IMPACT OF CONTROL STRATEGIES ON THE OFF-DESIGN OPERATION OF THE GAS TURBINE IN A COMBINED CYCLE GAS TURBINE (CCGT) POWER PLANT

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Abstract

The off-design performance of a combined cycle gas turbine (CCGT) plant for power generation is of great importance. In this work, a detailed mathematical model is developed for a triple-pressure reheat combined cycle power plant. Using the model, CCGT off-design operation and efficiency are studied under four gas turbine (GT) strategies. Controlling the turbine inlet temperature (TIT) gives better GT performance, but reduces the turbine exhaust temperature (TET) and the combined cycle efficiency. On the other hand, controlling the variable guide vanes (VGVs) along with TIT gives higher combined cycle efficiency for GT loads between 60% and 100%. Below 60% load, controlling the VGVs and TET outperforms the other operating strategies in combined cycle efficiency. Hence, a mixed operating strategy seems the best for improving the CCGT off-design performance.

Keywords

Modeling, Simulation, Gas turbine, Combined cycle power plant, Off-design performance.

Introduction

Combined cycle gas turbine (CCGT) power plants are widely used due to their high thermal efficiency and low emission. However, due to the frequent peak regulations in the power grid, they are often run at part-load conditions. The part-load operation decreases thermal efficiency, and the investigation of off-design performance of CCGT is an important topic.

CCGT usually comprises a gas turbine (GT) (topping cycle) and a heat recovery steam generator (HRSG) (bottoming cycle). Predicting their off-design performance requires an accurate simulation of the two cycles. Stone (1958) and Doyle and Dixon (1962) applied a stage-stacking method to simulate the off-design performance of multi-stage axial compressors with variable geometry angles. This method could obtain inter-stage parameters (pressure and temperature), and the compressor's overall performance. By adopting the stage-stacking method for the compressor and a stage-by-stage model for the turbine, Lee et al. (2011) developed a general off-design performance prediction program to simulate simple, recuperative, and reheat cycle GTs, which is useful when component maps are not available. Zhu and Sarvanamutto (1992) developed a mathematical model to predict the off-design performance of a three-shaft GT using component matching. They assumed choking in the GT, and began from the hot end to determine compressor operation. Al-Hamdan and Ebaid (2006) studied GT simulation for power generation by superimposing the turbine performance map on the compressor map. Haglind et al. (2009) developed one complex model (using component maps) and one simple model (using turbine constants) for predicting the part-load performance of aero-derivative

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GTs, and found both models to offer good agreement in terms of flow and pressure characteristics. Jimenez-Espadafor Aguilar et al (2014) applied an off-line component based program (GSP) to simulate the off-design performance of a two-shaft GT and studied the regulation methods of combined heat and power plants. Tsoutsanis et al (2015) proposed a novel method for modeling compressor and turbine maps and tuning their parameters to improve performance prediction and diagnostics under off-design, steady state, transient, and degraded conditions.

HRSG off-design modeling is mainly devoted to the correction of overall heat transfer coefficients. Kim and Ro (1997) and Kang et al (2012) correlated the overall heat transfer coefficients of different HRSG sections under off-design conditions with gas flow properties. Ganapathy (1990) developed a HRSG off-design performance calculation procedure, and proposed a detailed method for estimating heat transfer coefficients by considering HRSG design parameters, and turbine exhaust gas parameters. Zhang et al (2015) adopted Ganapathy's procedure to model the off-design performance of the bottoming cycle, and proposed some semi-empirical and semi-theoretical formulas. After overall heat transfer coefficients under off-design conditions are calculated, HRSG simulation starts by applying energy balance and heat transfer equations.

To achieve efficient off-design operation, GTs operating strategies have been of great concern over the years. Kim et al (2003) investigated the effects of variable inlet guide vane (VGV) on single-pressure combined cycle performances of single-shaft and two-shaft GT configurations, and found that VGV modulation increases single-shaft combined cycle efficiency, especially in high load range, but does not improve the efficiency of two-shaft engine, due to the GT performance degradation. Kim (2004) analyzed the part-load performance of GTs and combined cycles with different design parameters, and studied several load control strategies, and observed that a GT with higher design performance exhibits superior part load performances. Haglind (2010a, 2010b) analyzed the effects of variable geometry GTs on part-load efficiency of combined cycles for ship propulsion, and found that combined cycle part-load performance can be improved by variable area nozzle (VAN) and VGV control.

From our literature search, we concluded that no work has addressed the full complexity of a CCGT cycle. While many have modeled individual parts, a complete, detailed, modular, and high-fidelity simulation model for the entire CCGT plant does not exist in the open literature. This is particularly true as far as the use of actual compressor/ turbine performance maps and the study of real triple-pressure reheat combined cycles are concerned. Moreover, as the bottoming cycle is a passive system that utilizes GT exhaust heat to generate steam and power, such a model would provide the basis for studying various GT operating strategies and gaining insights into efficient operation. Lastly, no work has so far studied combinations of GT load control strategies to maintain efficient CCGT operation over a wide load range. This work aims to address all of these gaps in the literature.

System description

Figure 2 shows the schematic of a triple-pressure reheat combined cycle power plant. An air compressor (AC) compresses ambient air and injects it into a combustor. The fuel burns in the combustor, and the flue gas expands in a four-stage gas turbine (GT), of which the first three stages are cooled by bleeding air from the AC to prevent blade overheating. After expansion in the GT, the exhaust gas enters an HRSG that generates steams at three pressures, namely high (HP), intermediate (IP), and low (LP). The HP steam expands in the HP steam turbine (HP-ST), and mixes with the IP steam. The mixed steam is re-heated and expands in the IP ST (IP-ST). The IP-ST exhaust mixes with the LP steam, and expands in the LP-ST. The exhaust from the LP-ST is condensed in a condenser and pumped to an LP economizer to finish the bottoming cycle.

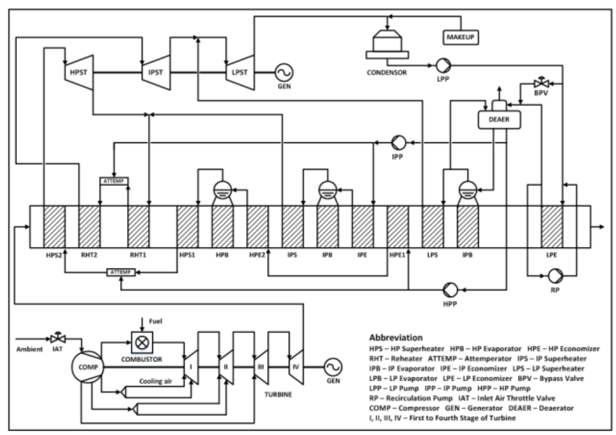

Figure 2. Schematic of a triple-pressure reheat combined cycle power plant.

Mathematical model

In what follows, VEC denotes vane angle correction factor; η denotes efficiency; $\Delta\alpha$ denotes relative change in VGV angle; m denotes mass flow; h denotes specific enthalpy; P denotes pressure; T denotes temperature; and LHV denotes lower heating value; κ is a constant; A denotes area; g denotes gravitational acceleration; R denotes gas constant; and γ denotes specific heat ratio; r is outlet pressure ratio; and ϕ denotes flow coefficient. Furthermore, subscript a is for air, c for AC, cc for combustion chamber, des for design condition, cl for cooling air, f for fuel, w for water, p for BFW pump, g for gas, t for turbine, in for inlet, s for steam, and superscript * for critical condition. We now write down the equations for modeling each component of the CCGT.

Air Compressor (AC)

We assume an *n*-stage, adiabatic, axial flow compressor with an identical pressure ratio for each stage. We use a generic performance map (Fig. 1, Palmer et al, 1993) for predicting its off-design operation. Three rows of VGVs at its inlet control the airflow. The VGV angle affects the AC efficiency as follows (Haglind, 2010a):

$$\eta_c = \eta_{c,map}\left(1 - \mathrm{VEC}\Delta\alpha^2\right) \tag{1}$$

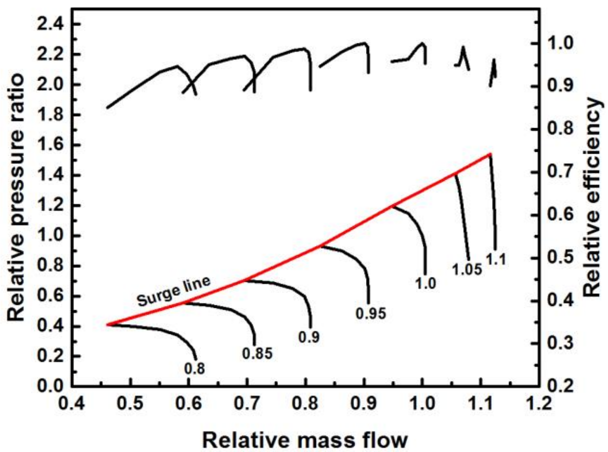

Figure 1. Relativized compressor map

Combustor

For the combustor, we write the following energy balance.

$$m_a h_a + \eta_{cc} m_f \mathrm{LHV} + m_f h_f = m_g h_g \tag{2}$$

Gas turbine (GT)

The off-design operation of a GT can be modeled by a constant swallowing capacity (chocking condition) described by a turbine inlet mass flow, temperature and pressure as follows (Streeter and Wylie, 1979; Palmer et al., 1993):

$$\frac{m_{t,in} T_{t,in}}{\kappa A_{nozzle} P_{t,in}} = C, \quad \kappa = \sqrt{\frac{g\gamma}{R_{gas}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{3}$$

During off-design operation, the distribution of cooling air flow to nozzle vanes and rotor blades is assumed to be unchanged, while the cooling air flow to each turbine stage is calculated from the pressure and temperature of the bleeding stages as follows (Erbes and Gay, 1989):

$$m_{cl} = m_{cl,des}\left(\frac{P_{cl}}{P_{cl,des}}\right)\left(\frac{T_{cl,des}}{T_{cl}}\right)^{0.5} \tag{4}$$

HRSG

The HRSG is just a series of heat exchangers. At design conditions, the energy balance determines the steam flow, temperature, and heat exchanger area. Under off-design conditions, we use the effectiveness-*NTU* method to model HRSG operation. The overall heat transfer coefficients (Erbes and Gay, 1989) and effectiveness (Kays and London, 1984) at off-design conditions are corrected as follows.

$$\left(\frac{U}{U_{des}}\right) = \left(\frac{m_g}{m_{g,des}}\right)^{0.8} \tag{5}$$

$$\varepsilon = f\left(NTU, \frac{C_{\min}}{C_{\max}}\right), \quad NTU = \frac{UA}{C_{\min}} \tag{6}$$

Then, the heat transferred is given by,

$$Q = \varepsilon Q_{\max} \tag{7}$$

The remaining state variables of the gas and water/steam flows are obtained from energy balances.

Steam turbine (ST)

An ST normally has three sections (HP, IP, and LP). We model each section separately using a modified form of Stodola's equation (Erbes, 1986).

$$m_s = \phi\sqrt{\frac{P_{in}}{v_{in}}}\sqrt{1 - \left(\frac{r - r^*}{1 - r^*}\right)} \tag{8}$$

BFW Pump

The isentropic efficiency of a water pump under off-design conditions is given by (Frank, 1995):

$$\frac{\eta_{pm}}{\eta_{pm,des}} = \left(2\frac{m_w}{m_{w,des}} - \left(\frac{m_w}{m_{w,des}}\right)^2\right) \tag{9}$$

Deaerator

The operation of a deaerator is modeled using the following mass and energy balances.

$$m_s + m_{fw} = m_{lpw} + m_{hpw} + m_{vent} \quad (10)$$

$$m_s h_s + m_{fw} h_{fw} = m_{lpw} h_{lpw} + m_{hpw} h_{hpw} + m_{vent} h_{vent} \quad (11)$$

This completes our full model for the CCGT power plant.

Operating strategies

The primary aim of a CCGT is to meet the power demand dynamically. Since GT is the primary source of power, the power is usually controlled by the GT. Two strategies are widely used in practice for tuning the GT load under off-design conditions.

- TIT Control: Adjust fuel flow to change TIT and thus GT output.
- VGV-TET Control: Manipulate the VGV angle at the AC inlet and the fuel flow simultaneously to change GT output, while maintaining constant the turbine exhaust temperature (TET).

However, several alternate strategies are possible. For instance,

- VGV-TIT control: Kim (2004) studied a single-pressure CCGT. They used VGVs to reduce air flow to 85% while maintaining TIT at its design value. Then, they adjusted fuel flow alone to reduce GT load further to 30%, while keeping VGV angle constant. However, modern GTs allow lower VGV angle (Haglind, 2010a), hence a straightforward modification is to continue the VGV-TIT strategy further lower than 85% air flow, leading to an increase in TET. This provides a better heat recovery performance in HRSG and increases ST power. Since TET usually has an upper limit, as the last stage of the GT has no cooling; we stop VGV-TIT strategy, when TET hits 650 °C. For lower GT load, we simply switch to TIT control, while keeping the VGV angle constant.

An alternative is to just replace the VGV control by Inlet Air Throttle (IAT) control in the above. In other words, adjust the IAT valve before the AC and fuel flow to reduce GT load until TET hits 650 °C, while keeping TIT constant. We will call this as IAT-TIT control. Thus, we study four operating strategies for GT control. In this work, we assume that the AC and GT share a common shaft, which has the same constant speed as the ST shaft over the entire load range.

Results and discussion

Figures 3-9 show the off-design performance of GT and combined cycle for the four GT operating strategies. First, consider the TIT control. To reduce the GT load to 30%, the fuel flow reduces by 51.6%, and TIT decreases from 1353.1 °C to 892.7 °C. As we see from Figure 4, the turbine exhaust flow (TEF) remains steady. This happens as the airflow increases by 0.9% to compensate for the decreasing fuel flow. However, TET reduces sharply from 605.2 °C to 393.8 °C, which lowers the bottoming cycle efficiency from 31.1% to 21.2%. This clearly lowers the total power output and overall efficiency of the plant.

In VGV-TET control, TET remains at its design value (605.2 °C), as VGVs close and the fuel flow reduces. In contrast to TIT control, both airflow and TEF decrease, but TET remains steady. The bottoming cycle efficiency remains nearly constant around 30.5%, suggesting the more critical role of TET versus TEF.

In VGV-TIT control, TIT can be maintained at its design value up to 73% GT load, when TET reaches 650 °C. TEF decreases by 20.1%, but the bottoming cycle efficiency increases. For GT loads below 73%, TEF reduces by 0.6% only, but TET decreases by 150 °C. Hence, the bottoming cycle efficiency decreases from 32% to 26.8%.

The behavior of IAT-TIT is similar to that of VGV-TIT. TET reaches 650 °C at about 70% GT load. TEF reduces by 18.9%, and the bottoming cycle efficiency increases by about 1% due to higher TET. For GT loads lower than 70%, the IAT angle is kept unchanged. The TIT control reduces the fuel flow and TIT from 1353.1 °C to 1025.3 °C. TEF remains nearly unchanged, but TET reduces from 650 °C to 495.5 °C. Hence, the bottoming cycle efficiency reduces from 32.0% to 26.6% as with VGV-TIT control.

A comparison of the four strategies suggests that the GT efficiencies are similar, but TIT seems more efficient. However, TIT seems the worst for the combined cycle efficiency. IAT-TIT and VGV-TIT controls are very similar across the load range in terms of both efficiencies. VGV-TET clearly outperforms others under 60% GT load, and has the best overall performance. VGV-TIT has only a marginal edge for loads exceeding 60%. This clearly suggests that a mixed GT operating strategy may be the best for a CCGT plant: VGV-TIT for GT loads above 60% and VGV-TET for loads below 60%.

Conclusion

In this work, a detailed and modular mathematical model is developed for simulating the part-load performance a CCGT power plant. The model was used to study the effects of four operating strategies on gas turbine and combined cycle performance. While TIT control may seem the best for GT efficiency, it is the worst for the combined cycle efficiency. IAT-TIT and VGV-TIT controls are very similar. Finally, VGV-TET control seems the best overall for the part-load control of a CCGT power plant. Our work highlights the disadvantages of using GT control as the preferred control mode for CCGT power plant operation. Clearly, the GT and ST cycles are two integral, significant, and interacting parts of a CCGT power plant, and focusing one at the detriment of the other as done in most literature is not a wise approach. This is the focus of our ongoing work, for which this work has laid the foundation by developing a rigorous high-fidelity model for CCGT plant simulation. Our work also suggests the possibility of mix of operating strategies that may prove the best as the power plant load varies over a wide range.

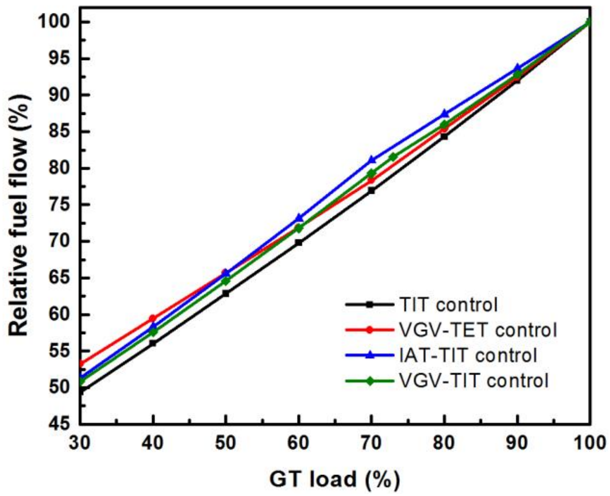

Figure 3. Fuel flow versus GT load.

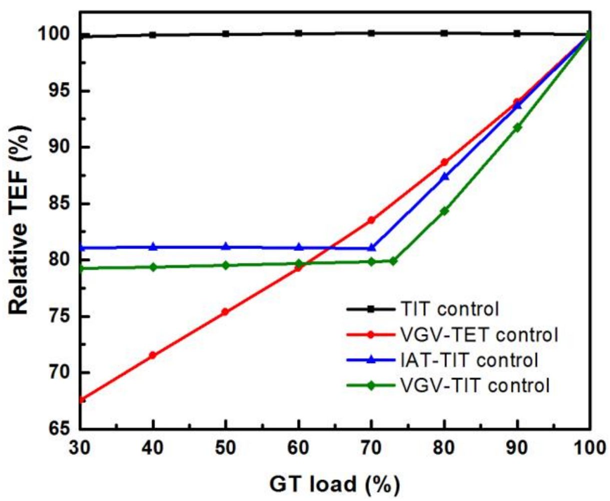

Figure 4. TEF versus GT load.

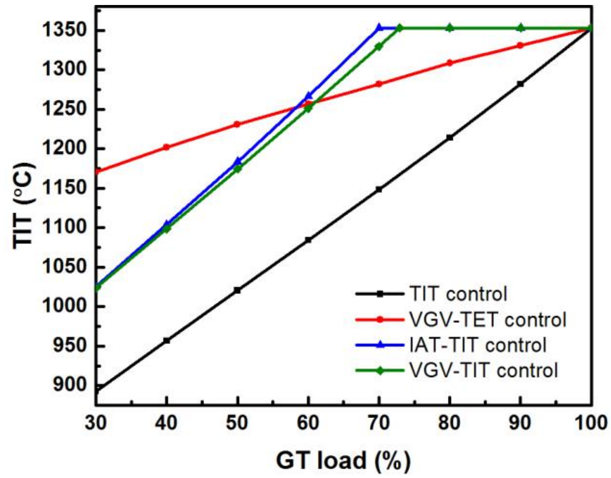

Figure 5. TIT versus GT load.

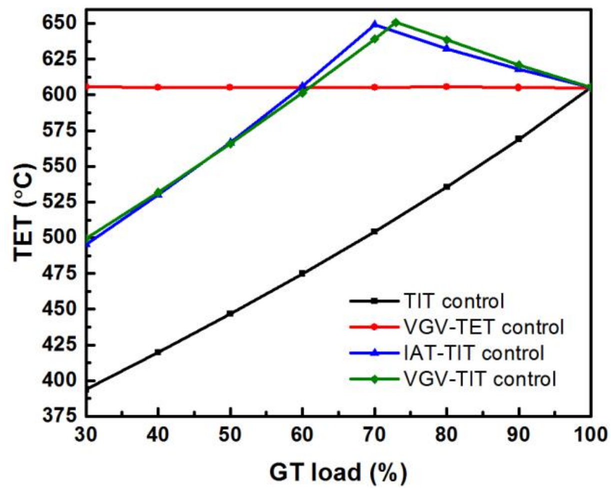

Figure 6. TET versus GT load.

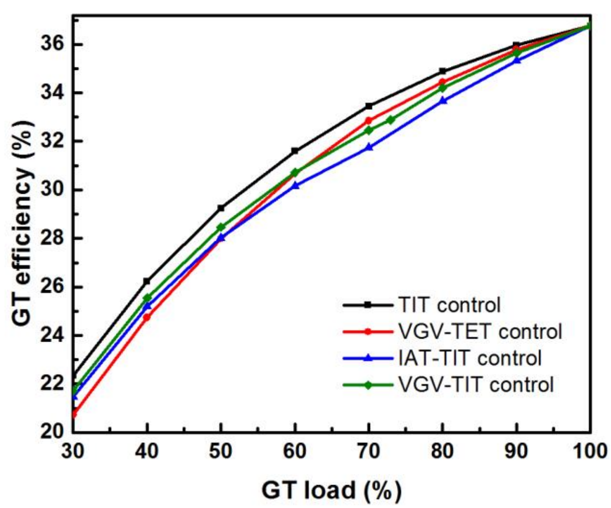

Figure 7. GT efficiency versus GT load.

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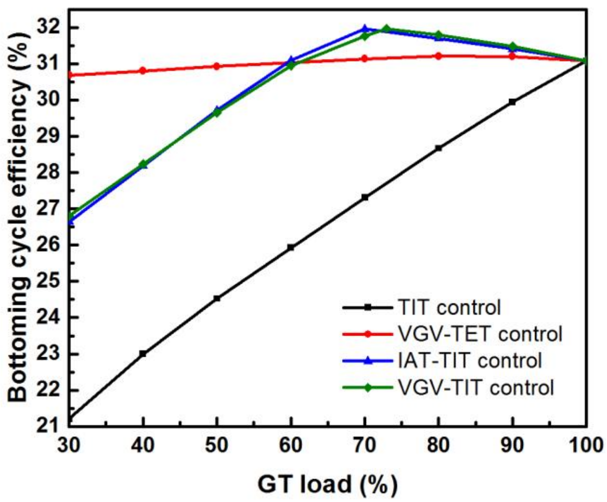

Figure 8. Bottoming cycle efficiency versus GT load.

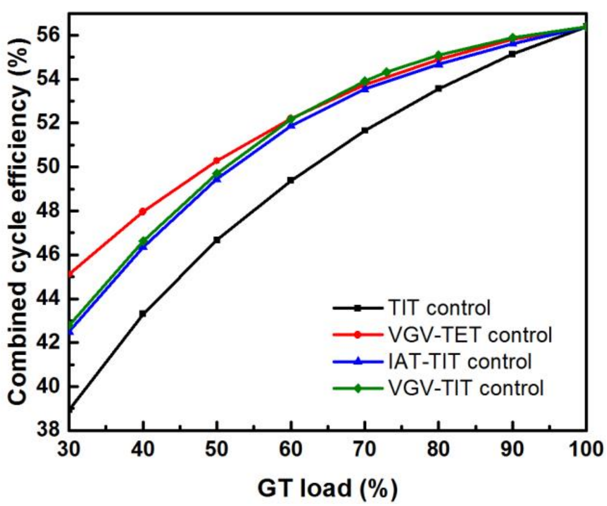

Figure 9. CCGT efficiency versus GT load.

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