actions.

This paper will describe the use and interdependence of some of these technologies that operators may utilize to maintain their plants.

NOMENCLATURE

CEMS	Continuous Emissions Monitoring Systems
PEMS	Predictive Emissions Monitoring Systems
EPA	Environmental Protection Agency
psi dpp	Pounds per square inch derived peak to peak
%LEL	Percentage of Lower Explosive Limit

ENVIRONMENTAL PROTECTION

Emissions Monitoring Systems

Currently, CEMS are mandated in many industrial countries for large stationary emissions sources[1]. The recent trend is toward expanded emissions monitoring for smaller emissions sources and offshore sites. While a CEMS affords many advantages, including continuous and direct measurement of emissions, it is costly; more so in remote or space constrained applications. PEMS are emerging as an alternative to CEMS. The EPA has been paving the way for this alternative by publishing the *Performance Specification 16 for Predictive Emissions Monitoring Systems and Amendments to Testing and Monitoring Provisions* [2]. The PEMS approach uses process measurements correlated to emissions and/or emissions models to compute the expected emissions values. Portable CEMS instruments are typically used to obtain periodic data for initial model calibrations. PEMS can approach the accuracy of a CEMS for as little as one-third the cost while adequately meeting the specific needs of the application.[3]

In Europe, there are various requirements (some based on the Kyoto Protocol[4]) with respect to gas turbine emission monitoring. The most important of these for the offshore industry is the Offshore Combustion Installations (Prevention and Control of Pollution) Regulations.[5] These regulations, and others in various stages of development or implementation, will require offshore operators to monitor, self-certify, and justify emissions of NO_X, CO, CO_2, SO_2, and unburned hydrocarbons to their country's Department of Trade and Industry (DTI).[6] An additional example can be found in Norway where a PEMS solution will be accepted as part of offshore platform emissions monitoring program to begin in 2011 (based upon author communication with Climate and Pollution Agency).

As the regulatory environment continually evolves, trading emissions is likely to become more prevalent. One can imagine the obstacles to trading emission credits based upon "predicted" levels. One would be unlikely to enjoy paying for a tank of petrol based upon the predicted amount that will be placed in the tank. Similarly, environmental organizations, corporations and countries will be hesitant to put such large financial stakes behind predicted emission levels. This requires that PEMS deliver reliable, accurate and verifiable measurements.

Desirable attributes of a PEMS system include:

- Precision and accuracy approaching conventional CEMS systems;
- Ability to utilize energy and mass balance computations to generalize its predictions;
- Sensitivity to major process inputs such as power load, ambient conditions, fuel composition and temperature, and total volumetric flow rate;
- Ease of updating/calibrating the algorithms.

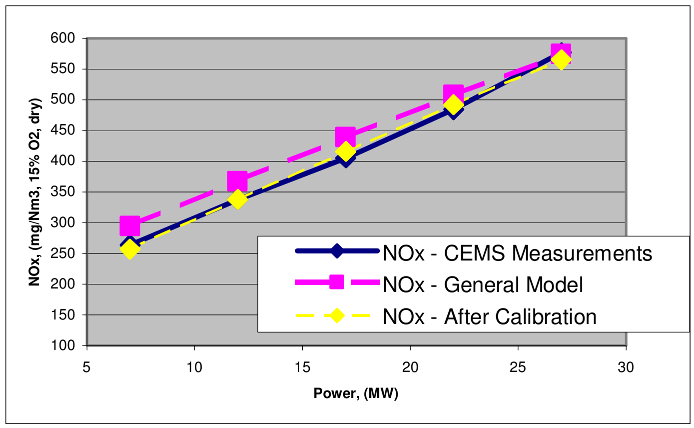

Figure 1 PEMS Model Calibration

A recent PEMS was developed using an emissions model based on a popular aero-derivative gas turbine used on offshore platforms. Figure 1. This model was developed in close cooperation with the OEM. The model calculates the gas turbine performance and predicts NO_X, CO, CO_2, SO_2 and unburned hydrocarbon generated based on ambient conditions, fuel composition and machine operating conditions, all while taking into account real time degradations. Models can be rapidly updated when new emissions test data is available. Typical accuracy limits are within 5-12% relative error compared to CEMS data although the data above shows possibilities to achieve greater accuracy with great care.

Temperature

Temperature is one of the basic measurements that are used prevalently in operating units. Bearing temperature, oil temperature rise across a bearing and oil drain temperature are all common references used to indicate mechanical issues.

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[1] http://www.epa.gov/ttn/emc/cem.html
[2] http://www.epa.gov/ttn/emc/cem.html
[3] http://www.automation.com/content/ge-energy-introduces-pems-solution-for-offshore-platforms-at-one-third-the-cost-of-similar-accuracy-cems
[4] http://unfccc.int/kyoto_protocol/items/2830.php
[5] http://www.legislation.gov.uk/uksi/2007/938/contents/made
[6] http://www.dti.gov.ph/dti/index.php?p=134

Changes in these values or abnormally high values should be investigated and correlated with load or other operation changes. Often, bearing temperatures changes can be compared to mechanical vibration changes as part of a root cause analysis. Gas turbines and especially industrial gas turbines have elevated bearing and oil temperatures (110 deg C or higher) due to the high temperature operating environment created by the turbine itself in addition to heavily loaded bearings due to the massive rotor.

Exhaust gas temperature (EGT) differential profiles, Figure 2, can indicate problems with the combustion system. These may be issues with fuel nozzles, combustion and cooling airflow, fuel quality, and combustor degradation among others. While the effects of this may immediately impact the lifetime viability of a hot combustor, transition piece or stator vane, cool sections may adversely affect the thermal expansion of a section of the circumference and cause distortion of the stator assembly. While this varied thermal growth can cause issues with the components themselves, it can also result in case distortion and alignment issues that can produce mechanical degradation of the rotating and stationary elements.

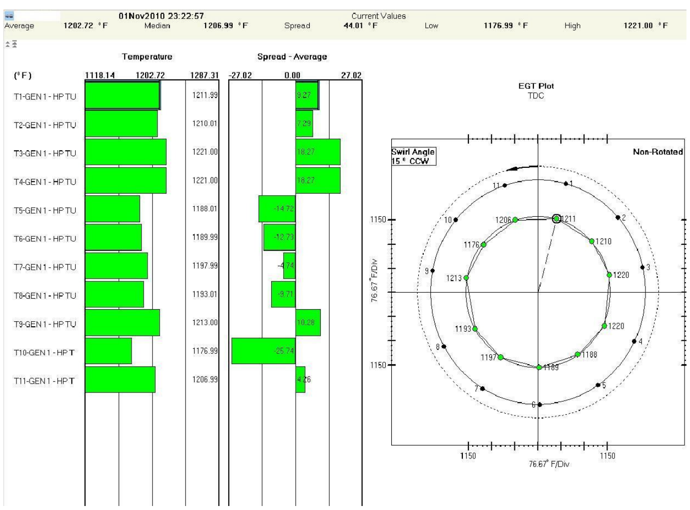

Figure 2 Aero-derivative Exhaust Gas Temperature Plot

Further, a hot spot or cold spot in the EGT spread may affect the emissions profile of the unit. As discussed in the PEMS description, many operators are being held to (or will be held to) tight operating limits on pollutants. An isolated hot section in the combustion system may produce more NOx[7] than is allowable, even though the average temperature spread is within normal limits.

Problems with combustion zones and fuel to air mixtures that may be due to combustion acoustics will likely present themselves in an uneven distribution in the EGT profile. Longer-term effects from prolonged combustion acoustic exposure may include combustion liner degradation,[8] and turbine blade or stator vane erosion that can manifest in EGT anomalies.

Dynamic Pressure (Combustor Acoustics)

While there are many types of combustor acoustics and stability issues, the most common is a flame stability issue sometimes termed “Humming”. With modern Dry Low NOx combustors, the fuel to air ratio has become very lean. This lean combustion can lead to conditions that set up an acoustic resonance in the combustor can, silo or annulus. If left un-checked, this resonance can create pressure pulsations in the combustor and/or transition piece large enough to create mechanical damage to the components. Once these components are liberated and are ejected downstream, severe mechanical damage can result.

Each gas turbine has different acoustic resonant frequencies of concern. The individualities usually boil down to the type of combustor for that specific machine. A similar model machine may have different kits for combustion depending on customer requirements and government regulators for emissions. All OEMs know what the frequencies of interest are for each model of combustor and can set up the monitoring systems to filter the dynamic pressure sensor signals for each frequency of interest. Often there are two or three frequencies of concern and band pass filters are set around them. During commissioning, the combustion control engineer will tune the fuel to air ratio and the combustion sequencing through different zones and nozzles to avoid operation that would excite the acoustic pulsations.

Figure 3 shows the process that a combustion engineer might face. The data on top is the band passed (120-1000 Hz) dynamic pressure signal with the Gas Generator Speed Reading on bottom for reference. It is observed on Startup that the Gas Generator experienced 7.3 psi dpp of pressure pulsations. The Waterfall data, Figure 4, shows the dominant frequency at ~300 Hz. The combustion system was tuned to eliminate these pulsations at these speeds. Similarly, it is observed that pulsations are experienced at different operating speeds and those conditions were also tuned out for stable long-term operation.

[7] Strakey, P., Weiland N., Richards G., *The Gas Turbine Handbook,* National Energy Technology Laboratory, United States Department of Energy. http://www.netl.doe.gov/technologies/coalpower/turbines/refshelf/handbook/TableofContents.html pp 203-207

[8] Oyediran, A., Darling, D., Rahakrishnan, K., *Review of Combustion-Acoustic Instabilities,* 31st Joint Propulsion Conference and Exhibit. 1995

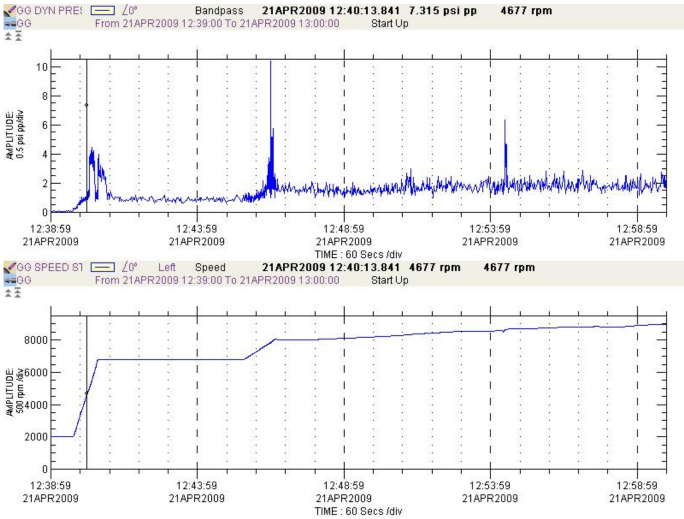

Figure 3 Aero-Derivative Combustion Acoustic Tuning Trend of Pressure pp and Gas Generator Speed

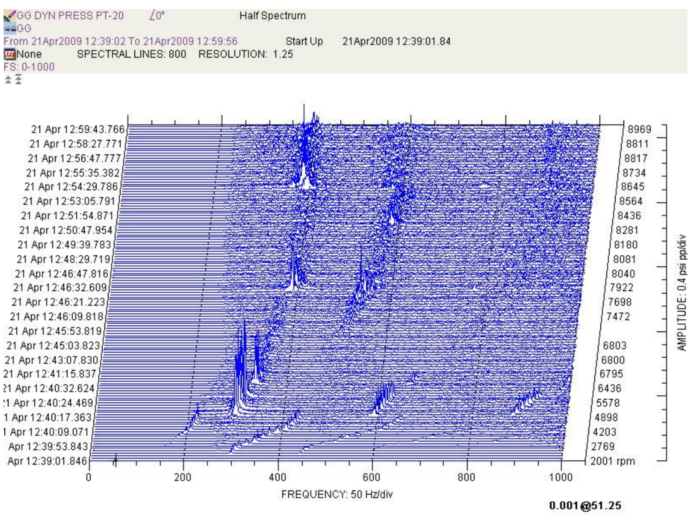

Figure 4 Aero-Derivative Combustor Acoustic Tuning Waterfall (Frequency Content)

Many systems are installed permanently since gas turbines need to be retuned for differing ambient conditions and fuel qualities and types. Experts can do remote tuning with access to the dynamics information as well as fuel flow, airflow, and other load conditions with the proper remote connection.

With the differing supplies of natural gas coming from new sources as well as LNG imports, fuel quality conditions such as the Wobbe Index can vary significantly more than previous variations seen by gas turbine operators.[9] On-line combustion dynamics monitors take on a role of more importance as these changes can cause a gas turbine to experience acoustic pulsations un-expectedly. The on-line system can be programmed to communicate with the control system to change the load on the unit to avoid the combustion acoustic pulsations that can be damaging to the unit. If necessary a shutdown can be initiated under extreme conditions.

Hazardous Gas Detection

The extreme power produced in relatively small packages delivers new challenges in protecting machinery and personnel. Hazardous gas detection systems are installed in and around gas turbine compartments to detect methane that may have escaped and created a hazardous gas environment that could lead to an un-expected explosion. These sensors are mandated by insurance and regulatory agencies and are most often with two and sometimes three probes in each location for redundancy. Once a level of gas is detected (%LEL) an alarm is issued and often the unit is tripped. These sensors are often very sensitive continuous catalytic bead technology based. These sensors must be designed to retain the calibration parameters over varying temperatures and time from last calibration, as many hazardous gas sensors are prone to drifting.

Due to the important safety considerations for explosive environments industrial practice dictates that the system be calibrated every 90 days due to the current catalytic bead technologies propensity to drift and become polluted. In many instances, this requires that the probe be removed from service or access to the probe be granted (which may require a machine shutdown) and a calibration gas be applied to the sensor head and the monitor readings validated. Two industry leaders supplying these systems have designed a remote calibration unit that can remotely apply calibration gas and clean air gas without removing the sensor or stopping the machine. Since the probes are often installed in a redundant fashion, calibrating the probes remotely can allow the unit to continue running. In addition, some of the sensors may require scaffolding or other access means to apply test gases to the sensors and this requires a prolonged shutdown to the unit for seemingly brief validation.

Other environments that use hazardous gas detection systems are generators (hydrogen cooled), hydrogen compressors, and pipeline compressors.

ECONOMIC-PERFORMANCE

Thermodynamic Performance Monitoring

Thermodynamic performance calculations in industrial use have been a common way to determine the condition of the machinery. Historically, calculations were done punctually by hand or even in a spreadsheet. On-line performance monitoring systems have become more prevalent in recent years and are offered by OEMs and third party vendors. Gas turbine performance calculations can be challenging given that the ambient conditions and part load conditions can change the performance characteristics of the unit. One way to account for the changes in initial conditions for performance

[9] Healy, T., Frederick, G., Tuning for LNG on the Fly, *Turbomachinery International* September/October 2007 Vol 48 No. 5

calculations is to make corrections for the changing conditions. While most performance monitoring systems use correction curves for items such as relative humidity and ambient temperature, some use actual first principle models to model the turbine.

For turbine generator applications, generator reactive capability curves are used to evaluate the operating point and efficiency of the generator. For combined cycle applications, the whole bottoming cycle of the HRSG, steam turbine, condenser, and cooling tower must be included in the calculations for thermodynamic performance.

For turbo-compressor applications, the compressor power must be calculated to not only determine the compressor performance parameters but also to determine the power delivered by the gas turbine. It is becoming more common that torque meters are placed on the coupling between the driver and driven machines to determine power delivered and power absorbed. This is helpful since many times field instrumentation can be suspect and using a compressor performance calculation to determine the gas turbine performance can be troublesome with typical in-accurate field instrumentation whose calibration is rarely validated. Often this instrumentation is not used for control and lays victim to reduced personnel capacity to validate non-essential sensors.

Once performance parameters are known, decisions about operations can be made with intelligent information. In a combined cycle power plant, the performance monitoring system should be able to make recommendations on a load profile for the power block based on dispatched load and existing degradation of the assets. For example, if a 2X1 power block is dispatched at 500 MW, the performance model system should be able to model how to best configure the GT1, GT2, ST1 and possible duct burners to maximize heat rate and minimize fuel consumption.

The performance monitoring system can also advise on the most economical wash schedule for the gas turbines and possibly the driven compressors also. This system should be able to determine what is recoverable degradation, which is often due to fouling and what is not recoverable due to surface finish of rotating and non-rotating components as well as seal degradation.[10]

The correlation of degradation in performance that may be due to seal rubs and mechanical manifestations may lead to operational limitations or other considerations such as turning gear and eccentricity settings to prevent seal rubs. This presents another instance where combining monitoring technologies and diagnostic methodologies can lead to more efficient and effective operations. If the mechanical behavior: un-balance response due to residual unbalance or a bow in the rotor upon startup or shutdown can be managed, the lifetime of the seals and other components critical to the performance of the machine can be extended.

Remaining Lifetime Calculations

Users do combustion inspections and turbine maintenance based on time intervals, operating hour intervals, equivalent operating hours intervals, and number of starts and stops and the condition at which the starts and stops occurred. Many OEMs provide the guidelines in the form of equations etc. to make the evaluations on when to perform maintenance. This data can be put on line and alert users on when an inspection is forthcoming. Some research has been done on Generalized Equivalent Operating Hour (EOH) and Equivalent Start (ES) Calculations Equation 1, Equation 2, [11],[12],[13]. This research suggests that there may be some simplifications in linear models for remaining lifetime calculations. However, many users have expressed interest in validating any remaining lifetime calculations with actual inspections to confirm that extending inspection (or shortening inspection) based on the actual condition or predicted condition of the machine. Once the validation of the calculations has been done, users would then have the confidence to trust the results of remaining lifetime calculations without physical inspections and maintenance. Often incidental mechanical issues can be introduced by the execution of these operations, so opportunities to avoid these inherent labor and parts costs and downtime, can also decrease the susceptibility to errors due to the inspections and maintenance.

$$EOH=\left\{\sum_{i=1}^{\#\,of\,category}\left(c_{gas}^{normal}\times c_{gas}^{peak}\times c_{gas}^{part}\times c_{fuel}^{other}\times c_{injection}^{water}\right)\times H_i\right\}+f\cdot ES$$

Equation 1 Generalized EOH Calculation

$$ES=g\cdot N_{starts}^{normal}+h\cdot N_{starts}^{part}+i\cdot N_{starts}^{peak}+j\cdot N_{starts}^{Emergency}+k\cdot N_{starts}^{fast}+l\cdot N_{trips}^{normal}+m\cdot N_{trip}^{low}$$

Equation 2 Generalized ES Calculation

Where

H = Categorized Operating hours

N = Number of cycles

c, f-m = Constants, some of which may be set to zero by some OEMs

*Each constant can be split into sub-categories by the OEM (i.e. fuel type, water injection, trip severity, etc)

[10] Meher-Homji, C., Matthews, T., Pelagotti, A., Weyermann, H. B., 2007 *Gas Turbines and Turbocompressors for LNG Service*, Proceedings of the Thirty-Sixth Turbomachinery Symposium, pp125-128.

[11] Townsend, R., Winstone, M., Henderson, M., Nicholls, J.R., Life Assessment of Hot Section Gas Turbine Components, Proceedings of a Conference Held at Heriot Watt University, Edinburgh, UK, October 1999.

[12] Pollard, M.A., Angello, L., On-Line Combustion Turbine Reliability Monitoring System Field Evaluation Results, Proceedings of ASME Turbo Expo (ASME Paper 2001-GT-0033), June 2001.

[13] Cooke, L. A., Component Management Programs for Reliable Life Extension Proceedings of ASME Turbo Expo (ASME Paper 2001-GT-0431), June 2001.

Correlating the remaining lifetime calculations with other technologies such as combustion acoustics, EGT profiles and performance can give the operators better confidence in the condition of their machines and managing the lifecycles of these assets.

MACHINERY MONITORING AND PROTECTION

Shaft Relative Radial Vibration

Shaft relative vibration, as we know it today has been around since the advent of the shaft rider systems and has become much more prominent since the application of the shaft relative proximity probe. Since the measurement is made with the shaft relative to the probe mounting, which is usually the bearing backing, this measurement must be interpreted as such on gas turbines with fluid film bearings: industrial gas turbines. Since most industrial gas turbines and especially large frame series, have a very heavy rotor with respect to the housing and casing, this data must be correlated with the casing vibration. More on this topic will be covered in the Shaft Absolute Radial Vibration section.

Aero-derivative gas turbines that have rolling element bearings generally do not have a shaft relative measurement since there is almost no shaft vibration relative to the casing. There are also installation limitations that make this measurement difficult in most installations.

The definition of this measurement is sufficiently described in the API 670[14] standard and it is assumed that most readers are familiar with this measurement in XY configuration. The API 616[15] standard defines the instrumentation specifically for Gas Turbines. The shaft relative measurements are very important when starting up and shutting down machines. Gas turbines are among the machines that exhibit the most temperature difference between startup and coastdown due to the extreme temperatures. These temperature differences are inherent due to the fact that the turbine aspires air at ambient conditions and compresses it to temperature ratios that can exceed 20:1. This compression heats the air and the combustion process elevates the temperatures to firing temperatures that can be on the order of 1600 deg C. Although the metal temperatures of the turbine do not nearly reach this temperature, the exhaust bearing, turbine housing and supports are subjected to extremely hot temperatures as compared to the compressor inlet and even the compressor discharge.

The data presented here as a hot coastdown as a baseline and a subsequent startup is shown in Figure 5. Often for analysis, a baseline will have a different color (red in this case) or line format (dashed etc.) than the most recent data of interest.

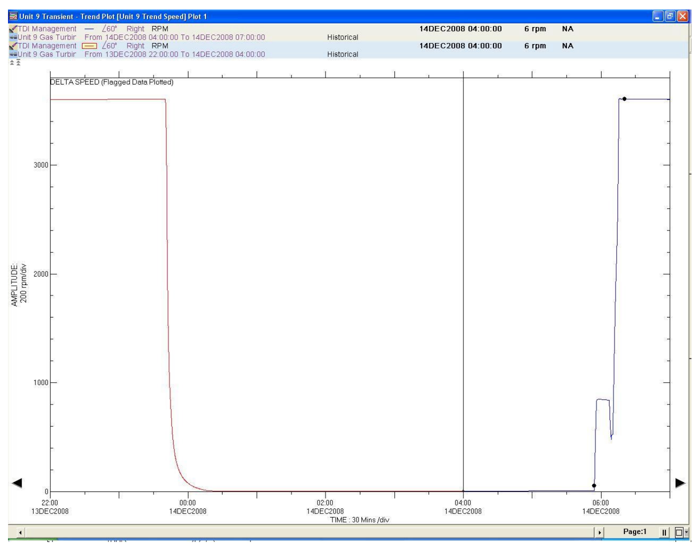

Figure 5 Speed Trend of Hot Coastdown (red) and Warm Startup (blue)

This extreme change between startup and shutdown often has significant effects on the balance condition, alignment, shaft stiffness, bearing stiffness, support stiffness, internal clearances and a myriad of other changes that happen as the unit undergoes its heating cycle. It is important to compare the

[14] American Petroleum Institute API 670 Standard for MACHINERY PROTECTION SYSTEMS

[15] API 616 GAS TURBINES FOR THE PETROLEUM, CHEMICAL, AND GAS INDUSTRY SERVICES

startup and coastdown profiles to characterize the differences. Differences can manifest with different unbalance direction response, different alignment issues, growing or reducing internal clearances, thermal expansion of probe supports (affecting shaft centerline readings), and differing resonant frequencies among many other changes.

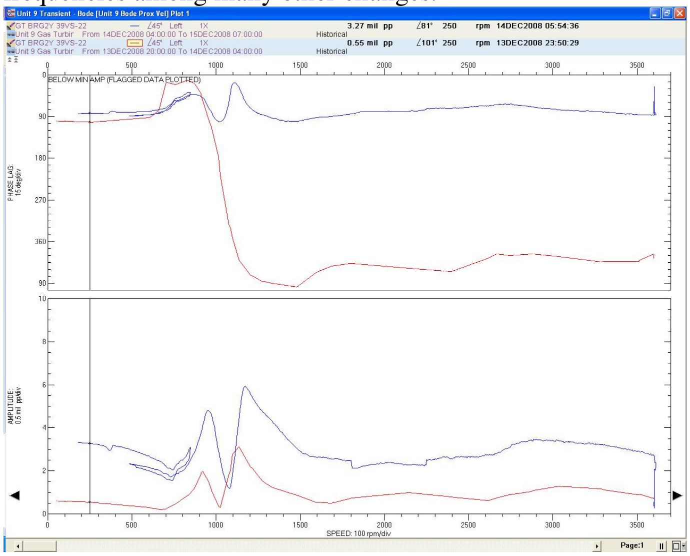

Figure 6 Data Presentation in Bode Format: Startup (Blue Curve) with Shutdown Baseline Overlay (Red)

Figure 6 depicts the difference between a typical and healthy coastdown after a day of power generation for a large frame series gas turbine. The subsequent startup shows a significantly larger 1X amplitude at slow speed (upwards of 4 mils pp) and almost twice the response in vibration through the startup phase of the engine. It would be evident from a simple inspection that the startup labeled 14 Dec 2008, which is the blue curve has a significant difference from the chosen baseline shown in red.

Figure 7 displays the same data set for the shaft within their estimated tilt pad bearing clearances as the machine goes from slow speeds to running speeds. This machine is a cold end drive gas turbine with bearing 2 as the exhaust and bearing 1 (coupling end) as the compressor inlet. It is observed that from the hot coastdown condition (red data presentation) it appears that there is not a significant difference (only 2-3 mils at full speed) between the baseline coastdown (red curve) and the subsequent startup curve (blue curve). The compressor end shows a little more sensitivity between startup and coastdown, as the startup ending position within the clearance is almost 4 mils lower than the hot coastdown. This may seem a little counter-intuitive as one would expect the large temperature difference between a hot coastdown and warm startup more so in the turbine section than the compressor section, but reflection on how the probes are mounted in each section may explain the difference. The probes mounted in the turbine section are mounted directly to the bearing face inside the oil cooled bearing housing. The probes mounted in the compressor section are mounted with an approximate 10-inch long probe extension on the bearing cover. This longer probe extension with possible thermal expansion differences in where they are mounted likely explains the difference.

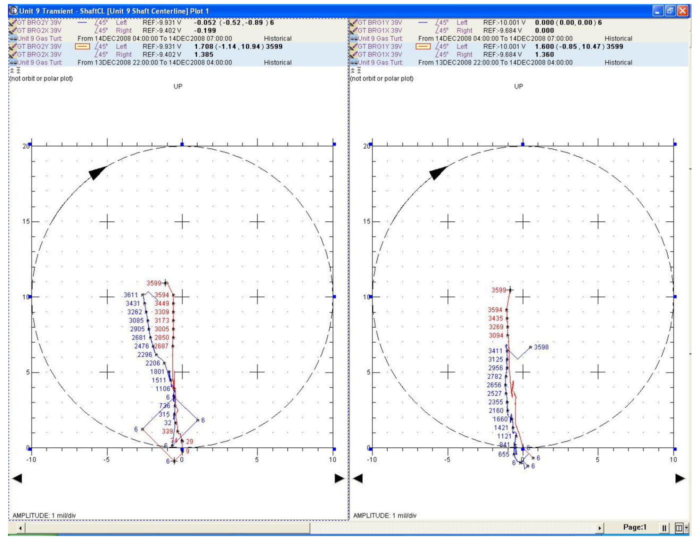

Figure 7 Data Presentation in Shaft Centerline Format: Startup (Blue Curve) with Shutdown Baseline Overlay (Red)

Once these differences have been established, a baseline should be determined for each a startup and coastdown. These may include a cold or warm startup, and a hot or warm coastdown. Once a baseline is established for each condition and it is determined normal and healthy, it can be compared to each successive startup or coastdown using overlay features. Since gas turbines rarely run for more than 90 days in the current operating environment, there are numerous opportunities to compare these transient speed events to a baseline.

If the site lacks the expertise to determine that a baseline is normal and healthy, a remote connection may be established that can link the data to an expert who may know the typical behavior for that make and model could establish a baseline.

If mechanical behavior as compared to baseline starts to deviate significantly, there may be issues with changing balance conditions, changing alignment or foundation issues, or other conditions that may affect the startup and coastdown behavior. The site personnel will likely have the expertise to compare the subsequent startups and coastdowns to the baseline and notice any significant changes as described above by simple curve comparison of the baseline to the most current data.

Along with the casing radial vibration and shaft absolute vibration, the shaft relative vibration should be monitored for changes during normal operation. Most often, the parameter surveyed over time is the overall vibration. Changes in overall vibration can be sensitive to load or even changes in ambient condition. The parameter that influences the overall vibration is often the 1X vibration. For a gas turbine driving a generator, both units could experience changes in 1X vibration due to normal (or abnormal) heating and cooling of the rotors as they run through their load profiles. A gas turbine may have a stacked rotor that changes its balance due to the extreme changes in temperature along the rotor and a generator may have a sensitivity to balance condition due to the MW or MVAR load demanded. Once this load profile is determined, it should be noted for future comparison. If any change in balance condition or load sensitivity is noticed, an investigation should follow. Some common issues found with gas turbines can be clogged cooling channels in turbine section, blade material loss and shifting rotor disks. The extreme firing temperatures generated by modern gas turbines necessitate film cooling and serpentine cooling of turbine stator vanes, which poses challenges to keeping the cooling air supplied and channels open. Issues with generators can be shorted windings on the generator rotor or other electrical issues.

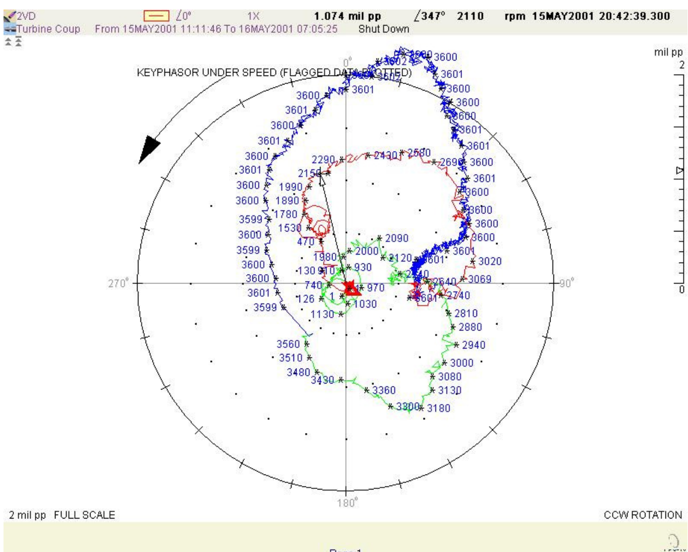

Figure 8 1X Vibration in Polar Format (Green Startup, Blue Loading and Unloading and Red Coastdown.)

Figure 8 shows a good example of the difference between a cold startup and a hot coastdown. It appears from the figure that during the loading of the unit to full load: ~80 MW, the balance condition of the unit changes as indicated by the approximate 3 mil pp change in vibration in the blue "Load Vector". While this vibration change looks impressive, the total vibration never exceeds 2.5 mils pp and is likely suitable for continued operation, even in a daily cycling mode.

While the diagnostics information for the shaft relative vibration is often the richest in content and ability to get to root cause of mechanical issues, it is often not used for machinery protection or automatic shutdown.

Casing Radial Vibration

Most gas turbines shut down on casing vibration, which should lead users to the greatest scrutiny of the systems. Often on large frame industrial units there are redundant sensors and monitoring channels for more reliable detection of mechanical issues. Casing transducers are either an accelerometer or a velocity transducer and usually have to be of the high temperature variety if they are mounted toward the turbine section of the unit.

A risk in using this methodology solely for machine protection is that the mechanical forces generated by the gas turbine rotor must be transmitted faithfully to the casing. It has been witnessed that shaft relative vibration upwards of 15 mils pp while casing levels remained under trip levels for industrial gas turbines.

Industrial gas turbines are often monitored for protection using velocity measurements on the bearing or bearing housing caps. It is also important to correlate this data with a Keyphasor® probe to ensure that startup; coastdown, steady state and nX vibration is trended. Much like the changes in shaft relative vibration from run to run and like operating loads are compared to one another, casing vibration should also remain consistent for the same conditions. Changes is casing radial readings can indicate degradations in alignment, foundation integrity, balance and support stiffness among others.

Accelerometers are often used on aero-derivative gas turbine units since their high temperature reliability in the high acceleration environment have proven to be more robust than moving coil velocity transducers. The acceleration signals are often integrated to velocity for band-pass filtered measurements that range from 25 Hz high pass to 350 Hz low pass depending on the OEM specifications for protection. Some OEM specifications require the signal be integrated to displacement and filtered to the running speed of the rotor(s). Most consider this measurement to be a safety measurement that prevents a catastrophic failure to the unit much like their aviation brethren.

The industry has found it difficult to use these signals for condition monitoring and forecasting maintenance.[16] Using measured vibration levels and oil condition has proved useful in many situations. Oil temperatures can often be an indicator of impending failures as well as particle or "chip" detectors that attempt to detect metal particles from bearings or other debris in the oil. Once these oil parameters are detected, often the unit is shut down for inspection.

[16]Harker, R.G., Betts, R.G., Hrabec, V., Swan, P., Using Internal Vibration Transducers to Improve Fault Detection on Aeroderivative Gas Generators, 46th Annual Gas Turbine Users Association Conference, 2001.

Many of the aero-derivative gas turbines do not have Keyphasor transducers on all their shafts. The most common speed measurement is made through a gear or set of gears, which loses the advantage of having a consistent physical phase reference on the shaft. However, many modern diagnostic systems can take the gear teeth or gear ratios and calculate a once per turn signal that can be used for startup, coastdown and even steady state signal analysis. Once the speed sensor measures the gear assembly signal, often a gear teeth combination or gear ratio of sufficient significant digits can be entered into the system for a reliable absolute phase for that run. That is to say that once a machine is started, the absolute phase reference is valid for the entirety of that run. Once the machine is stopped, the phase assembly of the speed gear changes and the absolute phase measurements are not repeatable from run to run. Figure 9 shows a constant speed data prior to a coast down in blue and constant speed data of a subsequent start up in red. It would normally be expected that the 1X amplitude and phase would be consistent run to run at similar conditions, but in this case, the system loses the gear assembly phase while it is stopped and a different resulting phase is observed. This is helpful in determining if the balance or other mechanical issue of an aero-derivative gas turbine is changing within a run. In addition for a consistent phase reading for a run, the startup and coastdown data can be collected to identify modal unbalance and resonance frequencies from run to run.

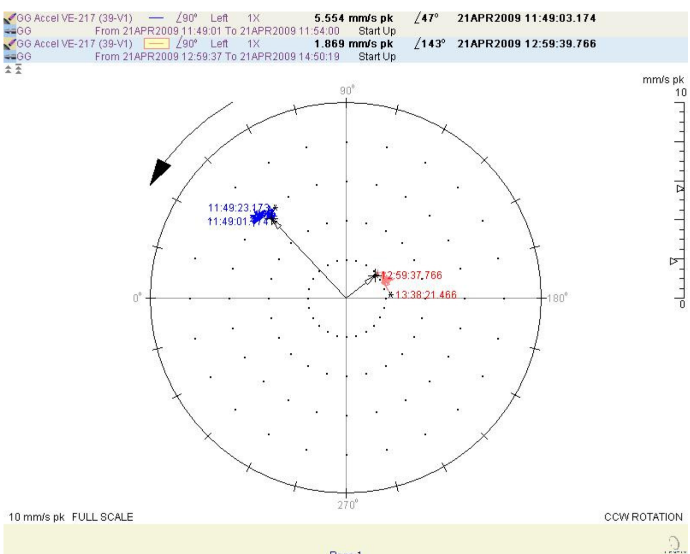

Figure 9 Generated Once-per-turn Reference from a 47X Speed Signal

It is important to record and trend the broadband signals if available. Most often, the raw and unfiltered acceleration signal is available. Less common is the availability of the velocity signal in the unfiltered version. Unfiltered data can be used to do analysis of blade and/or stator vane passing frequencies and combustion acoustics frequencies for example. Some of the first instances of combustion acoustics detection were found by analyzing data from the casing transducers on the unit.

Shaft Absolute Radial Vibration

Since the shaft absolute radial vibration signal is a heavily treated signal (integrated casing velocity signal + shaft relative signal), machine protection and automatic shutdown should be heavily scrutinized by ensuring that all components behave as expected in normal and abnormal operation. Some OEMs have experience from the shaft rider era and have confidence in the systems that provide machinery protection can do it with reliability and accuracy.

Large frame series gas turbines and steam turbines are the most common candidates for shaft absolute vibration measurements since the rotor to casing mass and stiffness ratios often create conditions where the rotor causes significant casing movement. A rule of thumb sometimes used qualifies that if the casing is vibrating on the order of 30% of the shaft relative reading, a shaft absolute reading is advisable.

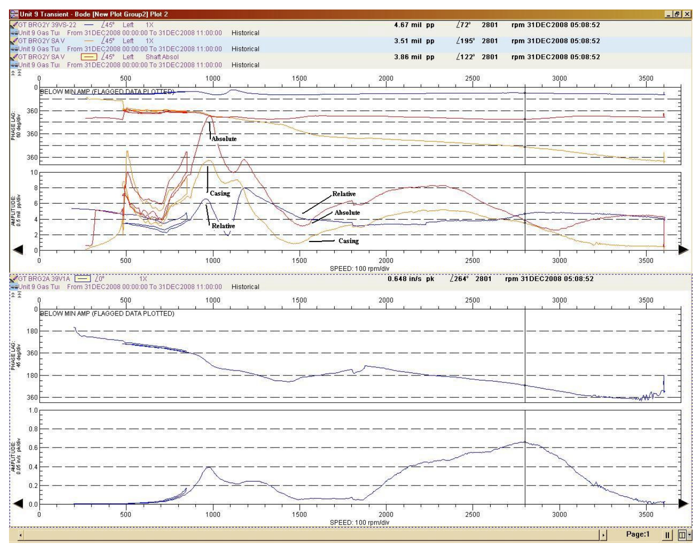

Figure 10 Bode Presentation Startup Data from Relative, Casing, and Shaft Absolute w/ Protection Casing Transducer on Bottom.

It is important to mount the shaft absolute probe systems in XY configuration. Since most machines exhibit anisotropic support stiffness, the shaft absolute measurement can prove useful in diagnosing issues related to axis stiffness discrepancies as well as possible rubs and alignment issues. Orbit analysis of the shaft absolute, casing absolute and shaft relative signals may provide the solution in their comparison. In addition, XY shaft relative probes are necessary for shaft centerline analysis. The data shown in Figure 10 shows that around 1000 rpm the shaft relative amplitude of ~ 6 mils pp generally adds to the casing absolute amplitudes of ~ 11 mils pp to result in ~ 17 mils pp of shaft absolute. Since these are all in phase, the addition of the two: shaft relative and casing absolute vectorially sums to the greatest value. At approximately 1500 rpm, it is observed that the shaft relative amplitude is ~ 4 mils pp while the casing absolute is ~ 3 mils with the shaft absolute resulting in ~ 1 mil pp. In this case, the shaft relative and the casing absolute are out of phase resulting in a much lower shaft absolute reading. All of these data sets are from the 45 deg Left probe from vertical.

By using the rich data set of orthogonally mounted shaft absolute probe sets, root cause of high levels of the top dead center protection seismic transducer readings (Figure 10 bottom plot) can be determined more efficiently.

Additionally, the data shown in Figure 11 shows the advantage of having a shaft absolute configuration in XY configuration. The machine has a protection system that has the velocity sensors mounted vertically. The data shows that there is significant motion, more than 15 mils pp, in the horizontal axis at full speed at approximately 350 cpm that the vertical sensors cannot detect because of their mounting axis. This cross-axis motion may actually perturb their readings, as many seismic sensors are sensitive to this cross axis vibration.

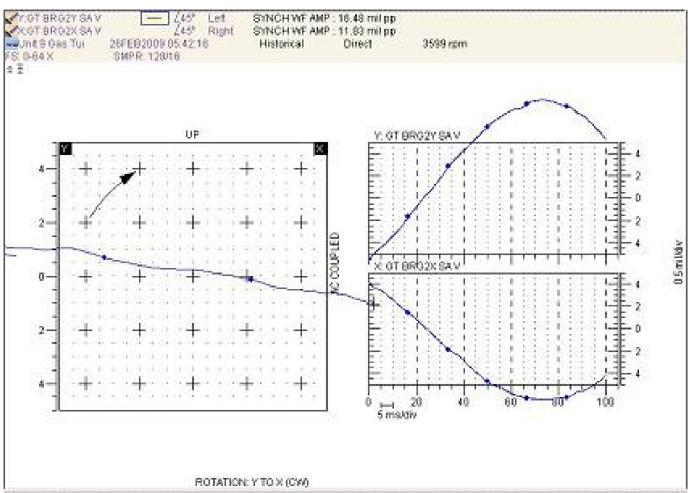

Figure 11 Shaft Absolute Orbit with Main Axis in Horizontal Plane.

Thrust Position/Axial Vibration

Thrust position is a well understood measurement in most turbo machinery applications. It is most often a protection parameter with automatic shutdown capabilities should the axial position within a thrust bearing exceed danger set points. Much like the casing vibration readings, the measurement is often a dual and sometimes a triple redundant reading that has voting associated with the signals. Thrust position within the thrust bearing clearance is most often correlated with the thrust bearing pad temperatures as well as the bearing drain temperatures for correlation.

Many installations that have well-established thrust monitoring equipment and methodologies do not include dynamic or vibration measurements in their condition monitoring or diagnostic systems. The protection systems that accept axial thrust signals use only the DC portion of the signal and go to great lengths to filter out any of the AC or dynamic portion. For any machine that has a thrust bearing, it is important to record and trend the dynamic portion of the signals. Gas turbines and compressors are especially prone to axial excitations. These axial excitations can be due to compressor stall, surge, axial rubs, coupling issues, and others. Pumps and gearboxes also have mechanical issues that can manifest in the axial direction.

Shaft Eccentricity (Shaft Bow)

It is this author's opinion that if a machine has a turning gear that machine is prone to a gravity bow and should have some provision to monitor the severity of the shaft bow. In normal practice, a standard Turbine Supervisory Instrumentation (TSI) system for large steam turbine generators includes a separate probe and channel mounted a certain distance from the bearings on rotors that are prone to bow. The bows on large gas and steam turbines (and some large compressors) can be attributed to a hot rotor that comes to a stop and bows due to gravity. The heavy rotors and relatively long bearing spans aggravated by a hot rotor with different elastic properties than when it is cold provide a perfect storm for gravity bows to set in on the rotors. If a gravity bow is not noticed and the machine is started, very severe rubs can result. The consequences of this can be severe vibration that exceeds trip set points that won't allow the unit to start. This is shown in Figure 12, that the unit starts with a 2-mil pp bow on the compressor end and shuts down with more than 11 mils pp of casing absolute readings before it can reach full speed. It has been witnessed that severe bows engage the seals so severely that it does not allow the unit to turn on turning gear. Less severe effects can be continued degradation in performance due to increased seal leakage caused by rotor to seal rubs. Although this performance degradation does not prevent operation, it can affect the economics of operation and have long term financial impacts especially in the volatile fuel markets operating plants are experiencing.

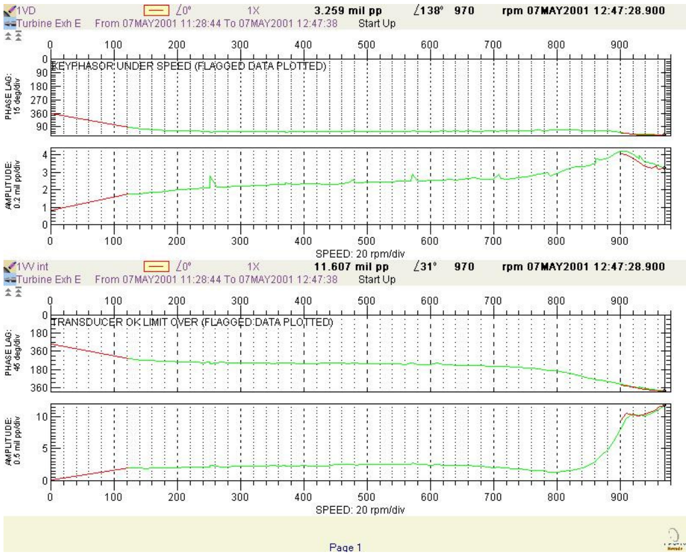

Figure 12 Turbine Start with a Bow and Trip

As described by Maalouf[17], gas turbines rarely have a separate eccentricity probe. There is a methodology to use existing shaft relative probes as eccentricity probes to detect abnormalities in the slow roll condition of the rotor. Often the gravity bow will manifest itself on the shaft relative probes mounted on the bearings, but this utilization is subject to the possibility of the probes being located close to a nodal point.

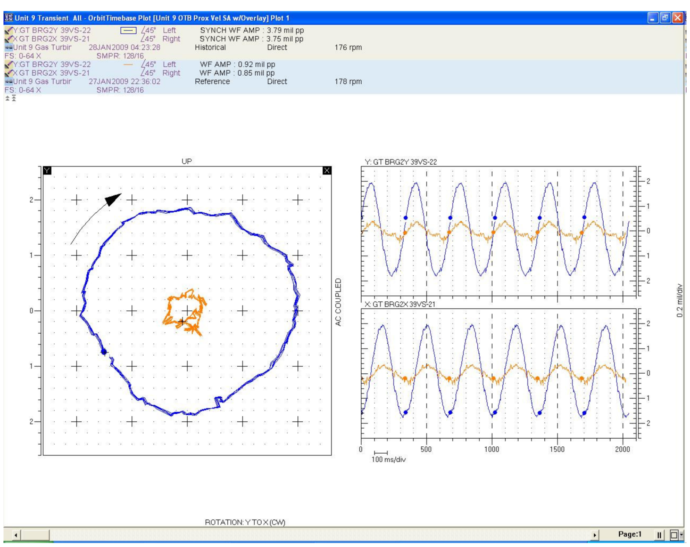

Figure 13 Industrial Gas Turbine Bow with Baseline Overlay.

As previously discussed, a baseline condition can be overlaid on a current operating condition for a very timely reference to what may be normal. As shown in Figure 13, this can apply to the bow or slow roll condition of the rotor. In this case, the orange un-filtered orbit time-base baseline data from a hot shutdown is overlaid on a startup orbit time-base with significantly larger amplitude. It should be noted that this data is at slow speeds: 178 rpm so there is no dynamic motion on the shaft and this is due to a bow in the rotor. Upon startup, this unit went into its first level of alarm as it passed through its balance resonance speeds due to the unbalance caused by the temporary bow.

Zero Speed

As discussed previously, many gas turbines are prone to a gravity bow while the rotor is hot. A zero speed measurement is often used to ensure that the rotor is turning on turning gear when it is appropriate. Often a zero speed indication is used to engage the turning gear after a machine has coasted down to turning gear speed or stopped. This is most often part of the control system or wired into the control system to ensure that the turning gear never engages while the unit is above a certain rpm.

Once the unit is at zero speed, it should be announced that the unit is not turning and the operators are made aware the rotor has stopped. In many situations, if the rotor is stopped, the unit cannot be started until it has been on turning gear for a specified amount of time. In the true condition monitoring sense, the unit should be on turning gear not based on time but the bow or eccentricity condition of the unit as previously discussed. The current operating environment of units sometimes requires quick starts and loading. OEM turning gear requirements are often time based and may be overly cautious while a condition-based system can ensure the unit can be started based on the suitable bow condition of the unit.

Over Speed Detection

Many of the mechanical bolt activated systems have either been replaced or complimented by electronic over speed systems. When the electronic systems are complimenting the mechanical systems the electronic system's trip point will be set slightly less than the calculated 10% mechanical overspeed point. In practice, it is best to have the electronic system trip the unit before the mechanical system engages the trip mechanism since the mechanical system may require some maintenance and machine disassembly. In addition, the electronic overspeed detection system is much more predictable.

Most often, electronic over speed systems have two or three speed sensors for redundancy. Some systems are designed to test the overspeed detection system while the unit is running. If the unit has multiple probes and voting is set correctly between the channels, one channel can be tested while the others are still providing overspeed protection. Some overspeed systems have an onboard function generator that can automatically sweep the input signals to send one channel into overspeed while the remaining channel(s) remain in normal operating condition.

[17] Maalouf, M., Slow Speed Vibration Signal Analysis: If You Can't Do It Slow, You Can't Do It Fast, Proceedings of GT2007 ASME Turbo Expo 2007: Power for Land, Sea and Air, GT2007-28252 2007.

ii http://www.epa.gov/ttn/emc/cem.html

Speed (Gear Teeth and Ratio)

Speed measurements are common and required on most machinery. The most common is a Keyphasor measurement that gives a once per turn signal that is key for making startup, shutdown and absolute phase measurements that are required when balancing and other diagnostics issues.

Other uses for speed can be startup and especially coastdown times. A machine should have consistent coastdown times from full speed to zero speed (or turning gear speed). Differences in coast down time can be due to rubs or other braking forces like back pressure variations due to bleed valves being closed or open when not necessary as well as other back pressure issues that may arise in pumps, compressors or steam turbines.

For mechanical drive units that may be driving compressors, speed variations at supposed constant loads can indicate an insufficiency in the speed control algorithm in the controller. Speed variations could also be an indication of a stalling or surging compressor.

Remote Monitoring Connectivity

Recent industry trends have shown that many of the people with the skills and experience to manage rotating equipment are getting stretched very thin. This is due to a combination of many global projects tapping into the resources of many established engineers to manage the installation, commissioning and operation of these new energy sites. This leaves existing plants short of resources to ensure their machinery is managed to standards that were established with the native experience. To aggravate the shortage in experienced rotating equipment engineers, many of these people are moving on to retirement age.

An ever-increasing trend is to enable the systems described in this paper to be accessed remotely. While this is not a new concept: "Move Data Not People"[18], many OEMs, End Users and third party suppliers are setting up centers commonly called "Remote Monitoring and Diagnostic" (RM&D) Centers. The key lever in setting up centers where sites around the world can be accessed over Internet connections is that key expertise can occupy their time in their specific domains of expertise, whether it is control system optimization, performance optimization, or mechanical diagnostics and management. These skill sets are highly refined and valued and if a business can use these skills without the time and expense of travelling, those skills are being utilized to their maximum benefit.

Setting up remote access must be done correctly. With the advent of many Cyber Criminals out there and their increasing sophistication, the implementation of remote access must exclude those who mean harm. In the event that the remote system may have access to affect the control or operation of the machines, it is ever important to establish security and access rules to ensure those who are qualified and those who have been granted access can work, while others who mean harm or may be harmful out of ignorance cannot affect these assets.

Many of the global companies have established their own security standards and designs for access. However there are few regulatory agencies that have started establishing standards to ensure the energy supplies are secure and safe. At least in the US the Federal Energy Regulatory Commission has certified the North American Reliability Corporation (NERC) as the Electric Reliability Organization.

"The role of NERC is to improve the reliability and security of the bulk power system in North America. NERC develops and enforces reliability standards, monitors the bulk power system, assesses future adequacy of the system. NERC also audits owners and operators of the bulk power system. NERC has created a Compliance Monitoring and Enforcement Program that will (along with the regional entities) monitor, assess, and enforce compliance with regulatory approved reliability standards."[19]

Some key definitions for establishing the Cyber Security are:

Critical Assets: Facilities, systems, and equipment which, if destroyed, degraded, or other rendered unavailable, would affect the reliability or operability of the Bulk Electric System.

Cyber Assets: Programmable electronic devices and communication networks including hardware, software, and data.

Critical Cyber Assets: Cyber Assets essential to the reliable operation of Critical Assets

Cyber Security Incident: Any malicious act of suspicious event that:

Compromises, or was an attempt to compromise, the Electronic Security Perimeter or Physical Security Perimeter of a Critical Cyber Asset

Disrupts, or was an attempt to disrupt, the operation of a Critical Cyber Asset

Electronic Security Perimeter: The logical border surrounding a network to which the Critical Cyber Assets are connected.

Physical Security Perimeter: The physical, completely enclosed border surrounding computer rooms, telecommunications rooms, operations centers, and other locations in which Critical Cyber Assets are housed and for which access is controlled.

NERC Critical Infrastructure Protection (CIP) standards provide the requirements and scope for compliance, but not a detailed procedure or methodology on how to achieve compliance.

[18] Roger Harker, Move Data Not People – Today's Productivity Tool, *Orbit Magazine,* September 1995

[19] http://www.nerc.com/

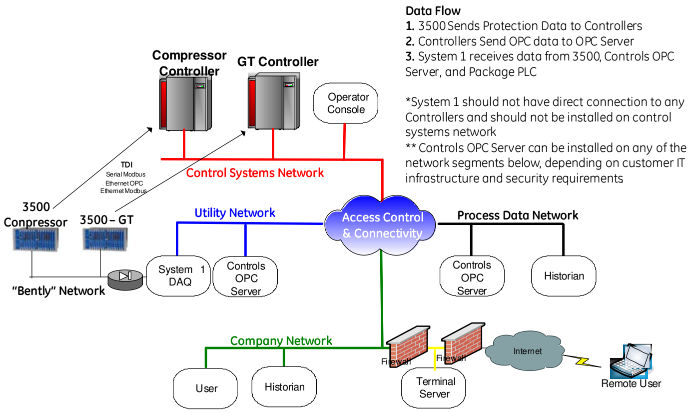

Figure 14 Typical Remote Access Configuration

Summary

The technologies available discussed in this paper can independently increase the ability of gas turbines and their driven asset to deliver reliable and efficient service. The wise use of each of these technologies in conjunction with one another can escalate their usefulness and ability to deliver value to the owners.

Enabling remote connectivity in a secure manner can allow resources around the world to advise local operators and maintenance staff to ensure that global best practices are available for all production assets.

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Keyphasor® is a registered trademark of Bently Nevada LLC a wholly owned business of General Electric.

The opinions and findings expressed in this article are those of the authors, and not in any way of Qatar Shell Services Company W.L.L.

ANNEX A

FULL SIZE FIGURES

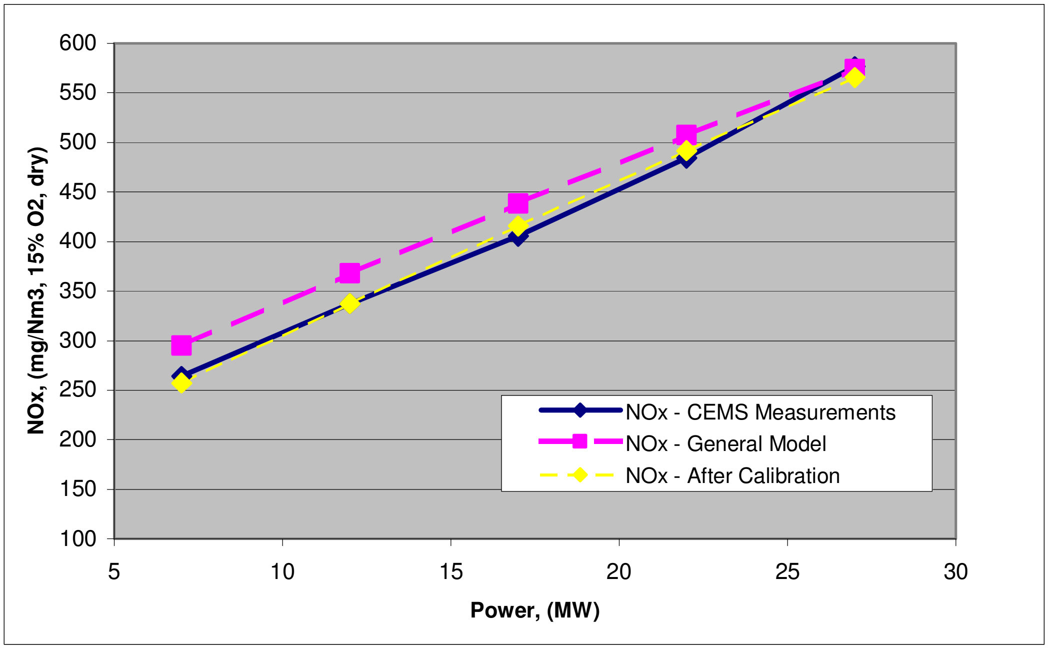

Figure 1 PEMS Model Calibration

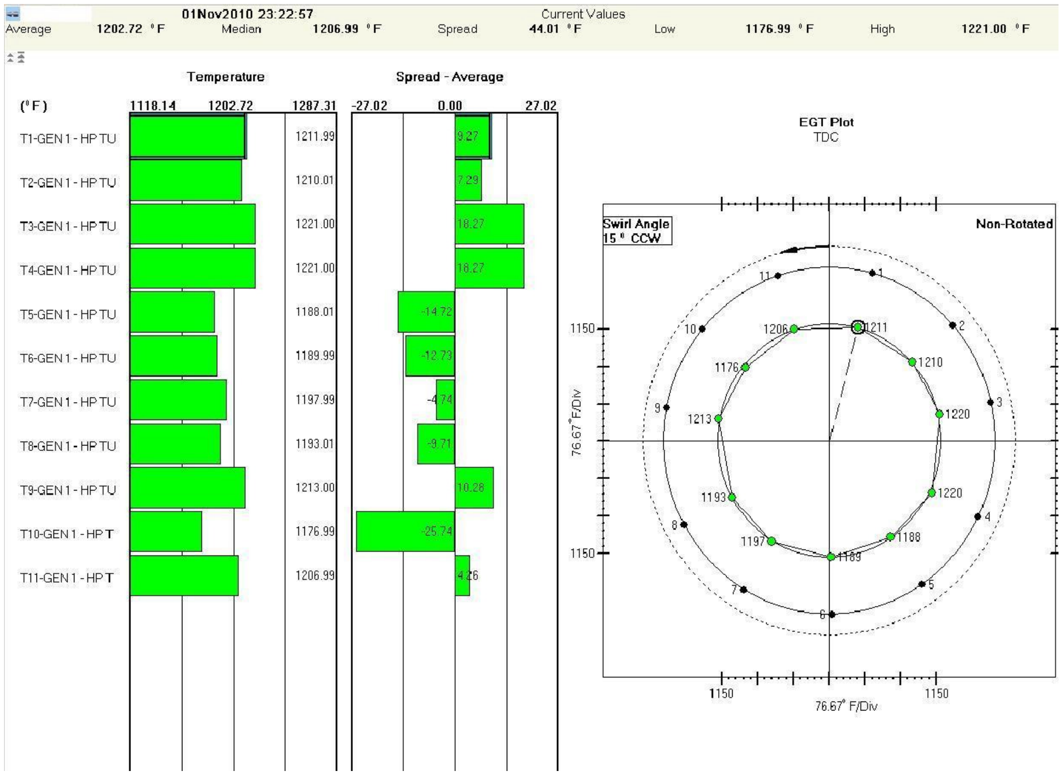

Figure 2 Aero-derivative Exhaust Gas Temperature Plot

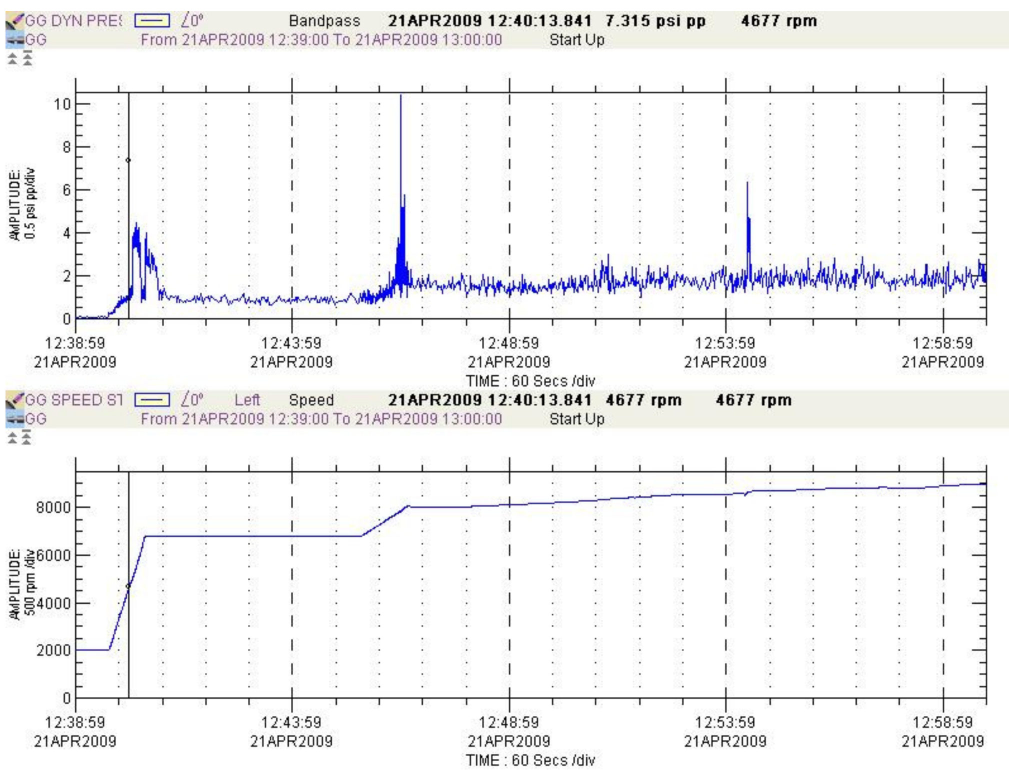

Figure 3 Aero-Derivative Combustion Acoustic Tuning Trend of Pressure pp and Gas Generator Speed

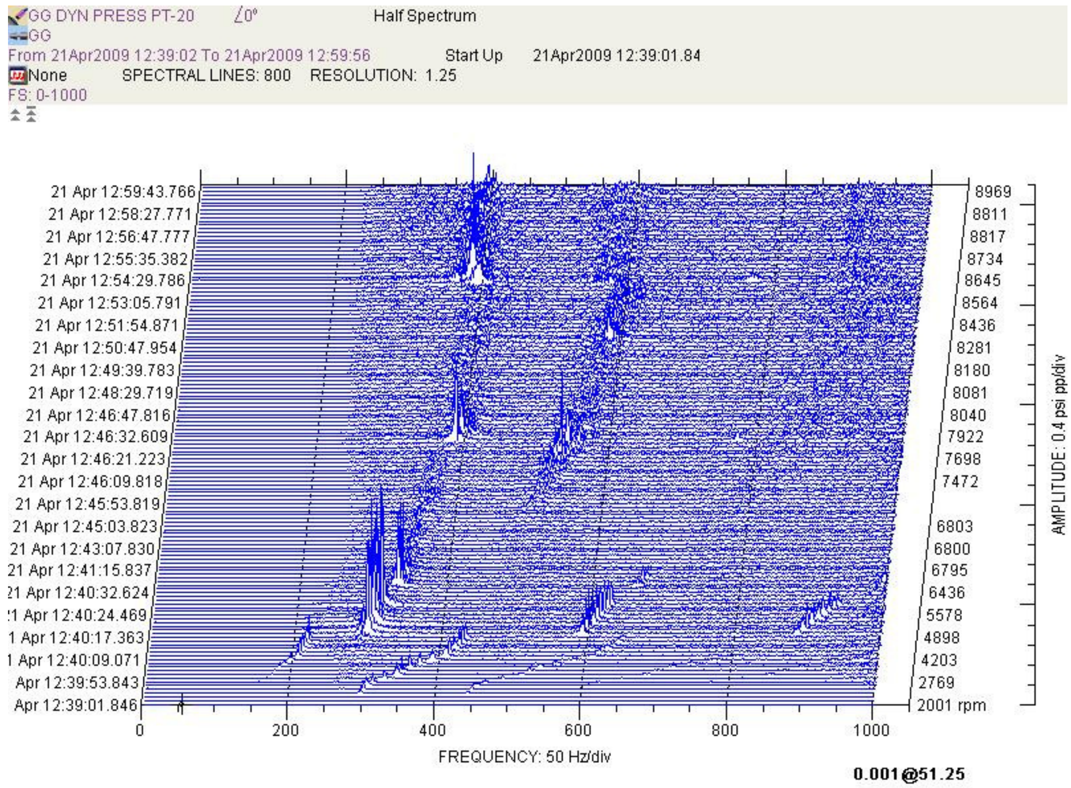

Figure 4 Aero-Derivative Combustor Acoustic Tuning Waterfall (Frequency Content)

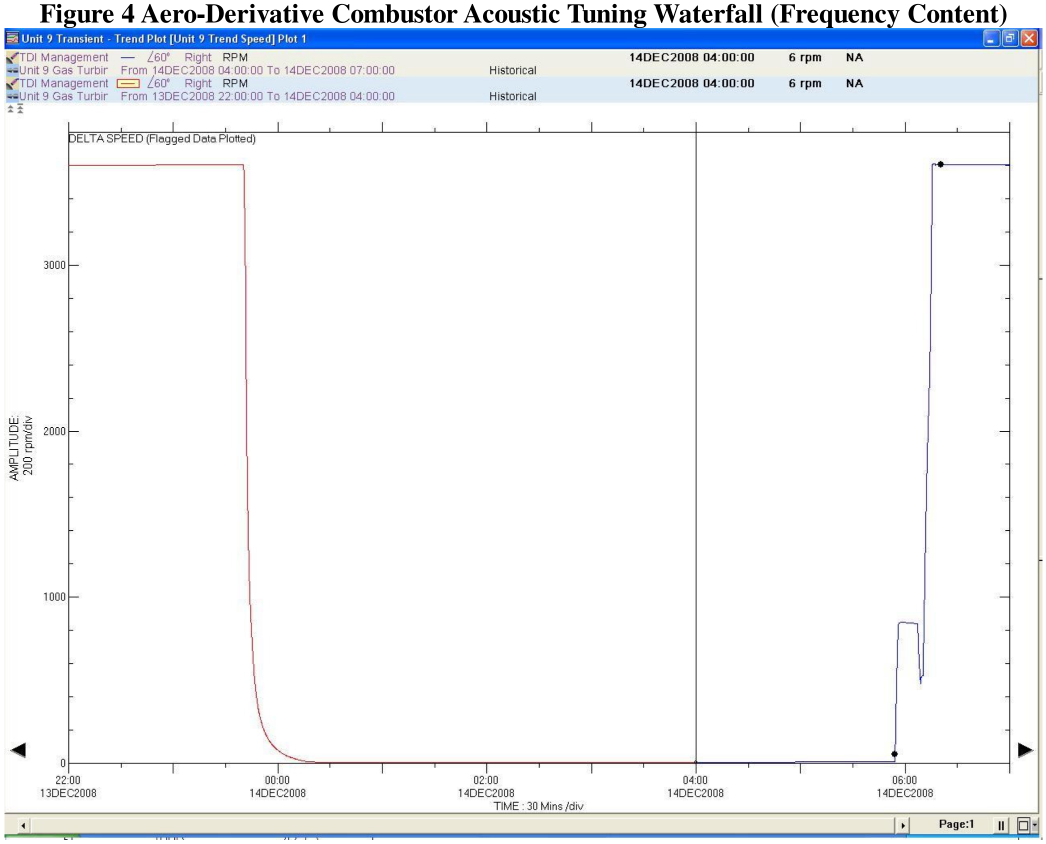

Figure 5 Speed Trend of Hot Coastdown (red) and Warm Startup (blue)

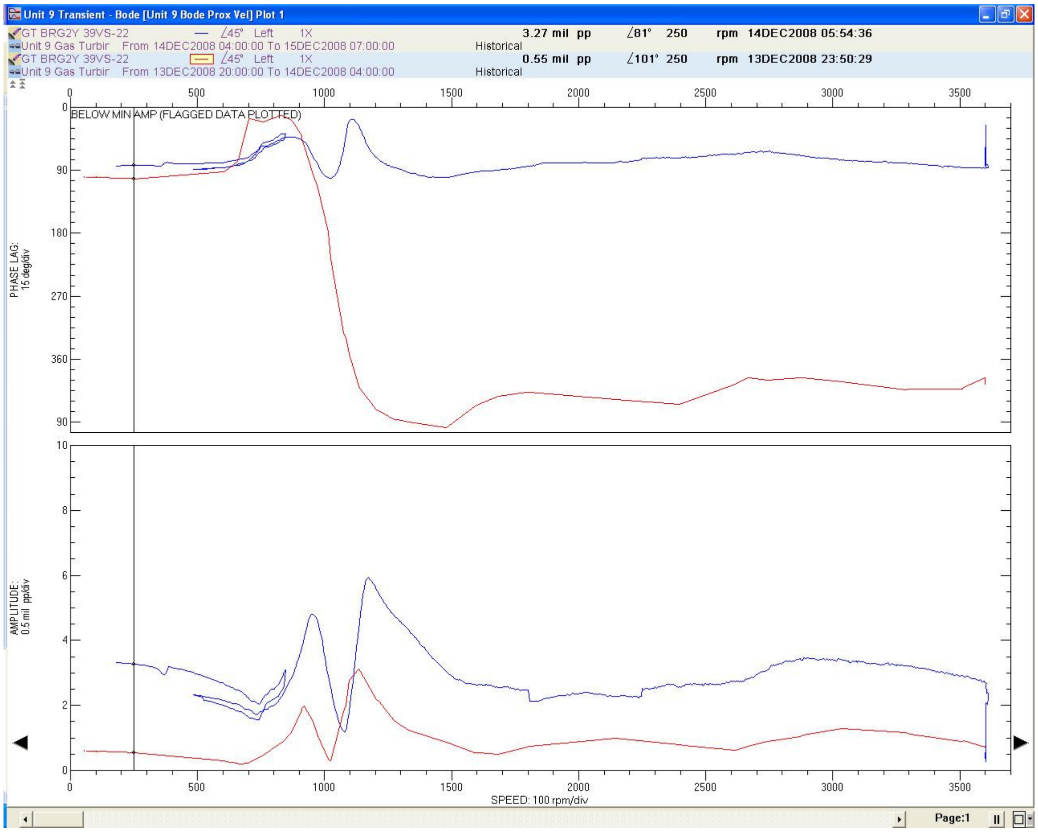

Figure 6 Data Presentation in Bode Format: Startup (Blue Curve) with Shutdown Baseline Overlay (Red)

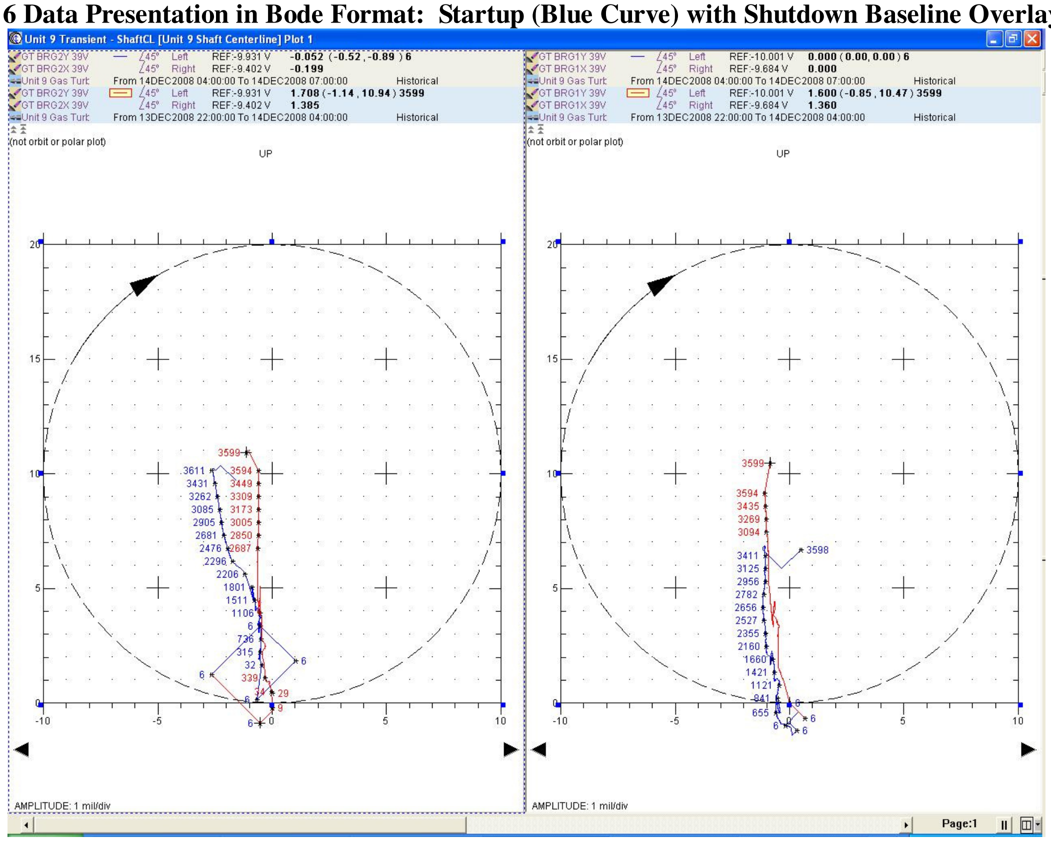

Figure 7 Data Presentation in Shaft Centerline Format: Startup (Blue Curve) with Shutdown Baseline Overlay (Red)

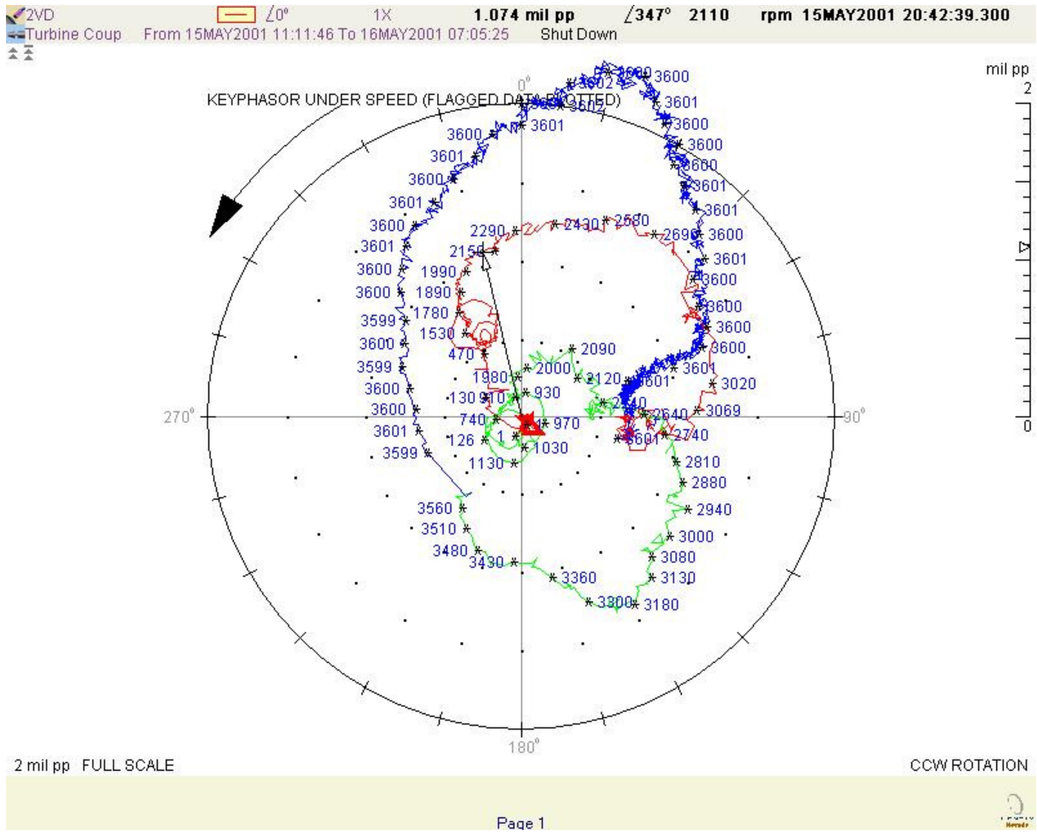

Figure 8 1X Vibration in Polar Format (Green Startup, Blue Loading and Unloading and Red Coastdown.)

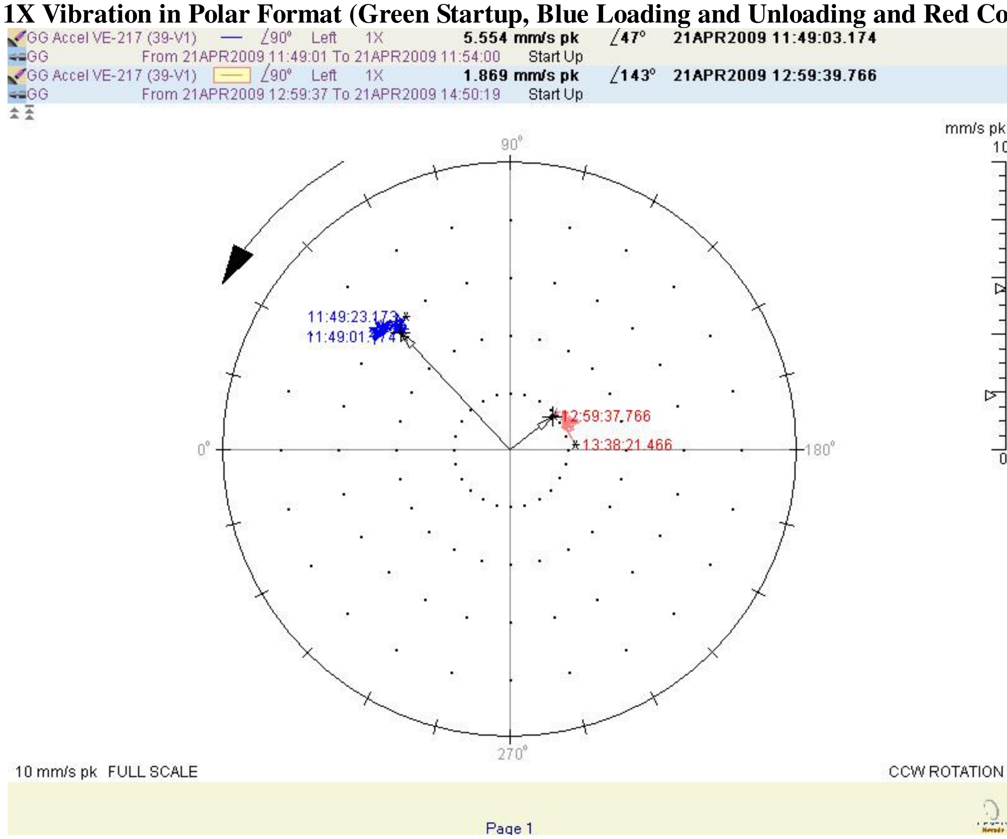

Figure 9 Generated Once-per-turn Reference from a 47X Speed Signal

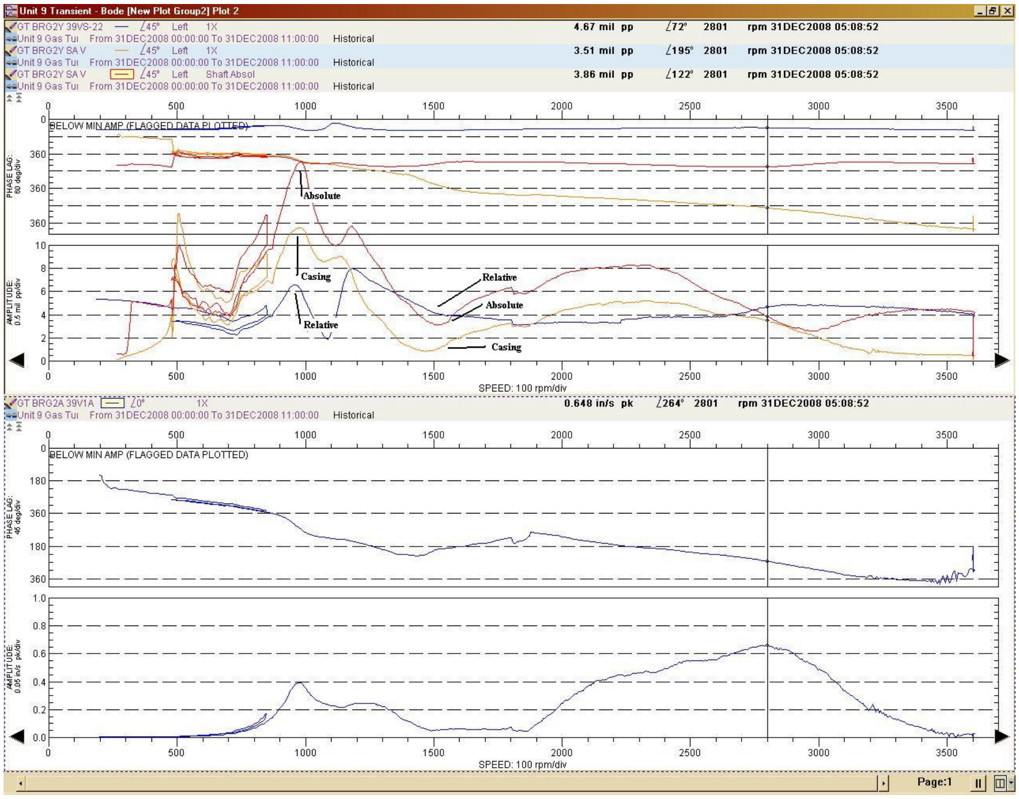

Figure 10 Bode Presentation Startup Data from Relative, Casing, and Shaft Absolute w/ Protection Casing Transducer on Bottom.

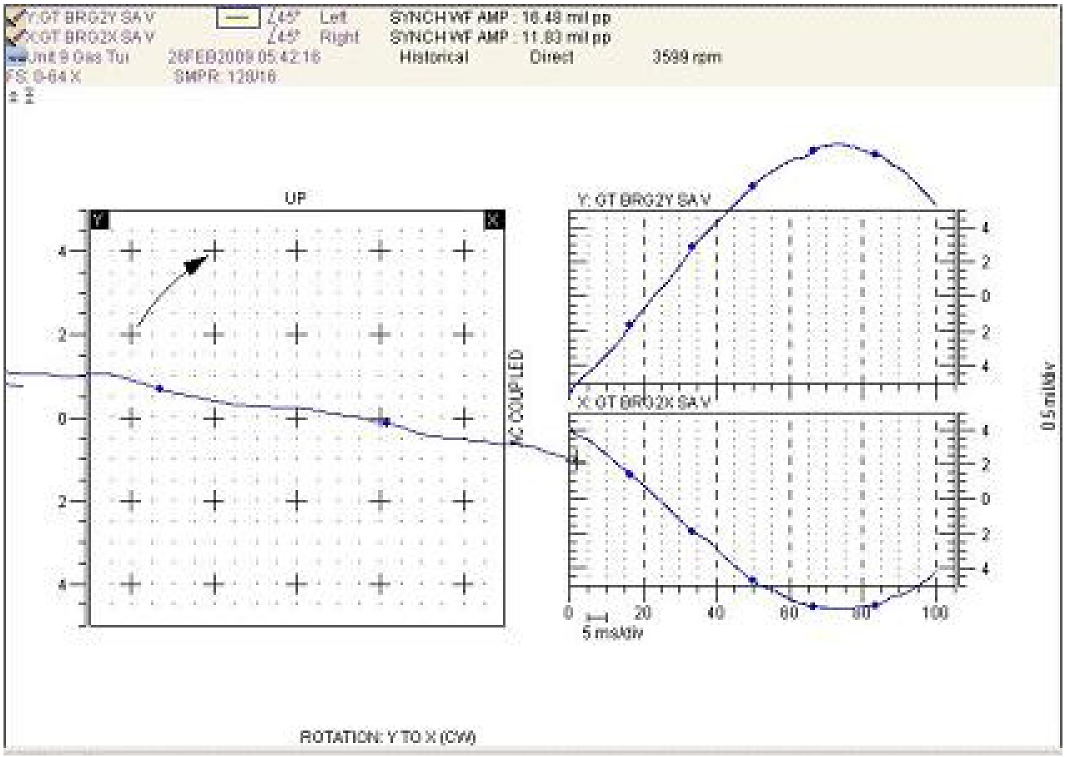

Figure 11 Shaft Absolute Orbit with Main Axis in Horizontal Plane.

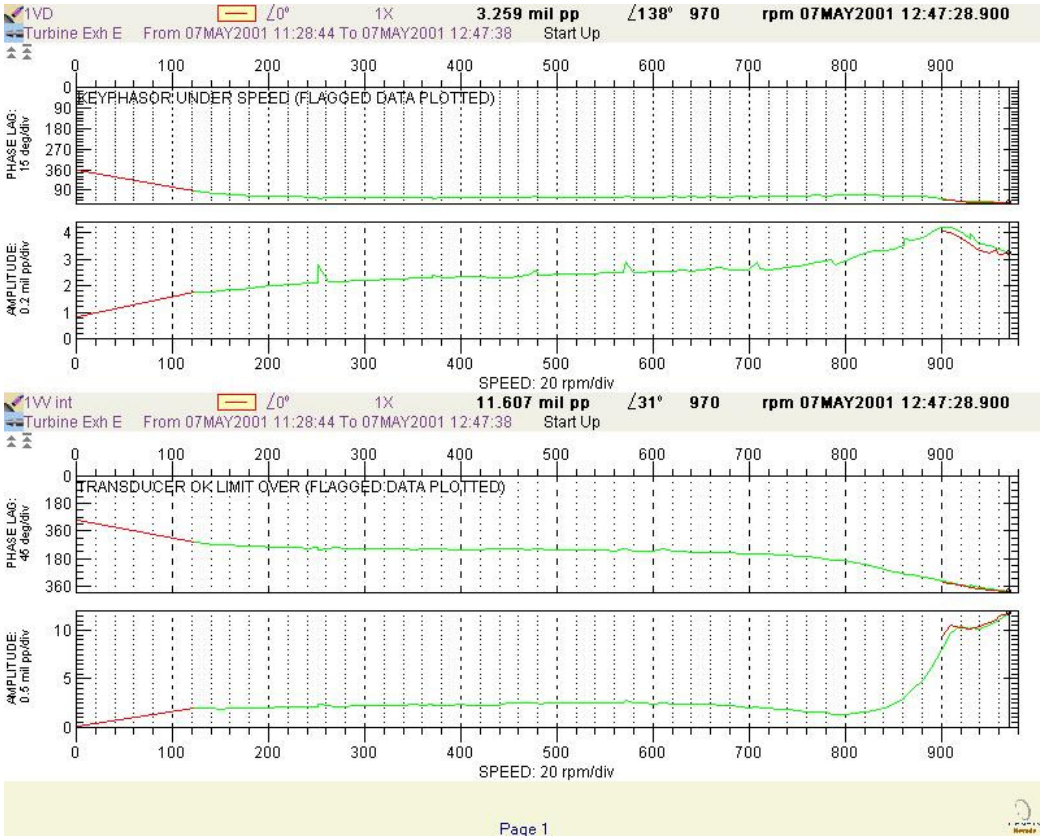

Figure 12 Turbine Start with a Bow and Trip

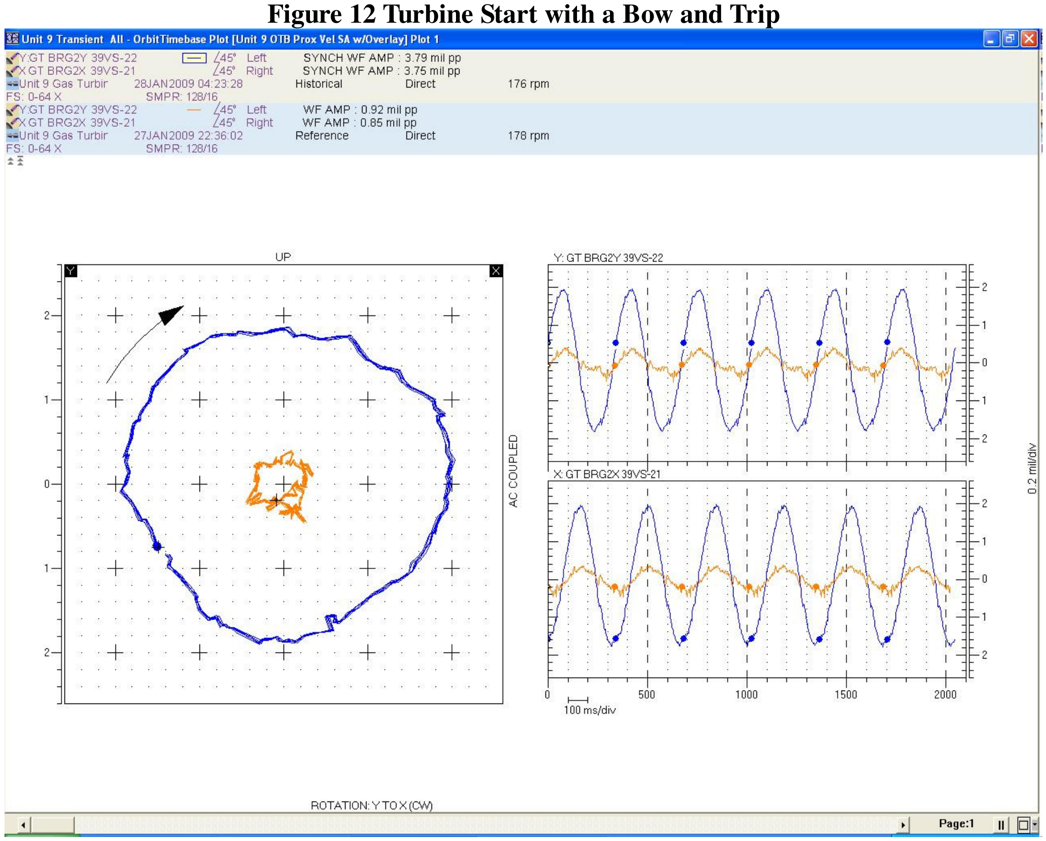

Figure 13 Industrial Gas Turbine Bow with Baseline Overlay.

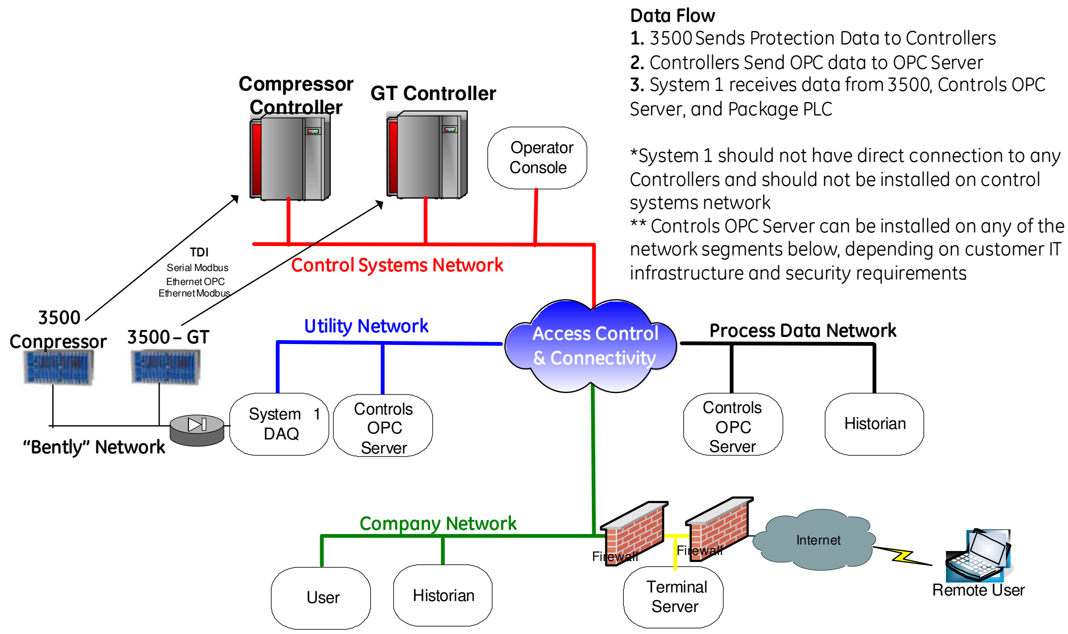

Figure 14 Typical Remote Access Configuration