ADVANCED TECHNOLOGY COMBUSTION TURBINES IN COMBINED-CYCLE APPLICATIONS

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1. INTRODUCTION

With the latest advancement of F, G, H, and J-Class combustion turbines, combined-cycle designs are affected when compared to the previous generation of F-Class machines. This paper provides an overview of some of the significant influences associated with advanced combustion turbine design for current combined-cycle installations.

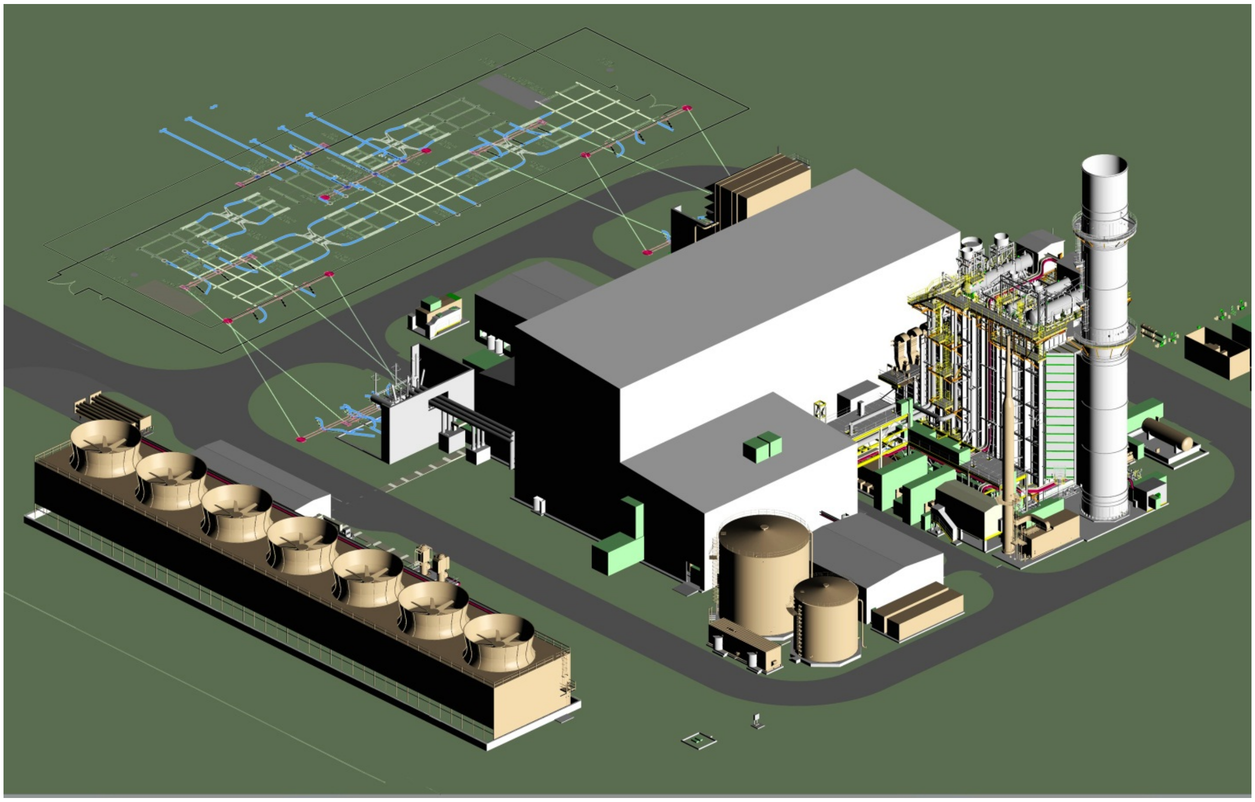
With the obvious consideration being larger combustion turbines and the additional power generated, there are other power block and balance-of-plant (BOP) design impacts. These impacts, as described below, include integration of the latest combustion turbine steam cooling or turbine air cooling, sizing of heat recovery steam generators (HRSGs) and steam turbines, complex designs related to the larger equipment, and material selection for higher-steam temperatures.

The current combined-cycle design incorporates a number of features to accommodate the requirements of today's market. There are numerous demands on the designs, including fast startup, both base load and cycling operation, high ramp rates, high efficiency, high reliability, lower emissions, and lower life-cycle costs, to name just some of the expectations. With all of these market demands, the combined-cycle plants are also trending larger in an effort to obtain economies of scale with the installed cost of the facility. All of these factors influence the design of the power block equipment, along with the integrated BOP equipment.

Figure 1 shows a representative model cut developed by Sargent & Lundy for an advanced combustion turbine facility project.

Presented at POWER-GEN Natural Gas, Columbus, Ohio, August 23-25, 2016

Figure 1. Advanced Combustion Turbine Facility Model Cut

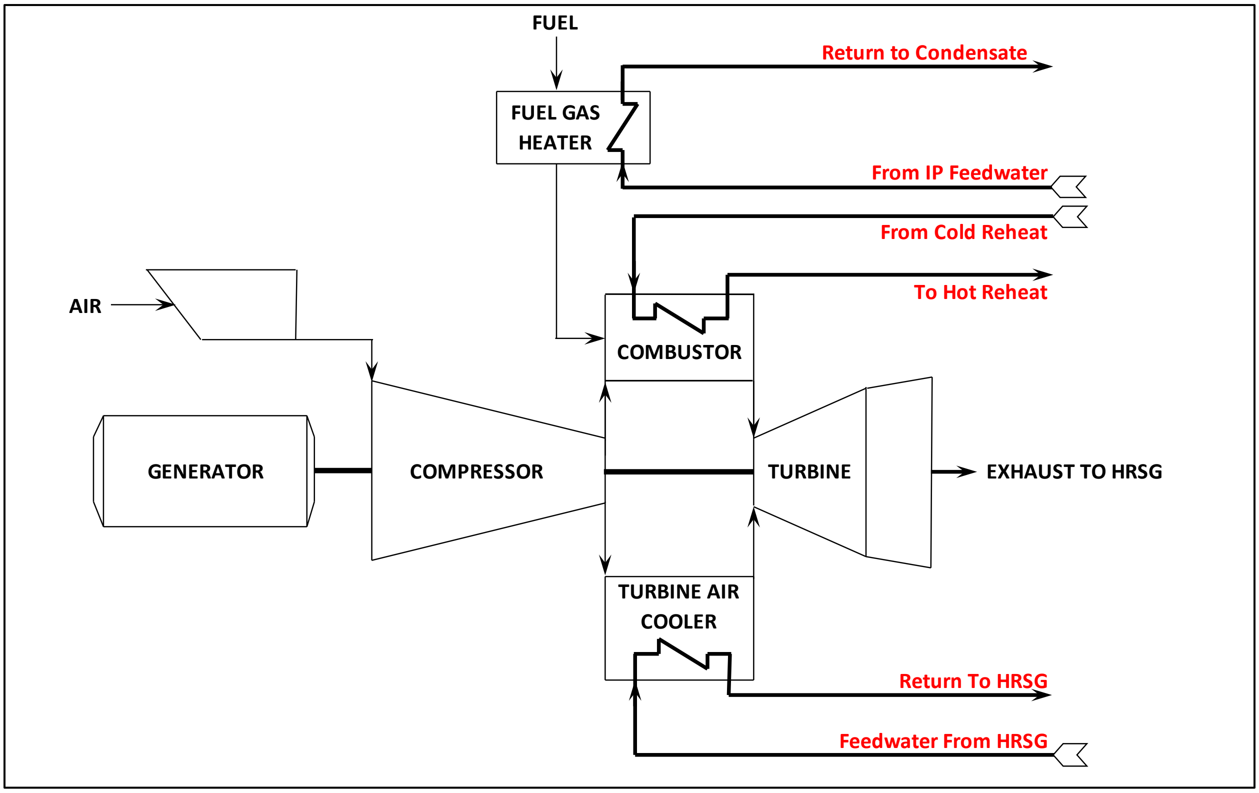


Source: Sargent & Lundy, L.L.C.

2. INTEGRATION OF COMBUSTION TURBINE(S) WITH THE STEAM BOTTOMING CYCLE BOP

There is opportunity with advanced combustion turbines to increase combined-cycle efficiency by integrating heating and cooling systems between both thermal dynamic cycles. These systems include: 1) fuel gas performance heating system, 2) steam cooling of combustor transitions; and, 3) energy recovery with the turbine cooling air (TCA) system. The advanced combustion turbines have an even greater design demand on the integrated systems providing these services. Figure 2 illustrates how these schemes between combustion turbine and steam cycle are integrated.

Figure 2. Integration of Combustion Turbine with Steam Cycle



2.1 Fuel Gas Performance Heating System

The fuel gas performance heater is not to be mistaken for a natural-gas dew point bath-heater, which increases the incoming fuel gas temperature by several degrees above the hydrocarbon dew point so the fuel remains in the gaseous state and no liquid drops impinge on hot combustor and/or turbine blade components causing pitting. The fuel gas performance heater provides significant sensible heat to the fuel prior to combustion and increases the efficiency of the combustion turbine. One method of accomplishing this is by extracting feedwater from the HRSG and directing this feedwater to a tube-and-shell heat exchanger. With high-temperature feedwater on the tube side and natural gas on the shell side, the temperature of the natural gas is raised significantly to improve combustion turbine efficiency. The cooled feedwater is then recirculated back to the condensate line entering the HRSG preheater.

The following criteria are guidelines for design of a standard gas fuel heating system:

- Ensure water pressure is higher than gas pressure during gas turbine operation and shutdown.
- The heated fuel must meet the requirements of the combustion turbine manufacturer's fuel specifications, including particulate coalescing filters and final-stage knockout vessels.
- Provide early indication of heat exchanger tube failure and prevent water from entering the gas turbine combustion system should a heat-exchanged tube leak.

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- Prevent gas fuel from entering the feedwater system following a heat exchanger tube failure.
- Provide overpressure protection to the combustion turbine gas fuel heating system piping and components.

Depending of the combustion turbine model and steam cycle configuration, the fuel gas final temperature requirements have increased over the combustion turbine requirements of the past, and can be greater than 600°F. The source of heat for the natural gas heaters is typically feedwater. While in the past, the feedwater could be sourced from intermediate pressure, the current advanced combustion turbines with ever-increasing pressure ratios require a high-pressure, high-temperature feedwater source. To satisfy these requirements, the source is usually from the outlet of the high-pressure economizer section of the HRSG.

2.2 Steam Cooling of Combustor Transitions

When introducing a new, advanced-class combustion turbine (G-Class with turbine inlet temperature of 1,500°C, or J-Class with turbine inlet temperature of 1,600°C), manufacturers will take a conservative approach, with extensive testing and validation programs. Third-party insurers also perform independent engineering reviews for the insurance market. This approach usually means some advanced combustion turbines are first introduced and tested with a steam-cooled combustor. The improved heat transfer effectiveness of steam cooling allows for better management of the initial unknowns associated with a new or advanced combustor technology.

The typical method of steam cooling is utilizing cold reheat directed to the combustor sections with the steam returned to the steam cycle as hot reheat steam.

As actual operating experience of the combustion turbine generator is obtained with testing and validation programs and/or commercial installations, cooling of the combustor has historically transitioned from steam-cooled to air-cooled. This is due to the design and operating complexity of integrating combustion turbine generator cooling requirements and matching temperatures with the steam cycle. Air cooling also improves operating flexibility with regard to quick start and load cycling.

It is worth noting that aside from the method of combustor cooling, the steam-cooled combustion turbine and the air-cooled versions are the exact same engines. There is no equipment difference between the two models.

2.3 Energy Recovery with the TCA System

The turbine cooling air system is designed to reduce temperature of the combustion turbine's compressor discharge air (utilizing feedwater), which is then used for cooling of internal turbine components, including the combustor, rotor, and blades. Heat transfer between the feedwater and turbine compressor air occurs across the TCA cooler heat exchanger. Design of this TCA feedwater system is in collaboration with design criteria provided by the turbine manufacturer. The turbine cooling air cooler feedwater interface typically utilizes supply water from the HRSG low-pressure drum as a branch from feedwater pump supply line or directly from the boiler feed pump discharge at the appropriate design pressure.

This extension of the feedwater system features the following interfaces:

- TCA supply water from the boiler feed pump discharge or a dedicated TCA pump
- TCA cooler heat exchanger (water side)
- TCA pump minimum flow bypass (upstream of TCA cooler, to condenser)
- TCA cooler bypass (downstream of TCA cooler, to condenser)
- Supply return piping system to the HRSG at the appropriate location

Turbine cooling air systems are also referred to as *once-through coolers* (OTCs) or *kettle boilers,* depending on the turbine manufacturer's nomenclature and function. Some TCA systems return the feedwater to the HRSG as high-temperature feedwater. Other designs return saturated steam to the HRSG, thus the name kettle boiler.

Due to the importance of maintaining the combustion turbine's cooling air flow at design temperature, redundant components are required (i.e., 2x100% TCA pumps). Manufacturer recommendations should be strictly adhered to with regard to TCA pump startup sequences and operating procedures, especially during upset conditions.

2.4 Natural-Gas Compression

Advanced combustion turbines have improved compressors with higher compression ratios. This means the fuel gas pressure delivered to the combustor must also be greater. The natural-gas pressure requirements can be greater than 600 psig for these advanced combustion turbines. While pipeline gas can provide these pressures, many U.S. locations will not guarantee the higher pressures will be supplied at all times. Therefore, fuel gas compressors may be required to achieve the gas pressure. There are several types of fuel gas compressors to consider, including reciprocating, screw, and centrifugal. Compressor installations of each type have been successfully used to supply compressed natural gas for combustion turbines. It is prudent to work with the combustion turbine manufacturer and to perform an evaluation for each application to achieve a consistent gas pressure to the machine that is pulse-free and oil-free. To assist in this verification, Sargent & Lundy, L.L.C. (Sargent & Lundy) uses internally developed software. The evaluation of the gas system also includes assessment of combustion turbine trips and the impacts on other combustion turbines at a multi-turbine site, and evaluation of natural-gas compressor trip on the fuel gas supply.

3. CHALLENGES OF DESIGNING STEAM CYCLES FOR ADVANCED COMBUSTION TURBINES

With the ever-increasing demands of the market and with the advanced combustion turbines being larger, design of the HRSGs and steam turbines must consider the higher combustion turbine exhaust gas flow volumes and temperatures, resulting in greater steam generating

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capabilities at higher temperatures and pressures. This presents the engineering team with many challenges when specifying equipment for design of the optimal combined-cycle configuration.

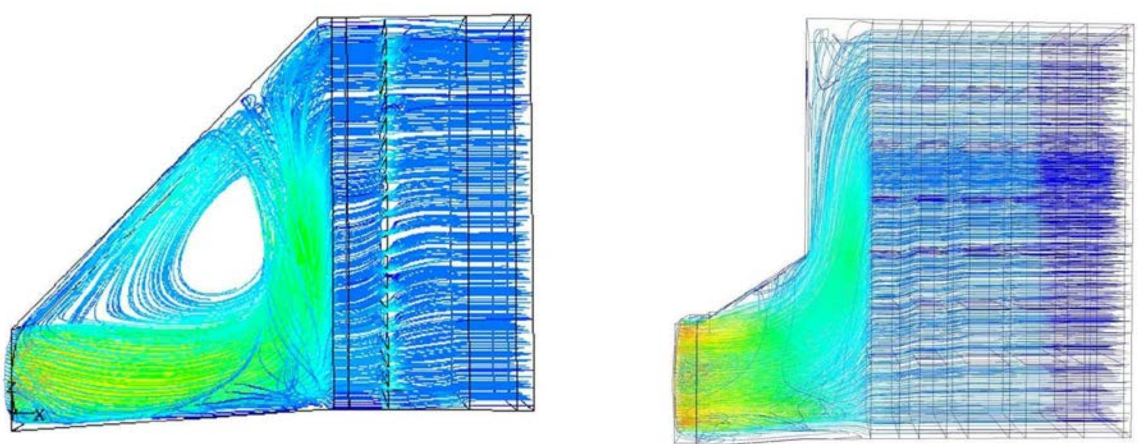
3.1 Heat Recovery Steam Generator

There are many factors that influence the design of today's HRSGs. These factors include the ever-increasing exhaust gas flows and exhaust gas temperatures from the combustion turbines, along with previously mentioned market requirements (i.e., fast start, frequent cycling, increased ramp rates, higher efficiency, lower emissions, etc.). As such, the design demands on HRSGs are ever increasing, with some of the more significant modifications noted below.

3.1.1 Transition Inlet Duct Design

The conventional inlet duct design for smaller combustion turbine exhaust gas flow rates included a gradual sloping inlet duct with no greater than a 45-degree slope. The inlet transition ducts of the past also included flow training components, such as turning vanes and distribution grids or flow-straightening devices. These earlier designs have proven to not work as well with the current advanced combustion turbines. The gradual inlet transition ducts tend to recirculate the exhaust gas flow within the duct area, causing additional turbulence and pressure drop, and thereby negatively affecting the combustion turbine and HRSG performance. Based on flow modeling and testing, the current transition duct design is much steeper and compact with minimal ductwork. This configuration has led to less turbulence, improved pressure recovery, and better flow distribution into the HRSG (as illustrated in Figure 3). This has helped improve not only the performance of the HRSG, but also the cost.

Figure 3. Flow Comparison – Conventional Duct versus Compact Inlet Duct



Conventional Duct **Compact Inlet Duct**

Source: Alstom Legacy (GE) HRSG Presentation to Sargent & Lundy

3.1.1 Thermal Design Considerations

With the fast start and high exhaust gas flows and temperatures, the HRSG's thermal design associated with today's advanced combustion turbines is impacted. Each HRSG manufacturer has determined its own strategies to accommodate the current demands. With the fatigue life of

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the high-pressure components (i.e., drums and superheaters), some manufacturers employ smaller-diameter longer high-pressure drums to reduce the metal wall thickness and minimize the thermal fatigue, while others have opted for once-through high-pressure designs to achieve the same or similar affect. However, the smaller-diameter drums reduce the retention time and the once-through designs require better water quality, and typically require the use of condensate polishers to achieve the water quality needed. Other thermal design strategies employed include single-row harp sections in the high-pressure sections and enhanced materials in the high-pressure sections to reduce tube wall thickness. Smaller-diameter tubes and headers are also used to minimize tube and wall thicknesses.

While these methods can have positive impacts on achieving the demands of additional steam flows, temperatures, and pressures while supporting fast startup, care needs to be taken not to sacrifice operability and maintainability of the equipment. As HRSG manufacturers are pressured to find cost-effective solutions for the current design requirements, minimizing the drum size and reducing the diameter of tubes and headers have helped achieve this goal. However, with smaller retention times in drums, the operability of the HRSG becomes more challenging. The smaller retention times leave less reaction time for upset conditions prior to potential high- or low-level trips, impacting overall facility reliability. Additionally, HRSG manufacturers desire to increase the tube density within modules to even greater values to reduce cost. The increased tube density impacts the ability to repair tubes located in the middle of the bundles.

HRSG manufacturers are also increasing the fin density and fin height, and lowering the fin thickness to reduce cost. These types of modifications significantly increase the risk of damage to the fins during maintenance. Additionally, these fin designs make it difficult to clean tubes/fins and restore HRSG performance.

3.1.2 Draft Loss Considerations

Due to the large exhaust flows from the advanced combustion turbines, HRSG manufacturers have incorporated designs utilizing three-wide module concepts. This is similar to the current designs for larger combustion turbines and is employed to reduce the overall HRSG pressure drop, improving the combustion turbine performance and, consequently, the combined-cycle facility performance.

3.1.3 Purge Credit and Stack Dampers

To support fast-start design, advanced combustion turbine designs have incorporated purge credit and stack damper designs. National Fire Protection Association (NFPA) purge credit is employed not only in the combustion turbine fuel gas supply, but also in the duct burner fuel gas supply. Purge credit is accomplished by purging the gas systems of gas during the previous plant shutdown and adding additional isolation valves in the gas supply system. This design allows the startup to occur without the volumetric purge required by NFPA 85, enhancing the fast-start capability of the facility.

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Another means of achieving faster starts is the use of stack dampers. Stack dampers allow the HRSG to be *bottled-up*, conserving the heat built up in the HRSG. Traditional design without stack dampers provides for the air to flow through the combustion turbine and HRSG, allowing the equipment to cool quicker. By employing a stack damper, the equipment is not cooled by convection of the air flowing through the machine. By using stack dampers, the ability to potentially improve the overall plant startup times is improved with the potential for changing a cold start to a warm start and a warm start to a hot start.

3.1.4 HRSG Balance of Equipment Design

The remaining HRSG components continue to be in use. These components include supplementary firing, post-combustion emissions control equipment (i.e., carbon monoxide [CO] and selective catalytic reduction [SCR] catalyst), gas baffles, drains, low-pressure condensate recirculation system, deaeration, etc. These systems continue to be used to improve the performance, operation, and maintenance of the HRSG.

3.2 Steam Turbine Generator

Steam turbine designs associated with advanced combustion turbines are similar to designs used for traditional Rankine cycles. These machines are capable of the increased steam pressures and temperatures associated with the latest combined cycles. The current combined-cycle steam turbine inlet main steam pressure is 2,400 psig and main steam and hot reheat temperatures are 1,112°F for both. The older combined-cycle designs with lower pressures and temperatures were able to use steam turbines without an inner shell. However, with the higher temperatures and pressures, steam turbine designs have to employ both an inner and outer shell design.

Traditional Rankine cycle steam turbine designs included steam chests with control stages. This allowed steam turbines to use partial-arc steam admission. However, combined-cycle units typically use full-arc admission, minimizing efficiency losses via throttling across the admission valves. This concept continues to be the preferred method for operating a combined-cycle facility.

Steam turbine designs are also incorporating multiple low-pressure turbine sections as the flow rates increase with multiple advanced combustion turbine/HRSG trains and incorporation of supplementary firing feeding a single steam turbine. These configurations require multiple condensers with the corresponding design challenges for large steam turbine pedestals and multiple-shell condenser design and operation. However, single-shaft combined-cycle configurations are also employed, allowing a smaller axial exhaust steam turbine configuration to be used.

3.3 Final-Stage Attemperation

One of the challenges with designs to reduce overall emissions is the matching of the HRSG steam production and the startup of the steam turbine. This is especially valid when trying to start the advanced combustion turbine and maintaining lower emissions. The traditional combustion turbine startup is to temperature-match the exhaust with the steam requirements for the steam turbine startup. This would require the combustion turbine to stay at very low loads, and not be able to control the CO and nitrogen oxides (NO_X) emissions. The current designs employ final-stage attemperators to allow the combustion turbine to start up independently from the steam turbine. The final stage attemperators allow the steam conditions to be matched to the steam turbine startup requirements regardless of the combustion turbine load. Therefore, the combustion turbine can ramp up to its emissions control load or greater and not be impacted by the steam turbine startup.

Sargent & Lundy has incorporated final-stage attemperators in combined-cycle designs to allow the combustion turbine to be started independently from the steam turbine to minimize emissions during startup at least as far back as 1997. At that time, Sargent & Lundy received special permission from the steam turbine manufacturer during the design process, since the ASME Turbine Water Induction Prevention guidelines in 1997 did not recommend water be added to the steam in route to the steam turbine without a downstream heat exchanger being employed. As interstage attemperators within the HRSG were not capable of controlling the steam temperatures for this configuration, final-stage attemperators were used. Today, this is a common design feature and the ASME Turbine Water Induction Prevention guidelines have been modified to allow for this configuration.

3.4 Auxiliary Boiler to Reduce Combined-Cycle Cold Startup Time

Auxiliary boilers are still being used to improve on combined-cycle cold startup times. Auxiliary boilers are used to generate steam for steam turbine seals and to establish condenser vacuum. Based on Sargent & Lundy's investigation, auxiliary boilers improve the cold startup time by approximately 15-30 minutes. This startup time requires the auxiliary boiler to be capable of starting quickly generating the steam needed for the steam seals and condenser vacuum.

4. CONCLUSIONS

The current advanced F, G, H, and J-Class combustion turbines impact the combined-cycle power block and BOP designs. These machines, along with various market demands, result in challenges in overall facility design. These challenges include integration with the steam bottoming cycle and BOP equipment, including the HRSG and steam turbine generator design, combustion turbine cooling air system, and the natural-gas supply system. In addition, the steam cycle is impacted through design requirements for fast startup, base load and cycling operation, high unit ramp rates, high efficiency, high reliability, lower emissions, and lower life-cycle costs, to name just some of the expectations. These techniques and their successful implementation have proven highly beneficial in the use of advanced combustion turbines in combined-cycle applications.

Presented at POWER-GEN Natural Gas, Columbus, Ohio, August 23-25, 2016