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MODELING OF GAS TURBINES AND STEAM TURBINES IN COMBINED CYCLE POWER PLANTS

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MODELING OF GAS TURBINES AND STEAM TURBINES IN COMBINED-CYCLE POWER PLANTS

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ABSTRACT

This document is the result of collaborative work among 27 members and contributors, representing organizations in 11 countries. The organizations included 7 utility companies, 4 manufacturers, 2 universities and research centers, and 4 consultants or engineering service providers.

The report presents a comprehensive document describing the dynamic characteristics of combined-cycle power plants, their control and protection, and modeling of such plants in power system stability studies. This includes:

- An overview of combined-cycle power plants describing the various configurations of these plants and their performance and characteristics.
- A description of the unique aspects of control and protection for gas turbines and steam turbines in a combined-cycle power plant, including a discussion on the frequency regulation capabilities of combined-cycle power plants.
- Dynamic models for modeling combined-cycle power plants, with an emphasis on the appropriate level of modeling detail for power system analysis.
- A discussion on field tests and grid code tests as they pertain to model assessment and validation.

The following provides a brief summary of each of the chapters.

CHAPTER 1 – INTRODUCTION

This chapter describes the scope and objectives of this document.

CHAPTER 2 – COMBINED – CYCLE POWER PLANT OVERVIEW

This chapter presents an overview on the types of combined-cycle power plants, the various configurations and presents qualitative descriptions of the controls and performance of combined-cycle power plants.

CHAPTER 3 – UTILITY PERSPECTIVE – THE NEED FOR BETTER MODELING AND UNDERSTANDING OF COMBINED-CYCLE POWER PLANT PERFORMANCE

This chapter is a concise presentation of the experience of various utilities, from around the world, with combined-cycle power plants. In addition, the need for better modeling and understanding of the performance of combined-cycle power plants is emphasized through examples of the types of system phenomena that require studying by utilities to ensure system security.

CHAPTER 4 – MODELING OF COMBINED-CYCLE POWER PLANTS FOR POWER SYSTEM SIMULATIONS

This chapter focuses on presenting new generic models adequate for the study of combined-cycle power plants in power system studies.

CHAPTER 5 – MODEL ASSESSMENT

This chapter presents an overview of field testing and model assessment for combined-cycle power plants.

CHAPTER 6 – SUMMARY AND RECOMMENDATIONS

This chapter summarizes the content of this report and presents recommendations for modeling of combined-cycle power plants for various types of system studies.

Five appendices at the end of the document provided material that complements the content of the chapters.

CIGRE TASK FORCE 38.02.25 ON
MODELING OF GAS TURBINES AND STEAM TURBINES IN COMBINED-CYCLE POWER PLANTS

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[1] Australian submission by Z. Bozic was prepared with the help of Alfred Li, Bob Stewart and David Tong.

List of Acronyms and Terminology

CAES	Compressed Air Energy Storage
Combined-Cycle	By a combined-cycle is meant the combination of both a gas turbine Brayton Cycle and a steam turbine Rankine Cycle into a single thermal cycle by using the exhaust heat from the gas turbine cycle to produce steam for the steam turbine cycle.
CCPP	Combined-Cycle Power Plant – this term refers to a combined-cycle power plant that encompasses a number of gas and steam turbines operating in a combined-cycle.
DLN	Dry Low NOx – this is a type of combustion system for gas fuel in gas turbines aimed at reducing the emission of nitrate oxides through fuel staging.
GT	Gas Turbine – this term refers to the entire unit i.e. the axial compressor, combustor and turbine assembly.
HRSG	Heat Recovery Steam Generator
IGCC	Integrated Gasification Combined-Cycle
IPP	Independent Power Producer
Multi-shaft CCPP	This configuration of a combined-cycle power plant is comprised of one or more gas turbines each with its own HRSG, feeding steam to a single steam turbine, all on separate shafts with separate generators. For smaller units it is possible to have the exhaust gas from a number of gas turbines all feeding into a single heat-recovery system.
Multi-shaft GT	This is an aero-derivative gas turbine where multiple-spooling is employed. That is, the axial compressor and turbine sections are separated into multiple sections that are mechanically separated, i.e. on separated shafts.
Simple-Cycle	Commonly used term when referencing gas turbines that are operated as stand alone units as opposed to being incorporated in a combined-cycle power plant.
ST	Steam Turbine
Single-shaft CCPP	This configuration of a combined-cycle power plant is comprised of a single GT, ST and generator all connected in tandem to a single rotating shaft. The exhaust of the GT is feed to a single HRSG that generates the required steam for the ST cycle.
Single-shaft GT	This is a heavy-duty gas turbine where the axial compressor, turbine and generator are all connected in tandem on a single rotating shaft. The phrase ‘heavy-duty’ is used to distinguish large single-shaft gas turbines from multi-shaft aero-derivative units.
VIGV	Variable Inlet Guide Vanes

CHAPTER 1

INTRODUCTION

In today's deregulated and competitive electric power market there is a significant demand for power plants with greater efficiency, maneuverability and low emissions. Due to their advantages in these areas, combined-cycle power plants have gained popularity and are beginning to account for a significant portion of the generation mix in many power systems around the world. Approximately two-thirds of the generation capacity in a combined-cycle power plant is produced by gas turbines. Gas turbines and their controls are significantly different from fossil-fuel steam-turbine power plants. In particular, the maximum power output of a gas turbine is highly dependent on the environmental ambient conditions because the gas turbine thermal cycle is an open cycle using atmospheric air as its working fluid. The maximum power output of the turbine is also dependent on the deviation of its operating frequency from its rated speed.

It is crucial to the electric power industry to have a source of information that clearly defines the characteristics of the controls and protection of combined-cycle power plants and their impact on system performance. Furthermore, the industry needs a source of information that can identify appropriate models for representing combined-cycle power plants in power system studies.

This document is the result of a collaborative effort by manufacturers, utility engineers, consultants and research organizations around the world on the subject of modeling combined-cycle power plants for the purpose of power system studies. Power system studies can be, but are not limited to, analyses of the following nature:

- The study of the impact of proposed new generating facilities on an existing power system.
- The study of system small-signal and/or transient stability.
- The study of large system frequency disturbances.
- The study of reactive/voltage stability of a power system.

In all of the studies mentioned above there is a need for an appropriate level of modeling detail. Some require a greater focus on the electrical components of the system and power plants while others require as much attention be given to appropriate modeling of the mechanical systems of power plants. In combined-cycle power plants the electrical components such as the generator and its excitation system are similar in nature to a conventional fossil-fuel power plant, and detailed description of these elements and appropriate models can be found in other documents [1, 2 & 3]. This document focuses on the controls and protection associated with gas turbines and steam turbines in combined-cycle power plants, and the modeling of these elements of the plant in power system studies.

The layout of the document is as follows:

- Chapter 2: an overview is provided of combined-cycle power plants. The various types and configurations of combined-cycle power plants are described together with the controls and protection associated with gas turbines and steam turbines in combined-cycle power plants.
- Chapter 3: several utilities from around the world describe their experiences with gas turbines and combined-cycle power plants, and their concerns with the models currently available in commercially available software. They also describe the phenomena often requiring simulation.

- Chapter 4: provides a brief historical account of models and modeling practices to date for the purpose of modeling gas turbines and combined-cycle power plants. The chapter provides a discussion of modeling combined-cycle power plants, and documents generic models that can be used to model gas turbines and steam turbines in combined-cycle power plants for most of the major manufacturers.
- Chapter 5: provides an account of model assessment with qualitative descriptions of sound practices for field-testing of units and expected unit performance.
- Chapter 6: provides a summary of the contents of this document together with recommendations for model usage.
- Five appendices are provided at the end of the document, which complement the material presented in the main chapters of the document.

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[1] IEEE Std 421.5-1992, IEEE Recommended Practice for Excitation System Models for Power System Stability Studies, IEEE, 1992.

[2] P. Kundur*, Power System Stability and Control*, McGraw Hill, 1994.

[3] IEEE Std 1110-1991*, IEEE Guide for Synchronous Generator Modeling Practices in Stability Analyses*, IEEE, 1991.

CHAPTER 2

COMBINED-CYCLE POWER PLANT OVERVIEW

2.1 Introduction

A combined-cycle power plant (CCPP), in its simplest form, consists of a gas turbine (GT), a steam turbine (ST), a heat-recovery steam generator (HRSG), and an electric generator. A variety of combinations exist, employing multiple gas turbines, HRSGs, and generators in several possible configurations.

A primary advantage of combined-cycle power plants is improved overall plant efficiency. The total thermal efficiency of a CCPP is significantly higher than that of a conventional fossil fuel plant. The higher thermal efficiency is due to the greater utilization of the total enthalpy produced by the combustion process in the gas turbine through the combination of the gas turbine Brayton cycle and the steam turbine Rankine cycle (thus originating the term combined-cycle). A typical simple-cycle conventional fossil fuel plant (e.g. simple cycle GT or coal burning ST plant) has an efficiency of 30-35%, while a CCPP can have efficiencies exceeding 55%.

Thus, the combined-cycle plant represents the integration of two cycles, one being the "topping" or high temperature cycle (the GT Brayton cycle) and the other being the "bottoming" or low temperature cycle (the ST Rankine cycle). The two cycles are coupled by means of a heat exchanger transferring the exhaust low-energy of the topping cycle to the bottoming cycle, hence producing additional work. In a combined gas-steam cycle, the heat out of the gas turbine exhaust gas is recovered in a heat recovery steam generator (HRSG) to produce steam for the bottoming steam cycle as shown in the thermodynamic cycle of Figure 2-1.

The development of gas turbines with higher turbine inlet temperatures has improved efficiency and made the gas-steam combined-cycle power plants a viable alternative to steam power plants. Figure 2-2 shows a typical implementation of the modern combined-cycled power plants.

The relationship of the thermodynamic cycle and the plant equipment is explained in a simple manner here using Figures 2-1 and 2-2. Point 1 of Figure 2-1 represents ambient conditions. Air is brought into the compressor and the compressor raises the pressure and temperature on the incoming gas to point 2. Point 2 to point 3 represents the addition of heat by the introduction of the fuel and its combustion to raise the temperature of the gas. The path from points 3 to 4 represent the expansion of the hot gas through the turbine, transferring energy to the turbine blades and to the shaft for conversion to electric power by the generator. The advantage of the combined-cycle plant is its ability to use the heat remaining in the gas turbine exhaust gas. The GT exhaust is supplied to the HRSG, where its heat energy is transferred to the working fluid of the steam turbine, as indicated by the path from points 4 to 1 and the heat transfer Qc.

The steam turbine process uses the Rankine cycle. The heat from the exhaust gas is transferred to the water in the economizer tubes, increasing the temperature of the water (points A to B). The drum boiler produces steam (B to C) and additional heat energy is transferred to the steam in the superheater (C to D). At this point, the steam is at a high pressure and temperature. The steam is then expanded through the steam turbine (D to E), transferring energy to the steam turbine blades and thus the shaft and electric generator. The steam is then condensed (E to A) and pressurized in a pump (A to B) to start the cycle again.

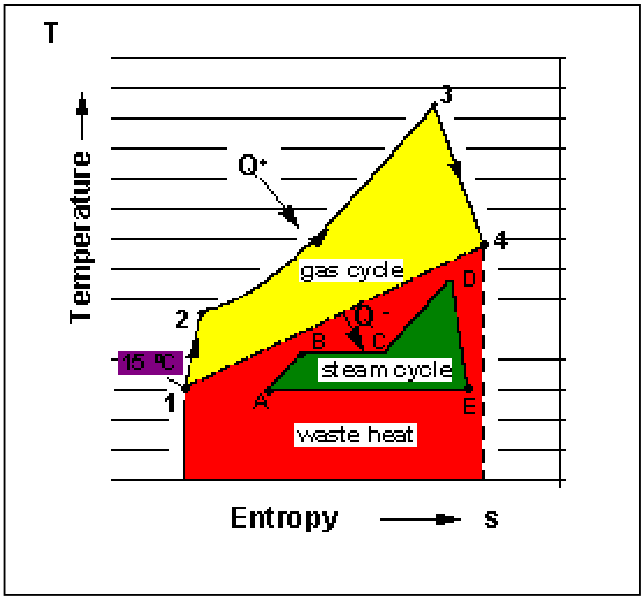


Figure 2-1: Combined-Cycle Diagram in Temperature / Entropy Coordinates

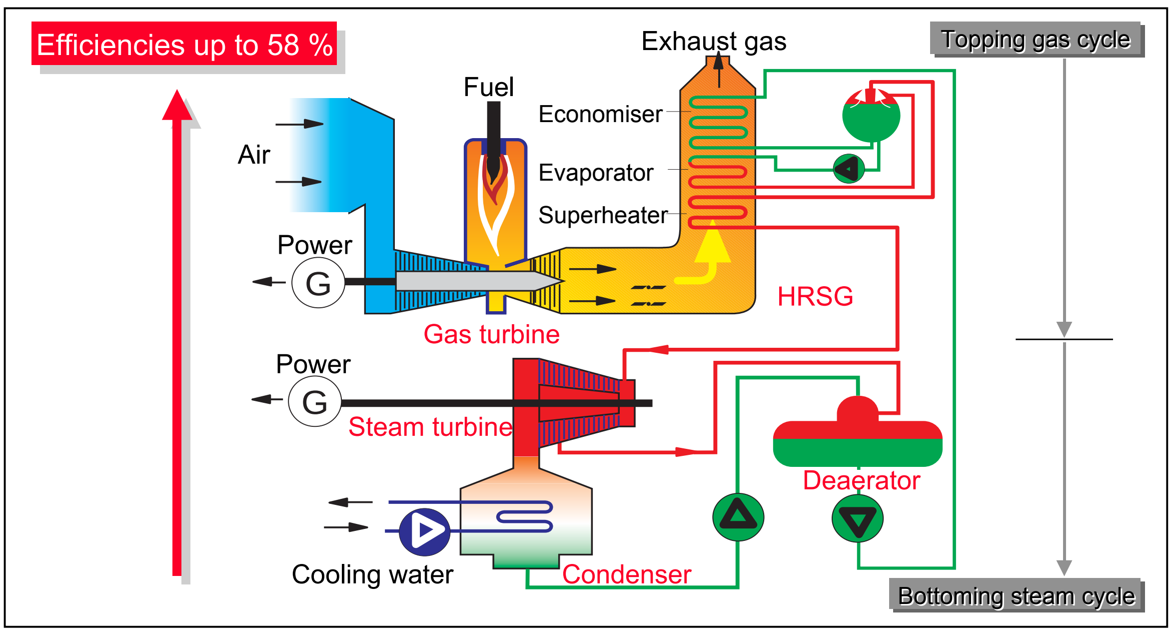


Figure 2-2: Typical gas-steam combined-cycle power plant (Source: Alstom Power AG)

2.2 Configurations of Combined-Cycle Power Plants

Combined-cycle power plants can be configured in a number of arrangements. These can be categorized into two main categories:

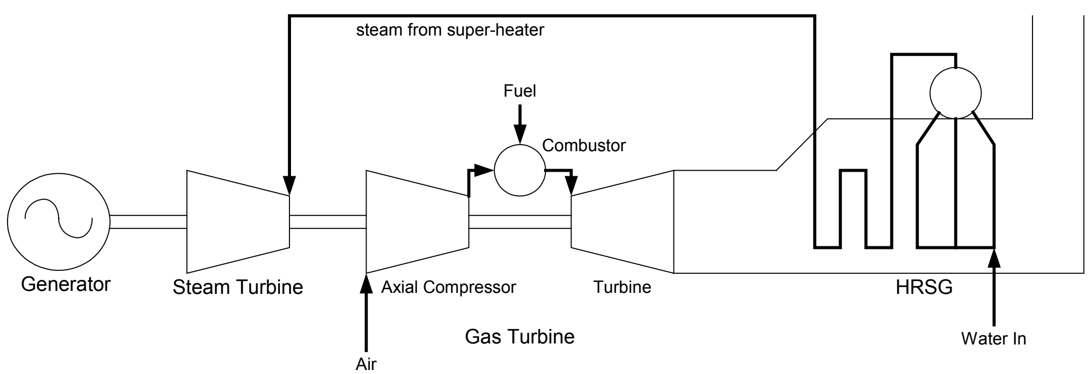
- Single-shaft units, and
- Multi-shaft units.

2.2.1 Single-shaft units

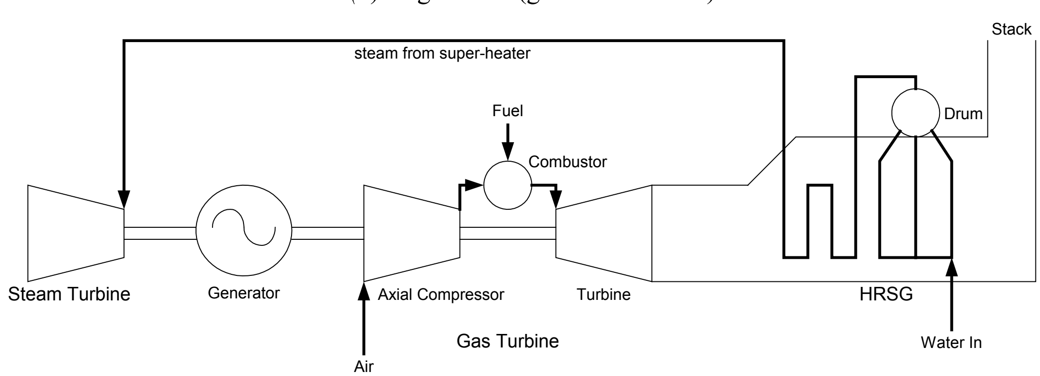
In a single-shaft application the gas turbine and steam turbine are driving the same generator. The advantages of a single shaft design are lower capital cost per MW compared to a single GT/ST multi-shaft unit, a single generator and simpler electrical connections, simpler controls, and a smaller footprint. Today, there are two common designs for single-shaft plants.

One design has the generator at one end, driven by both turbines from the same side. The steam turbine is rigidly coupled to the gas turbine on one side and the generator on the other. This design is shown in Figure 2-3 (a) and is used by several manufacturers.

A second design has the generator between the gas turbine and steam turbine, each turbine driving one end of the generator. The steam turbine engages and disengages with a clutch. This design is shown in Figure 2-3 (b).



*(*a) single-shaft (generator on end)



(b) single-shaft (generator between GT and ST); some manufacturers use this design with a clutch between the ST and generator to allow GT-generator operation independent of the ST

Figure 2-3: Single-shaft combined-cycle power plant configurations

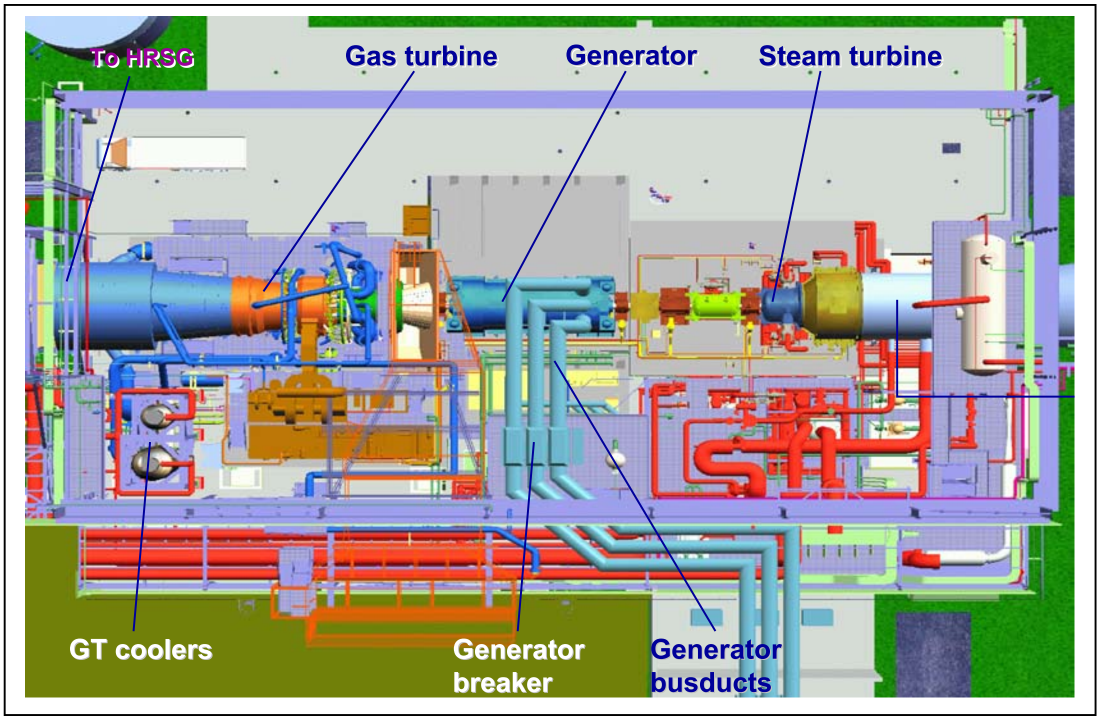


Figure 2-4: Single-shaft combined-cycle power plant (Source: Alstom Power AG)

A typical single-shaft plant layout is shown in Figure 2-4.

2.2.2 Multi-shaft units

A multi-shaft combined-cycle power plant consists of one or more GTs each with is own HRSG, feeding steam to a single ST, all on separate shafts with separate generators. By combining the steam production of all the HRSGs, a larger steam volume enters the steam turbine and in general increases the steam turbine efficiency. For smaller units, it is possible to have the exhaust gas from a number of GTs all feeding into a single heat-recovery system.

Different configurations of multi-shaft CCPPs are depicted in Figures 2-5 and 2-6, for a single GT plant and two GT plant respectively. The process is essentially the same, except for the two GT plant, the steam outputs of the two HRSGs are combined for supply to a single steam turbine. A typical plant layout with two GTs and HRSGs and one ST is shown in Figure 2-7.

Another application of combined-cycle technologies is the re-powering of existing gas turbine plants. The exhausts of existing simple-cycle GTs are used to fire a boiler to produce steam for establishing a steam cycle. Existing steam plants can also be repowered using combined-cycle technology. In re-powering designs the boiler and steam turbine are designed to couple with the GT exhaust, while in modern combined-cycle plant designs the gas and steam cycles in the plant are integrated in the design process in an effort to enhance the overall plant efficiency. Thus repowering schemes have lower overall efficiencies than new CCPP designs, but may still present an attractive option for existing installations.

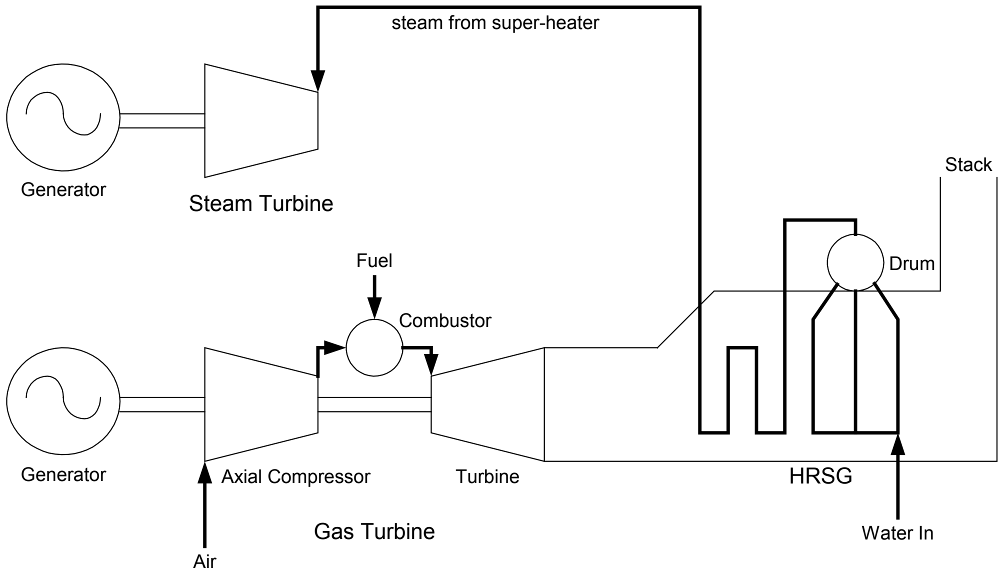


Figure 2-5: Multi-shaft combined-cycle power plant with a single GT

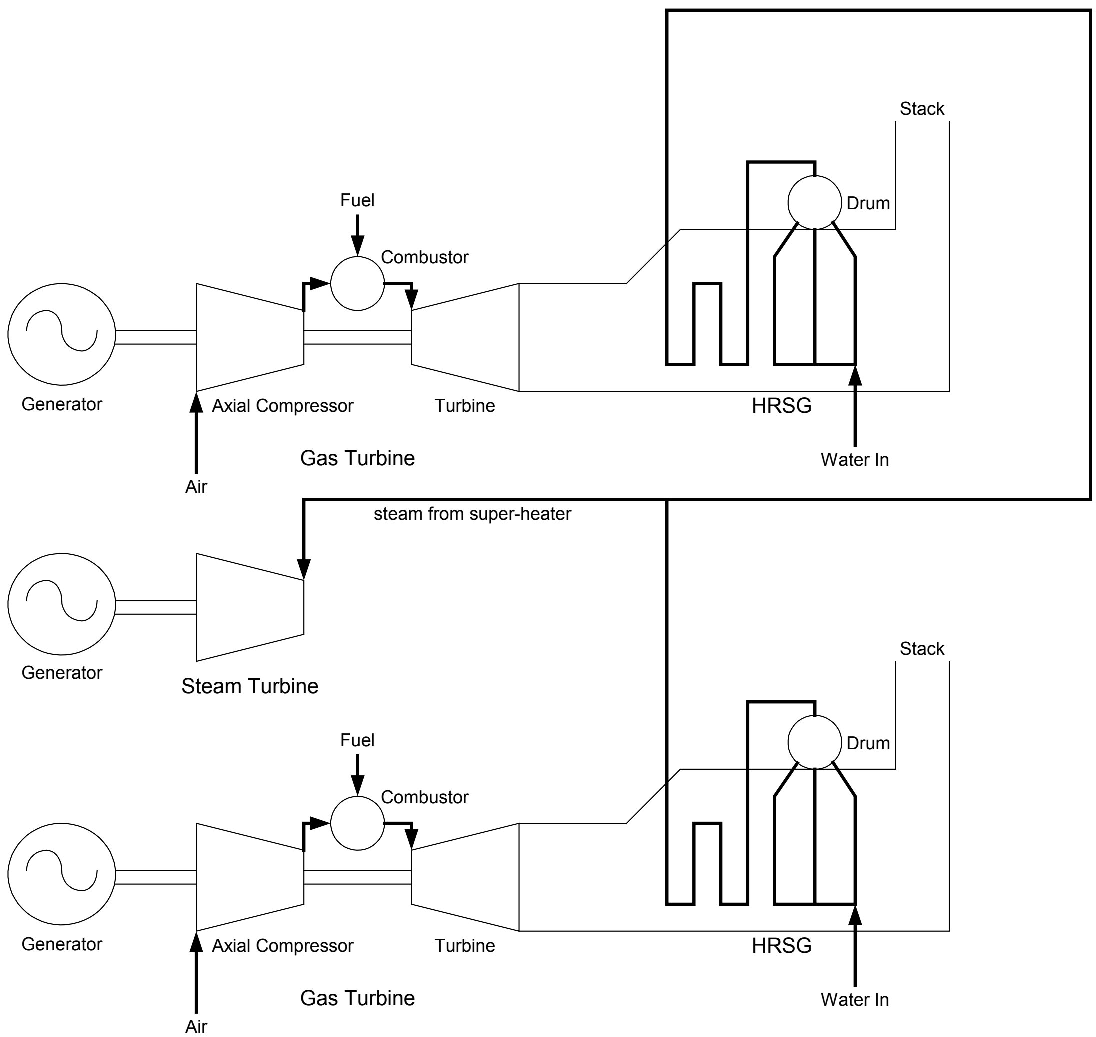


Figure 2-6: Multi-shaft combined-cycle power plant configuration with two GTs

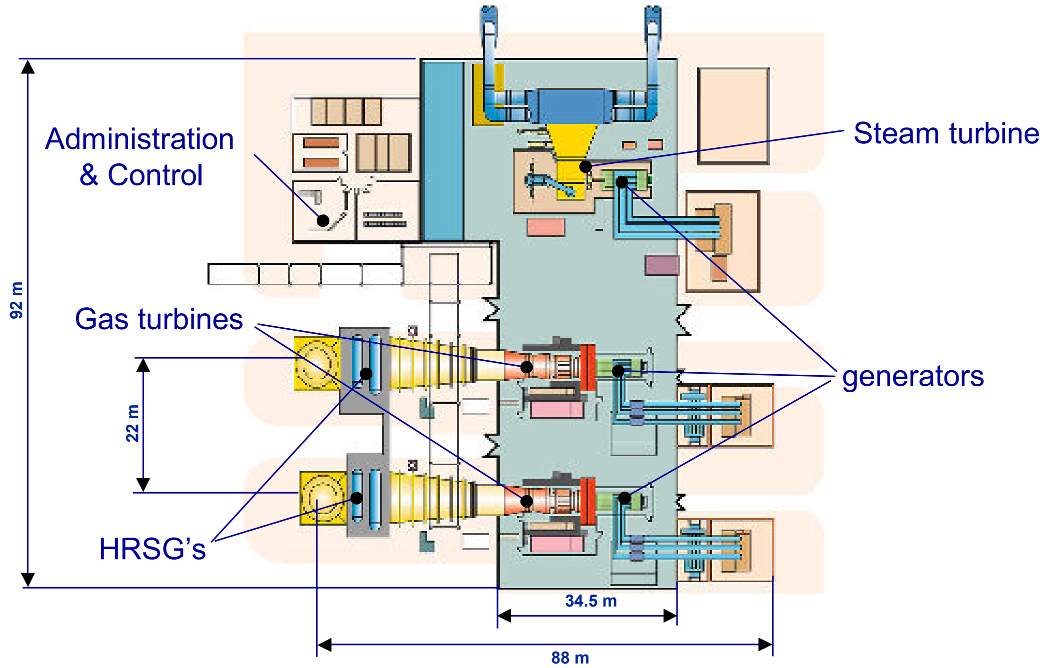


Figure 2-7: Multi-shaft combined-cycle power plant (Source: Alstom Power AG)

2.2.3 Comparison between Single- and Multi-Shaft CCPP Configurations

Single-shaft CCPPs are a modular design and typically the GT cycle cannot operate independently of the ST cycle. Some manufacturers incorporate a clutch system between the GT and ST shaft as described above, and thus can operate the GT independently of the ST. However, to take advantage of the high efficiencies boasted by these units, the GT and ST cycles must be operated together. Thus, the operation of the GT in a simple cycle regime may not be economically attractive.

The multi-shaft CCPPs can be, and frequently are, installed in multiple phases. That is, the GTs are installed and operated first before the ST is installed. This allows, for example, the plant output to grow as load demand increases. Furthermore, in a multi-shaft configuration, the GTs can be operated without the ST, provided exhaust gas bypass stacks and an HRSG damper have been incorporated into the plant design. Again, because of the reduced overall plant efficiency, and particularly in the US where emission constraints are quite strict, the operation of the GTs in a simple-cycle mode may not be economically attractive. Another advantage for multi-shaft configurations is that generally for units with multiple GTs, reduced capital costs result, based on the use of a single steam turbine.

2.2.4 Cogeneration / Supplementary Firing

Cogeneration is the sequential production of the heat necessary for industrial process (generally in the form of steam) and power production by means of recovery of energy from this production. Power can be "cogenerated" in either the topping or bottoming cycles. The steam requirements of the process (steam volume, pressure, and temperature) determine the optimal system configuration. In the combined-cycle systems described above, the steam from the steam turbine is generally condensed, but could obviously be used for an industrial process if the need existed. If steam requirements are different than those resulting from a standard cycle, for example if significantly more steam is required, supplemental firing (also known in Europe as 'post combustion') of the HRSG can be incorporated. Thus, supplementary firing is most often applied in combined-cycle cogeneration plants where the amounts of process steam must be varied independently of the electric power generated. Figure 2-8 shows a configuration that allows increased steam generation by adding a supplementary firing capability to the HRSG. Different fuels can be used for the supplementary boiler and the GT, providing an added flexibility. This option is particularly attractive where cheap alternate fuel, such as coal, is available. Supplementary-fired units will typically have an overall lower thermal efficiency than the standard CCPP and thus are commonly used only for cogeneration applications.

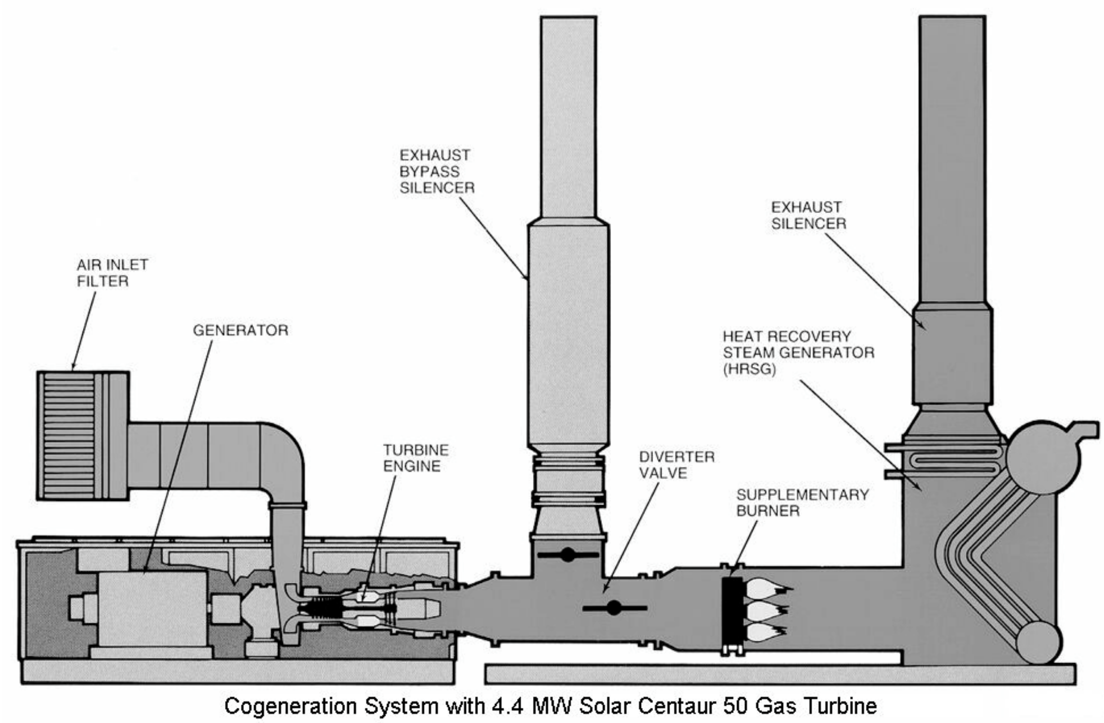


Figure 2-8: Typical plant process with supplemental firing (Source: Solar Turbines).

2.3 Main Components of Combined-Cycle Power Plants

An overview of the combined-cycle power plant was given in the previous section. This section gives more details on the major components of the combined-cycle power plant.

2.3.1 Gas Turbines

Gas turbines usually consist of an axial compressor, a combustion chamber and a turbine operating under the Brayton cycle. These three elements form the thermal block complemented by the air intake system, the exhaust system, auxiliaries and controls. At the compressor inlet casing, air is drawn into the axial compressor and compressed through multiple stages of stator and rotor blades. At each stage, the rotor blades add kinetic energy to the air while the stator blades convert the kinetic energy to potential energy by raising the static pressure of the air. The net pressure ratio of the entire axial compressor is typically between 15 to 20. The compressed air exiting the axial compressor is then mixed with fuel in the combustion chamber, where the combustion process takes place. The hot gas resulting from the combustion process is expanded through a multi-stage turbine to drive the generator and the compressor.

The exhaust gas flow and the temperature of the exhaust gas determine the power input to the heat recovery steam generator (HRSG). The fuel flow determines the power output of a gas turbine. The fuel flow and airflow together determine the firing temperature, which is the gas temperature at the exit of the combustion chamber. The fuel flow and airflow are adjusted based on the measurement of the exhaust temperature and the compressor pressure ratio in order to keep the firing temperature below a design limit. The compressor pressure ratio is determined from measurements of the inlet and discharge air pressures of the compressor.

The airflow can be adjusted by changing the angular position of the variable inlet guide vanes (VIGVs). Inlet guide vanes are essentially the first few stages of stator blades within the axial-compressor assembly. By reducing the airflow, the exhaust temperature is kept high at reduced loading levels to maintain the desired level of heat transfer into the HRSG and maintain an overall higher plant efficiency [1]. When the gas turbine is loaded close to base load, the VIGVs are wide open. The airflow is a function of the VIGV angle, ambient temperature at compressor inlet, atmospheric pressure, and the shaft speed.

There are in general three types of combustion chamber designs (i) annular, (ii) can type and can-annular, and (iii) silo combustors. The terms refer to the physical shape and layout of the combustion chamber. Figure 2-9 shows an example of an annular design where the combustion chamber is a single annulus shaped chamber. Figure 2-10 shows a can-annular design where the individual combustion cans are spaced equally apart around the casing in an annular pattern and are connected using crossfire tubes. In the silo design, Figure 2-11, a single combustion chamber is mounted vertically on the turbine thus resembling a ‘silo’.

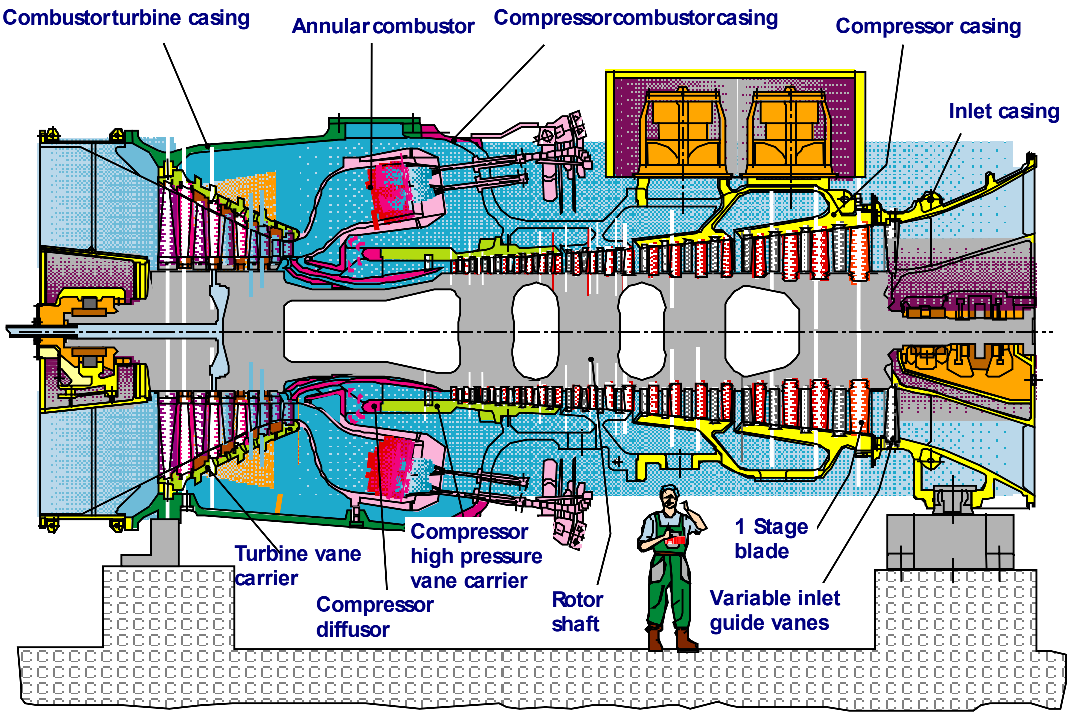


Figure 2-9: Heavy duty gas turbine with annular combustor (Source: Alstom Power AG)

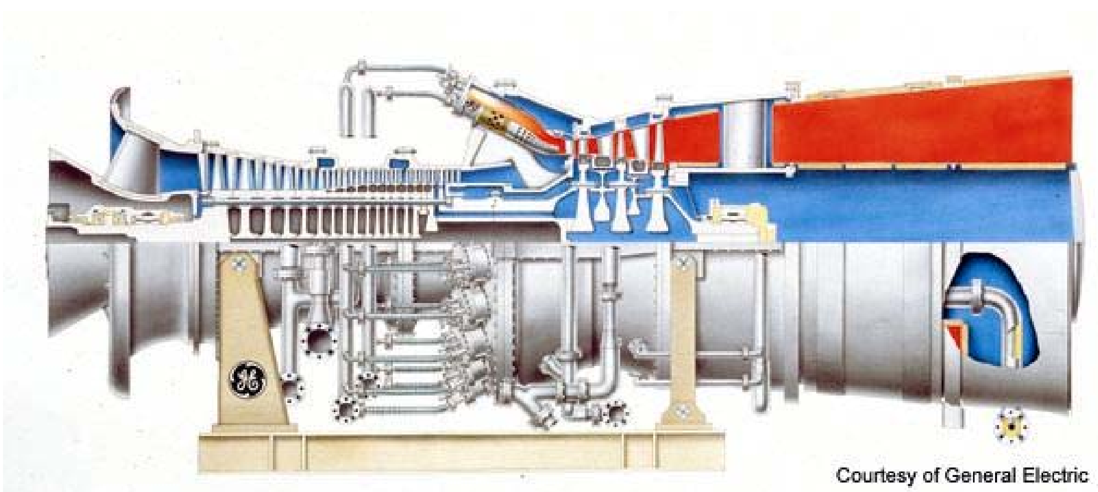


Figure 2-10: GE Frame 9 heavy-duty gas turbine (Source: GE Power Systems)

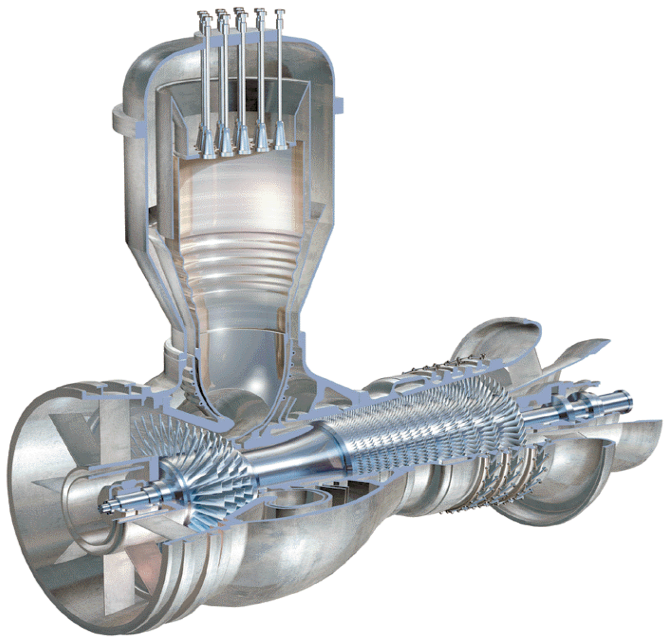


Figure 2-11: An example of a gas turbine with a silo combustor (Source: Alstom Power AG).

Heavy Duty versus Aero-Derivative Gas Turbines

Power generating gas turbines can be classified into two categories:

- Heavy-duty industrial (power) gas turbines.
- Aero-derivative gas turbines, consisting of an aircraft engine modified for industrial duty.

Modern heavy-duty power gas turbines used for large power generation applications such as the GE Frame 7 and 9 turbines, the Alstom GT26 & GT24, the Siemens-Westinghouse 501F & V94.3 and the Mitsubishi M701F have a multi-stage axial compressor, a multi-stage turbine and a generator all connected in tandem on a single shaft. The axial compressors on these turbines typically have a large number of stages (e.g., 17 or more) while the axial turbine will typically have 3 to 5 stages. Approximately 30% of the gross power output of the turbine is consumed by the axial compressor in the compression process. Modern machines generally have an annular chamber with multiple burners integrated into a thermal block. Heavy-duty power gas turbines range in size from about 25 to 250 MW. Figures 2-9 to 2-11 show examples of large heavy-duty gas turbines. Figure 2-12 shows an example of a smaller heavy-duty gas turbine.

For small power generation and industrial applications in the 10 to 50 MW ranges, aero-derivative gas turbines are commonly used. These gas turbines, as their name suggests, have been derived from aircraft engines. Aero-derivative gas turbines are normally a two or three stage turbine, with a variable speed compressor and driving turbine. The combustion chamber is generally of the can type. Multistage axial compressors designed to achieve high-pressure ratios are prone to aerodynamic instability if operated at rotational speeds that are widely removed from their designed operating point [2]. In order to overcome this difficulty, given the wide range of operating speeds required of aircraft engines, a design feature of these turbines is *multiple spooling*. In multiple spooling, the axial compressor and/or turbine is separated into multiple sections that are mechanically separated, i.e. on separate shafts. Figure 2-13 shows, schematically, an example aero-derivative turbine where the compressor is connected to the high-pressure turbine. Power is then extracted from the LP turbine, which is on a separate shaft. Figure 2-14 shows an aero-derivative unit. Examples of aero-derivative units are the GE LM2500, LM6000 and Alstom GT10.

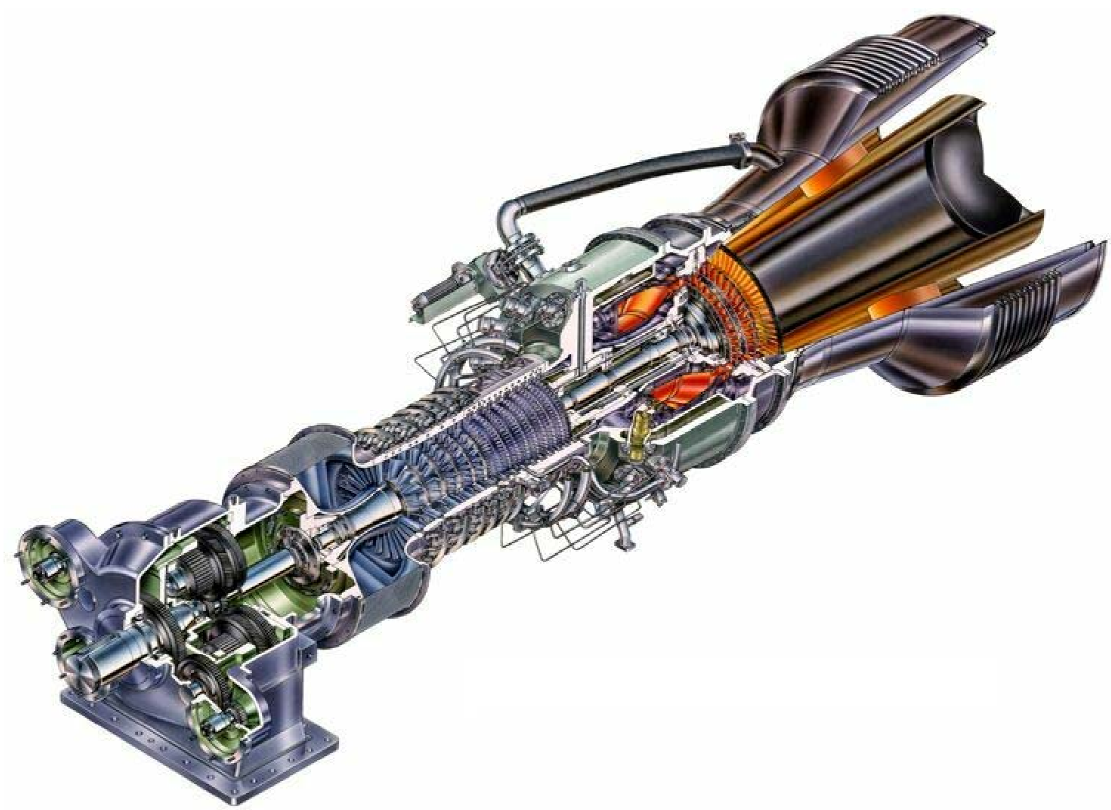


Figure 2-12: Heavy duty gas turbine, the Titan 130 gas turbine (Source: Solar Turbines)

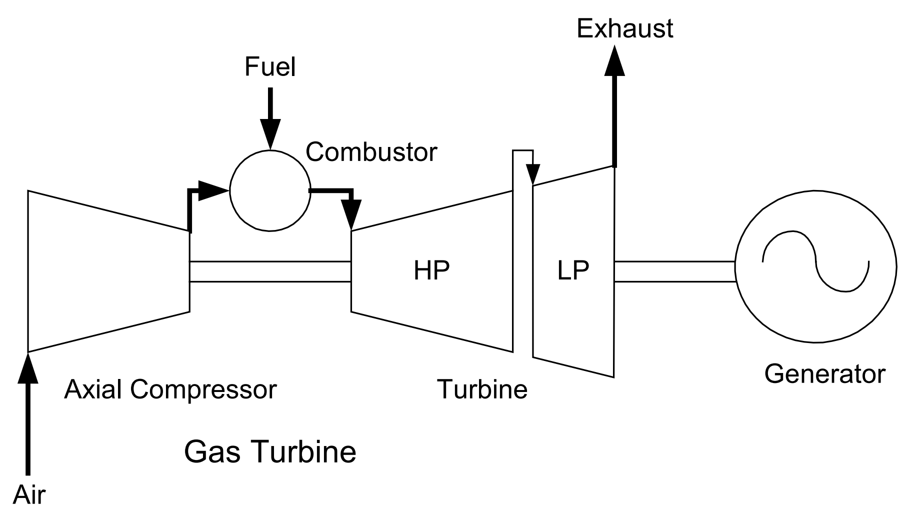


Figure 2-13: Multi-spool aero-derivative gas turbine

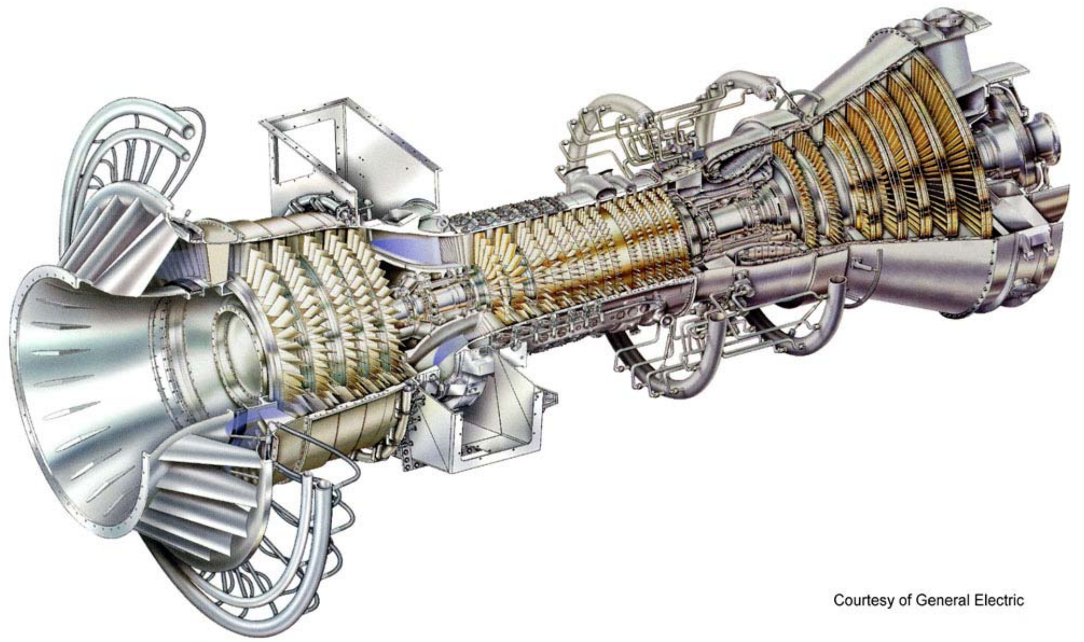


Figure 2-14: An example of an aero-derivative gas turbine, the GE LM6000 (Source: GE Power Systems)

The key differences between heavy-duty gas turbines and aero-derivatives are:

- As seen from the electrical network the combined moment of inertia of the turbine and generator (H) is significantly higher for a heavy-duty gas turbine as compared to a typical aero-derivative turbine. This is because typically for an aero-derivative unit the power turbine and generator are on a separate mechanical shaft than the high-pressure turbine and axial compressor. Thus, the combined moment of inertia of the multi-stage axial compressor and turbine for heavy-duty turbines is much larger than the relatively light LP turbine that drives the generator load on a typical aero-derivative unit.

- For cases where all compressor stages of an aero-derivative turbine are on a separate mechanical shaft than the generator and power turbine, the unit may not be as susceptible to operating constraints due to severe variations in system electrical frequency. The primary constraint on prolonged off-nominal frequency operation for heavy-duty gas turbines is the compressor characteristic [3]. By mechanically decoupling the compressor from the generator, the operating limits of the gas turbine become dominated by the operational characteristics of the high-speed shaft[1] and differ from the behavior of a heavy-duty gas turbine [4].

[1] By the high-speed shaft is meant the shaft connecting the HP turbine and compressor, which typically runs at a much higher speed than the shaft connecting the LP power turbine to the generator, in an aero-derivative gas turbine.

2.3.2 Heat Recovery Steam Generator

The heat recovery steam generator is the link between the gas turbine and the steam turbine process. There are three main categories:

- HRSG without supplementary firing. An HRSG without supplementary firing is essentially an entirely convective heat exchanger. No additional fuel is burned in the exhaust gases. The majority of the large CCPPs built and operated today use unfired HRSGs.

- HRSG with supplementary firing. Additional fuel is burned in the exhaust duct to increase steam generation. Supplementary firing is most often applied in combined-cycle cogeneration plants where the amounts of process steam must be varied independently of the electric power generated. In this case, supplementary firing controls the amount of process steam generated. For such cogeneration applications, at low loads the exhaust gases may be diverted and the steam is produced by independent firing [4].

- Steam generators with maximum supplementary firing. The gas turbine is in fact replacing the forced draught air blower, feeding hot combustion air into the boiler. Application is mainly for repowering of an existing power plant.

The function of the HRSG is to convert the exhaust energy of the gas turbine into steam. After heating in the economizer, water enters the drum, slightly subcooled. From the drum, it is circulated to the evaporator and returns as a water/steam mixture to the drum where water and steam are separated. The saturated steam leaves the drum for the superheater where it reaches the maximum heat exchange temperature with the hottest exhaust gas leaving the gas turbine. This is illustrated in Figure 2-2.

The heat exchange in an HRSG can take place on up to three pressure levels depending on the desired amount of energy to be recovered. Today, two or three pressure levels of steam generation are commonly used. Most modern HRSGs are of once-through boiler type. They are further classified into horizontal and vertical HRSGs, referring to the direction of the flue gases through the heat transfer section. The main advantage of horizontal HRSG is that no circulation pumps are needed. Figure 2-15 shows a typical horizontal HRSG. Figure 2-16 shows a typical combined-cycle power plant installation.

Figure 2-15: A horizontal heat-recovery steam-generator (Source: GE Power Systems)

Figure 2-16: A multi-shaft combined-cycle power plant showing two gas turbines, a steam turbine and HRSGs (Source: GE Power Systems)

2.3.3 Steam Turbine

The steam turbine consists basically of a casing supporting the stationary blades, a rotor with the rotating blades supported on journal bearings, and main stop and control valves.

Depending on the capacity of the steam turbine, it can be constructed with a single casing or have multiple casings. The type of cycle and the exhaust volume flow of the steam to the condenser will determine the number of casings to be used. Multi-casing steam turbine design is often the choice when considering reheat cycles, generally used in large combined-cycle power plants. Figure 2-17 shows a cross section of a two casing reaction type steam turbine. Note that this turbine has a high pressure (HP) turbine and a combined intermediate (IP) and low pressure (LP) turbine. In a reheat system, typical of large combined-cycle power plants, steam exiting the high pressure turbine will be routed back through the HRSG to receive additional heat energy before proceeding to the IP and LP turbine stages.

In a combined-cycle system, the steam turbine can be operated in two different modes, sliding pressure or fixed pressure control. In practice, a combination of these operation modes is common for combined-cycle power plants, depending on the level of power output.

During sliding pressure control, the throttling or control valves are fully open. The steam pressure is a function of the steam mass flow entering the steam turbine. The load (power output) of the steam turbine depends on the mass flow and is not directly controlled. Thus the load on the steam turbine can only be increased by increasing the steam flow, which, of course, involves generating more steam in the HRSG and generally requires an increase in heat from the GTs or supplemental firing, if present. Thus steam units operating in sliding pressure mode will not respond significantly to governor action in the first seconds following an event on the power system, and may take a minute to several minutes to respond with a significant increase in power. When operating near full power, most combined-cycle plants operate with sliding pressure control of the steam turbine

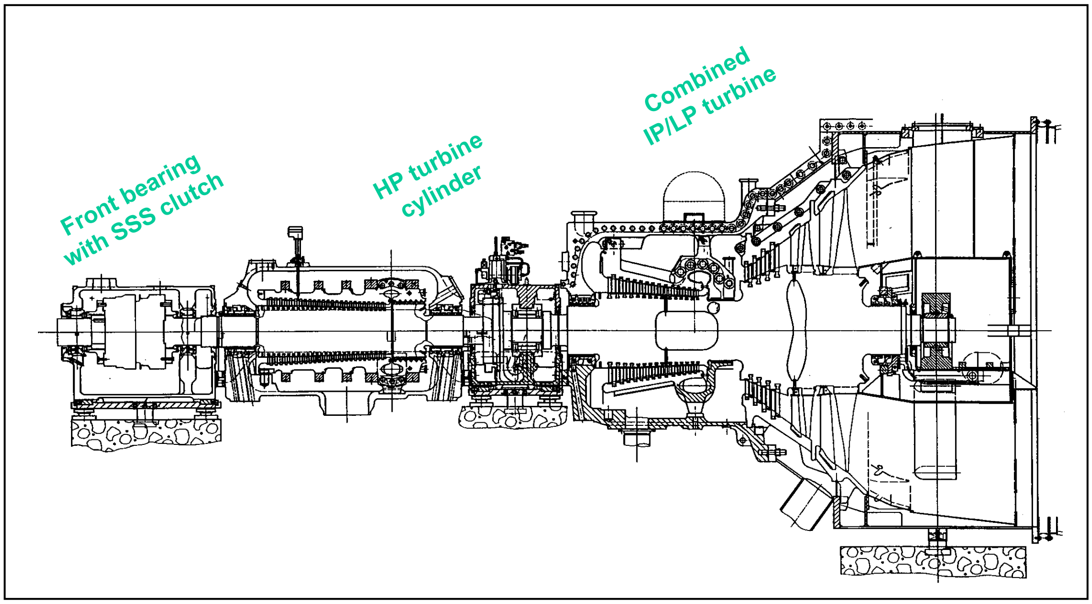


Figure 2-17: Cross section of a two casing reaction type steam turbine. For single-shaft CCPP, the Self-Syncrhonizing (SSS) clutch allows operation of the gas turbine, independently of the steam turbine.
(Source: Alstom Power AG)

When operating under fixed pressure control, the control valve position (valve opening) is changed to throttle the steam flow, thereby keeping pressure at the desired level. By partly throttling the steam flow, a better part-load efficiency of the steam turbine can be achieved.

2.3.4 Generator

Generators for combined-cycle power plants are essentially the same as any high speed generator. The electrical controls and protection associated with the generator are no different than that employed in a conventional fossil fuel power plant.

2.4 Controls of Combined-Cycle Power Plants

2.4.1 Plant Controls

The controls of a combined-cycle plant are quite complex. However, here we will address only those control loops which either directly affect the response of the power plant to power system disturbances or have an effect on the design or operation of the plant.

Load control and frequency response of a combined-cycle power plant are handled by the main plant control system. An overall plant load control system receives a load set-point signal and determines how the gas turbine should be loaded. The steam turbine is generally operated in sliding pressure mode with fully open steam turbine valves down to approximately 50% live steam pressure. Thus, the electrical output of a combined-cycle power plant without supplementary firing is controlled by the gas turbine only. The steam turbine will follow the gas turbine by generating power with whatever steam is available from the HRSG.

After a gas turbine load change, the steam turbine load will adjust automatically with a few minutes delay dependent on the response of the HRSG. There have been suggestions that independent load/frequency control of the steam turbine should be provided for sudden load changes. However, such systems would require the steam turbine to be operated under continuous throttle control, resulting in significantly lower efficiencies at full and part load conditions.

No distinction was made between load and frequency control above. An important aspect of the load/frequency control is the ability of a plant to react to rapid fluctuations in frequency that may occur due to some incident occurring in the electrical grid. This frequency response of the power plant must occur within seconds whereas the loading of the plant typically takes place over several minutes.

In order to sustain stable operation and extend the life of the gas turbines, a frequency dead-band may be introduced in the control system within which the plant will not respond (for example, in the US, typically a deadband of 0.025% is introduced into the speed governor control loop). Outside this dead-band, a droop setting is followed. The droop characteristic setting is defined during the planning phase by the grid operator and is in the range of 3 to 8 %, (typically 4 to 5 %). Combined-cycle power plants can be operated to supply frequency support (spinning reserve). For frequency support, the gas turbines are typically operated between 40 and 95% load, resulting in a proportionate partial loading of the steam turbine.

2.4.2 Gas Turbine Controls

Figure 2-18 shows the typical controls associated with a gas turbine in a combined-cycle power plant. The start-up and shutdown controls are numerous control loops and sequencing logic keyed to ramping the unit up during start-up and down during shutdown. The start-up controls ensure proper purging of the gas paths, establishing the flame, controlling acceleration and proper warming up of the hot gas paths before loading the turbine. These controls are not pertinent to power systems analysis. Typically, the acceleration control loop is active during start-up and shutdown periods as its set-point is varied through these processes. When the unit is on-line, the acceleration set-point is typically set in the order of magnitude of 1% per second, per second. Such acceleration is relatively unlikely in large systems such as the Western or Eastern interconnection in the US, even for an extreme system load/generation unbalance. Thus, the acceleration control can typically be ignored for power system studies for large interconnected systems. However, for islanding studies, smaller power systems, and particularly in the case of aero-derivatives, the acceleration control loop may need to be considered.

Combustion within the gas turbine cycle is a complex process and the detail design of the control aspects of the combustion process are outside the scope of this document. In this document, we will simply discuss some of the major concepts and issues concerning the combustion process that are pertinent to an understanding of the performance of gas turbines and thus combined-cycle power plants as they relate to the power systems. The key challenges of combustion design are [2]:

- To maintain a stable flame over a wide range of fuel/air ratios from full-speed no-load conditions to full-speed full-load conditions.
- To control the emission of CO, NOx, SOx and unburned hydrocarbons and particles such as soot or smoke.
- To ensure the structural integrity of the combustion chamber and components over the expected life of the unit.
- To maintain the temperature and spread of temperature of the gases after combustion at a known acceptable level to prevent thermally over stressing the turbine materials and thus reducing the operating life of the turbine.

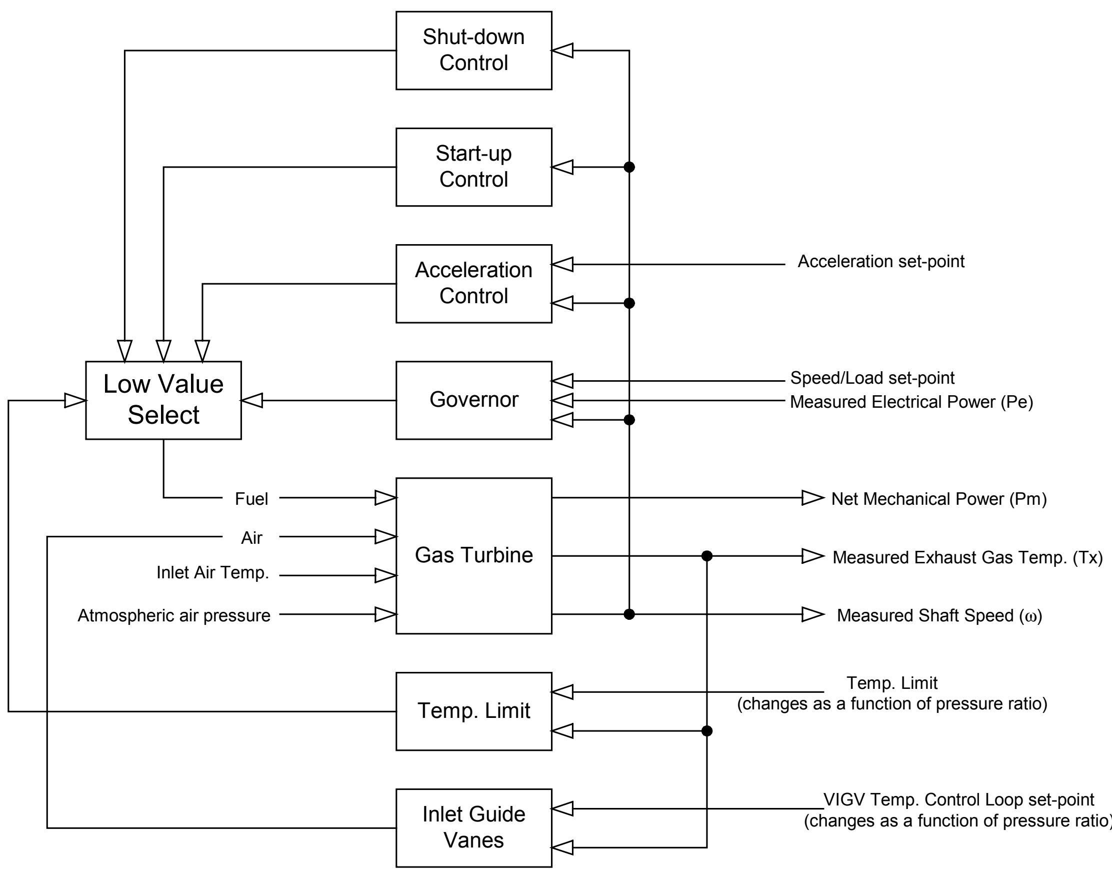


Figure 2-18: Gas turbine control diagram

Flame stability is an important design goal, but the details are not relevant to this document, as modeling the combustion process is too complex and not relevant to systems analysis. However, the planning engineer should be aware that some earlier designs of combustors and their associated controls were prone to flameout due to sudden abrupt control commands to decrease fuel flow (e.g. as a response to sudden increase in system frequency in a small islanded system). This was primarily due to sudden transitions through the numerous combustion modes, which under normal loading and unloading rates take many tens of seconds, resulting in low fuel to air ratio and thus quenching of the flame. It is incumbent on the host utility/industry and turbine manufacturer to consult and thus understand the potential for such transients and thus to protect against the possibility of flameout through control design and sound operating practice.

Control of emissions is also a very important concern. This can be achieved by steam injection, water injection, selective catalytic reduction or dry-low NOx combustion, the latter of these being applicable to gas fuel combustion. Again, modeling any of these processes may be inappropriate at the power system simulation level. However, the engineer should be aware that steam and water injection would tend to increase turbine output while decreasing thermal efficiency. Dry low NOx combustion involves moving through multiple modes of combustion until reaching a premix-mode. In premix-mode, most of the fuel is pre-mixed with air to achieve a lean mixture of air and fuel and thus lower peak flame temperatures, which results in a reduction in the formation of NOx.

Of particular relevance to power system analysis are the governor speed/load controls and the temperature limit control loops. In some cases, particularly for aero-derivatives, the acceleration control loop may also be of importance (note: for an aero-derivative the controls associated with the high-speed shaft may also be of relevance, particularly for islanding studies [4]). The limiting of gas temperatures at the exit of the combustion process is essential to limit stress to turbine components. This is done through limiting fuel flow and controlling airflow as a function of fuel flow and loading. It is important to capture the basic nature of this control in order to represent the temperature limit control on the gas turbine in system studies.

In addition, for combined-cycle operation an understanding of the VIGV controls is pertinent. The VIGV is controlled to maintain good steam conditions over partial load ranges by maintaining high gas turbine exhaust temperature at reduced gas turbine loadings. In many designs, it is the exhaust temperature of the turbine that is measured and thus controlled since it is more readily measured; the relatively higher temperature and temperature spread at the turbine inlet make the turbine inlet temperature less amenable to measurement. The exhaust temperature limit is not a constant and changes as a function of ambient conditions. Some manufacturers use gas turbine inlet temperature (TIT) control where the TIT is controlled by a combination of the fuel flow admitted to the combustor and the VIGV setting.

2.4.3 Steam Turbine Controls

As noted above, the steam turbine is generally operated in sliding pressure control. The control valves for the steam turbine are operated wide open and the steam turbine output is a function of the GT loading and related heat input to the HRSG. At reduced loads, in the order of 30 to 50% of full load depending on the design, the control valves may be partially throttled to maintain steam pressure.

Thus, the steam turbine generally does not respond quickly to system disturbances resulting in a drop in frequency (machine speed). Governor action for a rise in frequency will generally occur, resulting in a closing of the control valve, often with a deadband to prevent operation for small frequency changes. Though not commonly practiced, suggestions have been made to implement primary frequency control on the steam turbine governor in a combined-cycle power plant. Such systems would provide some immediate response out of the steam turbine following a sudden increase in load demand (decrease in system frequency). This control would be coordinated with the inlet pressure control and would result in constant throttling of the steam values thus significantly lowering the efficiency of the steam turbine as compared to operation in sliding pressure mode.

The control of the steam system involves many control loops similar to a conventional steam plant. For example, feedwater must be controlled to maintain proper drum levels; feedwater temperature must be controlled and feedwater hot well level. The boiler controls are simpler than a conventional steam plant in that the heat source is not controlled (i.e., fuel is not controlled to maintain steam conditions) but results directly from GT output.

Units with supplemental firing require additional controls to adjust the supplemental heat input, generally with outer loop controls to maintain steam conditions for external process use, in the case of cogeneration applications.

2.5 Emerging Technologies

2.5.1 Integrated Gasification Combined-Cycle (IGCC)

Combined-cycle plants generally are designed to use natural gas as the primary fuel source. In an integrated gasification combined-cycle (IGCC) plant, the fuel source can be heavy oil, coal, petroleum coke, or biomass for example. The ability to burn fuels with lower heat value in a high efficiency combined-cycle process is an attractive idea in areas with available fuel sources, such as large coal reserves. In addition, IGCC is attractive from an emissions viewpoint for burning of fuels such as coal.

Figure 2-19 shows a diagram of a typical IGCC system with an oxygen blown gasifier and integration of the air separation with the gas turbine. Net thermal efficiencies in the range of 45 to 49 % are achievable.

2.5.2 Compressed Air Energy Storage (CAES)

Compressed air energy storage (CAES) technology is a concept in power generation aimed at meeting market demands for electricity during peak hours by taking advantage of the fluctuations in peak and off-peak electricity prices [5, 6]. In CAES technology air is compressed by high efficiency industrial compressors during off-peak hours and stored typically in an under ground pressurized cavern. During peak hours the compressed air is released and mixed with fuel in a combustion system to run a gas turbine driving an electrical generator. Thus the key difference between a conventional GT and a CAES unit is that in a conventional GT the compressor and turbine are connected to the same shaft and operate in unison while in a CAES unit the compression stage is physically separate. Furthermore, CAES units have a relatively low inertia, similar to multi-shaft GTs, since the generator is driven only by a turbine with no tandem axial compressor. Also, in CAES technology control valves can be used to maintain a constant air mass flow condition from the pressurized cavern to the turbine, thus the unit sees the same air conditions irrespective of the ambient atmospheric conditions [5]. Consequently, CAES units do not suffer from base load variations due to changes in atmospheric conditions as do conventional open cycle GTs.

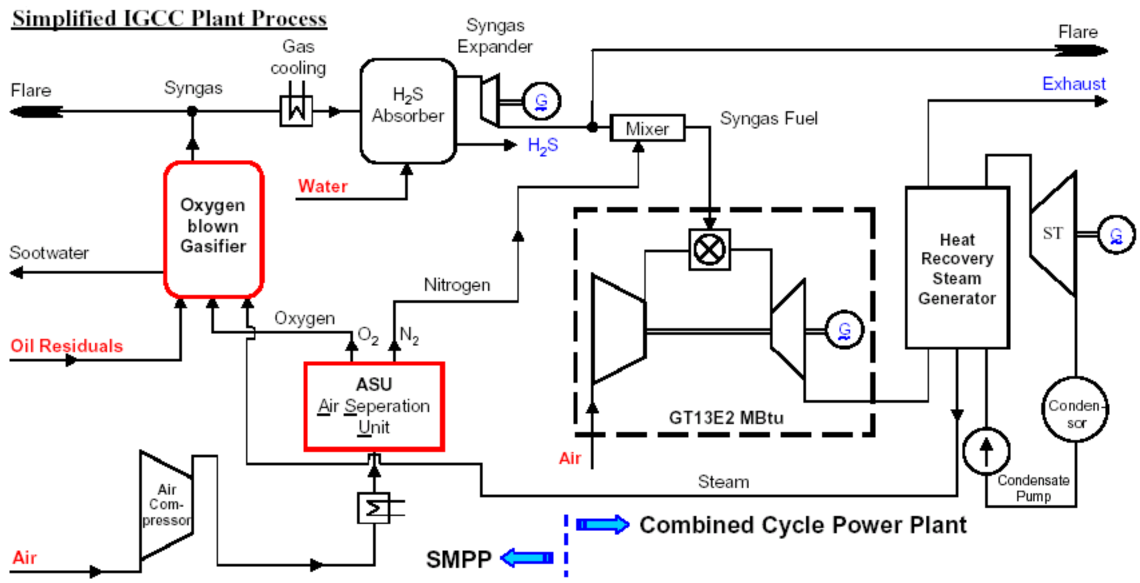


Figure 2-19: IGCC system with oxygen blown gasifier (Source: Alstom Power AG).

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CHAPTER 3

UTILITY PERSPECTIVE – THE NEED FOR BETTER MODELING AND UNDERSTANDING OF COMBINED-CYCLE POWER PLANT PERFORMANCE

3.1 Introduction

For many years utilities have had models for dynamic simulations of power systems. These simulations are needed to assess and plan the power system. The ultimate goal, at least from the utility perspective, is to provide reliable electric service to customers. Two recent changes affect these studies – a significant increase in gas turbine (GT) and combined-cycle power plant (CCPP) installed capacity, and a change from regulated or state owned utilities to more competitive electric markets commonly referred to as deregulation or liberalization.

Gas turbines and CCPP have become popular for a number of reasons. First, they are fueled by natural gas, for which prices have recently been at historically low levels. Second, compared to nuclear, coal-fired steam turbine or hydro plants, GTs and CCPPs encounter fewer environmental objections, have lower initial construction costs and a shorter construction time. Third, the overall thermal efficiency of a CCPP can be significantly higher than conventional coal-fired steam-turbine plants of similar size.

The introduction of competition in the electric industry has resulted in some fundamental changes. Competitive generation owners often have more freedom in locating new plants and retiring old ones. They often decide to run or not run a plant based on market conditions, not on power system requirements. Thus, the power system may be operated in ways never envisioned when built, and often pushed to its operational limits. These conditions result in an even greater need to study and understand the power system so that defensive strategies can be developed for the inevitable severe system disturbance. Because the same firm may not own generation and transmission, there is more organizational distance between those responsible for engineering the power plant and the transmission system. Adequate information on plant dynamics is needed by the transmission utilities at an early stage in the planning of plant development so that studies needed to ensure security of grid connections can be made appropriately in advance of plant commissioning.

In the following sections, several utilities from different countries describe their experiences with GT and CCPP, and their concerns with the models currently available. They also describe the phenomena often requiring simulation.

3.2 Utility Perspective

The following is a compilation of information from various utility sources describing the conditions in which they must operate, and their experience with simple-cycle and combined-cycle generation.

3.2.1 Australia

As of 30th June, 2000, gas turbine (GT) and combined-cycle power plants (CCPP) made 10.9% of the total installed generation capacity in Australia: 7.8% in the National Grid, 33% in Western Australia (South West Interconnected System), and 100% in Northern Territory (Darwin system). This percentage is expected to increase in the future due to replacement of old coal-fired plants with CCPP, construction of new GT and CCPP, and conversion to CCPP.

The Australian National Electricity Market Management Company (NEMMCO) 2001 Statement of Opportunities document provides a list of 'committed' projects, which are considered to be more certain. It shows that 872 MW CCPP (and another 43 MW of simple-cycle GT) and 1313 MW coal plant will be in service by 2002, and no further increases thereafter. The Reuter's Energy Bank Link (EBL) electricity market service provides a longer list of 'proposed' projects, which are less certain. It shows that projects for an additional 300 MW CCPP and 1020 MW of simple-cycle GT are in the 'advanced' state, while projects for an additional 5794 MW of CCPP are under consideration. These two sources quantify, in rough terms, increased reliance on GT based generation plants in Australia. In addition, in Western Australia, most if not all of the 900 MW of planned new generation up to 2010 are expected to be GT based (230 MW CCPP in 2002, etc). The same applies to Northern Territory.

In a number of isolated or weakly connected systems in Australia, GT and CCPP plants have emerged as a major type of generation, as they provide all or over 50% of the total generation capacity. Another trend is the introduction of higher rated and more efficient GT based generation plants, which have different dynamic MW response characteristics than their earlier counterparts. For these reasons, it is important for planning and operation of power systems in Australia to better understand how GT and CCPP behave under situations of low and high stress, and to accurately predict the reaction of GT and CCPP plant to severe credible disturbances.

There are three broad categories of GT and CCPP plants in Australia. First, cogeneration plants which are centered on a specific large industrial customer plant and are often physically located in close proximity of the customer's premises. They may or may not export surplus energy to the grid. These plants generally operate as base load units, since their customers typically have a flat load profile. The second category includes plants that export directly to the grid by contracting their sales with participants in the electricity spot market or through long term contracts. These plants lodge separate bids for ancillary services they provide. Those bids determine their role in the power system control. Third, plants which are major generators in a regional system. They supply the bulk of the load in a relatively small, islanded or weakly connected system. These plants are major control units in their respective areas.

Uncertainty of generation developments and new loads are the key system study issues in the deregulated electricity supply industry in Australia. Other issues include increased pressure to fully utilize utility investment, and the need to evaluate a wide range of alternatives including both regulated and non-regulated transmission, generation and demand side alternatives. Opportunistic generator behavior in the market environment makes the system study work more difficult. This applies to the often hard-to-predict dispatch and operation of the plant, and to the selection of suitable locations for the new plant and their type.

The growth of the interconnection and increased utilization of transmission assets, which has led to increased congestion in transmission networks, has exacerbated small signal stability problems. For example, while oscillatory stability limits have existed on the NSW-Snowy-VIC transfers for many years, these have become more prominent with the interconnection of Queensland. In a market context, power transfers from potential generators can be limited by transient stability, voltage stability, thermal or small signal stability limits. Some utilities have additional quality of electricity supply requirements.

Due to the uncertainties over the forthcoming deregulation, usually there is a period of very limited investment in the power sector during which reserve margins fall. A heat wave that follows may lead to power shortages in some areas and a rise in the wholesale electricity prices. These circumstances fuel the demand for GT based cogeneration and open cycle peaking plant, due to their short lead times. Some sections of Australia also experience significant load peaks. These types of demands can only be economically supplied through very low capital cost generation plants or demand side management. Other factors favoring GT and CCPP generation include concerns about greenhouse gas emissions and minimizing financial risk in generation investment.

An increasingly large proportion of generation from CCPP in Australia creates a need for more accurate models of GT and CCPP. Large gas turbines used in modern CCPP units have inferior maneuvering capabilities relative to older generations of smaller gas turbines for which computer models currently in use were originally developed. New designs and new plant characteristics create the need for new models. This also applies to steam turbines operating in combined-cycle, as they have substantially different dynamic MW response characteristics than the conventional steam plant, for which computer models were originally developed.

There is a considerable difference in dynamic performance of single-shaft and multi-shaft gas turbines. Single-shaft, heavy-duty gas turbines are robust, have high inertia, operate well on weak and islanded power systems (including oil platforms). Single-shaft GTs have sufficient maneuvering capabilities to meet all control requirements, the most onerous of which arise in the island operation with large fluctuating loads in mining, mineral processing and oil and gas industries. In contrast to single-shaft GTs, multi-shaft GTs are more prone to speed fluctuations due to their relatively lower inertia. Numerous operating problems could arise when multi-shaft gas turbine driven generators operate as major units on weak and islanded power systems. Combined-cycle operation aggravates these problems, as the percentage responsiveness to frequency excursions reduces typically by 1/3, due to the typical limited response of the steam turbine.

There is a need for better and more accurate models of all plants in the system because power systems are operated harder and closer to their physical limits. Greater pressure to defer investments has lead to the need for more refined and detailed studies. More accurate models would allow greater confidence to be placed on the results of computer simulations, therefore streamlining the work of utilities and other parties and assisting them to run their business more efficiently.

3.2.2 France

Gas Turbines Experience

France is one of the largest electricity markets in Europe. In 2000, the total production was 517 TWh, of which 76% was produced by nuclear power plants, 14% by hydro power plants and 10% by conventional thermal plants (Table 3-1).

Table 3-1: Power generation in France (2000)

	Maximum capacity (GW)	Generation (TWh)
Nuclear	63.2	395
Hydro	25.4	72
Thermal*	26.7	50
Total	115.3	517

* Conventional thermal plants (coal, fuel-oil, gas, etc.) including GT

The total installed capacity of the conventional thermal plants was about 26.7 GW in 2000. The main energy sources for these power facilities are coal and fuel-oil (Table 3-2). The conventional thermal power plants can be divided into two kinds of generation: centralized generation and decentralized generation. Both of the thermal generation categories include a few gas turbines (GT), but no combined-cycle power plants (CCPP), as of 2000.

Table 3-2: Conventional thermal generation in France (2000)

	Maximum capacity (GW)	Generation (TWh)
Coal	10.3	25.6
Fuel-Oil (≥250MW)	7.2	2.3
Others*	9.2	22.1
Total	26.7	50

* Fuel-Oil (<250MW), natural gas, industrial gas, waste and renewable energy.

The centralized thermal power plants are primarily made up of large power plants that are fuelled by coal or fuel-oil. A few centralized thermal power plants are gas turbines. Electricite de France (EDF) installed these GTs in the 1980's and in the 1990's. The total installed capacity of these GTs was 850 MW in 2000. These GTs are all heavy-duty units and range in size from approximately 50 to 200 MW. They are mainly fuelled by liquid fuel. Some units can be fuelled by either liquid fuel or gas.

The gas turbines operate as peak load units. They are placed in-service during peak load or during network constraints. They are used to supplement electricity generation to ensure balance between electricity consumption and generation. They also provide ancillary services (voltage support, black-start capability). These units are very flexible. They can start in less than 30 minutes and generate electricity to meet electricity consumption. Depending on the needs of the system and complying with the optimization between generation and consumption, these GTs provide power generation during peak load or after incidents on the grid such as a generation unit outage.

The decentralized thermal power plants are usually connected to voltage levels below 100 kV. The decentralized thermal power plants are primarily made up of auto-production, waste incineration and cogeneration. The cogeneration installed capacity was about 4200 MW in 2000. The cogeneration plants range in size between 10 to 50 MW and are usually aero-derivative GTs. Used with some industrial processes, the cogeneration plants generally operate as base load units.

The number of cogeneration plants has increased during the past few years, due to the technological progress and some incentives. Some fifteen cogeneration units representing a total capacity of about 400 MW were connected to the transmission network in 2001.

Gas Turbines and Combined-Cycle Power Perspective

The French electricity supply industry is undergoing profound changes following the French law in 2000 relative to the modernization and the development of the electricity public service. This law, which transposed the European Directive 96/92/CE, governs the network access procedure and tariffs. Independent power producers can build and operate new power plants according to some technical, financial and environmental requirements. Today, EDF has an obligation to buy, with a special tariff, the electricity produced by cogeneration plants and also by renewable energy plants that do not exceed 12 MW.

Two large combined-cycle plants are due for construction:

1. Integrated Gasification Combined-Cycle (IGCC). This power plant will integrate a gasification technology and a combined-cycle to product electricity. The electrical capacity of this IGCC will be about 250 MW. This power plant is scheduled to be in service in 2003.

2. Combined-Cycle Power Plant (CCPP). This power plant will be divided in two identical combined-cycles fuelled by natural gas and industrial gas, coming from the steel industry. The total electrical capacity of this CCPP will be about 800 MW (2 x 400 MW). This power plant is scheduled to be in service in 2005.

The French installed capacity seems to be enough to satisfy the future base load demand and semi-base load demand [1]. But after breaking up some conventional thermal power plants from 2008, new solutions will be needed to satisfy peak load demand. The development of new power plants depends on the energy policy:

- Electricity demand management,
- Renewable energy development,
- Reduction of greenhouse gases, and
- Air pollution control.

Complying with these principles, the number of GTs and CCPPs should increase in the future.

The Need for Models

New power plants are connected to the grid according to specific technical requirements. The procedure for the connection of new power plants depends on its installed capacity. Under the current regulations:

- Facilities with a capacity greater than or equal to 10 MW have to be connected to the transmission network.
- Facilities with a capacity below 10 MW have to be connected to the distribution network.

In the transmission connection procedure, the new power plants have to meet specific technical requirements that depend on the installed capacity and the technology of the power plant. In order to ensure the operating safety of the transmission network, the power plants have to provide ancillary services, such as voltage control and frequency control. Thus, the transmission system operators need accurate models of power plants to study power system stability.

3.2.3 Italy

In Italy, to date (year 2000), the generation from combustion turbines and combined-cycle plants is not very high. In fact, out of a total installed capacity of 78331 MW (27% hydro and 73% thermal) on 31st December, 2000 only 4700 MW of this was CCPPs. Moreover some old thermal plants have been re-powered for a total capacity of 8000 MW (2150 MW of gas turbines and 5850 MW of steam turbines fed by exhaust gas heat and reduced independent fuel). The re-powering has been made by water heating (4400 MW) and steam production (3600 MW). Finally, there is also 1700 MW of simple-cycle gas turbines in operation. So the total installed capacity including gas turbines and steam turbines using exhaust gas energy is about 12%.

The Italian electrical scenario is rapidly changing and the liberalization process is under development. In the year 2000, 82% of the energy generated was by thermal power plants and 18% by hydro plants. In the last few years, use of natural gas as a fuel source for electric generation has constantly increased in Italy, becoming, in 2000, the primary energy source for electric generation. In fact, the primary source of thermal power plant energy, in 2000, was 45% natural gas, 38% oil products, 12% coal and 5% others (see Figure. 3-1). Several thermal power plants are equipped with natural gas fed steam turbines. Due to this actual scenario, the energy prices are quite high and many opportunities for new generators are obvious. So it is easy to foresee in the future a very strong increase of CCPP generation characterized by very high total efficiency. The Italian independent system operator (ISO) has received requests to connect more than 70 GW of generation to the transmission network. It is reasonably foreseen, for 2010 that power from new CCPP plants will be in the 22,000-32,000 MW range (17000 by old-site renewed plants and 5000-15000 by new-site plants). With this prospective it becomes more and more important to have new and reliable CCPP models for the analysis of the power system.

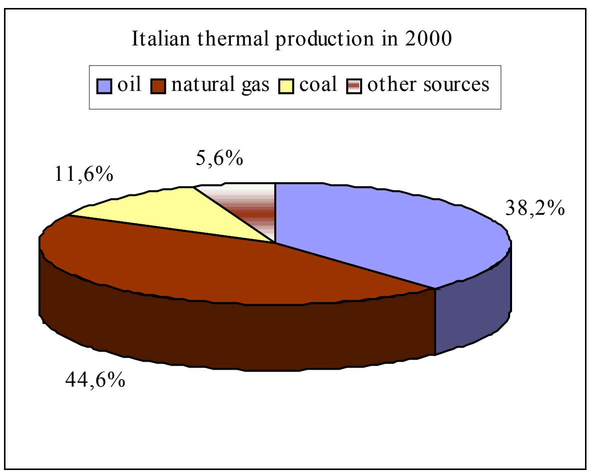


Figure 3-1: Italian thermal generation by fuel

Since, in the present and near future Italian market, the production cost of CCPPs, cogeneration plants and repowered power plants will likely be lower than the production cost of traditional thermal plants, CCPPs should be considered for base load operation. Nevertheless, when the percentage of total capacity from these power plants has increased to a significant level, power modulation will be required and plant flexibility will be extremely important. As regards the simple-cycle gas turbines, they are and will continue to be dedicated to load peaking.

The models for power system analysis will disregard the temperature dynamics of the steam section and will require only a simple model. On the other hand, very detailed models of the physical process, including the temperature dynamics of the steam section, will be needed in cases involving design and testing of control devices.

The incoming deregulation makes it more and more difficult to foresee the power dispatch, which depends on the "ask and bid" of the energy market and particularly on the bid prices. So the approach to power system analysis has to be more probabilistic than in the past, when a deterministic approach was favored. This new point of view has to be applied both to planning (impact on the grid of new generation) and to operation (power dispatch).

3.2.4 New Zealand

The electricity generation sources in New Zealand consist of a mix of plant types: hydro, steam, gas, CCPP and geothermal. The total installed generation capacity is approximately 8200 MW, and Table 3-3 shows the relative size of the installed generation mix. Maximum electrical load supplied in 2001 was approximately 6000 MW, of which 4000 MW was in the North Island and 2000 MW was in the South Island. The gas and CCPP plants in the North Island alone account for a total installed capacity of 1026 MW, a summary breakdown of which is shown in Table 3-4. The two single shaft CCPPs are rated at 360 MW and 395 MW and therefore each of these units represents a significant proportion of the total North island generation at any given time, ranging from approximately 10% during peak load periods to 20% during light load periods. South Island generation is solely based on hydro plants.

Table 3-3: Generation Mix in New Zealand

Generation Type	Installed Capacity	
	MW	%
Hydro	5170	64
Steam (coal /gas)	1488	18
Gas Turbines	1026	13
Geothermal	380	5
Total	8064	100

Table 3-4 Gas Turbine Based Generation in New Zealand.

Type of Generation	Installed Capacity (MW)
Single shaft CCPP plants (2 plants)	755
Multi-shaft CCPP plants (1 plant)	120
Cogeneration Gas turbine plants (4 plants)	151
Total	1026

The high voltage ac transmission networks in the two islands are connected through a bipole HVDC transmission link. The HVDC power transmission is predominantly from the hydro generation rich South Island to the industrial and commercial load centers in the North Island. The maximum capacity of the HVDC link presently stands at 1040 MW. During light load periods, the possible loss of HVDC transmission to the North Island due to a bi-pole failure represents a loss of approximately 50% of the total generation connected to the North Island power system. The dynamic behavior of the power system following such a contingency has therefore dominated the approach followed in setting up of power system planning and operating standards in New Zealand. Of particular importance in planning and operating CCPPs is the under frequency performance of the power system and the ability of CCPPs to withstand low frequencies historically encountered during such events. (Note that the nominal operating frequency in New Zealand is 50 Hz).

Under the present market environment, sufficient fast acting instantaneous power reserve is procured for ensuring the minimum system frequency would stay above 48 Hz during single contingent events (i.e. tripping of a single generating unit or one pole of the HVDC link). The under frequency performance of the power system during an HVDC bi-pole failure event is managed, mainly with the aid of load shedding arrangements, such that the minimum system frequency would stay above the frequency profile described in Table 3-5. Automatic under frequency load shedding will take place in two blocks at 47.8 Hz and 47.5 Hz. Each load-shedding block will constitute approximately 16% of the pre-event loading in the system. Table 3-5 is considered to be the under frequency performance that can be expected from future single shaft gas turbine units without incurring undue economic penalty for customizing their performance.

Table 3-5: Generating Plant Under Frequency Withstand Limits

	Frequency (Hz)	Minimum withstand time (s)
1	Above 47.5	Infinite
2	47.5	120
3	47.3	20
4	47.1	5
5	47.0	0.1
6	For frequency between those specified in above (2) – (5), the minimum generator withstand time at a particular frequency is derived by linear interpolation	

Depending on the system conditions and the reserve available, the rate of frequency decay in the New Zealand power system could be as high as 1 Hz per second. The modeling of CCPP primary governing response, ramp rates and temperature dynamics plays a significant role in predicting and assessing the plant performance during such events. Therefore in islanded power systems such as the New Zealand power system, accurate modeling of the generator output power variation with reducing frequency is important and would ensure adequate reserves are procured for maintaining the grid security under emergencies. Further, the accuracy of the plant models at the extremities of the operating frequency range and the impact of reduced model accuracy on the validity of the simulation results need to be clearly understood in assessing the system performance during under frequency events.

Given the operation of the New Zealand deregulated electricity market, the responsibility of achieving the plant performance as specified in the rules is an obligation on the asset owners. All CCPPs presently in service are capable of meeting the obligations, and no specific measures, such as de-rating of the plant etc., were required to be put in place for meeting the obligations.

3.2.5 United Kingdom

The generating plant mix on the National Grid Company (NGC) system has changed significantly in the last ten years with CCPP replacing old coal power stations. More CCPP is planned for connection to the NGC system. The table below illustrates the existing generating plant mix (2001) and that forecast for 2005 as given in NGC 2001 Seven - Year Statement.

Table 3-6: Contracted Generation Capacities (MW)

Year	CCPP	Simple-Cycle GT	Nuclear	Coal	Oil	Dual Fuel	Pumped Storage	Total*
2001	22711	1193	10109	23027	2519	2905	2088	64552
2005	40129	1478	9869	23027	2519	2905	2088	82015

* Excludes import from France (1976MW) and from Scotland (1200 MW in 2001 and 2200 MW in 2005)

As can be seen from the above table, CCPP generation forms a large proportion of the total generation on the NGC system (35% of total existing generation). This dominance of CCPP generation in the energy market in England and Wales is expected to increase further in the coming years (50% of total contracted generation in 2005). For a relatively small electrical system such as that of the UK, understanding and predicting accurately the behavior of CCPP is crucially important for planning and operation of the system. The NGC has, therefore, a need to model this type of plant with enough details to enable it to forecast the behavior of CCPP and simple-cycle GT plants under steady state and system disturbance conditions.

The performance of a CCPP depends largely on the design of the main plant and control systems adopted by manufacturers. The NGC worked very hard with generating companies and manufacturers to implement modifications on the initial plant design so that CCPP performance can meet the desired requirement on the NGC system.

Current NGC's concerns are mainly related to block loading capability, load rejection capability, maintaining the required active power under system frequency deviations and survivability of CCPP plant following a major system incident/blackout. Models are required to allow utilities to carry out steady state, dynamic and transient studies to accurately forecast system behavior in planning/operational time scales. Accurate models will help in the optimization on use of transmission assets and can reduce cost associated with operating the system.

The CCPP plant operation on the NGC system depends on the contracted position of their owner to deliver the required energy. The owner may have a portfolio of power stations that are not necessary CCPP, and decides whether a power station operates as base load or two-shift (part load) generation. Most CCPP operate usually as two-shift generation with some being base load. None of the CCPPs operate as peak lopping (peaking) generation. In the future, with the anticipated closure of old nuclear and coal plants, more CCPP are expected to run as base load generation. The simple-cycle GT plants on the NGC system are mainly used as peak lopping (peaking) generation.

In a deregulated electricity market:

- The transmission companies do not have control over where new generation is located or when existing generation is shut down. They may not even know whether a particular generator is going to be available the next day. This has led to major changes in power flows on the transmission system. In the case of the NGC, a probabilistic approach to its planning/operational studies has been implemented to assess the impact of the various uncertainties arising from the privatized electricity market.

- The transmission companies pay for ancillary services (reactive power + frequency response) and have financial incentives to reduce contract costs. Therefore additional studies (e.g. load flow/transient stability/voltage & frequency control studies) are performed to determine the capability to optimize trading/contract cost.
- The power system is running nearer to its capability limit. Therefore there is a need for detailed system defense studies (transient/dynamic stability studies) to assess the system under major contingencies (e.g. system split conditions/islanding). Accurate modeling is crucial to determine whether the system and generating plants will survive such major system incidents.

3.2.6 United States of America

In the United States (USA), there is no central electric utility or governmental body controlling the entire electrical system. Instead, there is a mix of public, investor and government owned utilities organized into several regional councils for security, operating under a mix of national, state, and local laws and regulations.

The following provides information from four different entities in the USA, and illustrate some of the diverse yet often common experience of the various regions.

3.2.6.1 American Electric Power (AEP), a member of the North American Eastern Interconnection

Prior to the onset of electric deregulation and transmission open access, relatively little attention was given to the modeling of the dynamic behavior of gas turbine-governors or the governing response of combined-cycle plants. This was primarily because most gas turbine applications had been small peaking facilities that were off-line most of the time.

The recent and tremendous surge in the development and commissioning of new natural gas powered generation projects in North America, due in part to the opportunities presented by transmission open access, has resulted in a need to better define appropriate dynamic models for the wide variety of gas turbines now being installed. These projects are mostly owned and operated by independent entities and may be on line at any season or time as favorable market conditions arise. This applies to peaking units and especially combined-cycle units, usually considered for base load operation due to their high overall thermal efficiency.

Until recently, the modeling of GTs and their governing controls in commercial software packages used for dynamic simulation has been limited to a few generic GT model types. These generic models were considered sufficient to approximately represent short-term dynamic behavior within the first few seconds following a system disturbance, but not long-term behavior, particularly that involving off-nominal frequency operation.

The modeling approach for CCPPs in most dynamic studies to date has been to independently model the speed governing control of the gas turbines with whatever generic gas turbine-governor models are available, and completely omit any response of the steam topping turbine, thus leaving the steam turbine power input as a constant throughout a simulation. In short-term dynamic studies such as those assessing transient and oscillatory stability, this approach is deemed acceptable. Nevertheless, the effect of load variations on the corresponding gas turbines will eventually affect the output of the steam turbine. In long-term dynamic studies, such as simulation of islanding and other disturbances involving off-nominal frequency operation, this behavior is important to capture.

3.2.6.2 Salt River Project, a member of the Western Electricity Coordinating Council

In Arizona, the environment for power system development has undergone drastic changes. During the past few years, Arizona maintained fast population and economic growth and the demand for electric power increased accordingly. Furthermore, the Western United States power market will be opened further to competition that will certainly have a great impact on operations and planning of the electric utility business. For the electric power industry in Arizona, there are eighteen independent power producer (IPP) projects proposed, some of which have already entered into the market in the year 2001. The majority of these projects are combined-cycle power plants.

Based on a survey conducted in 2000, a substantial number of CCPPs will be built in Arizona. These will significantly change the generation mix in Arizona. Installed capacity in 2000 was approximately 60% coal-fired steam plants, 10% natural gas and oil fired steam plants, 20% nuclear plants and 10% hydro plants. In the year 2003, only three years later, the combined-cycle plants will add about 4,800 MW of generation and will constitute almost 25% of the total capacity. An even faster growth rate is projected from 2002 to 2005 (7850 MW) resulting in CCPP potentially accounting for 40% of the total capacity. All of this new generation may not be built as projected if spot market prices become too low.

Influenced by deregulation, these CCPPs will be constructed and planned for serving a price-oriented market. Some will be used to replace older, less efficient power plants or plants requiring modifications because of new air quality regulations. The operations of these CCPP are “mixed”. They can be used as “peaking” or “base load” units depending upon the fuel availability, operation and maintenance costs, and short and long term power supply contracts.

Because of the market incentive, the majority (70%) of these combined-cycle plants will be located near or at the major EHV transmission network hubs. The balance (30%) will be scattered across the state and tied to the underlying local transmission systems. As of 2001, the new IPP generation owners will not construct new transmission lines, except building their own transmission to interconnect with the existing transmission networks. Since many of the utilities’ transmission systems will become increasingly limited by these external generation facilities, system planners need to make every effort to ensure an adequate and reliable power system. System impact studies, including power flow, transient stability, post-transient voltage stability, short circuit and subsynchronous resonance analyses, are required to meet the system performance requirements and criteria set forth by the transmission owners, and various electric regulatory agencies and reliability councils.

One particular concern is that new CCPP plants often have governor response characteristics and/or special control features different from conventional plants. Existing governor models in popular stability software are often not adequate for the governor models and the associated parameters provided by equipment suppliers. Therefore, it is desirable that adequate standard governor models be available for use in dynamic simulations.

3.2.6.3 ONCOR, a member of the Electric Reliability Council of Texas (ERCOT)

The deregulation of wholesale and retail electric markets in Texas has resulted in many changes. The old paradigm, where planning for new generation and transmission facilities was a coordinated effort, no longer exists. Today in ERCOT, generation is a competitive entity while transmission is regulated. The relationship between generation and transmission entities is mandated to be at "arms length". Consequently, the transmission system's "near term" or "existing" ability to transport the power to the load is often a secondary consideration when generation owners plan and locate new power plants. Simple-cycle and combined-cycle plants can be built much faster than the transmission lines needed to transport the power to the load. This has resulted in a few instances of congestion and limits on generation output while needed transmission lines are being built. Because the actual generation pattern, and consequently flows on the transmission lines, is now based on changing market contracts, the transmission system is being operated in ways never intended when originally designed. Over the last few years, lack of damping and small signal stability concerns have become more of an issue in ERCOT. Because spinning reserve (or load shedding in lieu of spinning reserve) can be a market commodity, and a transition from boiler follow to sliding pressure control has occurred over the years, frequency control following disturbances has become an issue as well. In conjunction with the above changes, there has been a marked increase in generation capacity from combined-cycle plants, as shown in Table 3-7. As the table indicates, the trend toward more CCPP capacity is expected to continue. These and other factors make it vital that system studies accurately simulate the power system performance, a necessity if system reliability is to be maintained. To obtain accurate studies, accurate models and data are required. While existing models for conventional steam power plants appear to have wide acceptance, the few models for GT governors appear to have less consensus, and their ability to accurately replicate actual performance under the broad range of conditions simulated has been questioned. Consequently, there is a great need for GT and CCPP models that accurately replicate actual field performance for a wide range of operating and power system conditions.

Table 3-7: ERCOT Generation Capacity [2]

	1995		2000		2005	
TYPE	Total MW	% Total	Total MW	% Total	Total MW	% Total
NUCLEAR	4800	8.9	4800	7.3	4800	6.6
HYDRO	436	0.8	470	0.7	470	0.7
ST.-COAL	14259	26.4	15450	23.6	15950	21.9
ST.-GAS	29082	53.9	30720	46.9	31331	43.1
GT	3036	5.6	3938	6.0	3512	4.8
COMB. CYC	1974	3.7	9671	14.8	15962	22.0
OTHER	333	0.6	498	0.7	671	0.9

3.2.6.4 Entergy Services Inc, a member of Southeastern Electric Reliability Council (SERC)

Entergy Services Inc is one of the major investor owned electric utilities located in the Southern part of the USA, and caters to the load demand in the states of Louisiana, Mississippi, Arkansas and Texas. Entergy's 2001 peak system load was 22,000 MW and installed generation capacity is 24,000 MW. The installed capacity consists of 78.7 % fossil, 21% nuclear, and 0.3 % hydro. However, with the onset of deregulation and issuance of FERC (Federal Energy Regulatory Commission) Order No. 888, Entergy has received more than 160 requests for generation interconnection on its transmission system. These requests equate to more than 60,000 MW of prospective new generation. To date, more than 23,000 MW of generation have signed an Interconnection and Operating Agreement with Entergy, which indicates a high level of commitment to complete the projects. Also, by the end of 2002, the construction of approximately 13,000 MW of new generation will be completed. Almost 90 percent of the proposed new generation is combined-cycle power plants (CCPP), the rest being mainly hydro units and some simple-cycle units. In the traditional utility environment, when coordinated resource planning was permissible, the goal was to strike a balance between generation and transmission resources in order to meet a certain level of reliability and financial goals. As a result, generation resources were normally located near major load centers and transmission facilities were constructed as required to deliver the power. However, in the deregulated environment coordinated planning between transmission and generation organizations is long gone. In this environment, new merchant plants may become network resources or they may also choose to sell their power off system. Transmission planners now face the complex task of planning a transmission network that is capable of serving load using numerous combinations of generation resources.

In order to integrate these new generating plants into the transmission system, Entergy has developed an interconnection study process. As part of this process, load flow, short circuit and stability studies are performed and the impact of the plant on the transmission system is assessed. In order to perform this study, accurate models for generator, exciter, turbine-governor and power system stabilizers are required. The main issue pertaining to the modeling of CCPP is the turbine-governor data. In order to better understand the problems associated with system separation, frequency excursions, and poorly damped oscillations, accurate turbine-governor models, as well as models of the electrical components of the plant, are needed. The turbine-governor models, even though they play a minor role in transient stability studies, are critical from the long term dynamic simulation standpoint, especially for frequency excursions. For example, when studying low frequency events, the main issue is the ramp rate of the gas turbine and the effect of frequency on the active power output. Under these conditions, the turbine-governor plays a critical role and their true effects should be reasonably represented through the models. Therefore, an accurate model of a CCPP is needed which includes turbine-governor, relevant controls and protective functions to simulate the CCPP response to power system disturbances in a reasonable manner.

3.3 Phenomena of Concern

3.3.1 Small-signal Phenomena

As noted in section 3.2, several utilities are noticing small signal problems. In general a governor is tuned to give an adequate droop and thus primary frequency control response based on grid code testing (see Chapter 5), while other plant controls, such as the automatic voltage regulator (AVR) and power system stabilizer (PSS), are tuned to ensure adequate transient and small-signal performance of the plant. However, there can be instances in which small, isolated power systems encounter relatively large cycling loads (such as a small simple-cycle GT plant serving primarily a heavy industrial load). In such cases the governor/turbine controls may interact with the load dynamics and thus result in very low frequency power oscillation on the system. In such cases where turbine/governors may interact with load dynamics, and usual measures such as a PSS do not adequately resolve the problem, it may be necessary to take a closer look at the turbine controls and recommend supplementary controls and/or other settings or control strategies to mitigate such interactions.

3.3.2 Transient Phenomena

Transient phenomena can be roughly divided into two segments, first swing (transient stability) and extended term. For first swing, it is generally assumed that the steam driven turbine in a combined-cycle plant will have constant mechanical power during the few seconds the transient occurs. This assumption is justified because of the long time constant associated with the heat recovery steam generator (HRSG), and because the steam valves are often operated in the wide open position, resulting in sliding pressure control. This assumption may not be correct when control features such as fast valving or special protection schemes are used to limit the output of the steam driven generator during transients. It is generally assumed that GTs, whether stand alone or as part of a CCPP, can make some change in mechanical output that can affect the transient response. The increase in speed following a nearby fault should result in some response by the speed governor to decrease turbine power during the first swing in machine speed. This initial governor response, even if small, will improve the ability of the unit or plant to remain synchronized. It is important that a detailed representation of maximum rate of fuel valve closing be included in models.

An extended term transient phenomenon essentially includes any event or simulation where the time focus goes beyond the first few seconds. Examples include reconstruction of major incidents, black start and system restoration, frequency response, cascading outages, and system breakup and islanding. These simulations can extend into minutes. It is expected that both combustion turbines and steam turbines will influence the outcome. It is important that models to be used to evaluate extended term phenomena accurately account for the gas turbine temperature control and the effects of system frequency on megawatt output. For the steam driven turbine, it is important that the HRSG dynamics be modeled (likely in simplified form) and the type of control, such as sliding pressure or inlet-pressure control, be included.

3.3.3 Generation-Load Unbalance Phenomena

The effect of gas turbine and combined-cycle plant speed governing controls during transmission system disturbances resulting in large system frequency deviations is a chief reason for desiring accurate models. Initiating events can include loss of a single large generator on a small system, the loss of several generators or a large generating plant on larger systems, or line tripping resulting in islanding part of the system. All of these disturbances may result in the formation of system islands, the tripping of load or generation, or both. Islanding events may lead to either over or under frequency conditions. Loss of generation events will result in under frequency conditions and loss of load events will result in over-frequency conditions. Events such as these may stretch out to tens of seconds or minutes, and are thus a subset of extended term dynamics phenomena.

The formation of an island with insufficient generation to balance the load will result in an under-frequency disturbance. The magnitude of the initial frequency decline, frequency recovery, if any, and possible frequency overshoot are all a function of the net island speed governing behavior, among other factors. A frequency overshoot situation may arise as a result of automatic under-frequency load shedding if over-shedding occurs, or there is insufficient downward speed governing action applied within the island as the frequency recovers.

It is critically important that models accurately represent the equipment behavior over the range of frequency operation allowed by the equipment manufacturer, and over an extended period of time. Similarly, the software implementation of the models must be robust enough to handle events needing simulation.

When GT and CCPP are connected to larger power systems, the models generally only need to have a load range of from approximately one-half load to full load, and short term overload capability. However, when connected to weaker and islanded systems, the models need to be accurate over a wider load range. Models should be reasonably accurate for speed deviations of up to +/- 5% from rated speed.

3.4 Data

The following table summarizes the types of studies utilities commonly make, the typical time frame for simulation, and the likely operating range of GTs or CCPPs.

Table 3-8: Modeling requirements

Model Requirements of GT and CCPP			
Type of Study	Operating Range		Time Frame
	Frequency	Loading	
Small Signal	nominal	Min. to Max.	steady state
Transient: first swing	± 5% of nominal	75 to100% of Max.	0 to 5 seconds
Transient: extended term	± 5% of nominal	Min. to Max.	1/2 to 10 minutes
Generation-Load Unbalance	± 5% of nominal	Min. to Max.	0 to 3 minutes
Note: frequency can be the range of operation of GT or CCPP			

The following table lists the generation capacity by type for the indicated utilities/regions in 2002.

Table 3-9: 2002 Data

	Utility System											
	New Zealand		United Kingdom*		American Electric Power**		Arizona		ERCOT		Entergy Services	
	MW	%	MW	%	MW	%	MW	%	MW	%	MW	%
Nuclear	0	0.0	9869	15.0	2060	7.4	3,990	21.2	4737	6.2	5451	20.3
Hydro Note 1	5170	64.0	2088	3.2	1057	3.8	1,650	8.8	475	0.6	155	0.6
Thermal Note 2	1488	18.0	28451	43.3	21391	76.5	9,165	48.8	45924	60.0	19717	73.5
GT	151	2.0	1193	1.8	3465	12.4	1,295	6.8	4409	5.8	1504	5.6
CCPP	875	11.0	24116	36.7	0	0.0	2,700	14.4	19893	26.0	0	0.0
Other	380	5.0	0	0.0	0	0.0	0	0.0	1124	1.5	0	0.0
Total	8064	100	65717	100	27973	100	18,800	100	76562	100	26827	100

Note 1. Includes pumped storage.
Note 2. Conventional thermal plants (coal, gas, oil, etc. fired) excluding GT and CCPP.
*** England and Wales only. Figures exclude import from France (1976MW) and Scotland (2200MW)**
**** ECAR only.**

Figures 3-2 thru 3-7 display the information from Table 3-9 (percent of total generation) in a graphical form.

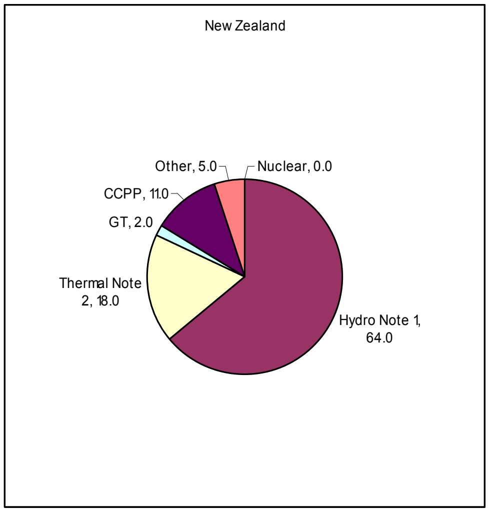


Figure 3-2

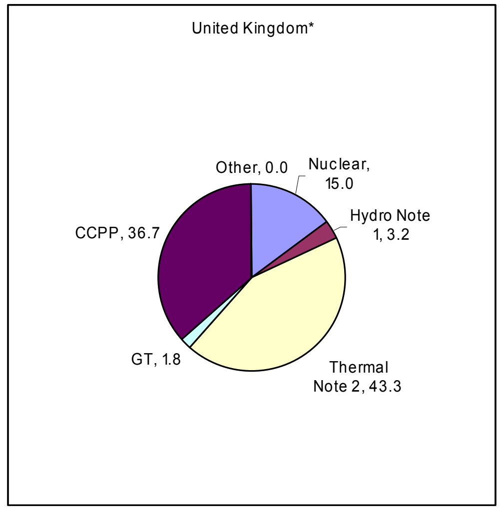


Figure 3-3

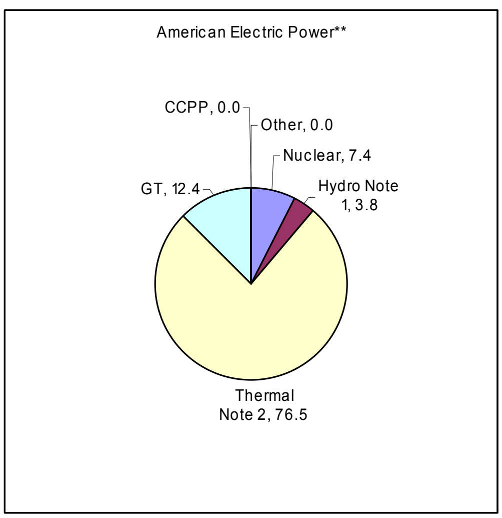


Figure 3-4

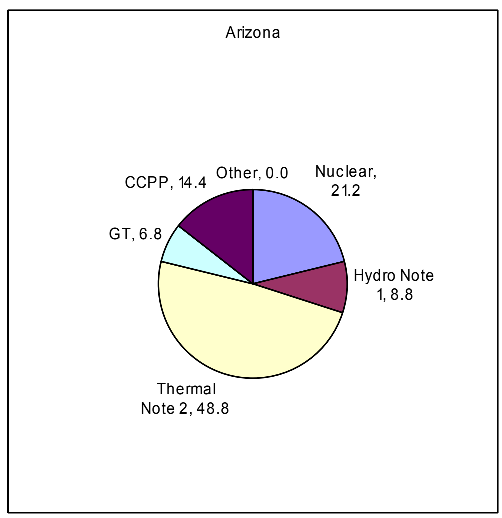


Figure 3-5

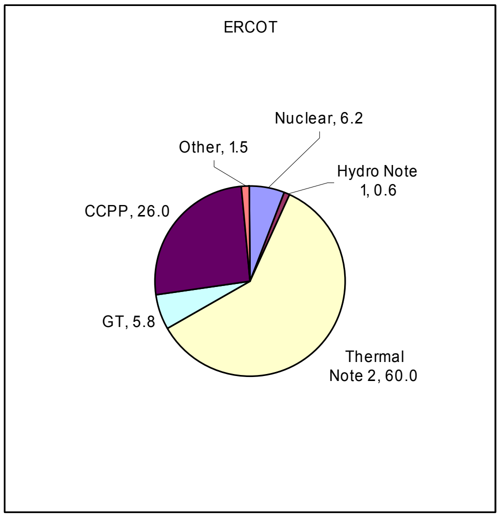


Figure 3-6

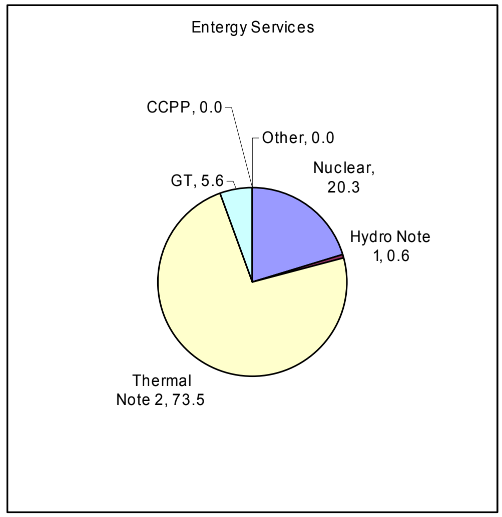


Figure 3-7

The following table lists the projected generation capacity by type for the indicated utilities/regions in 2005.

Table 3-10: Projected 2005 Data

	Utility System											
	New Zealand		United Kingdom*		American Electric Power**		Arizona		ERCOT		Entergy Services	
	MW	%	MW	%	MW	%	MW	%	MW	%	MW	%
Nuclear	0	0.0	9869	12.0	2060	5.2	3990	15.6	4737	5.7	5545	13.1
Hydro Note 1	5170	61.0	2088	2.5	1057	2.7	1650	6.5	475	0.6	155	0.4
Thermal Note 2	1488	17.0	28451	34.7	21486	54.2	9165	35.9	45924	55.7	19717	46.6
GT	151	2.0	1478	1.8	4095	10.3	1295	5.1	5033	6.1	4006	9.5
CCPP	1275	15.0	40129	48.9	10930	27.6	9400	36.9	24601	29.8	12921	30.5
Other	430	5.0	0	0.0	0	0.0	0	0.0	1731	2.1	0	0.0
Total	8514	100	82015	100	39628	100	25500	100	82501	100	42344	100

Note 1. Includes pumped storage.
Note 2. Conventional thermal plants (coal, gas, oil, etc. fired) excluding GT and CCPP.

*** England and Wales only. Figures exclude import from France (1976MW) and Scotland (2200MW)**
**** ECAR only.**

Figures 3-8 thru 3-13 display the information from Table 3-10 above (percent of total generation) in a graphical form.

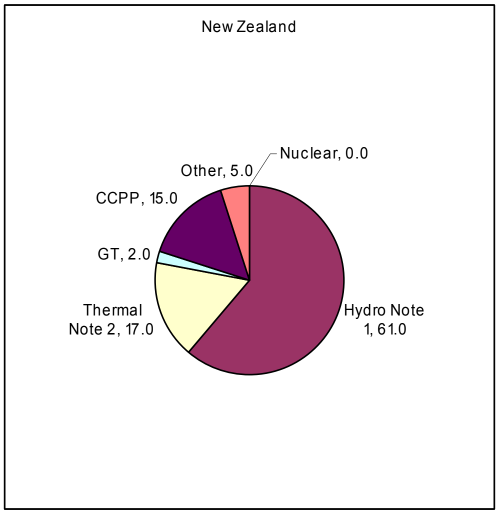


Figure 3-8

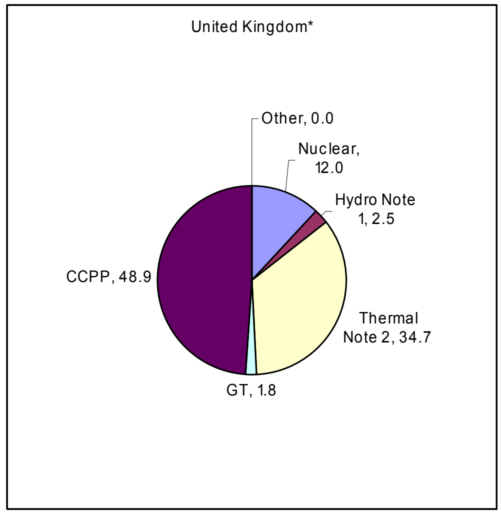


Figure 3-9

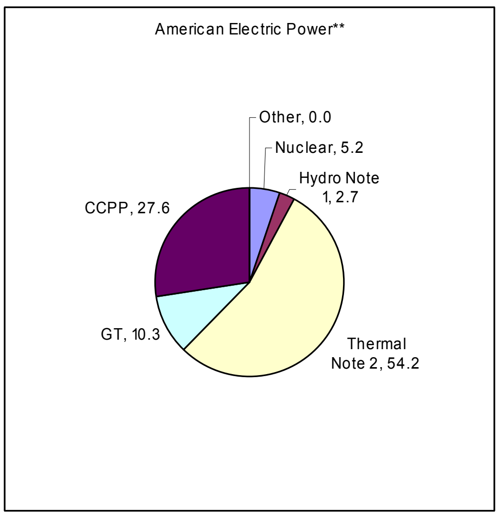


Figure 3-10

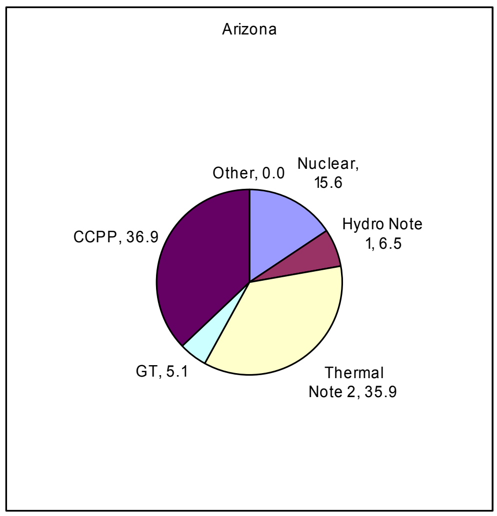
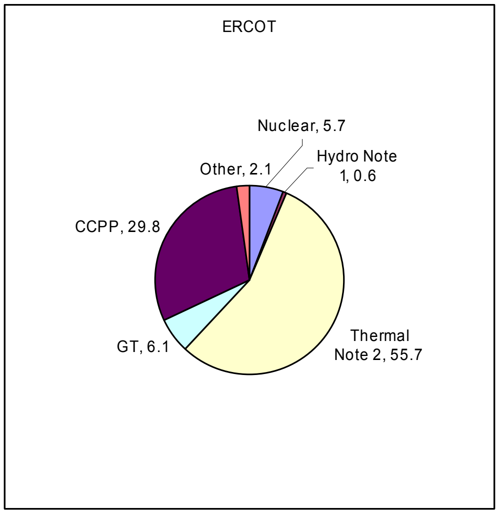


Figure 3-11



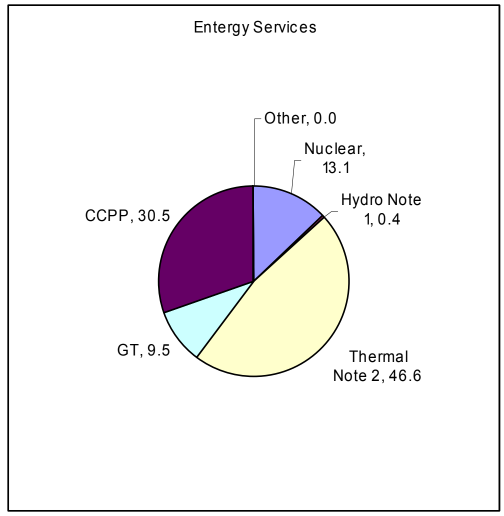


Figure 3-12 **Figure 3-13**

Figure 3-14 compares CCPP as a percent of total capacity for each utility/region in the figures above, for years 2002 and 2005. Figure 3-15 compares the MW capacity of CCPP for each utility/region in the figures above, for years 2002 and 2005. By almost any measure, the amount of CCPP is significant for four of the five entities, and clearly expected to grow in all five.

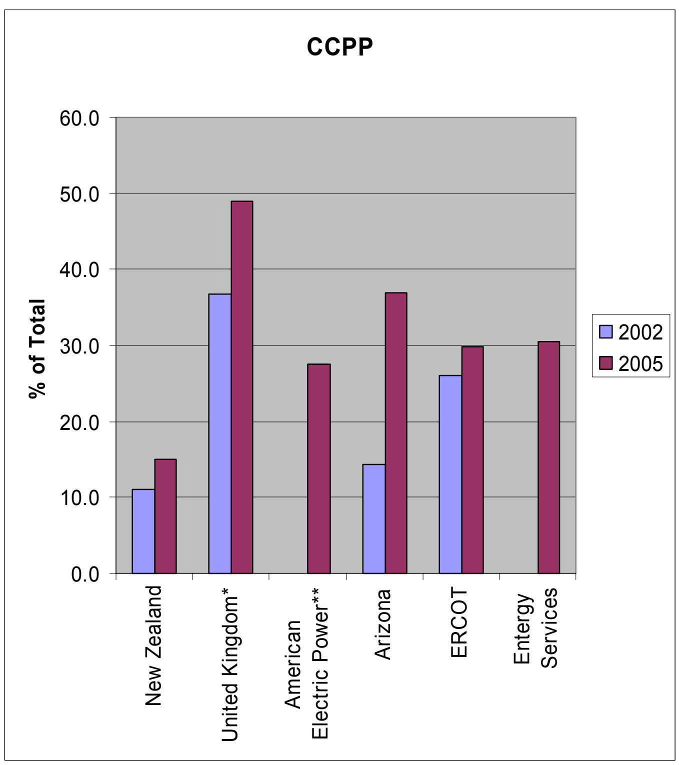


Figure 3-14

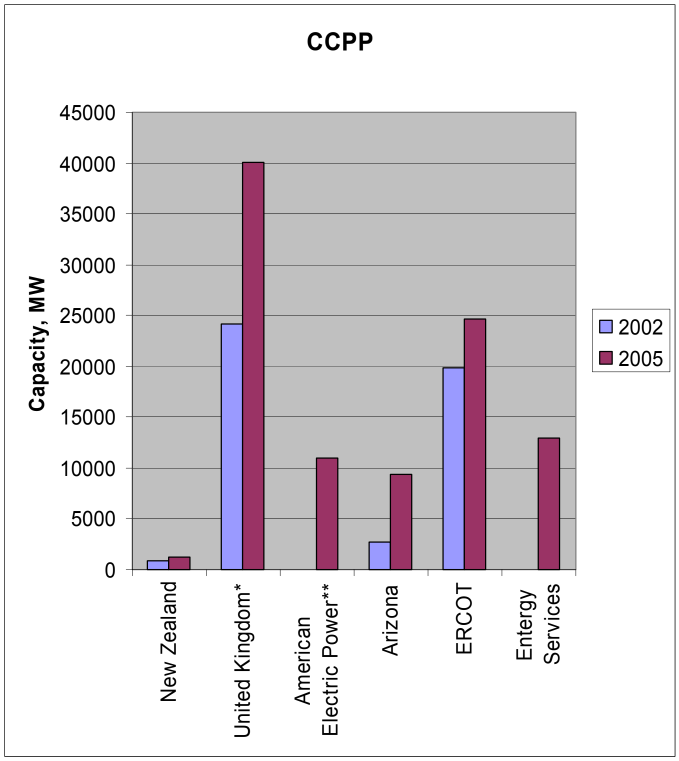


Figure 3-15

3.5 Summary

As is evident from the above, electric power systems vary from large and robust to very small, and from tightly interconnected to weakly linked. One thing that is clear is that GT and CCPP will be a significant part of total installed capacity for many systems. These units are operated in base load, load following and peaking duty. It is also clear that utilities are concerned about the security of the electric power system and the effect GT and CCPP generation will have on security. Newer GT and CCPP have different megawatt response characteristics than earlier versions. The models currently available do not necessarily replicate the actual performance of actual machines. It is thus evident that utilities and power system planners desire models of GT and CCPP that more accurately replicate the performance of the actual machines as they affect the power system.

It is also very clear that the introduction of competition into the electric power industry has significantly affected how the systems are planned and operated. Uncertainty in such basic information as the location of new generation or when units will be operating has made transmission planning even more difficult than in the past. In some instances, small signal stability and frequency control after disturbances have become more prominent problems. A major challenge facing utilities is to understand and accurately predict actual system behavior in light of new equipment (GT and CCPP) and in some cases new operating realities due to competition.

References

[1] "French long-term plan of investments in new generating capacity" ("Programmation pluriannuelle des investissements de production électrique"), Ministry of Industry Report, 29 January 2002.

[2] R Boyer, "Now That You have A Combined-Cycle Plant in your Backyard", presented at IEEE PES Winter Meeting panel session *Experience with Modeling of Gas Turbine and Combined Cycle Power Plants*, sponsored by the Power System Dynamic Performance Committee, Jan 2001, Columbus OH.

CHAPTER 4

MODELING OF COMBINED-CYCLE POWER PLANTS FOR POWER SYSTEM SIMULATIONS

4.1 Introduction

Since the 1990's, combined-cycle power plants have become a significant part of the generation mix in a number of systems around the world. Furthermore, until recently [1, 2, 3, 4, 5, 6] there have been very few publications on the modeling of combined-cycle power plants. Thus, the common practice hitherto has been to model the GTs in a CCPP using models available in widely used commercial programs with typical parameters and to model the ST as having constant mechanical power. Most of the existing gas turbine models have been based on [7]. Modeling single-shaft CCPPs has been achieved hitherto by developing user written models, or adjusting parameters in existing models in order to emulate the expected behavior of a single-shaft CCPP.

The level of modeling detail of CCPPs in a study is clearly driven by the type of study and the relative size and importance of the plant to the system. In most grid studies the important issue is the behavior of the power plants as perceived by the grid, and not the details of the internal behavior of the plants. For plant specific studies or studies related to small grids or fragments of a large grid the use of more detailed models specific to the plant in question may be appropriate [2, 5, 8], and in such circumstances it may be prudent to consult with the manufacturer of the specific turbine.

This chapter will present generic models appropriate for modeling the dynamic behavior of combined-cycle power plants. Modeling strategies and the limitations of the models presented here will be discussed. At the end of the chapter typical parameters are given for the models presented here.

Appendices C and D in addition to some of the references [2, 5] present manufacturer specific models, some of which are more detailed than the generic models presented in the following section.

4.2 Generic Models

The models presented here are intended for modeling any combined-cycle power plant in a grid study. Where studies are more focused on the performance of a particular plant or are focused on simulating specific test results pertaining to a specific unit, a more detailed model may need to be used and the advice of the manufacturer or other experts should be sought.

As noted in the prior chapters, the following points are of pertinence with respect to modeling CCPP for power system studies:

1. To represent the major control loops associated with the dynamic response of a GT during system disturbances, including the dependence of the GT maximum power output on large variations in system frequency.
2. To make appropriate representation of the response of the HRSG.

3. To represent the dynamics of the ST for long-term dynamic studies.

Note that:

- The electrical output of the combined-cycle power plant without supplementary firing is controlled by means of the gas turbine only.
- The steam turbine will always follow the gas turbine by generating power with whatever steam that is available.
- For a change in the gas turbine power the steam turbine power will adjust automatically with a few minutes delay dependent on the response of the HRSG.
- The steam turbine in a combined-cycle power plant is commonly operated in either sliding pressure (control valves are fully open) or fixed pressure control (control valves are throttled) mode.

Figures 4-1 and 4-2 show proposed simplified generic models for the GT and ST in a combined-cycle power plant, which capture all of the dynamic characteristics listed above.

Figure 4-1 shows a simplified generic GT model. The model is intended for use in a per unit system, per unitized on the turbine nameplate MW rating. The three major control loops represent the controls associated with the speed/load governor (Kpg/Kig/Kdg), the acceleration control loop (Kpa/Kia) and the temperature control loop (Kpt/Kit). The parameter Lset is the turbine speed/load set-point. The reset controller (Kmwp/Kmwi) acting on the set-point MWset represents the plant outer-loop control which can be in-service, particularly in modern plants, for the purpose of maintaining the unit's output at a pre-specified MW level as set by the plant operator. The deadband parameter 'dbd' may be used to represent the actual programmed deadband used in modern digital governors of GTs. The parameters Rv, Rp, Tp, Lset, Kpg, Kdg and Kig may be used, appropriately, to represent any desired mode of governor action. For example, setting Rv and Kdg to zero and Rp, Tp, Lset, Kpg and Kig to appropriate non-zero values effects a reset-controller with a droop of Rp acting on speed and electrical-power feedback. In contrast, setting Rv, Kdg, Kig and Rp to zero and Lset and Kpg to appropriate none-zero values effects a standard-droop governor with speed feedback alone, where droop is equal to 1/Kpg. The parameters max and min (on each of the three control loops) represent maximum and minimum fuel flow command. The parameter Wfo represents the amount of fuel flow at full-speed no-load. This is the fuel/power required to run the compressor. The minimum fuel flow (command) is typically less than Wfo but never zero. The parameters Vmax and Vmin represent the maximum and minimum fuel valve opening and thus by definition are typically set to 1.0 and 0.0 pu, respectively. The parameter Kt represents the turbine gain and is a scaling factor between the net fuel consumption (fuel flow – Wfo) to the net mechanical power output of the turbine (gross turbine power – power consumed by compressor). The time constant Tv represents the response of the fuel system and the time constant Tthcp, Tn and Td represent the response time of the exhaust (or inlet) temperature measurement system. The parameter Ta is the acceleration control differentiator time constant. The parameters Ttn1, Ttn2, Ttd1 and Ttd2 are time constants associated with the turbine dynamics. The variable Pe is the connected generator electrical power in per unit (on turbine nameplate MW rating) and speed is the unit's speed in per unit.

When the turbine is at its maximum power output it is operating against its temperature limit, and is thus under temperature control. Any fluctuations in ambient air conditions and/or turbine speed will result in a change in the airflow through the compressor. Therefore, the turbine output will change because of a change in fuel demand as determined by the temperature control loop, which is trying to observe the temperature limit. If the turbine is initially below full-load (i.e. below its temperature limit), then in the event of a frequency excursion there will be a transient overshoot in power output as the governor acts to increase power. After a few seconds (the time constant associated with the thermocouples measuring exhaust temperature) the temperature control loop will take control and start to decrease turbine output to maintain the exhaust temperature limit, if the governor attempts to push it beyond this limit. Reference [9] gives a more detailed description of the phenomenon. It is shown in [9] that:

- The maximum power output of the turbine is approximately proportional to changes in ambient atmospheric air pressure.
- The maximum power output of the turbine is a complex function of ambient temperature and drops off non-linearly as ambient temperature increases. This dependence is governed by the airflow characteristic of the compressor.
- The maximum power output of the turbine is also dependant on frequency (shaft speed) through much the same physical phenomenon as ambient temperature dependence, i.e. due to the reduced airflow.

In regions with year round warm to hot climates, inlet-air cooling systems may be installed in order to reduce and maintain the temperature of the air at the inlet of the compressor and thus increase the turbine MW capability.

References [5] and [9] give a detailed description of the dependence of the maximum power output[1] of the gas turbine on ambient conditions and system frequency. For the purposes of power system studies, the variation in maximum power output as a function of frequency is represented in the generic model using the parameter Tlimit and function F(x). Tlimit is the temperature limit of the turbine, which effectively limits the turbine power output. This parameter may be varied in order to effect variations in the megawatt capability of the turbine due to seasonal variations in ambient air conditions (see note 4, Table 4-1). The function F(x) may be implemented as either a look-up table or a piece-wise linear function. This function can be utilized, where necessary, and populated with appropriate data to emulate the dependence of the maximum power output of the turbine on variations in system frequency. To implement this function, manufacturer data is needed on the dependency of turbine peak load on system frequency, an example of this is shown in Figure C-1, in Appendix C. Note that this function differs for different ambient air temperatures.

The parameter aset is the acceleration set-point for the acceleration control loop; that is, if the unit begins to accelerate at a rate of aset pu/s^2 then this control loop acts to limit fuel flow. Finally, the low value select gate will have associated with it appropriate anti-windup reset logic in order to ensure smooth and proper transitioning between the various control loops being fed into the gate. The details of the implementation of this logic may vary among manufacturers and also among software vendors implementing this model in simulation programs. Where such details are important to simulation studies, the user should consult the manufacturer for detailed implementation information. This is depicted in Figure 4-1 by the dashed lines bounding the control loops affected by this logic and the inherent feedback from the low value gate output to each control loop.

Although the generic model shown in Figure 4-1 does not represent the frequency dependence of GT maximum power output based on physical principles, this model is a fair, simple and adequate representation for system studies. Where studies are more focused on the

[1] By maximum power output is meant the continuous base rating of the turbine, some times referred to as ‘base load’, as opposed to peaking operation. Peaking operation is where an option may be provided to allow for temporary operation of the turbine at power levels above the continuous base rating of the turbine by allowing a temporary higher temperature limit. Such operation, however, will result in greater stress on the hot gas path components of the turbine and thus increase maintenance cost [10].

performance of a particular plant or are focused on simulating specific test results pertaining to a unit, a more detailed model [2, 5, 8] may need to be used and the advice of the manufacturer should be sought.

In isolated operation, small systems or for units with low inertia such as aero-derivative turbines it is conceivable that the turbine acceleration control loop may come into play momentarily following a large generation/load imbalance. For these situations the acceleration control loop may be modeled as shown in Figure 4-1.

Figure 4-2 shows a simple model for the Heat Recovery Steam Generator(s) and steam turbine of a combined-cycle power plant. The model is per unitized on the turbine nameplate MW rating. The HRSG and turbine arrangements used in many combined-cycle power plants may generate steam at multiple pressures, have multiple steam drums, and steam admission at corresponding multiple points in the turbine. For purposes of grid simulation it is sufficient to represent this complex steam system by a greatly simplified model in which the entire boiler and turbine are treated as a single drum and single turbine admission valve. Modeling at this level has been shown to represent combined-cycle power plant behavior with accuracy that is appropriate for grid studies [2, 11].

The simplified model shown in Figure 4-2 gives recognition in principle of the thermal and pressure energy storage of the boiler drums, the presence of friction and throttling losses in the steam paths, and the possibility of controlling turbine inlet pressure with the turbine admission valves.

The heat provided by each gas turbine, Qg, is described by a function of gas turbine power, Pgt. The production of steam is taken to be proportional to the sum of the heat provided by the gas turbine (Qg) and that from supplemental firing (Qs). The variation of drum pressure as steam production and consumption are varied is characterized by the single time constant, Tdrum. The pressure loss due to flow friction in the boiler tubes and throttling losses upstream of the turbine inlet valves is characterized by the coefficient, Km. The resulting signal, Pt, represents steam pressure at the turbine admission valve.

The steam turbine power, Pst, and steam mass flow, qt, vary linearly with turbine inlet pressure, Pt, in the steady state. The transient behavior of the HRSG-turbine complex is represented by a lead-lag transfer function specified by Tn and Td. This gives an approximate recognition of the fact that the significant part of the turbine power developed in the intermediate and low pressure turbine sections lags the flow through the high pressure section.

In multi-shaft plants the HRSG part of this model (enclosed in dashed lines) may be replicated for each gas turbine. Diversion of steam for process use or directly to the condenser is represented by a single bypass valve, Bv. The flow of steam through the bypass valve is denoted by qb.

This model includes a simple proportional-integral controller to regulate inlet steam pressure. The controller can be made inactive by specifying a low value for the pressure reference, Pref. This allows the model to represent operation of the steam turbine in either controlled inlet pressure or sliding pressure mode.

For studies involving severe system frequency disturbances additional over-frequency/under-frequency controls may come into play, they may be added to the model as indicated in Figure 4-2. Where studies require such detail or are more focused on the performance of a particular plant or are focused on the simulating specific test results pertaining to a specific unit, a more detailed model may need to be used and the advice of the manufacturer should be sought.

For studies involving co-generation plants, the bypass valve parameter, Bv, may be set to a constant value to simulate the constant extraction of a portion of the steam generated by the

HRSG for process steam. Although the actual steam extraction process may occur at multiple stages/pressure levels, the representation here facilitates a simple generic model that attempts to captures this effect without attempting to model the exact details of the steam extraction process.

To model a single-shaft CCPP, the two models in Figure 4-1 and 4-2 may be connected as shown in Figure 4-3 to develop the resultant total mechanical power on the rotating shaft connecting both turbines and the generator. The constant K_{ST} represents the fraction, in per unit, of total nameplate mechanical power developed by the ST.

4.3 Typical Model Parameters and Modeling Guidelines

Table 4-1 presents a list of parameters for the generic gas turbine model in Figure 4-1. This set of parameters is provided for guidance only, and gives what might be expected to represent the behavior of a typical heavy-duty gas turbine whether in a simple-cycle or combined-cycle power plant. One set of parameters may only be applicable for a given range of operating conditions, for example from 80% to 100 % loading of the turbine.

Table 4-2 presents a list of parameters for the generic steam turbine model in Figure 4-2. This set of parameters are provided for guidance only, and represent what might be expected to represent the behavior of a typical steam turbine, following the gas turbines in a combined-cycle power plant.

Figure 4-4 shows a simulation of a multi-shaft combined-cycle power plant using the generic models. The load/speed reference set-point of the gas turbine was injected with a step increase of 1.5% (0.015 pu). The simulation was performed using the model parameters presented in Tables 4-1 & 4-2, respectively. The results show what would be the expected behavior of a typical combined-cycle power plant connected to a large power grid where system frequency would remain effectively unchanged due to such a step increase in the plant output. The gas turbine output increases until it is limited by the temperature control loop, transiently over-shooting its steady-state maximum power limit. The steam turbine, being essentially in sliding pressure mode, follows the gas turbine output with a delay of several minutes.

When modeling a single shaft combined-cycle unit, the models shown in Figure 4-1 and 4-2 may be connected by the blocks shown in Figure 4-3 and the resultant mechanical power used to drive a single generator shaft. For this case special care must be taken to use a consistent per unit system for the entire model. The most consistent approach is to per unitized the entire model on the nameplate rating of the entire shaft.

Table 4-3 presents a list of typical unit inertia constants for some of the major turbine manufacturers; the values provided are for total shaft inertia including both the turbine and a typical generator. It should be emphasized that total shaft inertia is dependent on the connected generator and thus the numbers in Table 4-3 are typical and provided for guidance only. The manufacturer's data should be consulted for the inertia constant of specific units.

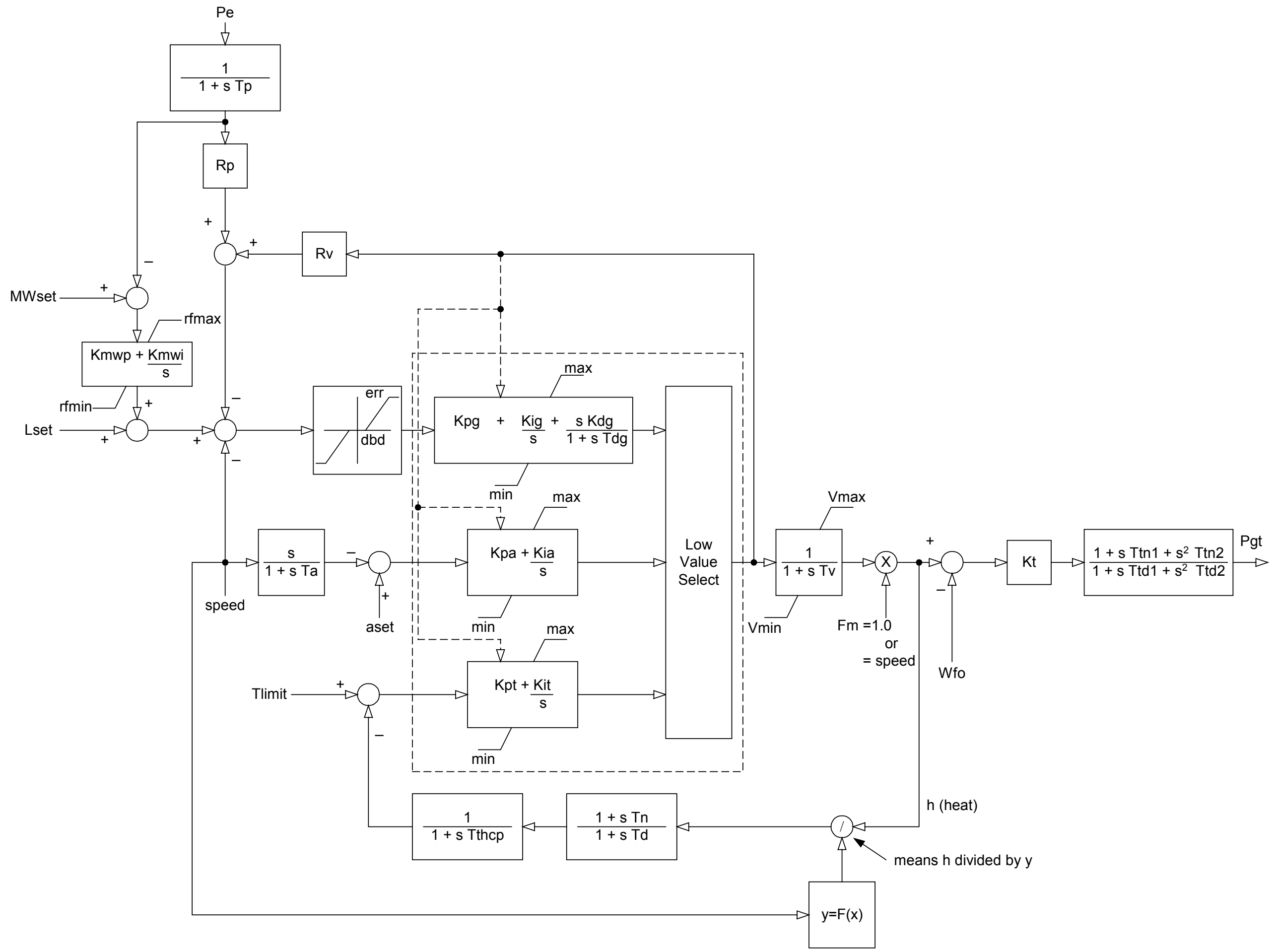


Figure 4-1: Generic GT model.

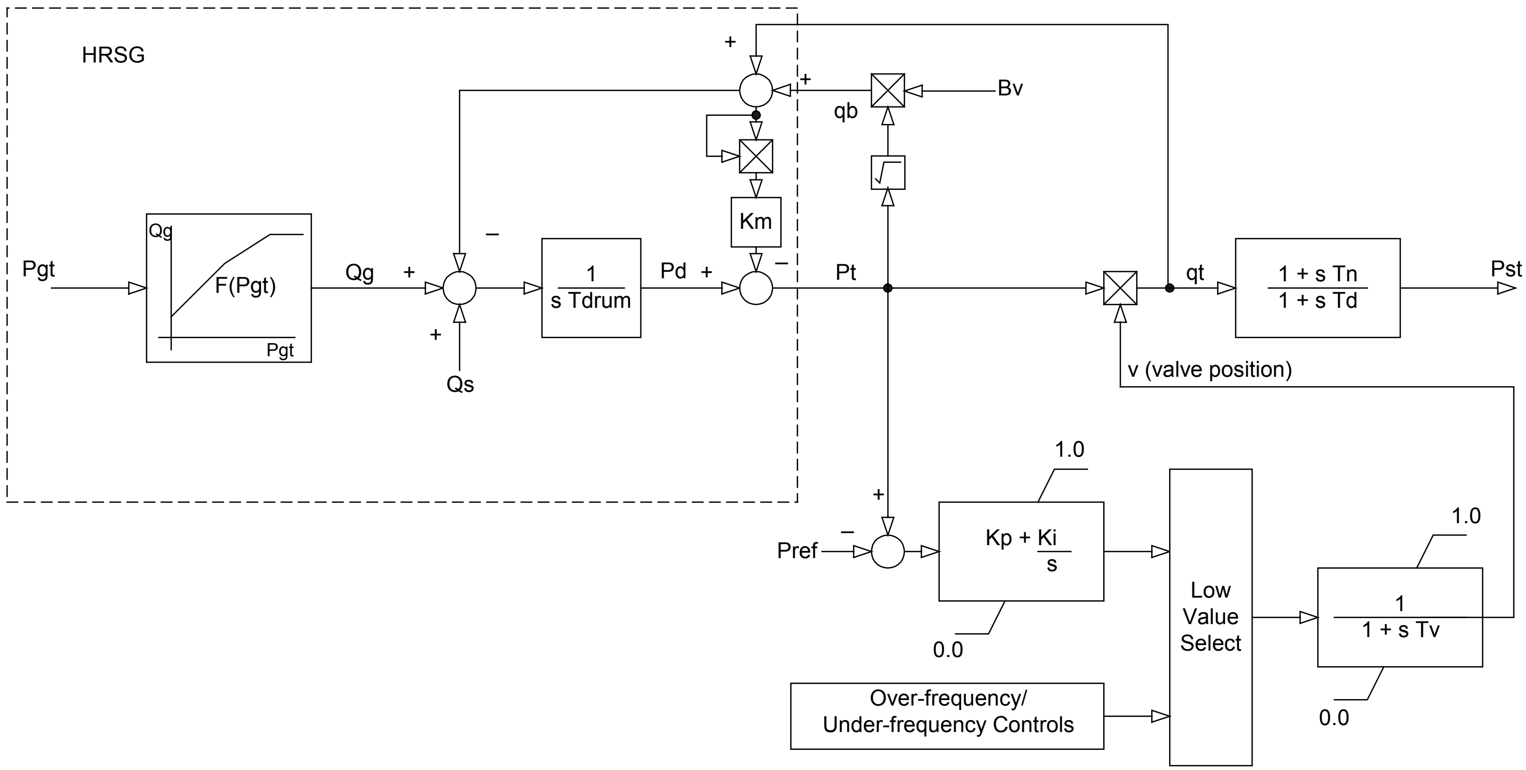


Figure 4-2: Generic HRSG/ST model.

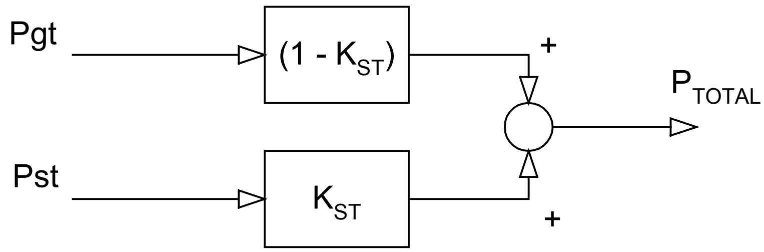


Figure 4-3: Connecting GT and ST model to represent a single-shaft CCPP model

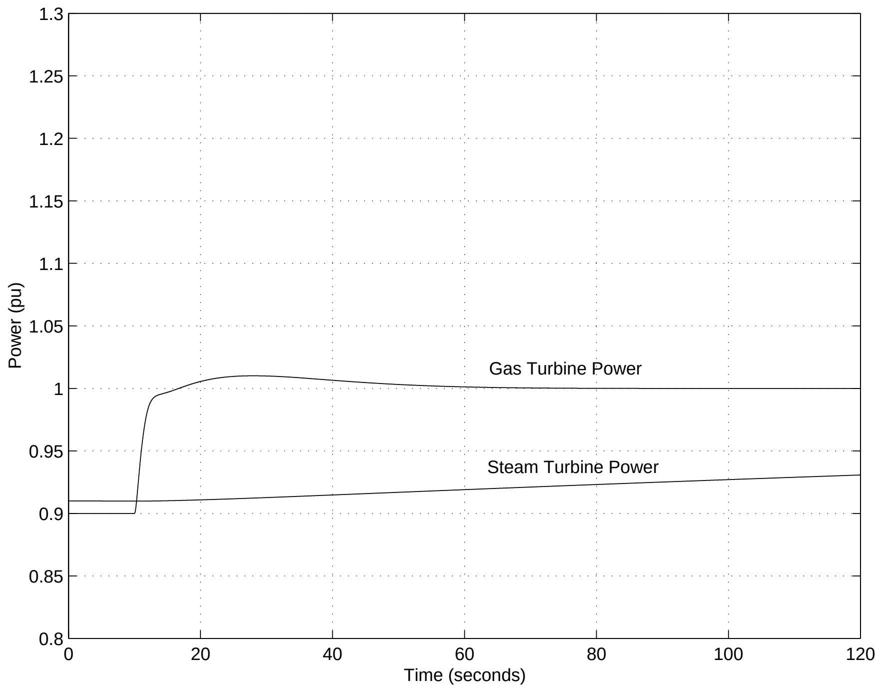


Figure 4-4: Simulated response of the generic models to a step increase in GT load/speed reference set-point.

Table 4-1: Example parameters for the generic gas turbine model.

Parameter Name	Value	Description
Rp	0.05	Electrical power feedback droop
Tp	5	Electrical power feedback time constant
Rv	0	Governor feedback droop
Kmwp	0 (1)	Proportional gain for outer loop MW control
Kmwi	0	Integral gain for outer loop MW control
rfmax	0	Maximum limit on outer loop MW control loop
rfmin	0	Minimum limit on outer loop MW control loop
MWset	0 (2)	Desired MW output of turbine in pu, when outer loop MW control is in-service (i.e. when Kmwp and/or Kmwi are not equal to zero)
Lset	(3)	Load/speed reference set-point
Dbd	0.0003	Intentional deadband
Err	0.005	Intentional error limit
Ta	0.1	Acceleration control differentiator time constant
aset	0.01	Acceleration limit set-point
Kpg	10	Speed governor proportional gain
Kig	2	Speed governor integral gain
Kdg	0	Speed governor derivative gain
Tdg	0	Speed governor derivative time constant
Kpa	0	Acceleration control proportional gain
Kia	10	Acceleration control integral gain
Kpt	1	Temperature control proportional gain
Kit	0.2	Temperature control integral gain
max	1.0	Maximum fuel flow command
Min	0.15	Minimum fuel flow command
Tlimit	0.9167	Temperature limit (in pu corresponds to fuel flow required for 1 pu turbine power i.e. = 1/Kt + Wfo) (4)
Tthcp	2.5	Thermocouple time constant
Tn	10	Heat transfer lead time constant
Td	15	Heat transfer lag time constant
Tv	0.5	Fuel system time constant
Vmax	1.0	Maximum valve opening
Vmin	0.0	Minimum valve opening
Fm	1.0	Fuel flow multiplier; typically set to 1.0. In some cases this is equal to speed (e.g. liquid fuel system with shaft driven fuel pump)
Wfo	0.25	Full-speed no-load fuel flow
Kt	1.5	Turbine gain
Ttn1	0	Turbine transfer function numerator time constant 1
Ttn2	0	Turbine transfer function numerator time constant 2
Ttd1	0.5	Turbine transfer function denominator time constant 1
Ttd2	0	Turbine transfer function denominator time constant 2
F(x)	(5)	Turbine characteristic curve

Notes:

(1) It is important to note that the plant outer control loop modeled by parameters Kmwp, Kmwi, rfmax, rfmin and MWset, if in-service, will greatly change the response of the unit. The unit, if not at its maximum power output (i.e. on its temperature limit), will initially respond through its governor to provide additional output during a system disturbance that results in a frequency drop. However, within tens of seconds to a minute (depending on the gain of the reset controller) the unit's output will be readjusted by the outer control loop to the initial output of the turbine prior to the disturbance.

(2) If the outer loop MW controller is in-service, MWset should be set equal to the initial steady-state value of turbine output during initialization of the model.

(3) Defined by user and/or by simulation program during initialization of the model.

(4) The value of Tlimit given in the table ensures that the turbine reaches its temperature limit once turbine output reaches the turbine nameplate rating (i.e. 1 pu mechanical power output). If, however, the user wishes to simulate a different ambient condition under which for example the maximum achievable turbine output is say 85% of nameplate rating, then Tlimit should be set to 0.85/Kt + Wfo.

(5) F(x) is defined based on data from the manufacturer. For example, a manufacturer's data may indicate that the maximum power output of the GT at 0.96 pu speed is 97% of its rated maximum power output at rated speed. We need to choose a value of F(0.96) to yield a steady-state GT output of 97% of its peak output. Some basic algebra can be done to show F(0.96) = (0.97*(Tlimit – Wfo) + Wfo) / Tlimit. (An example of the type of data needed to evaluate F(x) is provided in Appendix C, Figure C-1.)

Table 4-2: Example parameters for the generic steam turbine model.

Parameter Name	Value	Description
F(Pgt)	(1)	Heat versus gas turbine power, function or look-up table
Tdrum	300	Drum time constant
Km	0.15	Pressure loss due to flow friction in the boiler tubes
Tv	0.5	Actuator time constant for main steam
Kp	10	Governor proportional gain
Ki	2	Governor integral gain
Tn	3	Turbine lead time constant
Td	10	Turbine lag time constant
Qs	(2)	Supplemental firing
Bv	(3)	Bypass valve opening
Pref	0.5	Minimum steam pressure reference

Notes:

(1) A typical curve might be x=[0 1.0] y=[0.1 1.0]; otherwise this information should be sought from the manufacturer.

(2) This is the amount of supplemental firing, in per unit, applied to the boiler; it is to be defined by the user and/or program upon initialization of the model.

(3) This is the fixed position of the bypass valve defined by the user to simulate a fixed amount of steam extraction.

Table 4-3: Typical unit power output capability, polar moment of inertia, and inertia constant for some current turbine designs.

Turbine Type	Turbine Name	Frequency [Hz]	Polar Moment of Inertia [kg m^2]	Typical Generator MVA	Typical ISO Turbine MW	Generator Speed (rpm)	H Inertia on machine MVA base (pu)
GT	GT8C2	50	9550	68.9	57	3000	6.8
GT	GT11N2	50	22803	155.0	113	3000	7.3
GT	GT13E2	50	30256	210.0	165	3000	7.1
ST	KA13E2-3 –ST	50	22128	300.0		3000	3.6
GT	GT26	50	52160	300.0	265	3000	8.6
Single-shaft CCPP	KA26-1 SSPT	50	64136	500.0		3000	6.3
GT	GT11N2	60	15079	145.0	113	3600	7.4
GT	GT24	60	22919	208.0	183	3600	7.8
Single-shaft CCPP	KA24-1 DPRH	60	32338	300.0		3600	7.7
GT	GE-6B-60	60	3850	49	42	3600	5.6
GT	GE-6FA-60	60	6000	90	77	3600	4.7
GT	GE-7EA	60	10000	99	84	3600	7.2
GT	GE-7FA	60	15000	205	174	3600	5.2
GT	GE-6B-50	50	5600	49	42	3000	5.6
GT	GE-6FA-50	50	8590	89	76	3000	4.8
GT	GE-9E	50	19000	147	125	3000	6.4
GT	GE-9FA	50	35000	300	255	3000	5.8
GT (aero-derivative)	GE-LM6000	60	1250	49	42	3600	1.8
GT (aero-derivative)	GE-LM5000	60	750	41	35	3600	1.3
GT (aero-derivative)	GE-LM2500	60	765	25	21	3600	2.2
GT	Solar Titan 130	50	3850	15	14	1500	3.2
GT	Solar Titan 130	60	3000	15	14	1800	3.6
GT	Solar Taurus 70	50	1790	9	7.4	1500	2.5
GT	Solar Taurus 70	60	1320	9	7.4	1800	2.6
GT	Solar Taurus 60	50	1025	6.6	5.4	1500	1.9
GT	Solar Taurus 60	60	770	6.6	5.4	1800	2.1
GT	Solar Centaur 50	50	930	5.5	4.5	1500	2.1
GT	Solar Centaur 50	60	640	5.5	4.5	1800	2.1

It must be noted that:

- All inertias provided are referenced to the speed of the generator shaft.
- The moment of inertia and inertia constant values are for total shaft inertia including both turbine and generator.
- The turbine rating stated in MW is the ISO (15 deg C and at sea level) rating of the turbine.
- Many electric grid simulation programs require the inertia constant to be stated with reference to generator MVA base; this per unit value is shown in column 8 of the Table.
- The actual maximum power output of the gas turbines (and the single-shaft combined-cycle units) depends on ambient conditions; the generator MVA base, and inertia constant are independent of ambient conditions.
- The data provided is not binding and has been provided for guidance only; the turbine manufacturer should be consulted for the inertia constant of specific units.

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CHAPTER 5

MODEL ASSESMENT

5.1 Objectives

5.1.1 General

This chapter discusses the testing work needed in combined-cycle power plants to meet three separate but related sets of objectives, as follows:

1. Testing to meet grid interconnection code and licensing requirements.
2. Testing to identify characteristics of the plant that are needed to set up modeling parameters.
3. Testing to verify that plant models are correct and appropriately accurate.

It is recognized immediately that the testing discussed below is far from the complete range of test work that will be done in the start-up and commissioning of a new plant and that there will be valuable carry-over of results from other testing. It is recognized with equal force that the testing done for other purposes is mostly oriented towards the internal operational requirements of the plant itself, while the testing discussed here is primarily oriented towards describing the plant as it will be seen from the outside grid. Testing oriented to the direct operation of the plant, particularly to its safety and efficiency, always takes precedence over the 'descriptive' testing discussed here. Nevertheless, it is generally advantageous to introduce consideration of this descriptive test work into test planning as early as possible, both to ensure that its requirements are not missed, and because there are often opportunities for advantageous coordination in the scheduling of the different testing requirements.

5.1.2 Code Testing

It is common for the grid interconnection and licensing codes governing major plants to demand testing to verify several aspects of power plant performance.

The governing droop of the plant is considered by many grid operating entities to be critical to the proper operation of the grid load dispatch system and to proper assignment of frequency regulating duty; verification of droop by test is often required.

The primary and secondary power output response of the plant are viewed by the grid operating entities as an essential aspect of their normal and emergency control procedures; verification by tests involving a prescribed frequency disturbance is often required.

It is common for grid interconnection codes to prescribe the tests in considerable detail. One typical requirement is measurement of the response of the plant, in MW, to a synthetic frequency excursion of the form shown in Figure 5-1. The times t1 and t2, shown in Figure 5-1, should reflect the known characteristic of the system. The increase in power produced by the response of turbine controls to such a dip is often used as the measure of primary and secondary response. In many cases, while grid interconnection codes prescribe tests in considerable detail, these prescriptions are given in terminology that can only be related to a 'generic' power plant. This is done in the codes because of the administrative necessity of avoiding reference to any particular plant. The reconciliation of real test procedures that are practical and safe in a given plant with the wording of the applicable grid codes must be done with care and should be initiated as far as possible in advance of scheduled testing dates.

In most cases the testing required for compliance with grid interconnection codes is oriented to the observed behavior of the plant without any particular reference to analytical modeling. While the code testing is often useful for other purposes, the requirements spelled out in the codes may not cover what is required for model verification.

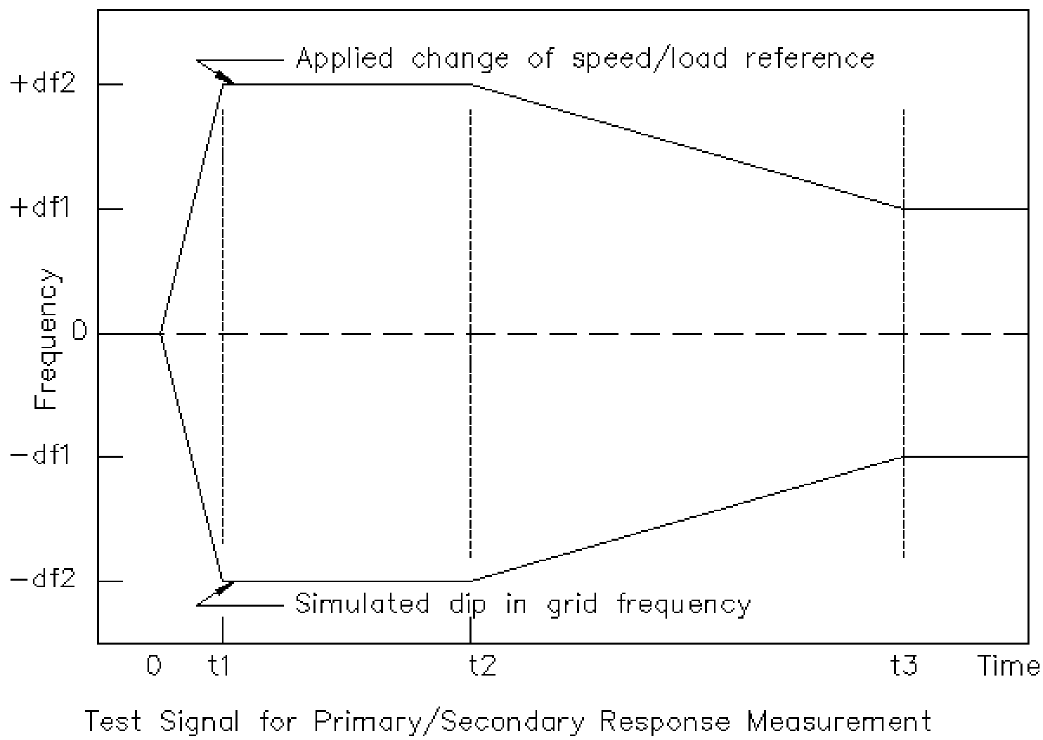


Figure 5-1

5.1.3 Testing for Model Verification

The ultimate purpose of the modeling discussed throughout this technical brochure is to provide the entities responsible for the electric transmission grid with trustworthy models of the plants on which the grid depends. Accordingly, the objective of all of this work, after a model has been developed and implemented in a simulation program, is to verify that the use of that model, in the particular grid simulation program, gives a reliable indication of what the plant will do in a grid disturbance. It is common for grid interconnection codes to include the delivery of a 'validated' model as a requirement for licensing.

The testing required for model verification can be both more extensive than, and different from, that needed to verify that plant performance meets code requirements. Verification of performance requires only that the plant do what is required, but verifying a model requires that it be shown how this performance is achieved.

Accordingly, it is generally necessary to plan for more extensive testing than is described specifically in grid interconnection codes. It is important to introduce this fact into test planning as early as possible.

5.1.4 Testing for Model Identification

The identification of dynamic models is a separate undertaking from that related to performance and model verification. The former can usually presume that a static and dynamic model of the plant has been developed and is ready for use; the emphasis there is on confirmation of work already done. (The model development is, of course, an integral part of the design of the plant). Model identification testing is prompted mainly by:

- The need to explain an unexpected behavior of a plant component.
- The need for continued advancement of power plant modeling in the academic and research sense.

The first of these is generally met by the application of skill and ingenuity during test sessions. In this regard it is important that the test staff be aided by flexibility and patience on the part of the plant and grid operations entities.

The second of the above considerations is usually handled best by expanding the monitoring of tests undertaken for the primary objective to cover signals of interest in the research and development activity.

5.1.5 Organization of Discussion

While the planning and administration of tests in advance must recognize the three sets of objectives explicitly, the execution of tests must be done in accordance with the requirements of minute-by-minute operation of the plant. The following discussion therefore follows a sequence related to operational matters, rather than to the objectives outlined above.

5.2 Steady Operation Tests

5.2 1 Governing Droop

Governing droop of a combined-cycle power plant must be clearly defined. Most combined-cycle power plants are relatively new, use electronic controls, and have their governing droop implemented as a relationship between turbine speed and generator electrical power. (Older mechanical-hydraulic turbine governors generally implement droop as a relationship between turbine speed and control valve position, with the relationship between speed and power then being dependent on variables such as steam pressure).

While policies relating to governor droop normally specify the droop in terms of percent speed change for a one per unit change in turbine power, this specification must be translated into a concrete measurable form for each specific plant. For a plant under test, the droop should be stated in terms of:

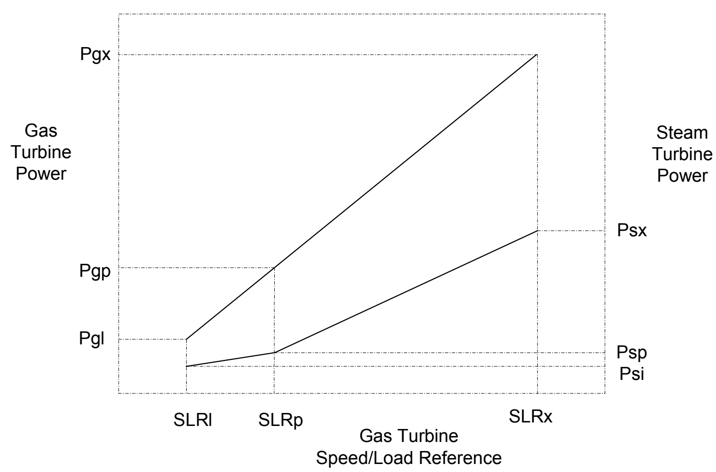
- either (percent speed change)/(MW of power change), or
- (Hertz of frequency change)/(MW of power change), or
- (rpm of speed change)/(MW of power change)

The governing droop of a combined-cycle power plant must, further, be defined with respect to where and how it is implemented. In a single-shaft plant the governor droop can, but does not necessarily, relate total power to shaft speed. In a multi-shaft plant, however, it is most common for the droop to be a characteristic of the individual gas turbines relating the power of their individual generators to their own speeds. In this case the effective droop of the plant is implied by the dependence of steam turbine power on gas turbine power, but is not a specific control function.

In both single-shaft and multi-shaft plants the dependence of steam turbine power on gas turbine power may not be simple or linear over the plant operating range. Factors in this variation include:

- The delivery of a varying fraction of the total steam production to users other than the steam turbine.
- The use of supplemental firing to increase steam production.
- Changes in gas turbine and steam turbine control modes.

An 'ideal' variation of steam turbine power with gas turbine output might be as shown in Figure 5-2. This particular relationship results in different plant effective droops at high and low outputs, even though the gas turbine governor droop is constant over its full operating range.



$$Rgt = \frac{(SLRx - SLRp)}{(Pgx - Pgp)}$$

$$Rplant = \frac{(SLRx - SLRp)}{K(Pgx - Pgp) + (Psx - Psp)}$$

SLRi, SLRp, SLRx - speed/load reference (pu).
Pgi, Pgp, Pgx - gas turbine power output at various speed/load reference set-points (pu).
Psi, Psp, Psx - steam turbine power output at various speed/load reference set-points (pu).
K - number of gas turbines per steam turbine.
Rgt - gas turbine droop in pu speed/pu power.
Rplant - total plant droop in pu speed/pu power.

Figure 5-2 Variation in overall plant droop. The plot shows steady-state power output of the gas turbine and steam turbine in a combined-cycle power plant versus speed/load reference set-point.

Because plant droop may be a variable quantity, it is recommended that testing for droop be aimed at stating the relationship between speed/load reference and power output graphically over the widest possible range of plant output, and that the outputs of the individual generators should be recorded and presented in the case of multi-shaft plants. Particular values of droop in load ranges and operating conditions of interest can then be derived as needed. It should be noted that while the formal definition of governing droop is usually made in terms of the derivative

$$\partial(\text{speed})/\partial(\text{power})$$

it is rarely possible to measure this directly. The more practical approach is to make separate measurements of

$$\partial(\text{speed})/\partial(\text{reference})$$

with the unit off line and with true speed varying and of

$$\partial(\text{power})/\partial(\text{reference})$$

with the unit on line and with true speed held substantially constant.

The signal (reference) in these two measurements is the speed/load reference of the governor.

The governing droop is then given by division

$$\text{Droop} = R = [\partial(\text{speed})/\partial(\text{reference})]/[\partial(\text{power})/\partial(\text{reference})]$$

The values of these two partial derivatives can be obtained as the slopes of graphs of speed and power versus governor reference.

Figure 5-3 shows an example of droop measurement on a single gas turbine running in simple cycle. The graph of power versus speed exhibits the normal imperfection of test work. The graph of speed versus reference in this case reflects the fact that the governor implements the droop through an integral control block and hence achieves an almost perfectly linear relationship between speed reference and power (i.e. droop is constant).

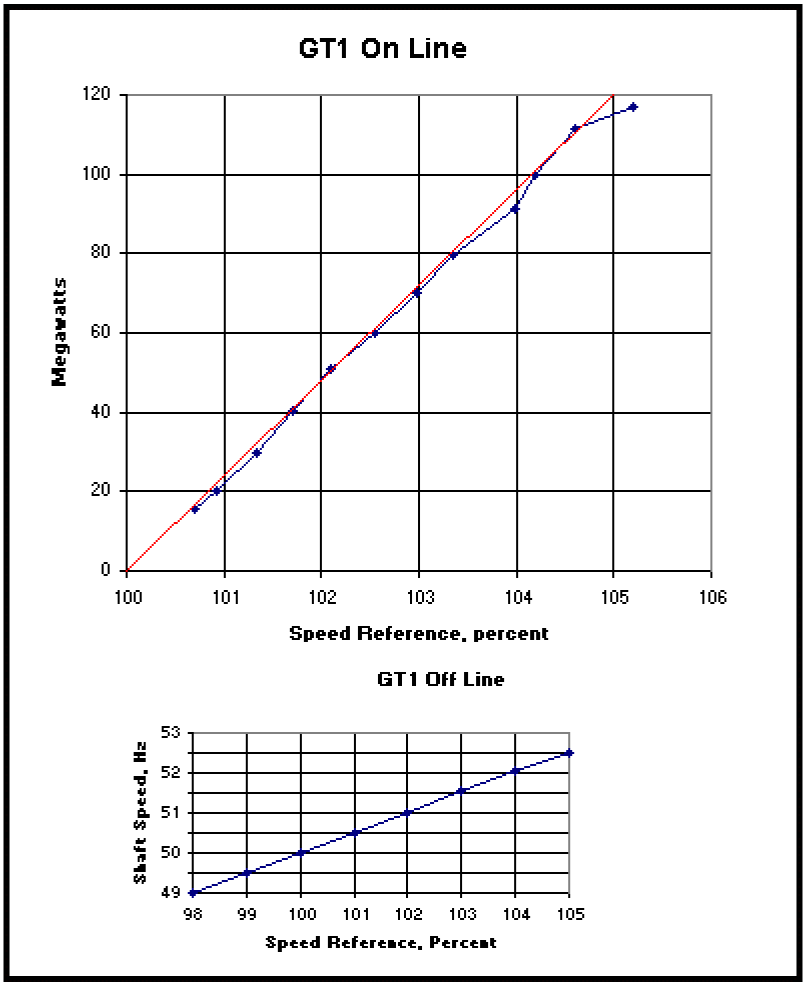


Figure 5-3 Droop of a simple-cycle gas turbine.

5.2.2 Mapping of Steady State Operation

The operation of a combined-cycle power plant depends on programmed relationships between many plant variables. These relationships are implemented by the gas turbine controls, the steam turbine controls, and the plant supervisory computer. While the nominal form of these control relationships are often available from the various control computers, the nominal forms do not always reflect true transducer gains, and imperfections in the way the plant follows the programs.

Many aspects of the modeling of a combined-cycle power plant refer to the relationships between plant variables. It is very desirable, therefore, to map the steady state operation of the plant as comprehensively as possible at the same time as dynamic response testing is undertaken. This mapping can be made, for example, by recording all significant variables during a normal run up of the plant from minimum to maximum load. The variables recorded should include, at least:

- Governor load reference(s)
- Gas turbine power output(s)
- Gas turbine governor fuel valve demand signal (governor output signal)

- Gas turbine fuel valve actual position signal (valve feedback signal)
- Gas turbine fuel flow
- Gas turbine guide vane opening
- Gas turbine exhaust temperature
- Gas turbine pressure ratio (or compressor discharge pressure)
- Boiler drum pressures(s)
- Superheater outlet steam temperature(s)
- Steam turbine throttle pressure (upstream of control valves)
- Steam flows to HP, IP, and LP turbines
- Steam turbine power output
- Steam turbine governor reference signal
- Steam turbine governor control valve demand signal

Figure 5-4 shows a representative recording of gas turbine variables taken during runup from minimum to maximum output. Note that while the load acceptance could be made at a uniform rate in this case, it is often necessary to pause for reasons such as changes in steam system control mode, and management of temperatures in the steam system and steam turbine. Careful advance planning is essential to ensure that the recording of plant variables is made in consistent conditions from beginning to end.

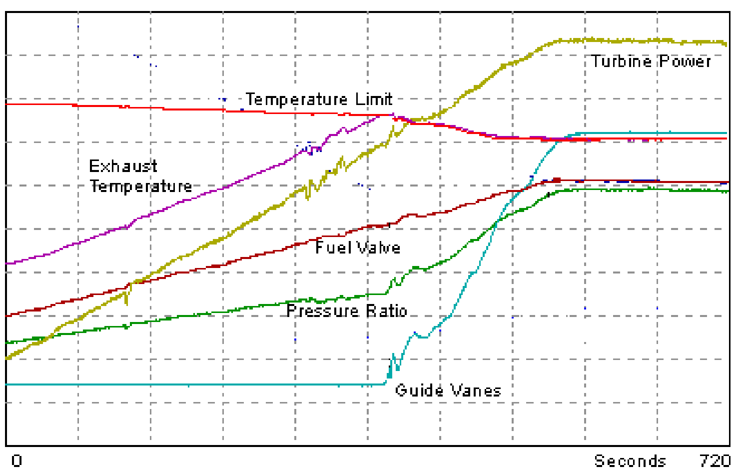


Figure 5-4 Loading of a heavy-duty gas turbine.

5.3 Dynamic Response Tests

5.3.1 Off Line Governor Dynamics

The dynamic behavior of the speed governor when the generator is off line is of no significance in grid studies because it is common practice to use different speed controller gains in on line and off line operation. Tests of the governor's response with the generator off line are, nevertheless, a very useful basis for verification of parts of the turbine governing model other than the speed control block itself. In particular, it is practical in an off line test to make the governor execute a small but rapid movement of the fuel valve; this is an excellent basis for verifying the phase lag and rate limit characteristics of the fuel valve. This test is also useful in the estimation of the time constants describing phase lag between the position of the fuel valve and the appearance of shaft power. These phase lags can be significant in engines where there is a substantial volume of manifold piping between the fuel valve and the combustors, for example.

Off line dynamic response can be stimulated either by making step changes to the governor speed/load reference or by manually opening the generator circuit breaker with the engine at a low initial output. The former approach allows both opening and closing initial motions of the fuel valve and is preferable, but depends upon the ability to gain direct access to the governor reference and requires meticulous safety precautions to ensure that the turbine is not inadvertently given an order for an excessive change of speed. Opening of the circuit breaker requires no penetration of the turbine controller but requires more time for each test and is of greater concern to the grid operator.

Figure 5-5 shows the recording of a sequence of tests by step changes to the speed load reference of a large single-shaft gas turbine. Figure 5-6 expands the recording of one of these step responses and clearly reveals a phase lag and rate limit between the speed governor output signal and the fuel valve position order signal.

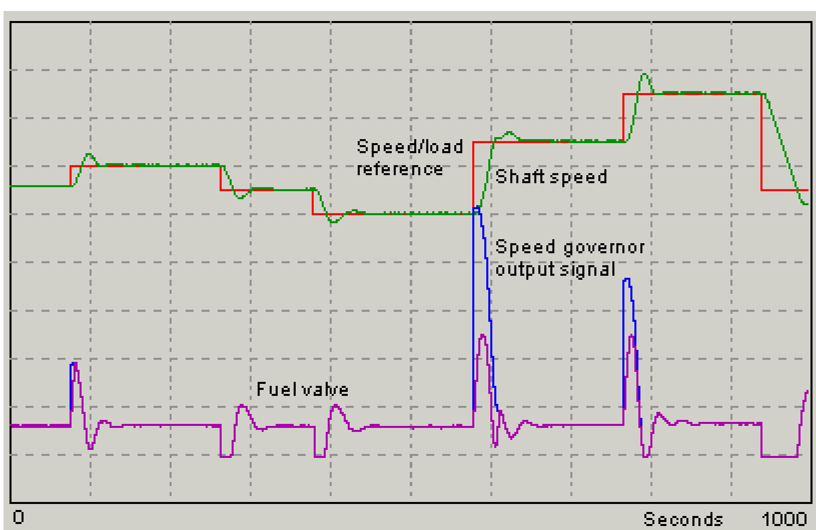


Figure 5-5 Off-line speed/load reference step tests on a heavy-duty gas turbine.

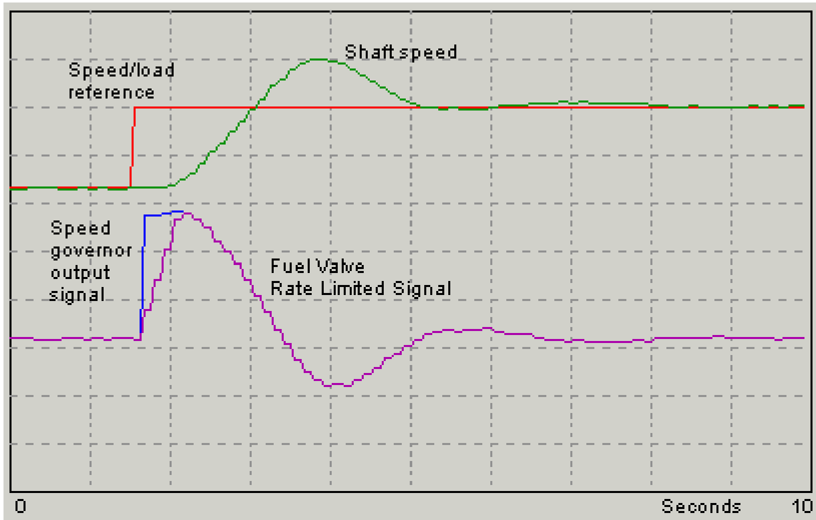


Figure 5-6 Off-line speed/load reference step tests on a heavy-duty gas turbine.

5.3.2 On Line Governor Dynamics

5.3.2.1 Speed/Load Reference Profile tests

The measurement of governing dynamics with the plant on-line is more difficult than when off line, but is mandated in many grid interconnection codes to verify primary/secondary response and to validate the ability of dynamic models to reproduce this behavior.

Several grid interconnection codes require a test where a severe dip in grid frequency is simulated by the injection of a signal with an equal-and-opposite upward profile into the governor as an addition to the speed/load reference. A typical requirement is for the governor reference to be raised by about 1 percent (.5/.6Hz) within 5 seconds or less. Primary response is then defined in several codes as the increase in output achieved in the first few seconds, typically about 10 seconds. Secondary response is typically defined as the response that is achieved in about 30 seconds and then sustained for a substantial period.

It is very desirable to execute tests of this type with varying amplitude of the test disturbance and from varying initial gas turbine outputs. At small test amplitudes tests of this type are a relatively mild disturbance to the gas turbine engine; useful test data can be obtained when the test signal amplitude is only of comparable amplitude to frequency variations that are encountered in normal operation.

Depending on the limiting and protection provisions of the engine control system, it can be practical to execute these tests with test signal amplitudes that will perturb the plant output by large amounts. Execution of any test that will perturb the plant output by more than about 10 percent must be planned with care and respect for safety, possible expenditure of fatigue life of engine parts, and impact on the grid.

Figures 5-7 and 5-8 show recordings of tests made by injection of the signal shown in Figure 5-1; with a small amplitude in Figure 5-7 and with a large disturbance in Figure 5-8. Figure 5-9 shows the response of the same machine to the same disturbance as in Figure 5-8 but from a higher initial load so that the output increase called for by the governor in accordance with its droop setting is overridden by the temperature limit.

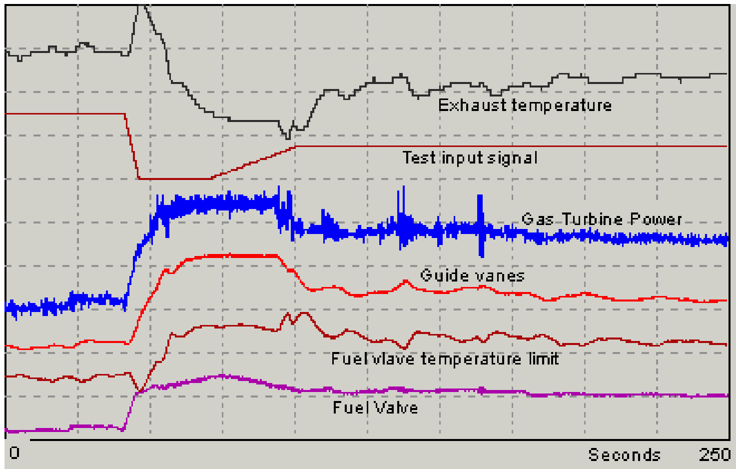


Figure 5-7

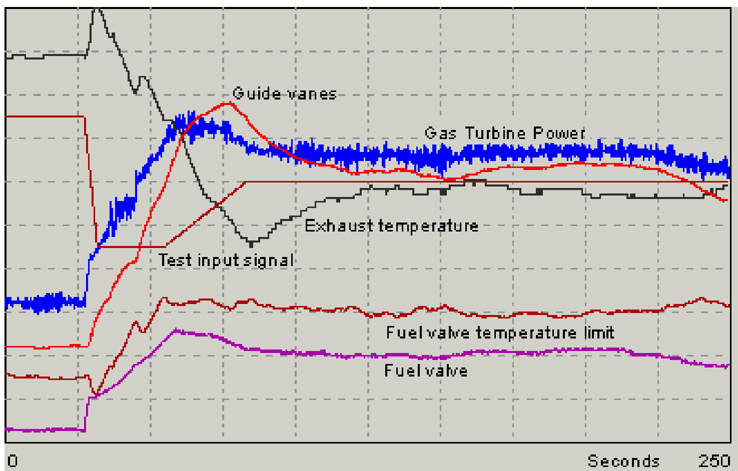


Figure 5-8

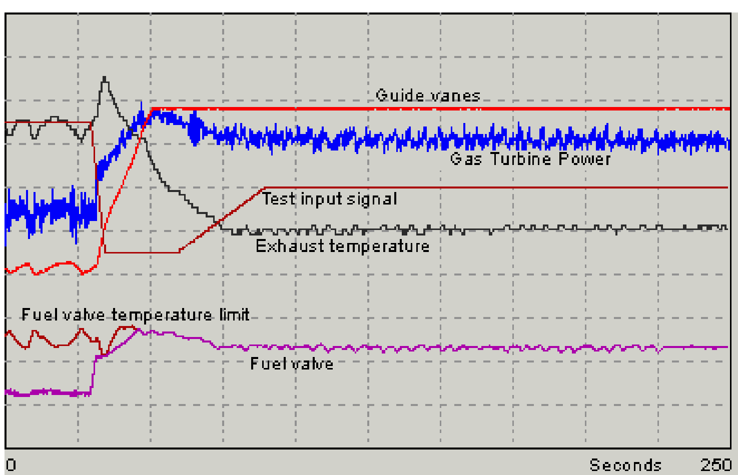


Figure 5-9 On-line speed/load reference step tests on a heavy-duty gas turbine.

5.3.2.2 Frequency Reference Step Tests

It is advantageous for verification of the model elements involving shorter time constants to complement the frequency dip tests with tests stimulated by small step changes of the governor speed/load reference. The methods and precautions needed for these tests are the same as for the frequency profile tests and it is generally advantageous to schedule both types of test in the same session. The difference between the two becomes more significant as the initial rate of change in the frequency profile test is reduced. When the frequency profile rate of change is slow, the step test gives a clearer confirmation of the dynamic behavior of the speed governor and temperature limit controllers.

5.3.3 Tests Related to Steam System Dynamics

5.3.3.1 Tests Performed by Changing Gas Turbine Power

The principal characteristics of the steam system is the time constant, or time constants, describing the storage of steam in the boiler drum(s) and in the steam headers between the boilers and the steam turbine. These time constants are very much longer (hundreds of seconds) than the time constants involved in the gas turbines (fractions of seconds). Because the steam system time constants are long, maneuvers of gas turbine power that are quite mild with respect to the gas turbine can be regarded as step changes with respect to the steam system.

Useful test information on the steam system can be obtained readily by recording steam signals during any operation that produces a clear change of gas turbine power in less than about 30 seconds and is followed by about 30 minutes of steady operation of the gas turbine. The step response tests described in 5.3.2.2 meet this requirement. However in many cases there is no need for any special test stimulus; the required change in gas turbine power can be achieved simply by having the plant operator change output decisively at his maximum normal rate. It is advantageous to make these tests both with the steam turbine control valves in effectively fixed position, full open for example, and with them in regulating range. The test with the steam turbine valves inactive gives a useful separation of the boiler dynamic characteristics from the dynamics of the pressure controller.

It is important to note that steam system tests should have plant load references, both the governor references and the plant load control references, held constant for an extended period both before and after application of the test input signal. This requires explicit coordination with plant operators and with the grid operating entity.

Figure 5-10 shows a typical response of a boiler and steam turbine to a quick change of gas turbine output. The gas turbine output was increased quickly over approximately 25 seconds and then held constant. The boiler HP drum pressure and steam turbine power then followed with a time constant of approximately 400 seconds.

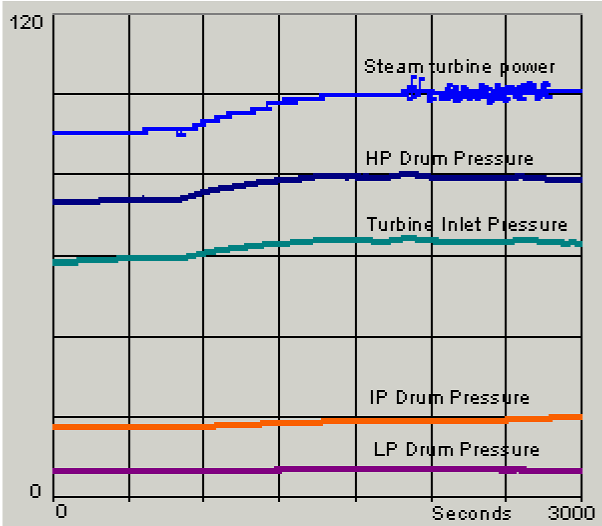


Figure 5-10 Steam turbine response to a change in gas turbine power; steam turbine control values are wide open.

5.3.3.2 Tests Performed by Changing Steam Turbine Pressure Reference

The dynamic characteristics of the steam turbine pressure controls can be determined by effecting rapid changes, or step changes if practical, on the pressure reference set-point of the steam turbine pressure controller. Such tests are not useful, alone, to reveal the characteristic time constant of the steam system. They are useful in conjunction with the tests described above in which the heat input to the boilers is changed quickly.

Figure 5-11 shows the response of the steam system of a large multi-shaft combined-cycle power plant when the steam turbine pressure reference is raised by a small step change. The movement of the steam turbine control valves under the direction of the inlet pressure controller is clear.

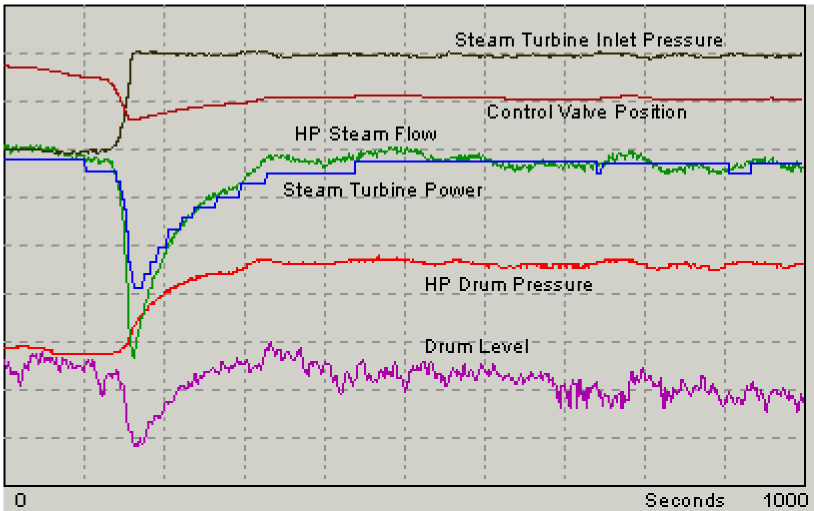


Figure 5-11 Steam turbine response to a step change in steam turbine pressure reference; steam turbine values regulating.

5.4 Operations and Precautions

5.4.1 Operational Scheduling

The coordination of test work in combined-cycle power plants reflects both their operational complexities and the fact that their outputs are often at the level where the maneuvering associated with testing is a significant concern to the grid operating entity.

Scheduling must recognize that, while gas turbines can be maneuvered relatively quickly, the timing of test steps must respect the slower maneuvering rate capabilities of the steam system and steam turbine. Steam and metal temperatures in the steam turbine are a major concern, even though these variables have no significant effect on turbine output and do not appear in the dynamic models considered in this document.

The possibility of an inadvertent trip during testing is a very real concern both to the grid and to the plant operator because:

- The sudden loss of the plant from a high output can create a grid emergency in situations where the plant is a significant fraction of the loaded capacity on the grid at the time.
- Restart of the combined-cycle after a plant trip can take several hours, leaving the grid in a capacity emergency for that period.

These concerns are often addressed by the grid by requiring that testing of a multi-shaft plant be done with only one of its two or more gas turbines in service. This minimizes the disturbance created by an inadvertent trip, but complicates the use of the test results in the verification of primary/secondary response and in the validation of simulation models with respect to full load conditions.

The verification of primary/secondary response, for example, can be handled with fair approximation with only one gas turbine in service, because the contribution of the steam turbine to the response in the period normally considered (less than one minute) is very small. Thus, even though the extrapolation of plant response as double the response achieved with a single gas turbine in service is undoubtedly an approximation with respect to the steam turbine contribution, the error in the overall result is minimal.

Where grid disturbances are a major concern, it is sometimes even requested that testing on the gas turbines should be done one-by one, and with the steam turbine unloaded. This is technically possible in some plants, but may represent an unacceptable duty for the condenser and/or bypass valves in others if carried on for any prolonged period.

5.4.2 Precautions

The normal precautions that are prudent in all power plant testing apply in combined-cycle power plants. Particular care must be given to operational factors that are unique to combined-cycles. Among these factors:

- Loading and unloading rates that are well within the safe capability of a gas turbine under test can be excessive with respect to the ability of the boilers and steam turbine to absorb the resulting change in heat input.

- While the pressure response of the steam system is relatively slow, the boiler drum level responds to changes of heat input much more rapidly than pressure and can approach tripping levels quickly if not managed carefully.

- Loading rates on the gas turbines are as likely to be limited by rate of change of steam temperature as by limitations of the gas turbine itself.

- Operation of a gas turbine at high output without loading the associated steam turbine may require full steam flow to be diverted through the bypass valves to the condenser; this can be unfavorable duty for the valves and condenser.

CHAPTER 6

SUMMARY AND RECOMMENDATIONS

6.1 Overview

Combined-cycle power plants are reported as having efficiencies exceeding 55%. Due to their high efficiency and other advantages, combined-cycle power plants have gained popularity and are beginning to account for a significant portion of the generation mix in many power systems around the world. This document has focused on describing combined-cycle power plants and their dynamics and control, and thus provides guidance for understanding combined-cycle power plants and modeling them for the purpose of power system studies.

Generally, there are two types of combined-cycle power plants

- Single-shaft units where the gas turbine and steam turbine assembly are in tandem on a single shaft, which drives the electrical generator connected to the grid.
- Multi-shaft units with one or more gas turbines each with its own heat-recovery steam-generator (HRSG) feeding steam to a single steam turbine, all on separate shafts with separate generators. For smaller units it is possible to have the exhaust gas from a number of gas turbines all feeding into a single heat-recovery system. For industrial or small municipality applications, the plant can be configured for co-generation where part of the steam produced is used for industrial processes or district heating. Higher efficiencies are possible with single-shaft units while multi-shaft units allow for greater flexibility where the gas turbines can typically be run independently.

Gas turbines in large combined-cycle power plants are heavy-duty turbines with relatively high inertia. For smaller industrial applications or peaking, simple-cycle aero-derivative turbines may be used.

Two emerging technologies should be noted:

- Integrated Gasification Combined-Cycle (IGCC), and
- Compressed Air Energy Storage (CAES) units.

In IGCC technology the fuel base of a conventional combined-cycle power plant is vastly expanded beyond natural gas to include lower cost and often more abundant resources such as coal. In CAES technology air is compressed during off-peak hours and stored typically in an under ground pressurized cavern. During peak hours the compressed air is released and mixed with fuel in a combustion system to run a gas turbine driving an electrical generator.

6.2 Performance, Control and Dynamics of Combined-Cycle Power Plants

Excluding combined-cycle power plants with supplemental firing, approximately two-thirds of the total power in a combined-cycle power plant is generated by gas turbines while one-third is generated by steam turbines. The steam turbine power output follows that of the gas turbines by generating power with whatever steam is available from the heat recovery steam generator(s). Thus, the electrical output of a combined-cycle plant without supplementary firing is typically controlled only by means of the gas turbines. The steam turbine is typically operated in sliding pressure control (control valves wide-open). At reduced loads, in the order of 30 to 50% of full load depending on the design, the control valves may be partially throttled to maintain steam pressure (inlet pressure control). Therefore, since the steam turbine is typically under sliding pressure control, the steam turbine generally does not respond quickly to changes in gas turbine power output. That is, a gas turbine load change will result in a steam turbine load change after a delay of a few minutes due to the time constant associated with the heat-recovery steam-generator (HRSG).

The maximum power output of a heavy-duty single-shaft gas turbine is highly dependent on the ambient air conditions (pressure and temperature) and the frequency of operation of the unit. This dependence is a consequence of the air-flow characteristics of the axial compressor and the enforced turbine temperature limit. Consequently, the maximum power output capacity of a simple-cycle gas turbine or a combined-cycle power plant is highly dependent on the climatic conditions of the installation and seasonal weather conditions.

If a CCPP, or for that matter a simple-cycle gas turbine, is not at its peak load (i.e. not on temperature limit) then the unit may participate in primary frequency control through the action of the turbine-governor. The governor on a gas turbine will control the turbine power output until the unit reaches its temperature limit. Once at temperature limit the unit will not yield any more increase in power output. In fact, for severe frequency excursions the maximum power output of a heavy-duty gas turbine will decrease as system frequency falls due to the decrease in air mass flow through the axial compressor. It is possible to allow a temporary higher temperature limit to facilitate overfiring and thus to maintain or increase the GT output during a severe frequency excursion. Such a capability, however, will result in greater stress on the hot gas path components of the turbine and thus increase maintenance cost. The steam turbine power output, as described above, will typically follow in a mater of minutes. During a severe frequency excursion over-frequency and under-frequency controls/protection can come into play, thus affecting the steam turbine power output.

The primary difference between simple-cycle operation and combined-cycle operation of gas turbine controls is in the controlling of airflow using the variable inlet guide vanes (VIGVs). In simple-cycle operation the VIGVs are wide open from a relatively low load level up to base-load. This allows for greater cycle efficiency and a lower heat-rate as the exhaust temperature of the turbine is kept lower during a greater portion of the loading cycle. In combined-cycle applications the VIGVs are typically modulated and become wide open only at or near the maximum power output of the unit. This is done in order to increase exhaust temperatures at part load conditions thus substantially increasing the exhaust energy recoverable by the HRSG. The net result is an overall higher efficiency for the combined-cycle power plant during part-load condition. The consequence of this variation in control strategy for similar units at part-load is that during a frequency excursion the resultant transient overshoot in turbine power would be higher on a simple-cycle unit since it would have started at a temperature significantly lower than its combined-cycle counter part. However, the final steady-state power output of both turbines would be the same once they reach, and settle down, at their temperature limit (or lower steady-state power output).

6.3 Modeling Recommendations

Based on the contents of this report, the following recommendations are presented as guidance for the purpose of modeling combined-cycle power plants in power system studies:

- Transient and mid-term time-domain stability analysis: Transient stability analysis typically requires 5 to 20 second time domain simulations. The objective of such analysis is to study the dynamic behavior of the system following a large disturbance and thus identify if any generating systems become unstable due to first-swing instability. For the sake of simplicity it is adequate to use a simple gas turbine model without representation of fluctuations in maximum power output due to frequency variations and to represent the steam turbine with constant mechanical power. Simple gas turbine models exist in most commercially available power system simulation programs. Alternatively, the model presented in Chapter 4 may be used. Mid-term time-domain stability studies require simulations over several minutes following a system disturbance. These simulations are often performed in relation to voltage-stability. If such studies involve disturbances that result in generation/load unbalance the simple models and approach for transient stability analysis may not replicate important GT and CCPP characteristics. Models of the form presented in Chapter 4 should be used to represent both the expected response of the gas turbine and the steam turbine.

- Small-signal analysis: Small-signal stability analysis is typically performed on a set of the power system equations linearized at a specific operating condition. Inherent in the analysis technique is the assumption that system perturbations are small and thus should not invoke any non-linearities in the system. If the gas turbine is on temperature limit (at its maximum power output), then small perturbations in electrical power and or system frequency will most likely have little to no effect on the output of either the gas turbine or steam turbine. Thus, it is appropriate for such cases to make no representation for either the gas or steam turbine-governor; that is, mechanical power is kept constant. Under part-load conditions a linearized version of a simple gas turbine model may be used. Since the steam turbine is most likely in sliding pressure mode, constant mechanical power input should be assumed for the steam turbine.

- Islanding studies or studies on small systems: Islanding studies are concerned with the separation of pockets of load/generation from a major power grid. This may occur as a result of the loss of a tie line between an industrial facility and a utility grid or as a result of a major disturbance that leave a large interconnected grid in fragments following major transmission line outages. Under such conditions the resulting islands may experience a large mismatch between generation and load. For such studies, particularly if the frequency in the island declines due to a lack of generation, models with adequate representation of the frequency dependency of the gas turbine maximum power output should be used. For mid-term simulations the output of the steam turbine must be linked to the gas turbine(s) through an appropriate steam turbine/HRSG model. The models presented in Chapter 4 incorporate these features, and are appropriate for islanding studies.

- Plant specific studies: Where studies are focused on the performance of a particular plant or the simulation of a specific test of a unit with measurement of internal plant variables, a more detailed model may need to be used and the advice of the manufacturer or other experts should be sought.

The generic models described in Chapter 4, if used properly, are suitable for the studies described in the first three items. Other models of this type may be available in simulation programs. Other manufacturer specific models may be used for any of the above studies, and are likely necessary to facilitate the objectives of a study of the nature described in the fourth bullet.

In general, when performing power system studies involving the modeling of combined-cycle power plants, questions to be considered include:

- Are units partially loaded and thus able to respond to an increase in megawatt demand due to a decrease in system frequency?
- Are units at their peak load (on temperature limit for gas turbines) or are they under outer-loop megawatt control (see Figure 4-1 and note 1 of Table 4-1)?
- Has the response of steam turbines in combined-cycle power plants been properly modeled?

Ensuring that such questions have been answered and valid assumptions have been employed in preparing system models for grid studies are essential to the success of a study.

APPENDIX A

THERMODYNAMICS OF COMBINED STEAM AND GAS CYCLES

A.1 General issues

The maximum of the Carnot efficiency, which could be attempted by an ideal thermal cycle, is:

$$\eta_C^{\max} = 1 - \frac{T_i}{T_s}$$

where T_s represents the maximum temperature of the cycle, and T_i the environmental temperature.

The thermal efficiency of a real cycle is obviously lower then the ideal one. The efficiency of an actual cycle is lower than this ideal maximum value because of the energetic and exergetic losses in a real cycle.

Figure A-1 shows the typical temperature intervals for which mechanical work is extracted for a conventional steam cycle (CSC), a gas turbine (GT) and a combined-cycle power plant (CCPP).

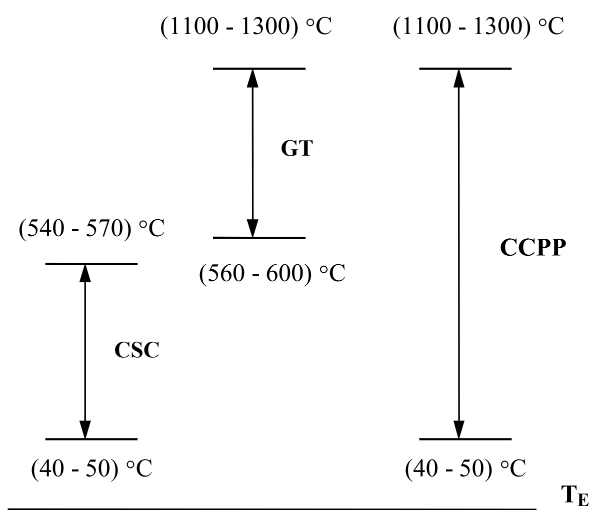


Figure A-1: Typical temperature intervals for extraction of mechanical work (T_E – ambient environmental temperature).

For the conventional steam cycles, the mechanical work extraction is realized between relative low temperatures. Even though temperatures that can be obtained after the fuel combustion can reach 1800 to 2000°C, the steam temperature does not usual exceed 540 to 570°C. The bottom temperature of this cycle is close to the ambient environmental temperature.

For a gas turbine cycle, the mechanical work extraction begins at the temperature obtained after fuel combustion. This temperature is significantly higher than that of steam in a steam turbine. Furthermore, the bottom temperature of the cycle is at a temperature much higher than the ambient environmental temperature. In a combined-cycle power plant the steam cycle is driven off of this exhaust heat emanating from the bottom of the gas turbine cycle. Table A-1 gives a comparison between the three cycles [1].

Table A-1: Comparison between GT, CSC and CCGT cycles

Parameter	GT	CSC		CCPP
		without Reheat	with Reheat	
Average high temperature, (in Kelvin)	950 - 1000	550 - 630	640 – 700	950 - 1000
Average low temperature, (in Kelvin)	500 – 550	320 - 350	320 – 350	320 - 350
Carnot efficiency, %	42 – 47	37 - 50	45 – 54	63 - 68

The combined-cycle works between the medium high temperature of the GT and the medium low temperature of the CSC. The result is a high Carnot efficiency, compared to the simple-cycles. In practice, the real thermal efficiencies achieved are not as high as that shown in Table A-1, due to the exergy losses, which appear at the connection between the cycles.

A.2 Combined-cycle classification

Depending on the primary energy input mode, and on the thermodynamic connection between the gas cycle and the steam cycle, there are three main types of combined-cycles (see Figure A-2) [2]:

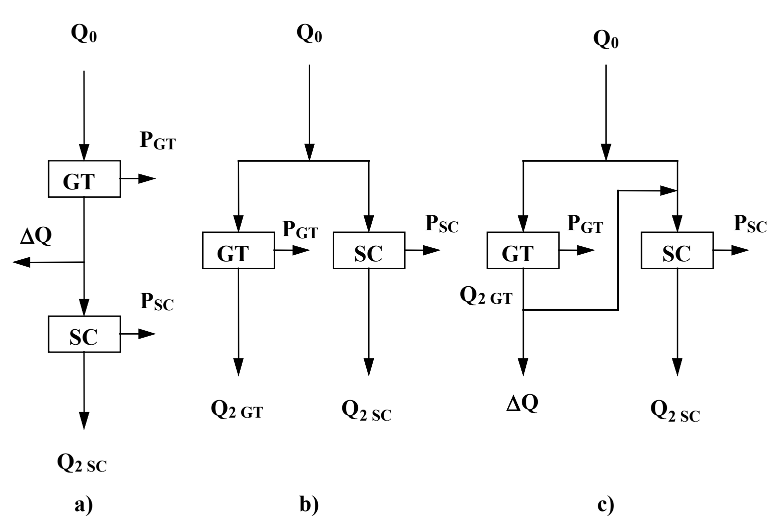


Figure A-2: Possible thermodynamic configurations of a combined-cycle plant (a-serial; b-parallel; c-serial-parallel). Q_0 – primary energy; ΔQ – heat losses at the connection between the cycles; P – power output; Q_2 – thermal power rejected at the cold source.

“Serial” Cycles

The primary energy is introduced only in the GT cycle, the steam cycle being only a recovery one. The heat from the fuel combusted in the GT cycle passes through both steps of the thermodynamic chain. The thermal efficiency of this type of cycle is the highest.

“Parallel” Cycles

The primary energy is simultaneously introduced in the gas cycle and the steam cycle. From a thermodynamic point of view, there is no real combined-cycle, the connection being only a strict technological one. Both the steam and the gas units work independently.

“Serial-Parallel” Cycles

In this situation a share of the primary energy passes through the whole thermodynamic chain while the rest is directly introduced in the steam cycle. The efficiency of this process is lower then the efficiency of a “serial” type.

The combined gas and steam cycle without supplementary firing

This is the variant providing the perfect overlap between the gas cycle and the steam cycle, where the fuel is injected only in the gas cycle. This is the typical configuration for modern combined-cycle power plants and is an example of a “serial” combined-cycle in which the gas cycle cold source represents the heat source for the steam cycle. The bottoming cycle is strictly dependent on the upper one (GT). By using the new generation of GTs, with steam-cooled components, a modern combined-cycle power plant is expected to reach efficiencies of up to 60 % [3].

The combined-cycle with supplementary firing

These types of plants, typically used in cogeneration applications, are an example of the serial-parallel type of cycle. A share of the primary energy is injected into the gas cycle, and another one is injected directly in the steam cycle. The supplementary firing in the steam cycle uses the excess air from the flue gases exhausted from the GT unit. In this case there is a heat quantity (obtained from the supplementary firing) that does not pass through the whole thermodynamic chain. In some conditions the steam cycle can function independently from the gas cycle, using a fan providing the necessary air for combustion. The combined gas and steam cycle with supplementary firing is usually applied for cogeneration. In these applications, the supplementary firing can be used for covering the heat demand peaks. Otherwise, the plant operates similar to a CCPP without supplementary firing.

The combined-cycle fully fired

As in the previous case, the high amounts of excess air in the flue gases exhausted from the GT is used for the combustion of a supplementary fuel.

Historically, when the rated power of early GTs and their efficiencies were limited, fully-fired combined-cycle represented the only solution to obtain a high efficiency combined-cycle. The advancements in gas turbine technology allowing the realization of high combustion temperatures and thus higher efficiency units has allowed the realization of the modern “serial” cycle combined-cycle power plants with higher efficiencies. At present, fully-fired combined-cycle is an attractive option only for repowering of existing conventional power plants.

Pressurized Fluidized Bed Combustion (PFBC)

In comparison to oil and natural gas, coal presents some advantages, like:

- The coal reserves are abundant. At the present annual production level, these can be sufficient for a period over 150 years, exceeding from this point of view the oil and natural gas reserves [4].

- The coal reserves are widespread evenly throughout the world, being exploited in over 100 countries.

- The coal price is relatively steady, not being influenced by political reasons.

For these reasons, the development of high efficiency technologies that use coal for energy generation can be attractive. One of these modern technologies is Pressurized Fluidized Bed Combustion.

Combustion takes place at high-pressure levels (generally over 10 bar) while the exhausted flue gas serves to run a gas turbine, after being first cleaned. Such a combination generates a type of combined-cycle, which is nowadays making the passage from demonstration to commercial stage.

Power plants equipped with PFBC have efficiency comparable to that of advanced conventional steam power plants. Moreover, PFBC shows real benefits concerning environmental impact.

Integrated Gasification Combined-cycle (IGCC)

The Integrated Gasification Combined-cycle represents another possibility for a high efficiency use of coal for energy generation. A brief description of this technology is given in section 2.5.1 of this document.

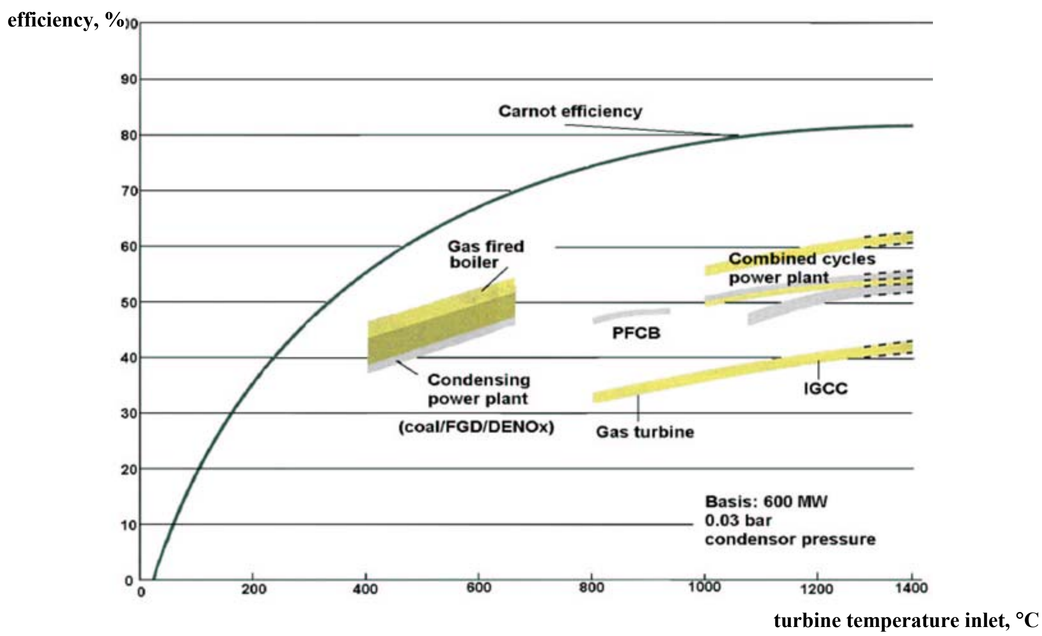


Figure A-3: Efficiencies actually achieved by different power generation technologies as a function of the turbine inlet temperature. FGD – flue gas desulphurisation; DENOx – nitrogen oxide reduction. (Source: Integrated Pollution Prevention and Control. Draft Reference Document on Best Available Techniques for Large Combustion Plants, European Commission, March 2001)

References:

[1] R. Kehlhofer, *Combined-cycle gas and steam turbine power plants*, Fairmont press, Lilburn, USA, 1991

[2] J.H. Horlock, *Combined power including combined cycle gas turbine (CCGT) plants*, Pergamon Press, Oxford, 1992

[3] S. Green, “Baglan Bay: an H showcase”, review Power Engineering International, September 1999

[4] *Green Paper – Towards a European Strategy for the Security of Energy Supply*, November 29, 2000, ISBN 92-894-0319-5.

APPENDIX B

CURRENT MODELING PRACTICE BY UTILITIES AND SYSTEM PLANNERS IN AUSTRALIA

B.1 Account of Modeling Practice in Australia

As a rule, the same gas turbine models have been used for simple and combined-cycle applications. In many cases this also applies to steam turbines, mainly because of lack of information on different characteristics of steam turbines when they operate in combined-cycle.

Manufacturers generally supply the models. Models are sometimes derived from tests and by relying on the knowledge of internal design and physical details of control systems, but this is rarely done due to the high costs involved. More often, the existing models in use are calibrated from test results.

In many cases simplification of the models originally supplied by manufacturers or those derived from tests has been necessary due to limited capability of the simulation software to model control systems. This has typically resulted in representing only the principal feedback loop, which is the turbine speed control loop. The study engineer must then represent the frequency control mode of the unit, DROOP or ISOCH. Auxiliary control and protection loops, such as the temperature control loop of gas turbines, are therefore not represented. Occasional attempts have been made to model the action of acceleration limiters of heavy-duty gas turbines with varying degrees of success.

When using these simple models, judicious adjustment of the maximum power output has been used to accommodate the combined action of the temperature control loop and of the selected mode of operation of the gas turbine (base load or peaking unit).

The use of these simple models requires sound engineering judgment to determine which control loop is active in the simulated timeframe, that is, which loop controls the response of the turbine, and to represent that specific loop in the turbine governor model for the simulation. This also depends on the type of phenomena considered.

Very complex models available in modern commercial software may be of no practical value unless a typical set of input data is provided that can be used when no specific plant information is available.

In conclusion, the adequacy of computer models should be assessed in the broader context of the simulations in which they are used. Their accuracy should be proportional to the relative importance of the unit to the power system in which it operates.

B.2 Current Models in Australia

Most popular commercial power system analysis packages offer limited but standardized libraries of gas turbine models, typically for single-shaft gas turbines. Models for multi-shaft gas turbines are less common and are often implemented as ‘user defined models’ (UDM).

Western Power's Practice

Western Power uses PSS/E software, and PSS/E GAST and GAST2A models for single-shaft gas turbine and has used the software vendor's UDM model for the LM6000 two-shaft gas turbine generators.

Until recently, Western Power has not specifically represented the sliding pressure mode. This practice has changed with the first major combined-cycle power plant in Western Australia. Two options have been considered for a new 230 MW combined-cycle power plant: single-shaft CCPP and two-shaft CCPP. Both options assume the use of a heavy-duty gas turbine and a three-stage steam turbine. Their models are described below.

Single-shaft CCPP

For a large modern single-shaft combined-cycle power plant the following UDM model was developed and implemented in PSS/E in Western Power. The standard library model for a heavy-duty gas turbine, GAST2A, was modified in a manner that the governor controls only two third of the total shaft mechanical power which corresponds to the full output of the gas turbine. The remaining one third of the shaft mechanical power, supplied by the steam turbine operating in the sliding pressure mode, is kept constant throughout the transient stability simulations.

Two-shaft CCPP

For the two-shaft combined-cycle power plant option, Western Power has developed and implemented the following model. One standard library model, GAST2A, is used for the heavy-duty gas turbine generator. The steam generator is modeled without a governor, for consistency with the single-shaft CCPP model described here.

For future two-shaft CCPP projects we may use a less conservative model for the steam set by allowing a limited governor response. A few alternatives have been considered. First, we could use one standard steam set model, say IEESGO, with a large droop, in order to 'weaken' the governor response. This is a practical alternative since the IEESGO model cannot represent the deadband. Another alternative would be to use a governor model that can represent the deadband. Finally, a sophisticated governor UDM model for the steam turbine could be created which would activate the governor control only if the frequency exceeds an upper limit, of say 50.5 Hz. That sophistication would enable modeling of the effect of closing of the turbine valves under the rising system frequency conditions and/or fast valving.

APPENDIX C

ALSTOM SIMPLIFIED DYNAMIC MODELS FOR COMBINED-CYCLE POWER PLANTS

C.1 Alstom Power Simplified Dynamic Models for Gas Turbines

The Alstom simplified dynamic models of gas turbine-governor systems have been developed for application in power system dynamic studies. These models are valid for the current Alstom fleet of heavy-duty gas turbines, with power outputs ranging from 57MW (GT8C2) to 265MW (GT26) and are valid for operating the gas turbines in parallel with a grid.

As described in Chapter 2, actual control systems of gas turbines are highly complex and include many functions that are not of interest in power system dynamic studies such as start-up, process protection etc. Therefore, the philosophy behind the Alstom set of simplified models is to provide a phenomenon representation capable of accurately capturing the dynamic behavior of the unit, rather than a detailed physical representation. Dynamic behavior in this regard principally refers to the response of the gas turbine-governor to system frequency excursions.

While connected to the grid, typically, the main gas turbine controllers that will be in operation are the frequency controller, the temperature controller and variable guide vane controller. The dynamic behavior of these controllers is incorporated in the Alstom's phenomenon model representation of the gas turbine-governor. The base power output from a gas turbine also depends on the fuel mix, humidity, backpressure of the turbine exhaust, the speed of the gas turbine and temperature. The last two factors are incorporated in the gas turbine-governor model via correction curves. An example of a correction curve is shown in Figure C-1. The shape of the correction curve is dependent on the GT type and ambient conditions.

The configuration of the gas turbine-governor comprises, in general, the following modules as shown in Figure C-2:

- frequency control (static and dynamic deadbands)
- power limitations
- power distribution
- gas turbine dynamics

Frequency Control

The frequency control module (more commonly referred to as speed-governor) provides system frequency support and has the same common structure for the entire Alstom GT fleet. This module consists of a prefilter, a deadband and a gain (K_D). The deadband can be either a standard static deadband (Figure C-3) or an enhanced dynamic deadband (Figure C-4). Any frequency excursions outside the specified deadband would generate a frequency error signal (Δf). The gain (the reciprocal of which is more typically known as droop) converts this resulting frequency error signal into a power demand signal (ΔP_c) based on a droop characteristic. In general the power output from the GT will increase or decrease according to ΔP_c. The power demand signal is feed into the power limitations module shown in Figure C-5.

Power Limitations

The power limitations module, shown in Figure C-5, basically models the restrictions place on the GT power response based on physical constraints of the combustion technology. Frequency response of the GT is limited to a power output range between a maximum (FRMX) and a minimum (FRMN) load.

The power demand ΔP_c is limited to a particular rate by the load rate limiter function, and is dependent on the fuel used and the operating point in question. The rate limiter prevents inadvertent burner pulsation, lean combustion flame out or excessive over-firing by controlling the unloading and loading rate of the GT.

The grid frequency has a major impact on plant behavior, as it determines the generator speed and therefore the gas turbine speed. The gas turbine compressor's speed defines the airflow entering the gas turbine, which is significant for plant performance. This characteristic of the GT is incorporated in the turbine-governor control model via a set of correction curves that provide the base load as a function of frequency and ambient temperature. For the Alstom GT type GT11N2 and GT13E2, this function is implemented in this module whilst for the GT type GT8C2, GT24 and GT26 this is implemented in the power distribution module. The output from this module is the relative commanded load change, represented by the parameter CPCL in Figure C-5.

Power Distribution

Physically, the relative commanded load change signal is used to manipulate the fuel mass flow and the airflow. The airflow depends on the GT shaft speed and the position of the VIGV. The resulting fuel to air ratio determines the quality of the GT combustion process in terms of efficiency and emission levels. For a given airflow, there is an upper limit of fuel to air ratio that is determined by a maximum allowable GT temperature, either the turbine inlet or the turbine outlet temperature.

This physical characteristic is represented in the control model by power contribution factors for the combustor units and compressor/VIGV unit. These factors inherently represent also the allowable temperature limit for a particular airflow.

Figure C-5 shows the generic power distribution module applicable for the either Alstom GT types GT8C2, GT24 or GT26. The GT8C2, has no SEV combustors, therefore parameters related to this combustor should be set to zero.

For the above mentioned types of GT, the base load as a function of frequency and ambient temperature is taken into account in these modules.

Gas Turbine Dynamics

The gas turbine is represented by the dynamics of the combustor and the compressor/VIGV units in this module. The dynamics of the environmental (EV/SEV) combustors are represented by first order lag functions. The dynamics of the compressor/VIGV unit is represented by a second order transfer function of the form

$$\frac{\omega_0^2}{s^2 + 2\zeta\omega_0 s + \omega_0^2}$$

where ω_0 is the undamped natural frequency and ζ the damping ratio of the compressor/VIGV unit.

For the turbine type GT11N2 and GT13E2 a total accumulated delay is used to represent combustion reaction time, turbine and exhaust system transport delays, etc.

Power limiter function consists of an upper and lower limit. The output from the power limiter function is the mechanical power of the GT in per unit based on the GT rating. After suitable per unit conversion, this will be the mechanical power of the associated generator in power system studies.

The gas turbine dynamics modules are shown in Figures C-6 and C-7.

C.2 Alstom Simplified Dynamic Model of the Heat Recovery Steam Generator and Steam Turbine-Governor

HRSG and Steam Turbine-Governor Model Structure

A generic model, shown in Figure C-8, allows the frequently encountered HRSG and steam turbine-governor configuration used in the Alstom CCPP to be represented in a single structure, assigned to the steam turbine, for dynamic studies. The HRSG in common use today, have either two or three pressure levels. The heat recovery boiler subsystem is modeled by steam generation and steam storage functions. Steam from the boiler subsystem then feeds the steam turbine. The steam turbine can be modeled with up to 3 pressure sections namely high pressure (HP), intermediate pressure (IP) and low pressure (LP) sections as well as of the reheat or non-reheat types.

The generic model provides the possibility of including turbine speed governor and pressure control systems. A static frequency deadband as well as a droop factor is part of the speed control function. The net effect of this control is to adjust the position of the control valves to control the admission of steam to the steam turbine sections.

When the CCPP is operated at high loading levels and efficiency, the steam turbine is normally in sliding pressure mode with the control valves fully opened. The turbine speed governor and pressure control (valves) systems can therefore be neglected for most dynamic studies.

To increase the efficiency of the CCPP, manufacturers often have optimized water-steam cycle configuration to suit project specific and ambient conditions. This results in different HRSG and steam turbine configurations even for GT of the same type. This must be accounted for when modeling a CCPP for dynamic studies. The majority of the components in this model have similar functionality as those of conventional steam power plants, details of which can be found in [1].

C.3 Combined-Cycle Power Plant

As described in Chapter 2, there are two categories of plant (i) single-shaft, and (ii) multi-shaft. In both these configurations, the flue (exhaust) gas from the gas turbine(s) is utilized to heat the HRSG boiler. Since the output from the gas turbine model is the GT mechanical power a conversion factor is needed to obtain the thermal power from the flue gas. This conversion can be approximated as follows:

$$FGEF \cong cg2s * (PTGT-1) + 1$$

where:

FGEF is in per unit and is the thermal power input to the HRSG boiler
cg2s is the GT power to flue gas energy flow factor.
PTGT is the total mechanical power output in per unit from the contributing gas turbines.

Figure C-9 shows the implementation of the equation as a set of control block functions.

Single-Shaft Combined-Cycle Power Plant Model

The total mechanical power output from a single-shaft CCPP is the sum of the power contributions from both the gas and steam turbines. In general, if all the steam generated is used for driving the steam turbine, and if the power plant is operated at high loading levels, then the gas turbine will contribute to approximately two-thirds of the CCPP output and the remaining one-third is from the steam turbine. Therefore, the total steady-state power output of the single-shaft CCPP can be expressed as:

$$PTOT_{SS} \cong PTGT*CGT + PTST*CST$$

where:

$PTOT_{SS}$ is the mechanical power output of the single-shaft CCPP in per unit.
PTGT is the total mechanical power output of the GT in per unit.
PTST is the total mechanical power output of the ST in per unit
CGT is the contribution factor of the GT to total power output of the CCPP.
CST is the contribution factor of the ST to total power output of the CCPP.

For single-shaft CCPP the subsystem connecting the GT to the ST input is shown in Figure C-9.

Multiple-Shaft Combined-Cycle Power Plant Model

In a multi-shaft CCPP each generator is driven by its own turbine. For multi-shaft CCPPs where there is no steam extraction requirement for external processes, the exhaust gas from all the gas turbines in operation will be used to supply the thermal energy to the HRSG for steam generation. It is common practice, when all the gas turbines are in operation, that each turbine contributes an equal amount of thermal energy to the HRSG. Considering a multi-shaft CCPP with 3 gas turbines and 1 steam turbine, the total thermal power from the 3 GTs to the HRSG can be expressed as:

$$FGEF_{MM} \cong cg2s * (PTGT_{MM} - 1) + 1$$

where:

$FGEF_{MM}$ is the total thermal power in per unit as defined previously
$PTGT_{MM}$ is sum of the power output in per unit contributed by each individual gas turbine to the total thermal energy expressed as:

$$PTGT_{MM} = k_{GT1}*PGT_1 + k_{GT2}*PGT_2 + k_{GT3}*PGT_3$$

PGT_i is the power output of each individual gas turbine in per unit.
k_{GTi} is the contribution factor of each individual gas turbine to the total thermal energy. The sum of all k_{GTi} must equal unity.

The steady-state power output from the steam turbine in per unit is:

$$PTST_{MM} = FGEF_{MM}$$

For multi-shaft CCPP the subsystem connecting the gas turbine to the steam turbine input is shown in Figure C-10.

Typical Data for Modeling Combined-Cycle Gas Turbine Power Plants

Tables C-1 and C-2 list typical data for the different Alstom gas and steam turbines. For simulating an actual combined-cycle power plant it is necessary and important to obtain project specific data from the manufacturer in order to achieve accurate modeling and results.

References:

[1] IEEE Working Group Report, "Hydraulic Turbine and Turbine Control Models for System Dynamic Studies," *IEEE Trans*, Vol. PWRS-7, No.1, pp. 167-179, February 1992.

Table C-1: List of typical gas turbine parameters.

Parameter	Description	GT8C2	GT11N2	GT13E2	GT24	GT26	Unit
T_{f1}	Pre-filter time constant	0.5	0.5	0.5	0.5	0.5	s
T_{ZETA}	Trend filter time constant	30.0	30.0	30.0	30.0	30.0	s
F_{REF}	Reference frequency	50/60	50/60	50	60	50	Hz
F_{STP}	Speed set-point	1.0	1.0	1.0	1.0	1.0	p.u.
L_{FS}	Frequency sensitive mode	1	1	1	1	1	-
U_{PPDB}	Upper static deadband[1]	0.015	0.015	0.015	0.018	0.015	Hz
L_{OWDB}	Lower static deadband[1]	-0.015	-0.015	-0.015	-0.018	-0.015	Hz
D_{YNDB}	Relay upper dynamic deadband[1]	0.015	0.015	0.015	0.018	0.015	Hz
D_{YNRC}	Relay lower dynamic deadband[1]	0.001	0.001	0.001	0.001	0.001	Hz
S_{TDB}	Relay upper static deadband[1]	0.4	0.4	0.4	0.48	0.4	Hz
S_{TDBRC}	Relay lower static deadband[1]	0.015	0.015	0.015	0.018	0.015	Hz
DROOP	Governor droop D[2]	0.02-0.1	0.02-0.1	0.02-0.1	0.02-0.1	0.02-0.1	p.u.
T_{f2}	Post filter time constant	1.0	1.0	1.0	1.0	1.0	s
P_{GTM}	Max. gas turbine power output (rating)[3]	57	113	165	183	265	MW
P_{STPG}	Initial load setpoint[4]	Variable	Variable	Variable	Variable	Variable	p.u.
F_{RMX}	Max. power level for frequency response	1.0	1.0	1.0	1.0	1.0	p.u.
F_{RMN}	Min. power level for frequency response	0.5	0.7	0.6	0.4	0.4	p.u.
L_{DSPMX}	Max. load set-point	1.0	1.0	1.0	1.0	1.0	p.u.
L_{DSPMN}	Min. load set-point	0.5	0.7	0.6	0.4	0.4	p.u.
L_{DRT}	Rate limiter	0.024	0.009	0.024	0.014	0.014	p.u./s
C_{EV}	EV capacity	0.3	-	-	0.15	0.15	s
C_{SEV}	SEV capacity	0.0	-	-	0.25	0.25	s
C_{VGV}	VGV capacity	0.7	-	-	0.6	0.6	s
T_{EV}	EV dynamics time constant	5.0	-	-	5.0	5.0	s
T_{SEV}	SEV dynamics time constant	0.0	-	-	5.0	5.0	s
T_1	Set-point-to-Fuel dynamics time constant	-	0.1	0.1	-	-	s
T_T	Accumulated delays	-	0.32	0.32	-	-	s
Z_{ETA}	Relative damping coefficient	0.45	0.30	0.30	0.45	0.8	p.u.
W_0	Undamped natural frequency	0.22	1.05	1.05	0.22	0.22	rad/s
P_{TMX}	Max. thermal power output	1.1	1.2	1.1	1.1	1.1	p.u.
P_{TMN}	Min. thermal power output	0.0	0.0	0.0	0.0	0.0	p.u.

Notes:

[1]The deadband should be converted into per unit by dividing the value by the reference frequency FREF. For relay deadbands, the output of the relay is set to 1, if input is greater than the relay's upper deadband and is set to 0 if input is less than the relay's lower deadband.
[2]The power gain KD is the inverse of the governor droop D.
[3]The gas turbine rating is project specific. Per unit power is on this base.
[4]The initial load set-point is usually calculated from a load flow case.
[5]The base load as a function of frequency and ambient temperature correction curves can be obtained from the manufacturer based on the gas turbine designed operating temperature.
[6]SPD is the generator speed/grid frequency deviation.

Table C-2: List of typical HRSG and steam turbine parameters.

Parameter	Description	KA24 DPRH ICS	KA26 TPRH SSPT	KA13E2-3 TP OK	Unit
T_G	Speed governor time constant	-	0.04	0.1	S
F_D	Frequency deadband in load mode	-	0 to 10	0 to 10	±%
K_{GST}	Speed governor gain	-	12.5-33	15-30	p.u.
P_{STP}	Power set-point of steam turbine	-	1.0	1.0	p.u.
T_{SM1}	HP servo motor time constant	-	0.1	0.25	S
T_{SM2}	IP servo motor time constant	-	0.1	-	S
C_{OP1}	Maximum opening speed of HP valves	-	0.1	0.1	p.u./s
C_{CL1}	Maximum closing speed of HP valves	-	-5.0	-5.0	p.u./s
C_{OP2}	Maximum opening speed of IP valves	-	0.1	-	p.u./s
C_{CL2}	Maximum closing speed of IP valves	-	-3.7	-	p.u./s
f(m1)	IP steam command=f(HP steam command)	-	1.3*(m1-3.5%)	-	p.u.
T_{B1}	Boiler time constant	8-12	5.0	5.0	S
T_{B2}	Boiler time constant	8-12	30.0	30.0	S
T_{B3}	Boiler time constant	40.0	45.0	40.0	S
T_{HP}	HP turbine time constant	-	0.1	-	S
T_{IP}	IP turbine time constant	-	0.1	-	S
T_{LP}	LP turbine time constant or Cross over piping time constant	1.0	0.2	0.5	S
T_{RH}	Reheater time constant	5	9	-	S
F_{HP}	HP power contribution	0.21	0.21	0.43	p.u.
F_{IP}	IP power contribution	0.28	0.33	-	p.u.
F_{LP}	LP power contribution	0.51	0.46	0.57	p.u.
C_{GT}	Power contribution of GT to CCPP	0.65	0.65	-	p.u.
C_{ST}	Power contribution of ST to CCPP	0.35	0.35	-	p.u.
LIN HP/IP	Valve linearization HP/IP	-	See Note 1	See Note 1	
P_{STM}	Maximum steam turbine power output (rating)[2]	-	-	-	MW
k_{Gti}	Power contribution factor for GTi [3]	-	-	1/N	-
Cg2s	Conversion factor for gas turbine power to flue gas energy flow	0.6	0.6	0.6	p.u.

Notes:

[1]Valve linearization curves can be obtained from the manufacturer.
[2]The maximum steam turbine power output (rating) is used for per unit conversion. Per unit power is on this MW base.
[3]The sum of all power contribution factor should equal unity. $\sum k_{GTi} = 1.0$, for i = 1 to N where N is the number of GT in operation.
KA24 DPRH ICS – Single-shaft Double Pressure Reheat CCPP.
KA26 TPRH SSPT – Single-shaft Triple Pressure Reheat CCPP.
KA13E2-3 TP OK – Multi-shaft Triple Pressure Non-Reheat CCPP.

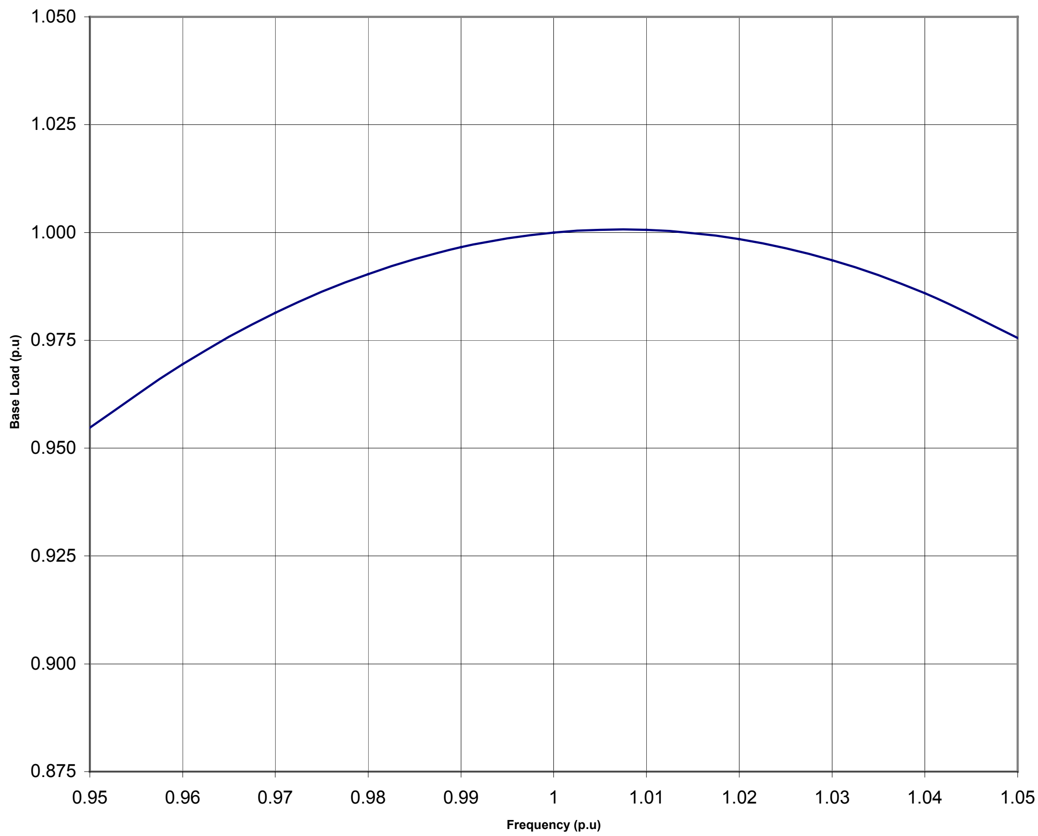


Figure C-1: An example of the GT base load output as a function of frequency correction curve for a particular type of Alstom heavy-duty turbine and ambient temperature.

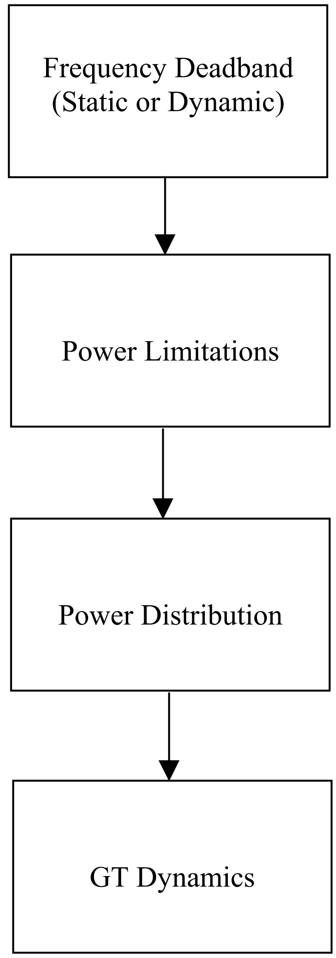


Figure C -2: Configuration of the Alstom gas turbine-governor system.

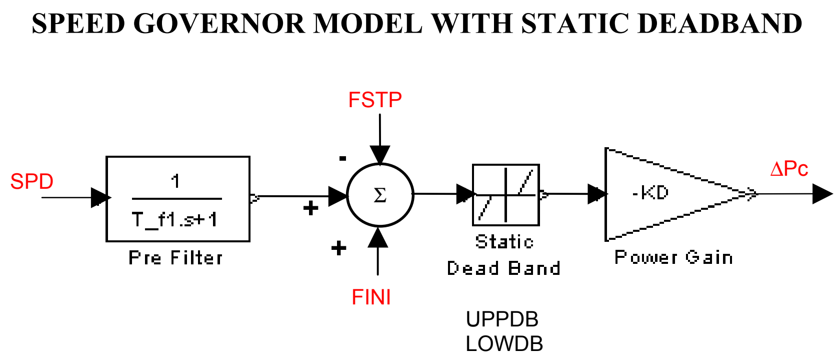
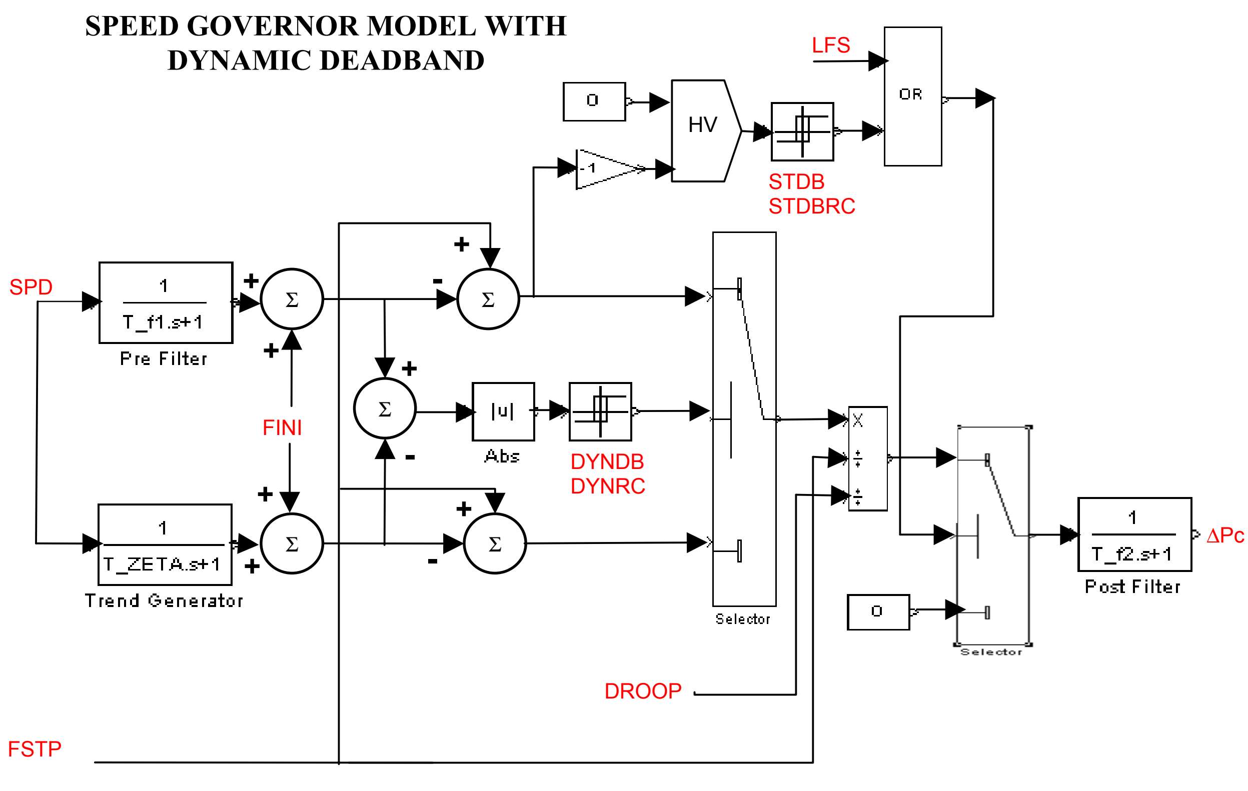


Figure C-3: GT speed governor model with static deadband valid for all Alstom GT.



FINI = Initial measured speed

Figure C-4: GT speed governor model with dynamic deadband valid for all Alstom GT.

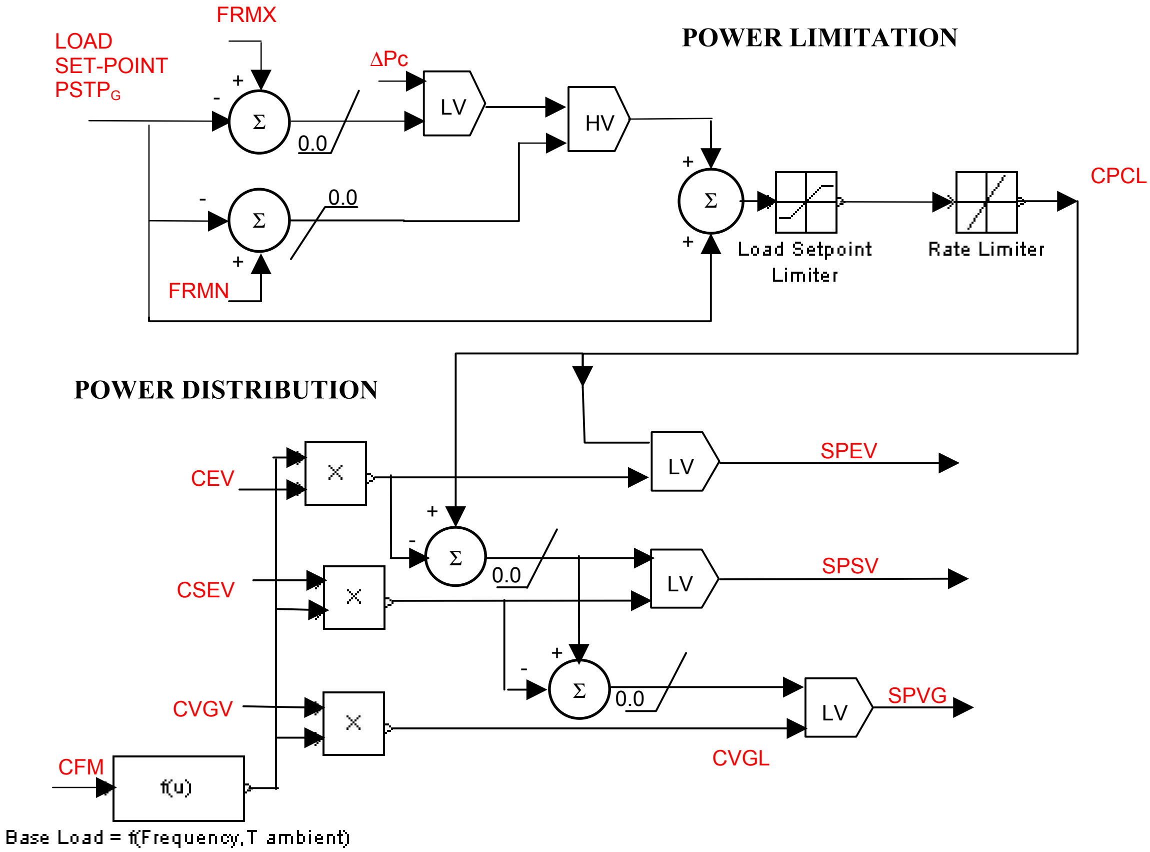
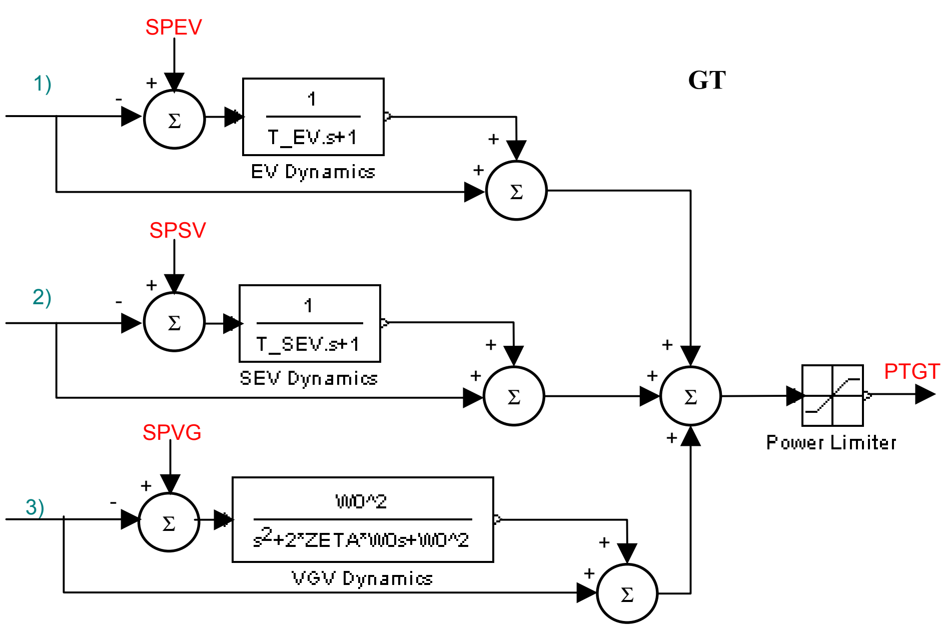


Figure C-5: Power limitations and distribution modules valid for Alstom GT type GT8C2, GT24 or GT26.



1) MIN(CEV, PSTP)
2) MAX(0, MIN((CSEV, PSTP-CEV))
3) MAX(0, MIN((CVGV, PSTP-CEV-CSEV))
Note: CSEV and T_SEV is 0.0 for GT8C2

Figure C-6: GT dynamics module valid for Alstom GT type GT8C2, GT24 or GT26.

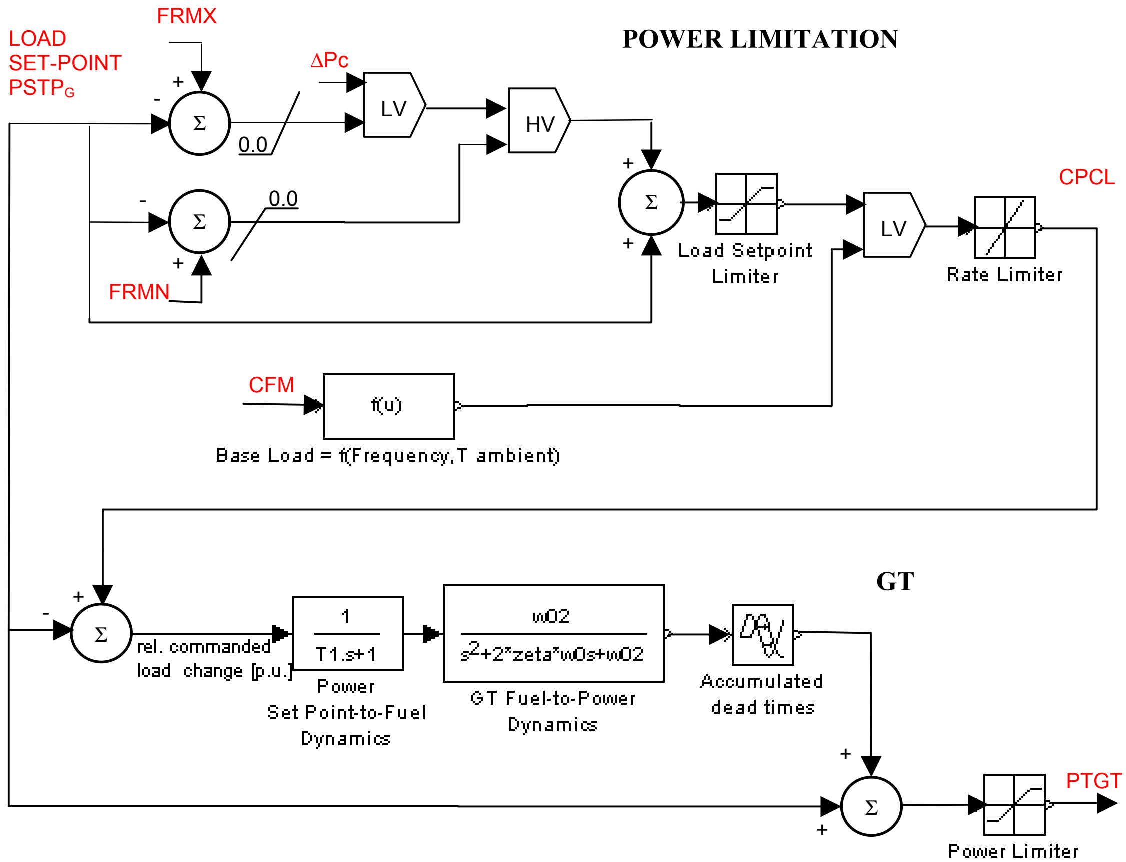


Figure C-7: Power limitations and GT dynamics modules valid for Alstom GT types GT11N2 or GT13E2.

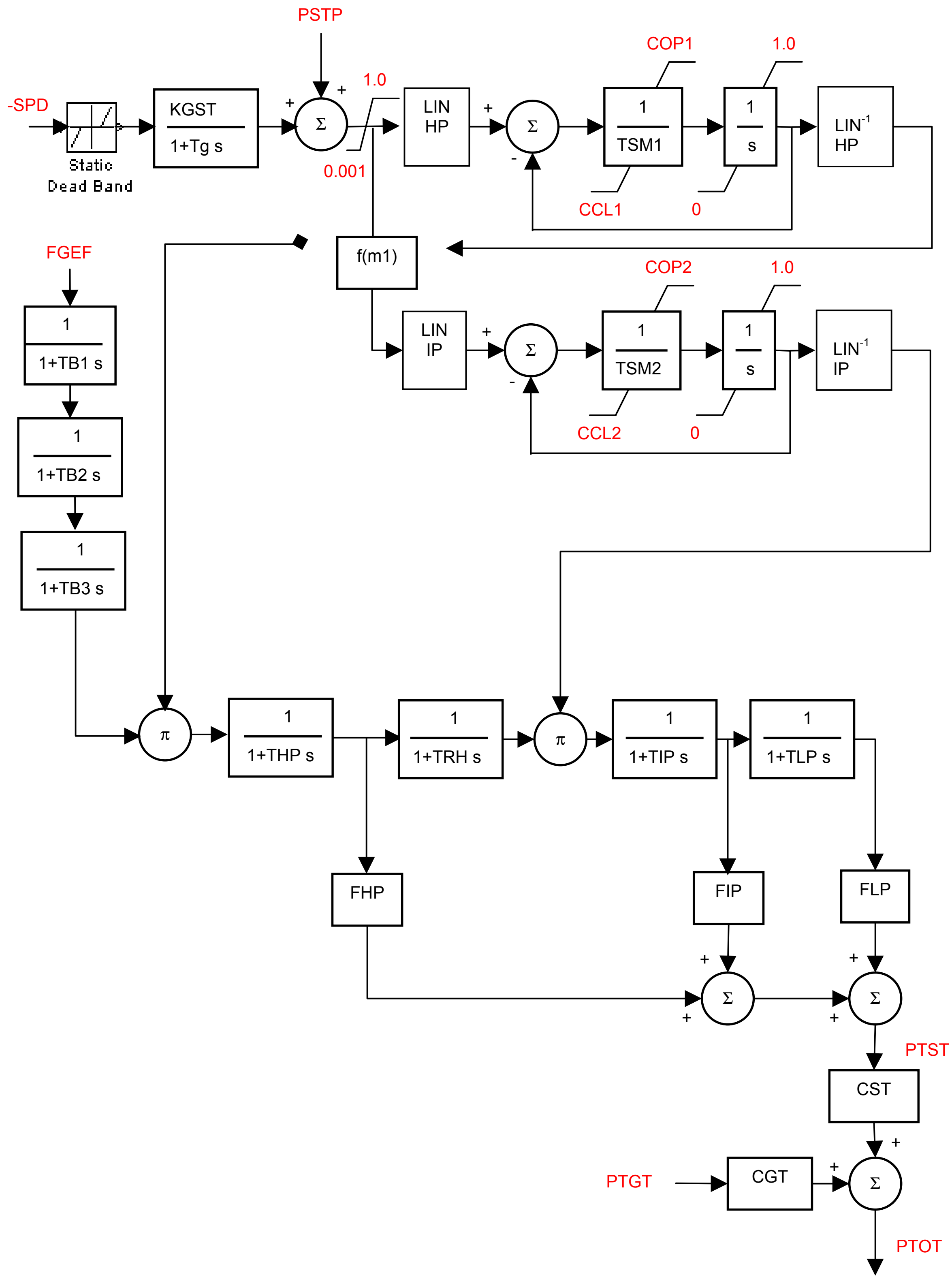


Figure C-8: Alstom generic HRSG and steam turbine-governor model.

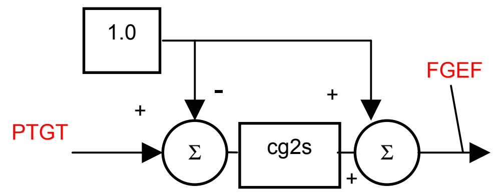


Figure C-9: Subsystem connecting the gas turbine to the steam turbine valid for the Alstom single-shaft CCPP.

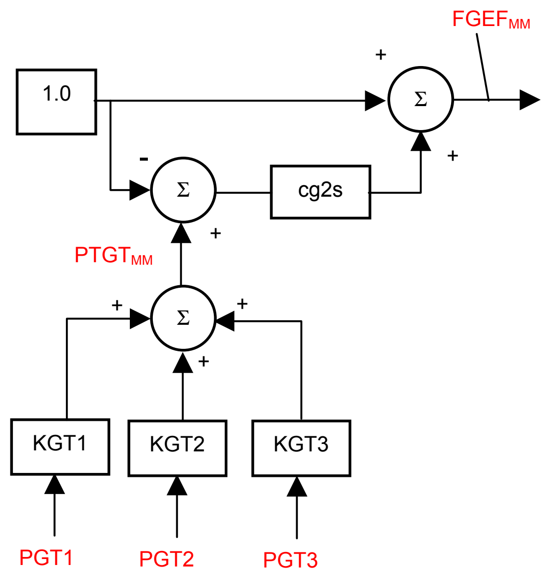


Figure C-10: Subsystem connecting the gas turbine to the steam turbine valid for Alstom multi-shaft CCPP with 3 gas turbines and 1 steam turbine.

APPENDIX D

SOLAR TURBINES SIMPLIFIED DYNAMIC MODEL FOR GAS TURBINES

D.1 Solar Turbines Simplified Gas Turbine Model

The model for Solar Turbines fleet of units is shown in Figure D-1. This model can be used to predict speed transient response to changes in load when the turbine generator:

1. is not operating in parallel with any other generators, or
2. is operating in parallel load sharing mode with other identical turbine generators.

By removing the rotor net power, rotor inertia and rotor speed blocks and changing the temperature set-point to 1.0, this model can be used to predict turbine generator power output response to changes in utility frequency when the turbine generator is connected to a utility.

D.1.1 Limitations of Model

The model is relatively accurate at speeds between 95% and 103.5%. At speeds above 103.5%, some Solar turbine generators have additional over speed controls that are not accounted for by the model. At speeds below 95%, the performance prediction is less accurate than between 95% and 103.5%. The model should not be used for speeds above 105% or below 90%.

For conventional combustion applications, this model accurately predicts response to step load transients up to 100% rated power. For Dry Low NOx applications, the predicted response to step loads over 10% rated power may be less accurate. For step loads over 10% rated power on Dry Low NOx applications, the more elaborate model in section D.2 should be used.

The per unit base for the turbine and load power is the turbine rating for the given inlet air temperature and pressure (ambient temperature and pressure).

Import / Export control and kW load control options are not included in the model.

D.1.2 Protection Systems Pertinent to Power Systems Simulations

1. The turbine generator shuts down due to under frequency at 90% speed.
2. The turbine generator shuts down due to over speed shutdown at 108% speed.
3. The turbine generator shuts down due to over temperature shutdown if it exceeds rated temperature (1.0 pu) for more than 20 seconds when operating in island mode.
4. The turbine generator shuts down due to reverse power if the generator load is less than –5% for more than 3 seconds.

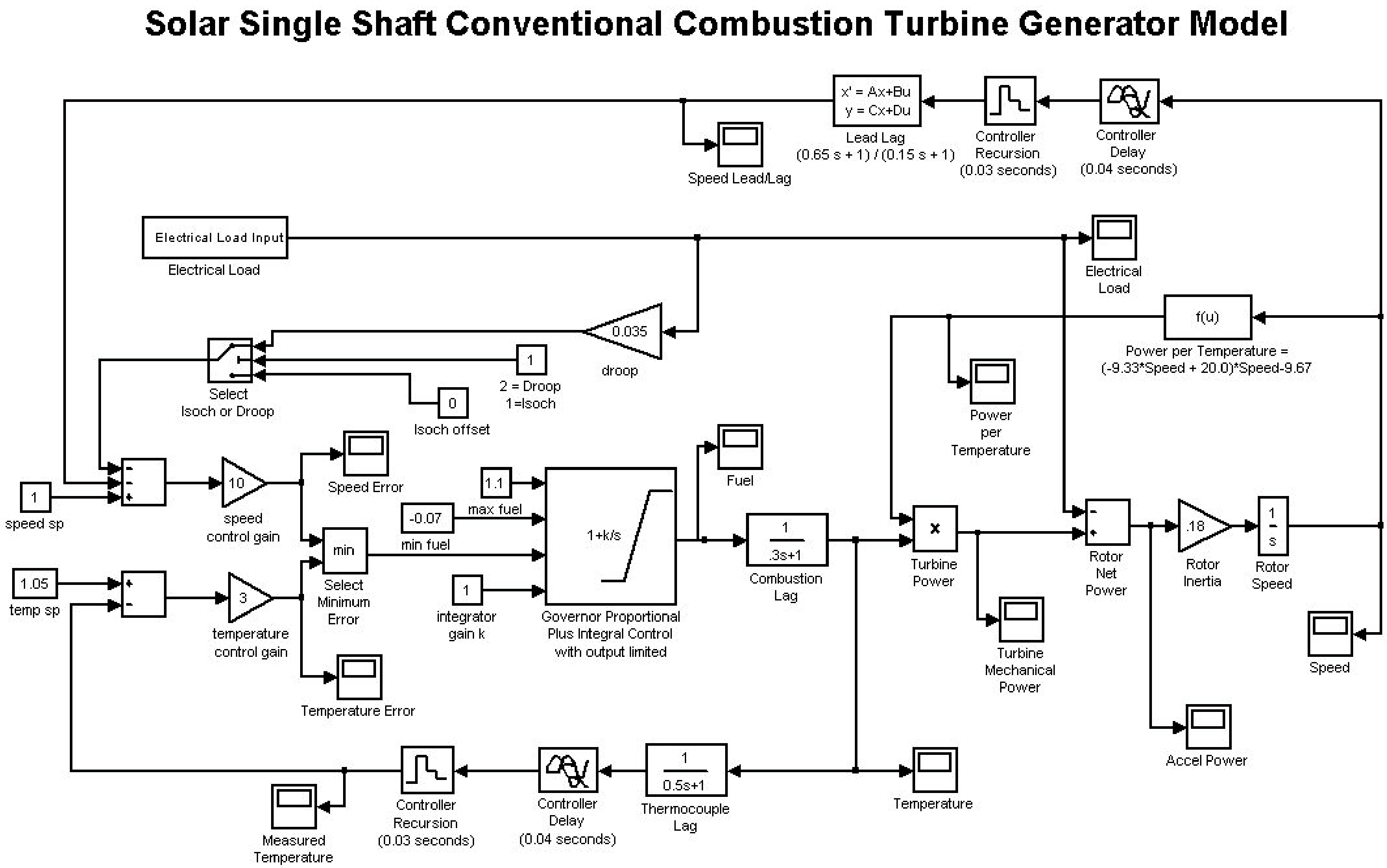


Figure D-1: Solar Turbine's gas turbine model block diagram.

D.2 Solar Turbines Single-shaft Dry Low NOx Gas Turbine Generator Set Model

The model for Solar Turbines Single-shaft Dry Low NOx gas turbine is shown in Figure D-2. This model can be used to predict speed transient response to changes in load when the turbine generator:

1. is not operating in parallel with any other generators, or
2. is operating in parallel load sharing mode with other identical turbine generators.

By removing the rotor net power, rotor inertia and rotor speed blocks and changing the temperature set-point to 1.0, this model can be used to predict turbine generator power output response to changes in utility frequency when the turbine generator is connected to a utility.

D.2.1 Limitations of Model

The model is relatively accurate at speeds between 95% and 103.5%. At speeds above 103.5%, some Solar turbine generators have additional over speed controls that are not accounted for by the model. At speeds below 95%, the performance prediction is less accurate than between 95% and 103.5%. The model should not be used for speeds above 105% or below 90%.

For SoLoNOx applications, this model accurately predicts response to step load transients up to 100% rated power. For conventional combustion applications, the simplified model in section D.1 should be used.

The per unit base for the turbine and load power is the turbine rating for the given inlet air temperature and pressure (ambient temperature and pressure).

Import / Export control and kW load control options are not included in the model.

The proportional plus integral governor must not be allowed to windup when the output is saturated.

D.2.2 Protection Systems Pertinent to Power Systems Simulations

1. The turbine generator shuts down due to under frequency at 90% speed.
2. The turbine generator shuts down due to over speed shutdown at 108% speed.
3. The turbine generator shuts down due to over temperature shutdown if it exceeds rated temperature (1.0 pu) for more than 20 seconds when operating in island mode.
4. The turbine generator shuts down due to reverse power if the generator load is less than –5% for more than 3 seconds.

D.2.3 Notes on Modeling of the Air Flow Control

The modeling of the turbine airflow control is shown in the lower right hand section of the model. The turbine airflow set-point can come from either of two values depending upon whether or not the turbine is operating in low emissions mode. During low emission mode, the airflow is calculated from the Low

Emissions Temperature Set-point and the fuel flow. During conventional combustion mode (not low emissions), the combustion airflow is 1.0. There are two requirements to operate in low emissions mode. The first requirement is that the turbine must be operating above 50% load (0.5 per unit fuel flow). The second requirement is that the fuel flow must not have had a rapid increase in fuel flow within the previous 20 seconds (fuel flow rate of change less than 0.2 per unit per second fuel flow for 20 seconds). The time delay on pickup function block has the following characteristic:

The Timer Done output is on (true) when the Timer Enable input has been continuously on (true) for a time greater than the Timer Delay input value. Otherwise, the Timer Done output is off (false).

D.3 Rotor Inertia in Solar Turbine's Models

The rotor inertia shown in the model, 0.18, is an average value for Solar single-shaft generator sets. The following Table lists typical inertia constants by turbine model. The units of the inertia constant are speed / second / power where speed and power are scaled "per unit". A speed of 1.0 represents rated speed and a power of 1.0 represents turbine rated power. If the inertia constant is 0.18, the speed will change by 18% of rated speed each second when the rotor net power (acceleration power) is 100 % of turbine rated power.

Table D-1: Turbine/Generator inertia constant for Solar Turbine Generator Packages

Turbine Model	1800 rpm	1500 rpm
Centaur 40	0.18	0.19
Centaur 50	0.19	0.21
Taurus 60	0.19	0.22
Taurus 70	0.16	0.17
Titan 130	0.13	0.15

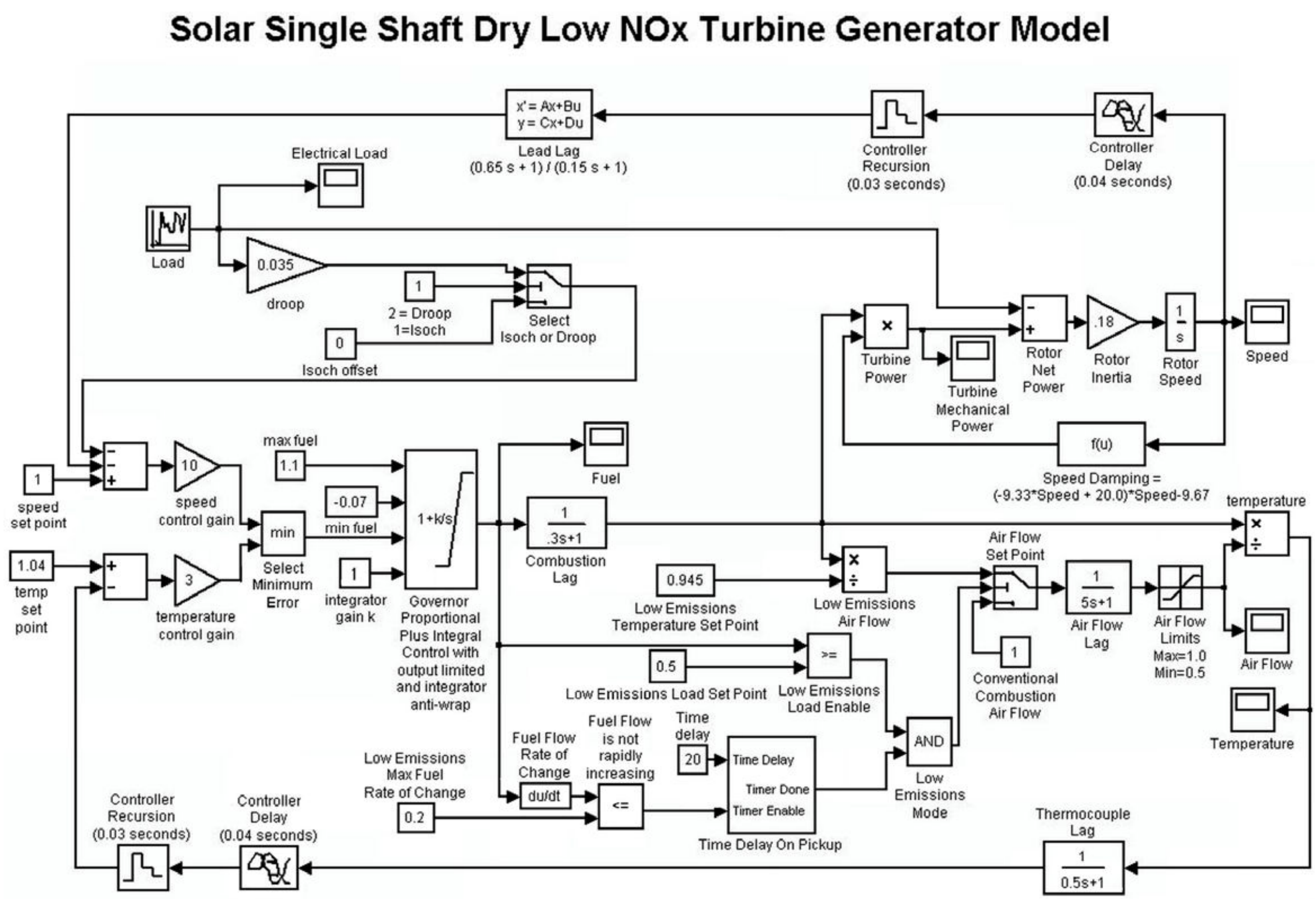


Figure D-2: Solar Turbine's gas turbine model block diagram for single-shaft, dry low NOx units.

APPENDIX E

SIMULATION AND TESTING OF THE DYNAMIC BEHAVIOR OF AN AERO-DERIVATIVE GAS TURBINE FOR ISLANDING STUDIES[1]

E.1 Introduction

Due to their high efficiency and low emission of pollutants, an increasing number of recent industrial cogeneration units are based on aero-derivative gas turbines in Belgium, France and Spain in the chemical and paper mill sectors.

This appendix underlines through an example of islanding scheme implementation, the need for deep understanding and appropriate modeling of aero-derivative gas turbines to achieve operational security and adjust the protective and primary control systems.

It is illustrated by a case study derived from a 90 MW Electrabel CHP plant of a large chemical plant in Belgium. This plant was commissioned by Tractebel Energy Engineering in June 2000 and consists of two General Electric LM6000PD dry low emission gas turbine gensets with associated HRSG's, and one back-pressure steam turbine genset.

E.2 Objectives and constraints of the islanding scheme

To take advantage of the high reliability of the Western Europe interconnected grid, most of the CHP plant gensets are operated in synchronous mode. As a consequence, the supply of the industrial plant loads is secured by two power source types, which are the public network and the CHP plant gensets. In the event that the plant is isolated from the interconnected grid, if the size of the cogeneration plant permits, it is highly desirable to operate the industrial installations in islanded mode.

In such operating conditions, the main objective of the islanded operation is to achieve an effective independence of the gensets operation to prevent the loss of the power supply of the essential load following sudden severe disturbance or collapses on the public grid.

To be feasible, a number of necessary conditions must be satisfied and verified by dedicated electrical stability studies. These conditions are briefly summarized:

1. Compatibility of the specific islanding protections with the industrial installations load-shedding systems and other protection systems such as the undervoltage contactor drop off for motor loads.
2. Consistency between the steam system adjustments and those of the electrical network following isolation from the public grid. Following an islanding, the gas turbines output level is often reduced. The associated exhaust gas flow reduction could influence the quality of the steam of the HRSG drums. To prevent a general CHP plant delayed tripping, it may be needed to automatically

[1] This appendix is based on the paper by K. Karoui and J-L. Vandesteene, "Simulation and Testing of the Dynamic Behavior of a 40 MW Aeroderivative Gas Turbine Genset in Islanding Situation", Powergen 2001 Europe conference, May 2001, Brussels.

compensate the power level reduction by a back-up boiler demand increase or a reduction of the steam consumption.

3. Checking that the cogeneration plant and the islanding protection meet the requirements imposed by the public grid operator.

E.3 Intrinsic performances of aero-derivative gas turbines

Aero-derivative engines feature rotors that are composed of several aerodynamically coupled shafts. The small diameter and the absence of mechanical coupling lead to a low inertia, mainly concentrated in the generator. This results in faster acceleration and deceleration and larger speed variations for a given power/load imbalance on the turbine shaft. The margin with respect to the under and over frequency generator relay thresholds are reduced accordingly.

This results also in lower transient stability margins (higher risk of loss of synchronism). This risk is increased by the lower performance of the grid protection plan, on account of the grid connection being made at comparatively lower voltage levels. These factors raise the likelihood of the isolation protections being activated if the system is equipped with such protections and the major role played by the cogeneration plant's control systems.

Modern controls are often entirely digital. The traditional speed, acceleration and exhaust gas temperature channels of the fuel control are augmented by the gas generator controller composed of a set of additional control loops having a higher priority and which permanently control the higher speed shafts and the flame temperature limits. The air-flow is constantly adjusted in order to ensure compressor stability.

Following a sudden demand change, the speed governor controller response often interacts with the gas generator controller that might counteract and lead to a lower dynamic response.

The General Electric LM6000PD engine

The two-spool CF6-80C2 aircraft engine is converted to the industrial aero-derivative engine LM6000PD through removal of the large fan from the low-pressure rotor. The external load and the synchronous generator are coupled to the cold end of the turbine.

For 50 Hz operation, the load comprises a synchronous generator with a reduction gearbox. Unlike the aircraft engine, the low-pressure rotor runs at a constant speed of 3600 rpm from no load to maximum power.

Its main actuator characteristics are variable inlet guide vanes at the inlet to the low-pressure compressor, variable stator vanes on the high-pressure compressor, a Dry Low Emission (DLE) combustor and a modulating bleed system.

The DLE combustor is composed of three concentric circles of burners equipped with separate and independent burners. The central ring is referred to as the pilot ring and is always on. Unlike the pilot ring, fuel to the burners in the inner and outer rings has to be turned "on" and "off" by means of staging valves. Sections of the combustor are turned "on" and "off" as functions of the delivered power. The staging sequence comprises several configurations of combustion. At no load, only the pilot ring and half of the inner ring are active, when, near full load the burners of the three rings are active.

These different combustor configurations cover only a limited operating power range. There are "gaps" between the configurations, i.e. power regions in which the DLE engine could not run. This is overcome by using compressor bleed to increase or decrease the air-flow and to permit an additional adjustment of the bulk flame temperature.

Focus on the LM6000PD fuel control

The fuel control refers to the part of the digital control system that determines the total fuel flow demand. It comprises a set of controllers and fuel flow limiters, which through a series of minimum-maximum selects and through priority selection logic, choose a single fuel flow demand.

The fuel controller adjusts the fuel flow to regulate an engine quantity like the power turbine speed, gas generator speed, etc.; whereas fuel flow limiters apply upper and lower fuel flow limits to the flow demand. Only one regulator or fuel flow limiter can be active at any time. The controlled or limited variables are:

1. Power turbine and gas generator speeds,
2. Gas generator acceleration or deceleration speed rates,
3. Maximum and minimum fuel flows,
4. Maximum turbine output temperature
5. Maximum flame temperature

The combustor staging logic controls the opening and closing of the outer rings staging valves. The inner and outer staging valves are opened and closed in accordance to the required combustor configuration. The transition between two successive configurations requires increasing or decreasing bleed in conjunction with opening and closing staging valves. Due to the finite response time of the airflow control and because of the small combustor flame temperature windows, it is not possible to switch immediately from one combustor permanent configuration to another permanent configuration and a series of intermediate (or partial) configurations are required. Therefore, the LM6000PD includes additional staging logic to optimize speed holding and load drop capability. This logic anticipates the need to change combustor configuration before the normal conditions are satisfied. As a result, the variations in power turbine speed can be minimized when large changes in load occur. As an example load drops can cause the staging control to switch from full load fuel flow to no load fuel flow mode almost instantaneously. On the contrary, following a rapid load increase, the stating logic imposes to go sequentially through the all combustion configurations between initial and final load demand levels.

E.4 Modeling requirements

The elements listed hereafter are critical and have to be known and modeled with accuracy, especially for the scenarios involving large disturbance leading to the activation of the on line islanding protection.

The public transmission grid and the base and back up times of the protections that are to eliminate short-circuits in the network.

The industrial facility's network and load: Particular attention will be given to induction motors. The great many of smaller asynchronous motors, which cannot reasonably all be modeled individually, are aggregated on the basis of sufficiently common characteristics including their driven loads. Particular significant loads like electrolysis cells are modeled with their specific protections.

CHP plant generator, excitation systems and their controls: The aim of modeling the generator and its exciter system is to obtain an adequate time evolution of the excitation voltage applied to the generator following a fault cleared in back-up.

Protective systems: The real behavior of systems under large disturbances is obviously greatly influenced by the operation of various protections and security systems. While the expected operation of selective protections is often included when defining the simulated scenarios (therefore without actual modeling of the relays), complex scenarios involving cascaded operation of protections may require substantial modeling efforts. Accordingly, the model usually includes protections against overloads, impedance, loss of synchronism, high or low voltage or frequency, etc. The generating unit's internal protections may also play a defining role. Without being explicitly modeled they need to be included in the interpretation of the simulated scenarios.

Gas turbine modeling requirement

Following islanding, the system's dynamic behavior and in particular the frequency essentially depend on the gas turbine's dynamic behavior.

Two modeling approaches are possible:

1. Using a simplified model and identifying it based on real tests or on the results of simulations run with a manufacturer-validated detailed model [1].

2. Integration into the power system simulation tool of a detailed model of the gas turbine, manufacturer validated.

For the Tractebel Energy Engineering LM6000PD cogeneration projects, mainly the second approach is adopted as the General Electric LM6000PD Engine and Control Simulation Program has been coupled with the EUROSTAG software for the simulation of power systems dynamics. The model was then validated by comparing its output with data from transient tests performed by the manufacturer on a load bank test bed (see Figures E-1 and E-2)

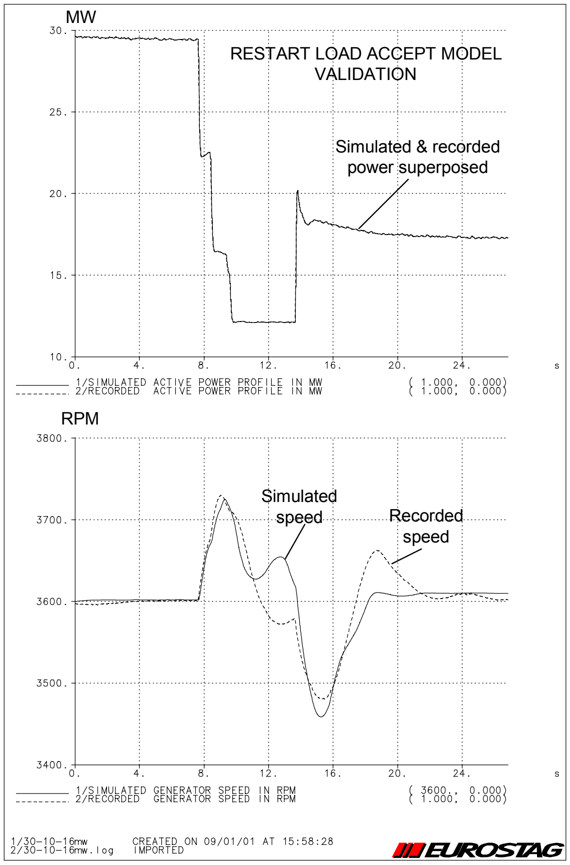


Figure E-1

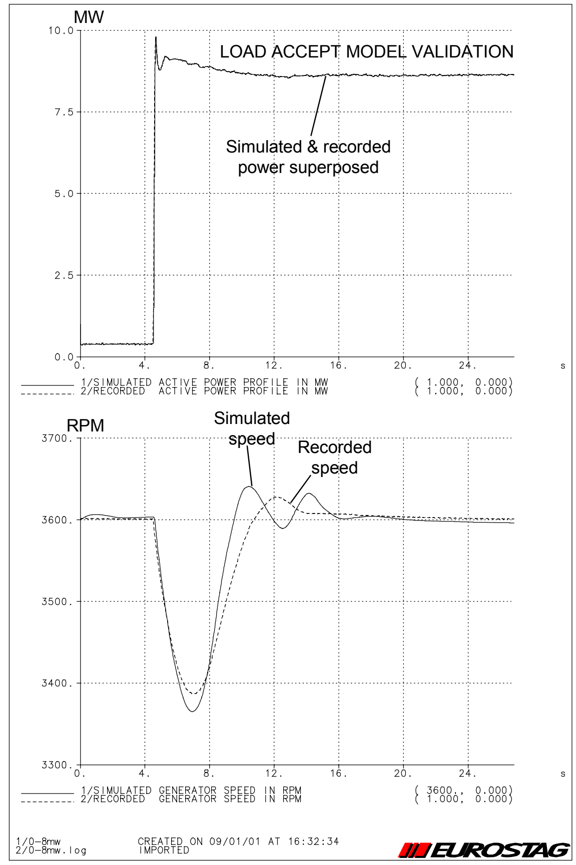


Figure E-2

E.5 System description

An Electrabel CHP plant of 90 MW was commissioned in June 2000 at a large chemical plant in Belgium. The CHP plant is composed of two LM6000PD Dry Low Emission gas turbines gensets with associated HRSG's, and one back-pressure steam turbine totaling approximately 125 MVA. The load of the plant consists of 80 MVA asynchronous motor and 75 MW electrolysis cells. In its normal configuration the system is connected to the public grid via two 150/70 kV transformers (see Figure E-3) [2], [3].

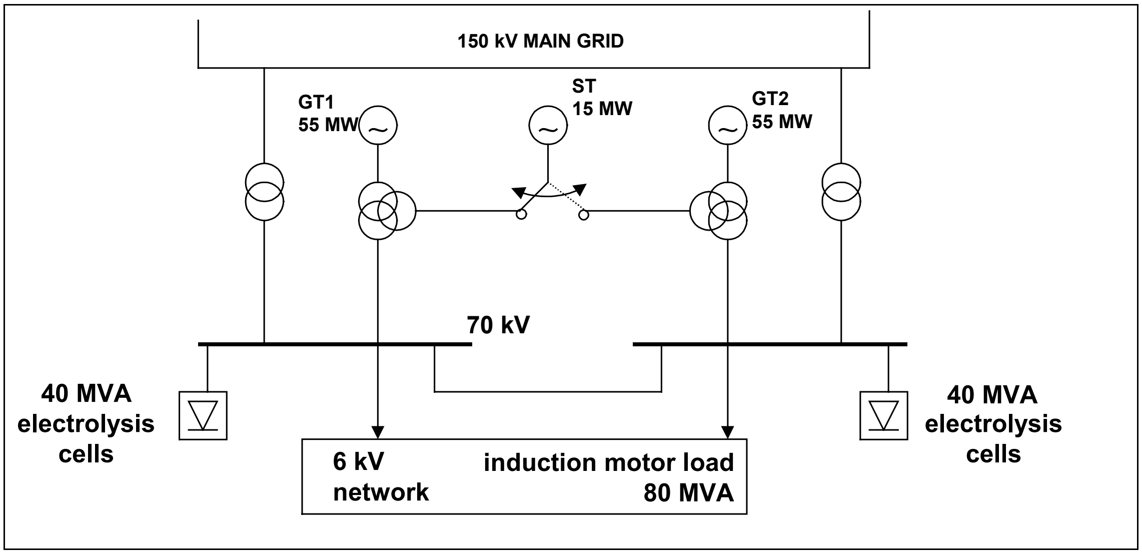


Figure E-3

E.6 Analysis of the islanding requirements

A number of asynchronous motors play a vital role in the functional continuity and security of the chemical processes. Any stops of these motors lasting more than a few seconds would result in a prolonged stoppage of the chemical plant's production processes. Hence the operator wishes to ensure ways to keep their vital loads serviced or to automatically restore supply in the event of operation of the protective systems.

Islanding criteria

The adopted islanding criteria are three fold:

- Islanding in case of frequency collapse in the main grid
- Islanding in case of voltage collapse in the main grid
- Islanding in case of long lasting external fault

Initial simulation findings showed that with standard controller settings following partial load rejection, initiated by a severe fault in the public grid, the frequency behavior displayed a large transient frequency dip. This was found to be incompatible with the islanding process since the frequency dip would result in the gensets tripping due to the action of their under-frequency relays. The performance of the turbines where improved thanks to an optimization by the manufacturer of the over-speed and the acceleration logic settings of the power turbine and gas generator shafts control as well as an increase of the rate of staging at the combustor level (see Figure E-4).

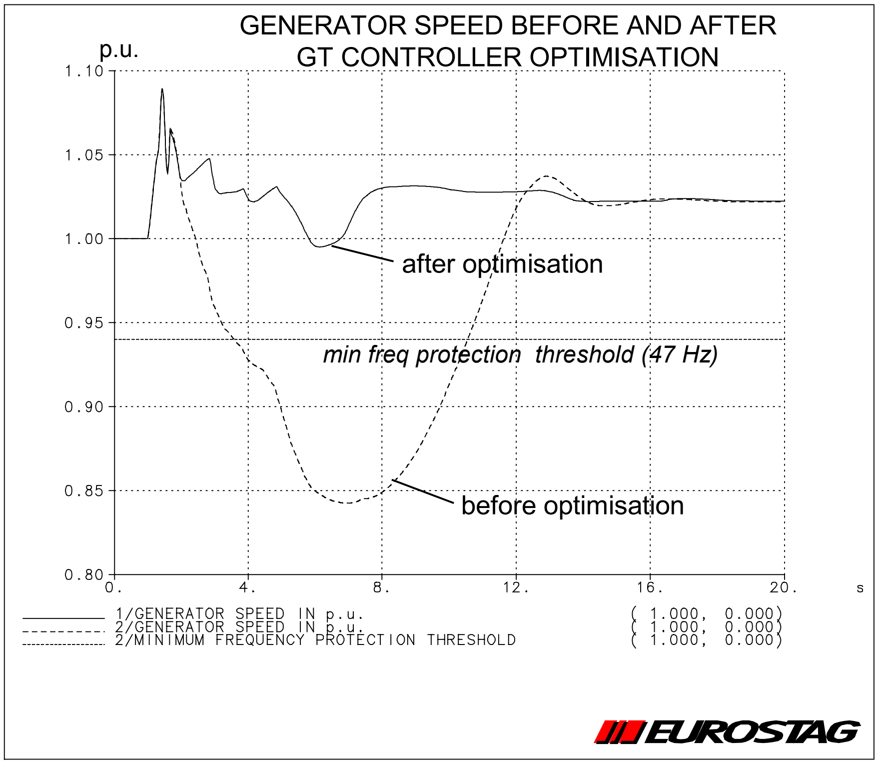


Figure E-4

E.6 Conclusions

The main focus in designing an islanding scheme is the transition from the interconnected to islanded mode following large disturbances. New cogeneration plants, based on low-inertia aero-derivative gas turbines pose a particular challenge in ensuring robust speed control following an islanding event. The discussion here has shown an example of collaborative work with a turbine manufacturer for optimizing the turbine controls on aero-derivative units for the purpose of a smooth transition into an islanding mode of operation.

Site tests and the collaboration of the GT manufacturer remain most of the time a necessary step to bring a final confirmation of the simulation results related to partial load drop and high sudden restart load accept capabilities.

The modeling work nevertheless requires high skills and sustained effort for keeping the models updated. Major returns of this effort are certainly a deeper knowledge of system behavior allowing the enhancement of system reliability and an adequate equipment sizing to match the required performance.

References

[1] M. Stubbe, C. Merckx, K. Karoui and J. Dubois, “Optimization Method for Parameters Identification and Controller Tuning”, 12th CEPSI Conference, Pattaya, THAILAND, 2-6 November 1998.

[2] M. Stubbe, "EUROSTAG, outil pour la maîtrise de la sécurité des systèmes électriques industriels", Revue E - n°3/4 - 98 (in French).

[3] J.P. Bécret, "Dynamique des grands réseaux industriels", Revue E - n°3/4 - 98 (in French).