Development and adaptation of dynamic models for new power generation sources

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Abstract

This dissertation's main aim was to adapt a generic gas turbine and combined cycle power plant dynamic model for use in the power system simulation software, DigSilent PowerFactory. Due to the advantages in overall efficiency and lower emissions compared to conventional coal fired power plants, combined cycle power plants have gained popularity. Combined cycle power plants have become a significant portion in power generation across the world in recent times.

Due to changes in the world to minimise carbon-dioxide footprints, there is demand for cleaner methods of power generation. In South Africa, the main power source is still coal fired power stations, but in recent times, gas turbine power plants were added to the power system.

Approximately two-thirds of the generation capacity in a combined cycle power plant is produced by the gas turbines. The other third is generated by the steam turbine. Using the steam that is available means the overall efficiency of the power plant is improved and the emissions are decreased.

Gas turbines and their controls are significantly different from the controls of a conventional steam turbine plant. In particular, the maximum output power of the gas turbine is very dependent on the deviation of its operating frequency from the rated frequency (or speed of the gas turbine), and the ambient conditions in which the gas turbine operates.

In an effort to provide the industry with a single document and simulation model that summarises the unique characteristics, controls and protection of combined cycle power plants, the Cigre Task Force 25 was formed [1]. The aim of this Task Force was to develop an open cycle gas turbine (a more detailed model than existing models) and a combined cycle power plant simulation model, as no detailed models existed in any power system simulation software.

The aim of this dissertation was to adapt the Cigre simulation models, enabling their use in the DigSilent PowerFactory power system simulation software and validate their performance.

Opsomming

Hierdie tesis se hoofdoel was om ŉ generiese gasturbine en saamgestelde-siklus elektriese kragaanlegmodel om te skakel vir die gebruik in kragstelsel simulasie-sagteware, DigSilent PowerFactory. As gevolg van die voordele in totale rendement en laer vlakke van uitlaatgasse in vergelyking met konvensionele steenkool-kragstasies, het saamgestelde-siklus kragaanlegte in gewildheid toegeneem, en het die aanlegte ŉ belangrike komponent in kragopwekking in die wêreld geword. In Suid-Afrika is die hoofkragbron steeds steenkoolkragstasies, maar onlangs is daar ook gasturbine-aanlegte toegevoeg tot die netwerk. As gevolg van die verandering in die wêreld om koolstofdioksied vlakke te verminder, is daar ŉ aanvraag na skoner metodes vir kragopwekking.

Ongeveer twee-derdes van die opwekkingsvermoë in saamgestelde-siklus kragaanlegte word deur die gasturbines gelewer. Die ander een-derde word deur die stoomturbine gelewer, en met die dat stoom beskikbaar is, word die totale rendement van die kragaanleg verbeter terwyl die uitlaatgasvlakke verminder word.

Gasturbines en hul beheer verskil beduidend van die beheer van konvensionele stoomturbine-aanlegte. In besonder, die maksimum vermoë van die gasturbine is afhanklik van die afwyking van die frekwensie-werkspunt van die ontwerpsfrekwensie (of die ontwerp-spoed van die gas-turbine), en die omgewingstoestande waarin die turbine funksioneer.

In ŉ poging om die industrie van ŉ enkele dokument en simulasiemodel te voorsien wat die unieke eienskappe, beheer en beveiliging van saamgestelde-siklus kragaanlegte modelleer, was die Cigre-Taakspan 25 gevorm [1]. Die doel van die taakspan was om ŉ gasturbine model (‘n meer gedetailleerde model as die bestaande modelle) en ‘n saamgestelde-siklus kragaanlegsimulasie model te skep, aangesien daar geen detail modelle beskikbaar is nie in die simulasie pakkette.

Die doel van hierdie tesis was om die omskakeling van die Cigre-modelle te doen om in DigSilent PowerFactory-kragstelsel analise-sagteware gebruik te kan word.

Declaration

I, Johannes Hendrik Grobler, (ID Nr: 700710 5269 08 6) hereby declare that all the material incorporated into this dissertation is my own original unaided work, except where specific reference is made by name or in the form of a numbered reference. The work here-in has not been submitted to any other university to obtain a degree.

………………………………...
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August 2011

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Glossary of Terms

Combined-Cycle	The gas turbine Brayton cycle and a steam turbine Rankine cycle are combined 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 (CCPP) – a CCPP consists of a number of gas and steam turbines operating in a combined-cycle. CCPPs are configurable in a variety of combinations of the gas and steam turbines.
GT	Gas Turbine – a machine consisting of an axial compressor, combustor and turbine assembly, and auxiliary equipment, used to produce rotating mechanical energy and heat from various types of fuel suitable for use in the gas turbine.
HRSG	Heat Recovery Steam Generator, used to generate steam from the heat of the exhaust gas of a gas turbine.
Multi-shaft CCPP	A type of configuration for combined-cycle power plants comprising one or more gas turbines, each with its own Heat Recovery Steam Generator, feeding steam to a single steam turbine, all on separate shafts with separate electrical 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. Multiple spooling is having the axial compressor and turbine sections mechanically separated into multiple sections of the shaft.
Simple-Cycle	A simple-cycle refers to gas turbines that are operated as stand-alone units as opposed to being incorporated in a combined cycle power plant.
ST	Steam Turbine - A steam turbine is a rotating engine that extracts energy from steam (at high pressure and temperature) and converts it into useful mechanical work. Steam turbines have a casing around the blades that contains and controls the working fluid. Modern steam turbines frequently employ both reaction and impulse in the same unit, typically varying the degree of reaction and impulse from the blade root to its periphery. When an electrical generator is connected to the shaft of the steam turbine, the mechanical energy is converted into electrical energy.
Single-shaft CCPP	In this configuration of a combined cycle power plant, a single gas turbine, a single steam turbine and single electrical generator are connected in tandem to a single rotating shaft. The exhaust of the gas turbine is supplied to a single heat recovery steam generator that generates the required steam for the steam turbine cycle.

Single-shaft GT	A single-shaft gas turbine 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 between large single-shaft gas turbines and multi-shaft aero-derivative units.
VIGV	Variable Inlet Guide Vanes - vanes used to control the airflow into the gas turbine inlet system.
r/min	Rotational speed of equipment, revolutions per minute
rad/s	Rotational speed of equipment, radians per second
HP	High pressure
IP	Intermediate pressure. “IP” also refers to the medium pressure in the steam turbine.
LP	Low pressure
CO	Carbon monoxide
NO_X	Nitrogen oxides
SO_X	Sulphur oxides

List of Abbreviations

V - Voltage (volt)
I - Current (ampere)
AC - Alternating current
DC - Direct current
K - Temperature rise (kelvin)
°C - Temperature (Celsius)
VA - volt-ampere
var - volt-ampere reactive
W - watt
OC - Open cycle
CC - Close cycle
CCPP - Combined cycle power plant
HRSG - Heat recovery steam generator
ST - Steam turbine
GT - Gas turbine
υ - Speed (m/s)
E_k - Kinetic energy (J)
m - Mass (kg)
ω - Angular velocity (rad/s)
T_m - Mechanical torque (Newton-meter, N.m)
AVR - Automatic voltage regulator
r/min - revolutions per minute
rad/s radians per second
MPa - mega-pascal
ppm - parts per million

- Chapter 1 -

1 BACKGROUND

1.1 INTRODUCTION

In today's deregulated and competitive electric power market there is a significant demand for power plants with greater efficiency, controllability, manoeuvrability and low emissions. Currently, the bulk of South Africa's generation of electricity is from coal-fired power stations that are relatively slow to respond to demand changes, such as the sudden addition of a bulk load to the network, or the disconnecting of a major part of the network. Gas turbines and combined cycle power plants have the ability to respond quicker to the demand changes in the power system, compared to coal-fired power sources.

Due to their 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. In a typical combined-cycle power plant, approximately two-thirds of the generated power is produced by gas turbines, while the other third is produced by steam turbines.

Gas turbines and their controls are significantly different from fossil-fuel steam turbine power plants. In particular, the maximum power output of the gas turbine is highly dependent on:

- The environmental ambient conditions, as the gas turbine thermal cycle is an open cycle using atmospheric air as its working fluid.
- The maximum power output of the turbine is dependent on the deviation of its operating frequency (or operating speed) from its rated speed.

Gas turbines are mainly classified as [11]:

- Aero-derivative gas turbines,
- Light industrial gas turbines and
- Heavy-duty industrial gas turbines.

The size of aero-derivative gas turbine units varies from 8 MW to 25 MW and in fact are aircraft engines that are used as "gas generators". These units have efficiencies in the range 35% to 45% and are used extensively in the oil and gas production industry.

The size of light industrial gas turbines range from about 5 MW to 15 MW. This type of turbine is used extensively in many petrochemical plants for compressor drive trains, and has efficiencies just above 30%. Due to the size of the units, they are popular in offshore applications.

The heavy-duty industrial gas turbines are found in refineries, chemical plants and power utilities. Their size range from 3 MW to 480 MW in simple open cycle configuration, and have efficiencies that range from 30% to 46% depending on the size. These units do however lend themselves to various heat recovery methods, e.g. exhaust gas heat exchangers and recuperators on the inlet air.

Existing power system simulation software currently contains gas turbine models that represent only the gas turbine and its associated generator [2], [3]. The models only represent electrical generators that are directly connected to gas turbine shafts. Most of these models were created several years ago, while new technology and techniques were recently developed to control gas turbines and combined cycle power plants [2], [3].

It is therefore not possible, with presently existing library models in commercial power system simulation software packages, to model combined cycle power plants accurately, as the existing models do not make provision for the steam generation component, i.e. the heat recovery steam generator.

The development and utilisation of combined cycle power plants made it necessary to model these plants correctly in power system simulation software.

1.2 OBJECTIVE

The aim of this dissertation is to develop or adapt new dynamic models for gas turbine and combined cycle power plants for use in industrial grade power system simulation software packages and validate their performance.

The deliverables of the dissertation are dynamic model libraries for combined cycle power plants:
- Gas turbines – open cycle, single shaft units
- Combined cycle power plants

Multi-shaft units, i.e. combined cycle power plant configurations with more than one shaft, were not considered in this dissertation.

1.3 OVERVIEW OF DISSERTATION

This dissertation is divided into the following chapters:

Chapter 1 is the introduction and provides background and the objective to the dissertation.

Chapter 2 contains the literature study and provides the background information and theory required to model the various equipment in power system simulation software. The equipment includes the various types of steam turbines, the various types of gas turbines and combined cycle power plants.

Chapter 3 provides a detailed description of the theory of the existing gas turbine models used in power system simulations. This chapter includes a comparison between the existing gas turbine and the new combined cycle power plant models. The theory includes the various building blocks for simulation models and the control theory of these models.

Chapter 4 provides a detailed description of the existing gas turbine dynamic simulation models. The new adapted dynamic models for the gas turbine and combined cycle power plants are included in this chapter.

Chapter 5 contains the simulation results of the adapted dynamic models.

Chapter 6 presents the findings, recommendations and conclusions of this dissertation.

Chapter 7 includes the list of references.

Appendix A includes the model data used for the simulations.

Appendix B includes the basic building blocks for the simulation models.

- Chapter 2 -

2 LITERATURE STUDY

2.1 BACKGROUND

In the literature study, the various types of turbine driven generation sources and their characteristics are discussed. This will give the reader a better understanding of the various types of generation sources.

Various power generation sources are in use today. These sources include steam, wind, hydro and pumped storage schemes. Renewable sources are solar, wind and wave energy based. Other generation sources are gas turbines and combined cycle power plants.

Only the steam turbine, the gas turbine and heat recovery steam generator (HRSG) will be discussed in detail in this dissertation. Combined cycle power plants are a combination of gas turbines, heat recovery steam generators and steam turbines.

In the literature study, a description of the characteristics of the following equipment is included:

- Steam turbines,
- Gas turbines – open cycle power plants,
- Combined cycle power plants (made up of gas turbines and heat recovery steam generators (including steam turbines)).

The characteristics and the dynamic response of the various equipment are different, e.g. a steam turbine will react differently than a gas turbine to the same disturbance in the power system due to the differences in design, functioning, construction, control systems and various other factors. It is therefore important to understand the characteristics of each of the components of a combined cycle plant separately before the characteristics of a combined cycle power plant can be understood. A typical combined cycle power plant consists of gas turbines and steam turbines. Various combinations are possible, and will be discussed. In the simplest form, the gas turbine, the steam turbine and the generator are connected to the same shaft.

The inclusion of this information provides some insight into the characteristics of equipment to allow better understanding of the dynamic modelling of this equipment. The detailed characteristics of each of the components fall outside the scope of this dissertation.

2.2 STEAM TURBINES

The steam turbine is used to recover the heat produced by the gas turbines and makes a significant improvement to the efficiency of the combined cycle power plant. It is a key component of the system and various configurations will be considered in the coming paragraphs.

2.2.1 BACKGROUND

A steam turbine falls in the category of equipment that is generally known as “prime movers”. A prime mover is a source of rotating mechanical energy. The energy is generated by burning coal, gas or nuclear fuel. The generated heat is used to heat pure water to superheated steam and very high pressure in a boiler. The temperature of superheated steam is typically between 300 ºC and 540 ºC. The steam’s pressure ranges from 3 MPa to 15 MPa, depending on the type of turbine [13]. The high-pressure superheated steam is channelled through a series of pipes and the steam turbine casing. The turbine casing encloses the rotor of the turbine. The rotor consists of blades that are designed to rotate between the stationary blades (positioned on the inside of the casing). When the steam passes through the blades, the energy of the steam is transferred to the rotor. This in turn forces the rotor to turn (i.e. rotating mechanical energy is created). The mechanical energy is available on the shaft of the rotor. An electrical generator that is connected to the shaft will convert the mechanical energy into electrical energy. Valves regulate the amount of steam that flows into the turbine. By regulating the valve opening, the amount of mechanical energy transferred from the boiler to the turbine is controlled. The transfer of the mechanical energy from the turbine to the generator is therefore controlled [9]. Figure 2.1 shows a cutaway view of a steam turbine.

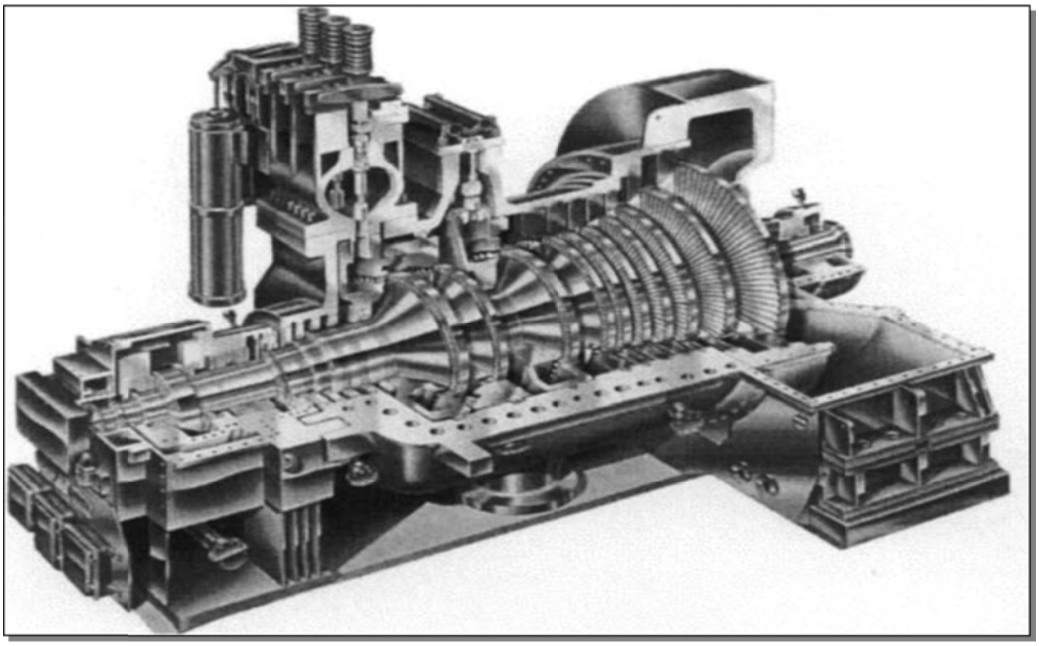


Figure 2.1: Cutaway view of a steam turbine [1]

2.2.2 TYPES OF STEAM TURBINES

Various types of steam turbines are available. The types are [13]:

2.2.2.1 STRAIGHT CONDENSING TURBINE

The steam enters the turbine at one pressure, and leaves it at a pressure below atmospheric pressure at the turbine exhaust. The steam is converted back into liquid form in the condenser. From the condenser the liquid is pumped to the boiler for reheating. In the boiler, superheated steam is generated and sent to the turbine in a continuous cycle. Figure 2.2 shows a straight condensing turbine.

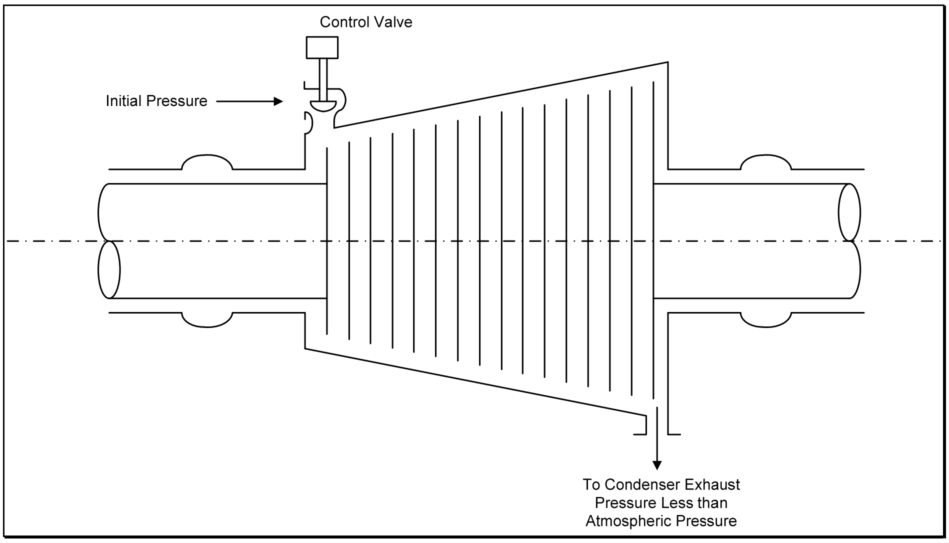


Figure 2.2: Straight Condensing Turbine [13]

2.2.2.2 STRAIGHT NON-CONDENSING TURBINE

The steam enters the turbine at one pressure, and leaves it at a pressure equal or higher than atmospheric pressure at the turbine exhaust. The remaining steam energy is used elsewhere and can then be circulated back to the condenser, the boiler and back to the turbine. This is also known as a backpressure turbine. Figure 2.3 shows a straight non-condensing turbine.

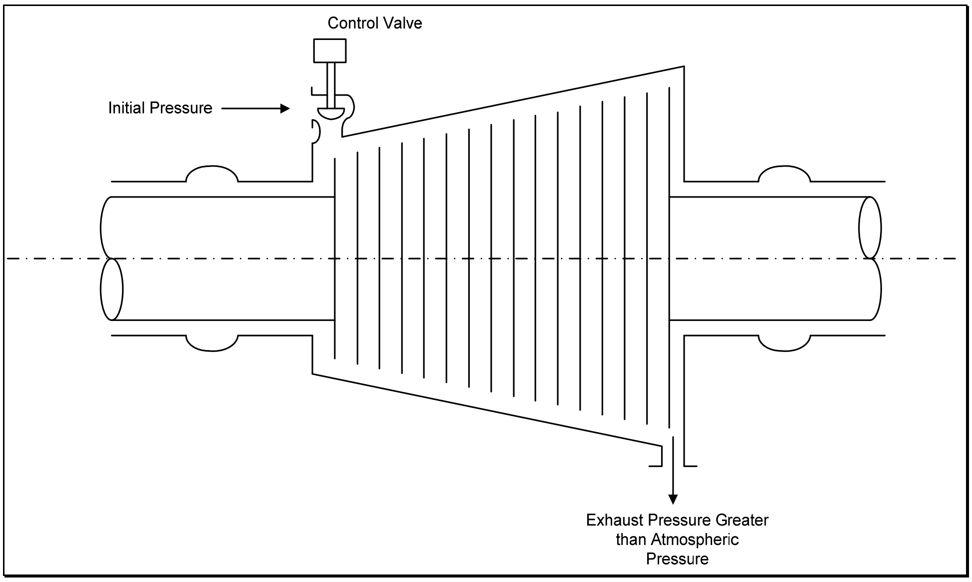


Figure 2.3: Straight Non-condensing Turbine [13]

2.2.2.3 NON-AUTOMATIC EXTRACTION - CONDENSING OR NON-CONDENSING TURBINE

The steam enters the turbine at one pressure and is extracted at one or more of the stages. The pressure of the extracted steam is not controlled. The extracted steam is used elsewhere and then sent to the condenser, the boiler and back to the turbine. Figure 2.4 shows a non-automatic extraction, condensing or non-condensing turbine.

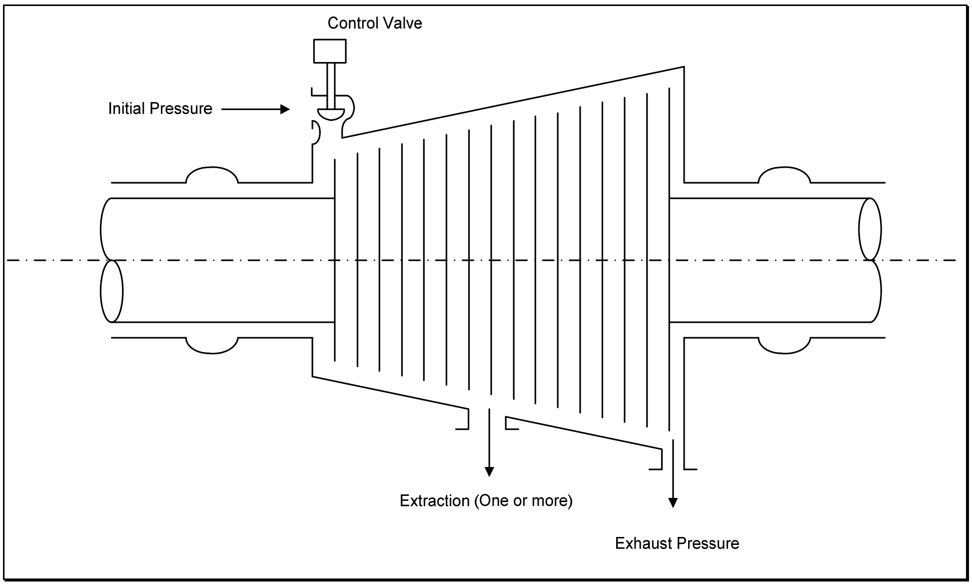


Figure 2.4: Non-automatic Extraction Turbine [13]

2.2.2.4 AUTOMATIC EXTRACTION - CONDENSING OR NON-CONDENSING TURBINE

The steam enters the turbine at one pressure and is extracted at one or more of the stages. In this setup, control valves regulate the pressure of the extracted steam. The extracted steam is used elsewhere and then sent to the condenser, the boiler and back to the turbine. Figure 2.5 shows an automatic extraction turbine.

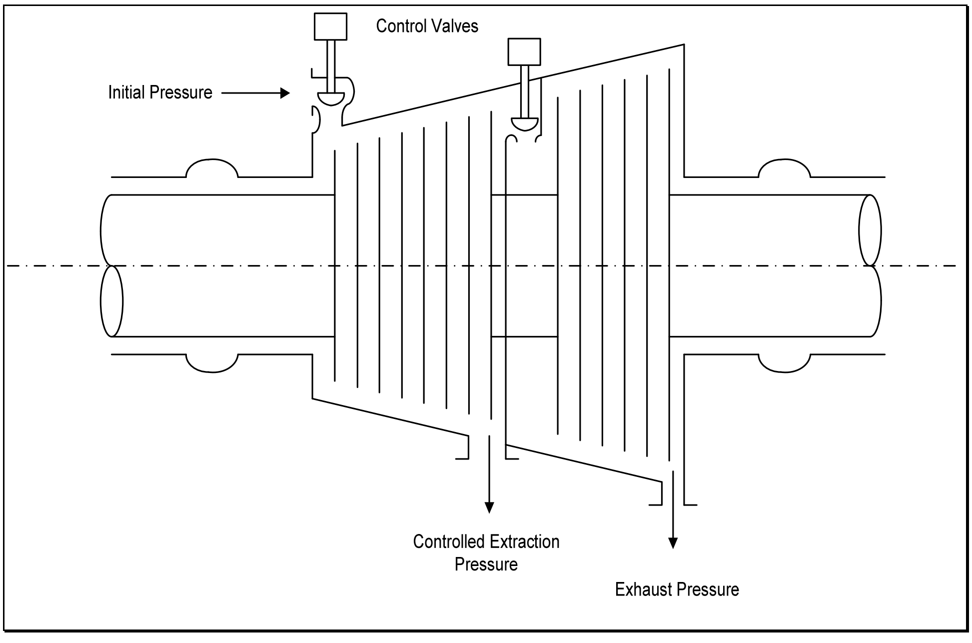


Figure 2.5: Automatic Extraction Turbine [13]

2.2.2.5 AUTOMATIC EXTRACTION / INDUCTION - CONDENSING OR NON-CONDENSING TURBINE

Steam is extracted or inducted into the turbine at more than one of the stages, while the control valves control the pressure. The extracted steam is used elsewhere and then sent to the condenser, the boiler and back to the turbine. Figure 2.6 shows an automatic extraction / induction turbine.

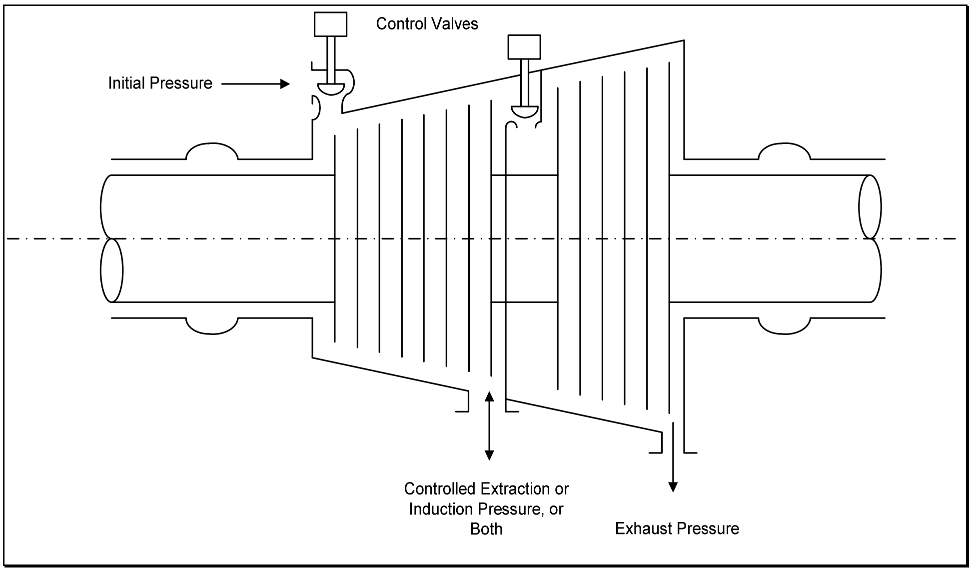


Figure 2.6: Automatic Extraction / Induction Turbine [13]

2.2.2.6 MIXED PRESSURE - CONDENSING OR NON-CONDENSING TURBINE

Steam enters the turbine at more than one of the stages of the turbine, through separate openings. By means of control valves, the pressure of the steam at each opening is regulated separately. The extracted steam is used elsewhere and then sent to the condenser, the boiler and back to the turbine. Figure 2.7 shows a mixed pressure turbine.

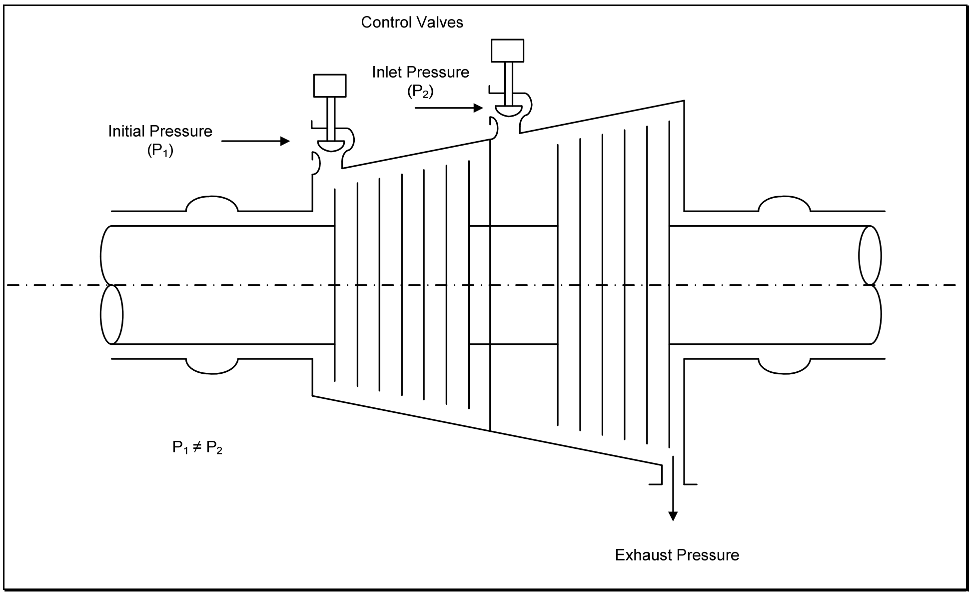


Figure 2.7: Mixed Pressure Turbine [13]

2.2.2.7 REHEAT TURBINE - CONDENSING OR NON-CONDENSING TURBINE

Steam enters the turbine at one pressure and is extracted at a lower pressure and temperature. The extracted steam is reheated and re-admitted to the turbine. Steam pressure is controlled at both inlets by means of control valves. The extracted steam is used elsewhere and then sent to the condenser, the boiler and back to the turbine. Figure 2.8 shows a reheat turbine.

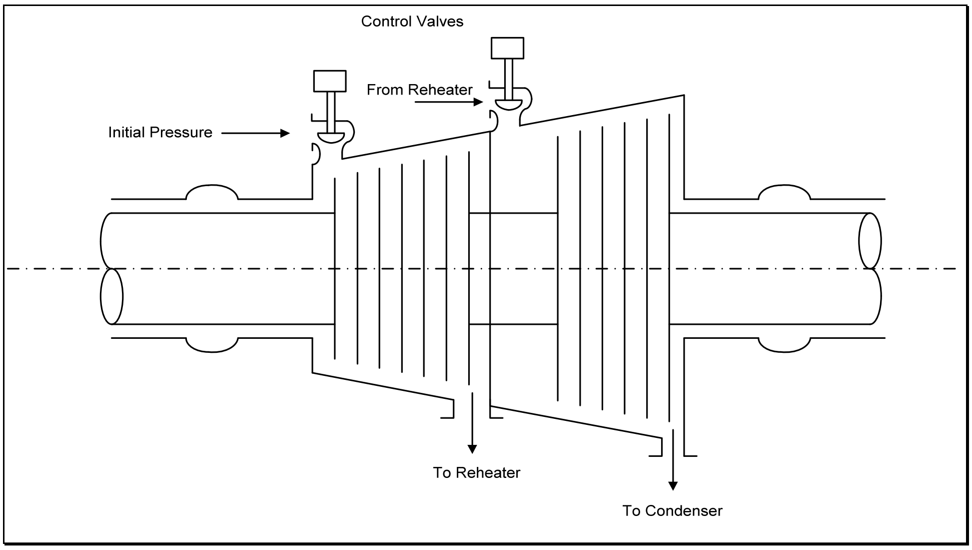


Figure 2.8: Reheat Turbine [13]

The physical construction of the above steam turbine types can be a single-stage or a multiple-stage turbine [10].

A single-stage turbine is a turbine in which the conversion of kinetic energy (of the steam) to mechanical energy occurs with a single expansion of the steam in the turbine, i.e. from the inlet steam pressure to the exhaust steam pressure.

Multiple-stage turbines consist of more than one stage in which the conversion of kinetic energy to mechanical energy takes place.

2.2.3 STEAM TURBINE CASINGS

Casing designs employ either single- or double-shell constructions [9]. Both construction types are used in many applications. The double shell construction prevents initial steam being in contact with the outer casing.

The required capacity, the type of cycle and the exhaust volume flow to the condenser will determine the number of casings. Single- and multiple-casings are used. A multiple-casing turbine usually consists of high- (HP), intermediate- (IP) and low-pressure (LP) stages [24].

Multi-casing turbines are often used when reheat cycles are used. In a reheat system (typical in large power plants), the steam from the high pressure stage is routed back to the boiler or heat recovery steam generator to receive additional heat energy before proceeding to the intermediate- or low-pressure stage of the turbine.

In a reheat system, found in large combined cycle power plants, steam exits the high-pressure turbine and is routed back through the heat recovery steam generator to receive additional heat energy before being routed back to the intermediate- and low-pressure turbine stages.

In a combined cycle system, the steam turbine is operated in two different modes. The one is “sliding pressure” and the other is “fixed pressure” control.

Sliding pressure control is achieved by keeping the control valves in the fully open position. The steam pressure entering the steam turbine is therefore a function of the steam mass flow entering the steam turbine. The power output of the steam turbine depends on the mass flow of the steam, and cannot be directly controlled. The only way to increase the power output from the steam turbine is to generate more steam from the heat recovery steam generator. This is achieved by increasing the heat generated from the gas turbine. Supplemental firing, if it is present, will also produce an increase in the generated heat from the combined cycle power plant. Supplemental firing is additional heating of the steam in the heat recovery steam generator. This is sometimes required if the amount of required steam differs from the amount required to generate electrical power, i.e. to be used for a process elsewhere. The additional heat is also used for generating more mechanical power from the steam turbine in the heat recovery steam generator. The dynamic response of a steam turbine operating in sliding pressure control mode is slow as it will not respond significantly to governor action in the first seconds following a power system disturbance, and could take several seconds to several minutes to respond with a significant increase in power. Most combined cycle power plants are operated in sliding pressure control when operated near full load [1].

Fixed pressure control is achieved by controlling the inlet valve position, and this in turn controls the steam flow, thereby keeping pressure at a desired level. The main objective of fixed pressure control is to obtain a better part-load efficiency of the steam turbine. Fixed pressure control is used to keep the steam turbine output constant. Any change in the gas turbine output will not change the output of the steam turbine as with sliding pressure control [1].

2.2.4 THERMODYNAMIC CYCLE

The complete process of converting the fuel’s chemical energy into mechanical energy is called a thermodynamic cycle. The most efficient thermodynamic cycle for an ideal fluid is the Rankine cycle (see Figure 2.9) [15], [18]. Engineers generally employ this cycle as a standard for comparing the performance of actual steam engines and steam turbines. Figure 2.9 shows the Rankine thermodynamic cycle.

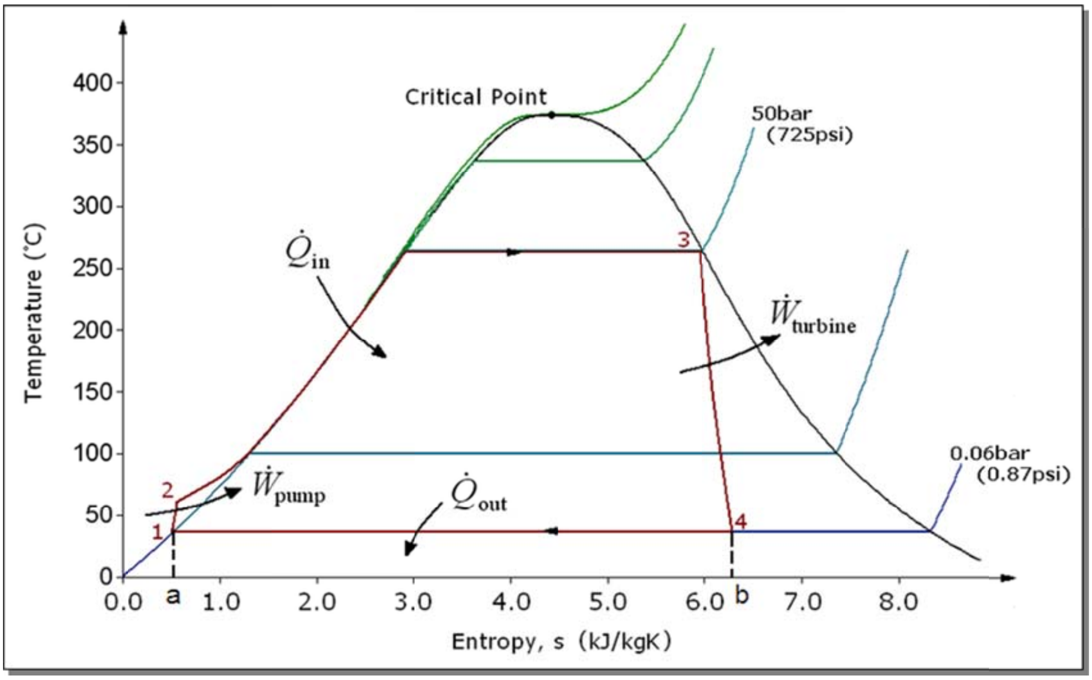


Figure 2.9: Rankine Cycle [18]

The Rankine cycle consists of an isothermal heat input (heat addition at constant pressure), an isentropic expansion, an isothermal heat rejection (at constant temperature) and an isentropic compression.

With reference to Figure 2.9, the four processes of the cycle are:

Process 1 – 2:	The liquid (water) is pumped from a low pressure to a higher pressure. As the fluid is a liquid at this stage, the pump requires little input energy (W_{pump}). This is also known as reversible adiabatic pumping process in the pump [18], [19].
Process 2 – 3:	The liquid that is under pressure enters the boiler. In the boiler, the liquid is heated at constant pressure by an external source. The liquid becomes a dry saturated gas. The amount of input energy (Q_{in}) can be calculated from an enthalpy-entropy chart for the liquid [18], [19].
Process 3 – 4:	The dry saturated gas (or super-heated steam) leaves the boiler and enters the turbine. Here, the steam expands through the turbine, transferring energy to the turbine. During this process the steam (or gas) temperature and pressure decreases. The amount of energy ($W_{turbine}$) can be calculated with an enthalpy-entropy chart for the liquid / gas [18], [19].
Process 4 – 1:	During this process, the gas enters the condenser where it is condensed into a liquid and the process repeats itself. A constant pressure transfer of heat in the condenser takes place [18], [19].

The efficiency of the Rankine cycle is given by (with reference to Figure 2.9):

$$\eta = \frac{w_{net}}{q_H} = \frac{area(1-2-3-4-1)}{area(a-1-2-3-4-b-a)} \qquad (2.2.4.1)$$

In analysing the Rankine cycle, it is helpful to think of the efficiency as depending on the average temperature at which the heat is supplied and the average temperature at which it is rejected [18]. The efficiency can be increased by increasing the pump outlet pressure, increasing the boiler outlet temperature and decreasing the turbine outlet pressure [18], [19]. However, the Rankine cycle efficiency is lower than the Carnot thermodynamic cycle for the same maximum and minimum temperatures. Two reasons to prefer the Rankine cycle over the Carnot cycle are given in [19]. The first is the pumping process. It is difficult to build a pump that can handle the mixture of liquid and vapour at point 1, and deliver saturated liquid at point 2. It is much easier to condense the vapour completely and handle only liquid in the pump and the Rankine cycle is based on this characteristic. The second reason involves the superheating process at point 3. In the Rankine cycle, this takes place at constant pressure, while in the Carnot cycle this takes place at constant temperature. In the Carnot cycle, this takes place during a drop in pressure, which means that the heat must be transferred to the vapour as it undergoes an expansion process. To achieve this in practice is very difficult. This makes the Rankine cycle the preferred cycle to be applied in practice [19].

2.3 GAS TURBINES

The other major piece of equipment in a combined cycle power plant is the gas turbine. It determines the configuration of the system and is instrumental in the plant’s dynamic performance. Various types of gas turbines are discussed in this section.

Gas turbines also fall in the category of “prime movers”. Gas turbines are a source of rotating mechanical energy. The use of a gas turbine to produce mechanical energy is in many respects the most satisfactory method to generate mechanical energy. The absence of reciprocating and rubbing members (e.g. the internal combustion engine as found in vehicles) means that the balancing of the moving parts is relatively simple, that the consumption of lubricating oil is exceptionally low and reliability is high [12].

The disadvantage of a steam turbine is therefore the installation of bulky and expensive steam generating equipment, whether it is a conventional boiler or a nuclear reactor [12]. A significant feature of these generating plants are that the hot gases that are produced never reach the steam turbine, it is merely used to produce an intermediate medium for heat transfer, namely steam [12]. A more compact system results when the water-to-steam step is eliminated and the hot gases are used directly to drive the turbine [12]. The heat for a gas turbine is generated inside the turbine, while the medium that carries the heat is air.

Gas turbines are mainly classified as [11]:

- Aero-derivative gas turbines,
- Light industrial gas turbines and
- Heavy-duty industrial gas turbines.

2.3.1 AERO-DERIVATIVE GAS TURBINES

The size of these units varies from 8 MW to 25 MW. This is the typical size required in the oil and gas production industry. These gas turbines are in fact aircraft engines that are used as “gas generators”. This is a source of hot high velocity gas. This gas is directed into the power turbine, which is placed close to the exhaust of the gas generator. The power turbine drives the electrical generator. These units have efficiencies in the range 35% to 45% [22].

The benefits of this arrangement are [11]:

- Easy maintenance since the gas generator can be removed as a single and simple module,
- High power-to-mass ratio,
- Can be easily designed for a single lift modular installation,
- Easy to operate,
- Small area required for the installation.

The main disadvantages of this arrangement are [11]:

- Relative high costs of maintenance due to short running times between overhauls,
- Lower fuel economy than other types of gas turbines,
- Expensive to replace.

The aero-derivative gas turbines are generally used in combined cycle power plants for power generation, especially in remote areas where the power requirements are less than 100 MW. The petrochemical industry uses this type of gas turbine on offshore platforms, mostly due to their compactness and ease of replacement for maintenance purposes [22]. It is thus clear this type of gas turbine is mainly used in smaller isolated power systems for power generation.

2.3.2 LIGHT INDUSTRIAL GAS TURBINES

The size of these units range from about 5 MW to 15 MW. This type of turbine is used extensively in many petrochemical plants for compressor drive trains. Their efficiencies are just above 30%. These units are similar in design to the heavy-duty industrial gas turbines. However, their casings are thicker than the aero-derivative casings, but thinner than the heavy-duty gas turbine casings. They usually are split-shaft designs that are efficient in part load operations [22].

Some manufacturers utilise certain advantages of the aero-derivative units, i.e. high power-to-mass ratios and easy maintenance. The high power-to-mass ratios are achieved by running the units with high combustion and exhaust temperatures and by operating the primary air compressors at relative high compression ratios. To achieve the high power-to-mass ratios, a minimum of metal is used and this necessitates a more frequent maintenance program. Easier maintenance is achieved by a modular design of the turbine, in order to remove the combustion chambers, the gas generator and the compressor turbine easily [11].

As these units rarely drive electrical generators, they will not be studied.

2.3.3 HEAVY-DUTY INDUSTRIAL GAS TURBINES

These units are found in refineries, chemical plants and power utilities. Their size range from 3 MW to 480 MW in simple open cycle configuration. Their efficiencies range from 30% to 46% depending on the size [22]. The main reason for their selection is the long running intervals between major maintenance overhauls. These units can also burn a variety of fuels in liquid and gas form, including the heavier crude oils. These units can also tolerate a higher level of impurities in the fuels. Heavy-duty industrial units are unsuitable for offshore applications due to the following reasons [11]:

- High power-to-mass ratios required stronger and larger support structures,
- Longer maintenance intervals, as the machine must be disassembled into many separate components, as a modular design approach cannot be followed for these machines,
- Poorer thermodynamic performance than the smaller units.

These units do however lend themselves to various heat recovery methods, e.g. exhaust gas heat exchangers and recuperators on the inlet air [11].

Heavy-duty industrial gas turbines are usually found in power plants where they form part of large power systems. Gas turbines and combined cycle power plants have the ability to respond quicker to the demand changes in the power system, compared to coal-fired power sources. Combined cycle power plants operate with greater efficiency, controllability, manoeuvrability and lower emissions when compared to coal fired power stations.

2.3.4 GAS TURBINE MAIN COMPONENTS

The three main components of gas turbines are [12]:

- Axial compressor,
- Combustion chamber, and
- Turbine (operating under the Brayton thermodynamic cycle)

The three main components form the thermal block, while the air intake system, the exhaust system, the controls and auxiliaries complement them.

Air is drawn into the axial compressor through the air intake system. The axial compressor compresses the air by several 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 increasing the pressure of the air. The typical pressure ratio of the air through the axial compressor is 15:1 to 20:1, but can be as high as 35:1 [12]. The pressure ratio is the ratio of the compressor outlet pressure to the inlet pressure.

The air exiting from the axial compressor is mixed with fuel in the combustion chamber. Combustion takes place, and the hot gas from the combustion process expands through a multi-stage turbine. The compressor requires energy to compress the air.

The addition of energy in the form of fuel to the point where the turbine spins at full speed, but with no connected external load, is called the no-load, full-speed of the turbine. The amount of fuel required to operate the gas turbine at no-load and full-speed, is called the no-load, full-speed fuel flow of the turbine.

Additional energy (or fuel) will allow the turbine to drive external connected loads (i.e. load connected to the shaft of the turbine). The turbine therefore drives the axial compressor and the generator connected to the shaft of the gas turbine. This is the single-shaft configuration. Refer to Figure 2.16 for a single-shaft configuration.

The air- and fuel-flow determines the power output of a gas turbine. The fuel-flow, airflow and air temperature together determine the firing temperature in the combustion chamber, as this determines the temperature at the outlet of the combustion chamber. The exhaust temperature is measured, and this determines the fuel- and airflow, the compressor pressure ratio and controls the firing temperature [1], [12].

The airflow is regulated by changing the angular position of the variable inlet guide vanes. The inlet guide vanes are essentially the first few stages of stator blades in the axial compressor. The exhaust temperature is kept high at reduced loading levels, by reducing the airflow. This is done to maintain high overall plant efficiency in combined cycle power plants. When the gas turbine is loaded close to design ratings, the vanes are fully open. The airflow is a function of the angle of the vanes, the ambient temperature at the compressor inlet, atmospheric pressure and the compressor shaft speed [1].

Two types of systems are in operation today, the open cycle and the closed cycle gas turbine [12].

In the open cycle system, fresh atmospheric air is continuously drawn into the turbine, compressed by the compressor and energy added to the air by the combustion of fuel. The air is expanded through the turbine and released into the atmosphere through the exhaust of the turbine [12].

In the closed cycle system, the same air is circulated continuously through the system. However, this system requires a cooler to cool down the hot air from the turbine’s exhaust. The fuel is also not burnt in the circulated air, but a heat exchanger is required. The combustion gases therefore do not pass directly through the turbine [12]. It is claimed that the closed cycle does have advantages over the open cycle system [12]. The possibility of using high pressure (and hence high gas density) throughout the cycle could result in reduced size of the turbine for a given output. The power output can also be regulated by changing the pressure level in the circuit [12]. This means that the power output can be regulated without changing the cycle’s working temperature, hence with little change in overall efficiency.

In general, three types of combustion chambers are in use in gas turbines [1]:

- Annular
- Can type and can-annular
- Silo

The three types refer to the physical design (shape and layout) of the combustion chambers. Figures 2.10 to 2.12 show gas turbines with the different types of combustion chambers.

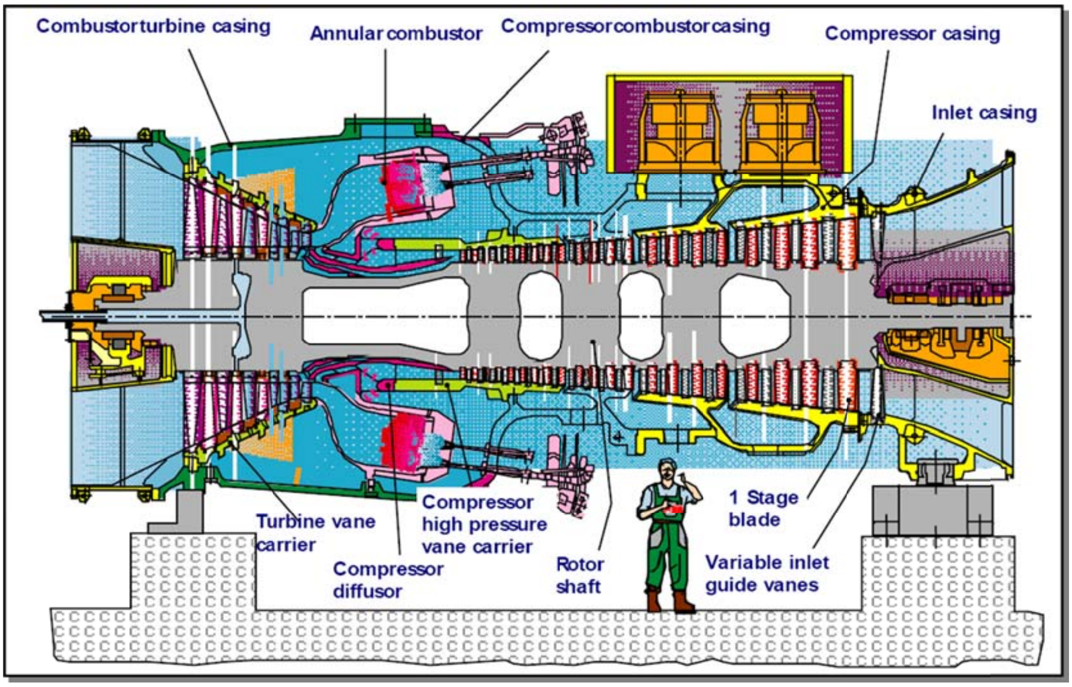
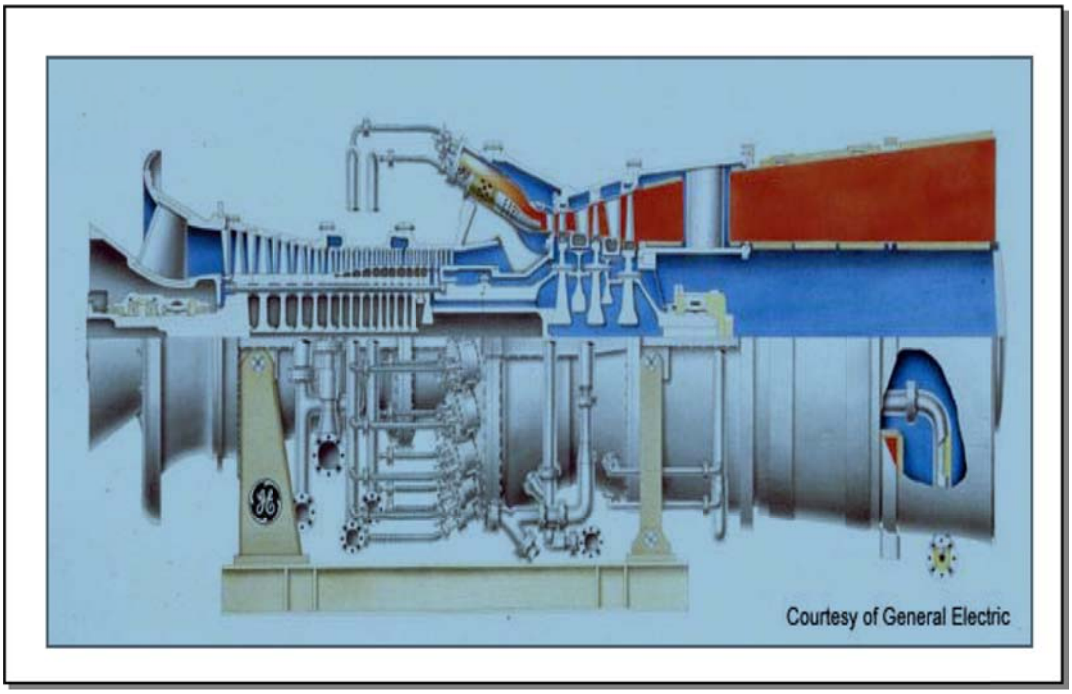


Figure 2.10: Gas turbine with annular combustion chamber [1]



Courtesy of General Electric

Figure 2.11: Gas turbine with can-annular combustion chamber [1]

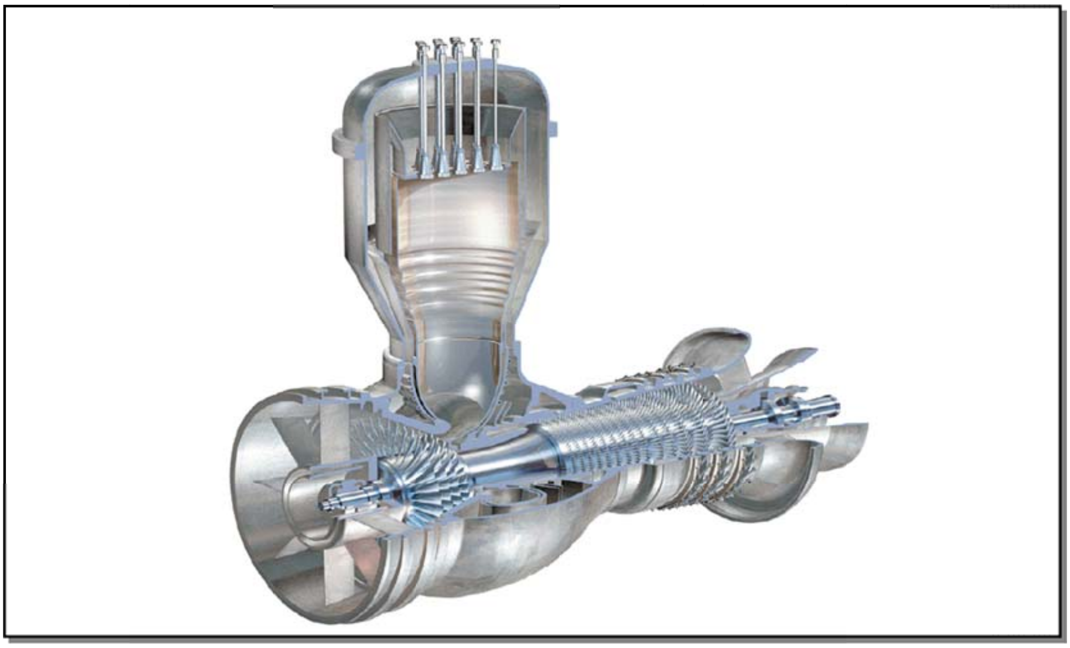


Figure 2.12: Gas turbine with “silo” type combustion chamber [1]

Heavy-duty power gas turbines range in size from 3 MW to 480 MW, while small power generation and industrial applications are in the 10 MW to 50 MW range. The small units are usually aero-derivative gas turbines. As the name suggests, these gas turbines are derived from aircraft turbine 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 usually of the can-type. If a multi-stage axial compressor is designed to achieve high-pressure ratios, it can lead to aerodynamic instability if the turbine is operated at rotational speeds that are widely removed from the designed operating point. To overcome this problem, a design feature of these turbines is multiple spooling used to achieve the wide range of operating speeds required of aircraft engines. In multiple spooling, the axial compressor and/or turbine are mechanically separated into multiple sections, in other words on separate shafts, known as the multi-shaft setup [1].

Single-shaft gas turbines are inherently simpler than multi-shaft turbines, since there are fewer and more accessible bearings, and no variable geometry in the gas turbine hot section. These constructional advantages are somewhat offset by the operational disadvantage of a significantly narrower operating speed range, the latter requiring unique process controls. Nevertheless, the single shaft gas turbine is the only means of achieving the very high output power levels in a “dry” cycle, since the largest heavy-duty multi-shaft gas turbines are rated at less than 30 MW [1]. “Dry” cycle refers to the reduction of NO_X levels by the use of dry low NO_X combustors. New developments have goals of reducing the NO_X levels to below 9 ppm. Catalytic convertors are also used in conjunction with the dry low NO_X combustors to further reduce the NO_X levels [22].

In short, utilisation of single-shaft gas turbines for variable speed mechanical drive service requires careful co-ordination of the prime mover, driven equipment and connected process valve control and other controls [1]. Figure 2.13 shows a heavy-duty gas turbine, the Titan 130 [1].

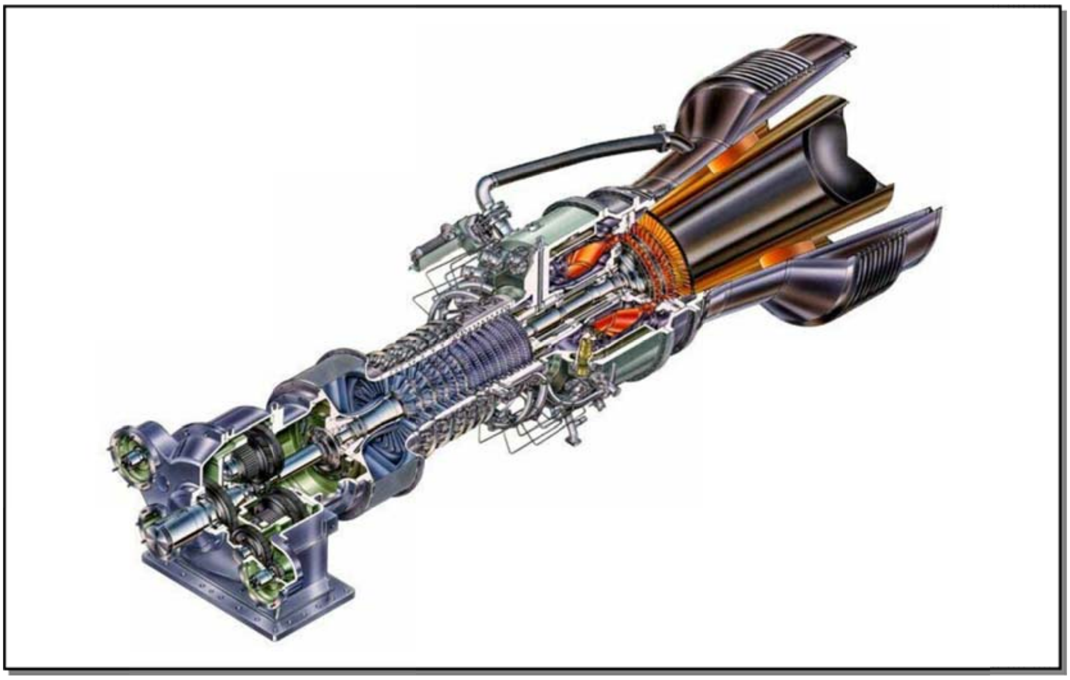


Figure 2.13: An example of a heavy-duty gas turbine, the Titan 130 [1].

2.4 COMBINED CYCLE POWER PLANTS (CCPP)

2.4.1 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 electric generators in a variety of combinations. The primary advantage of a combined cycle power plant is the improved overall efficiency, as the total thermal efficiency is significantly higher than that of a conventional fossil fuel plant or the gas turbine alone [1]. The higher efficiency is attributed to the greater utilisation of the total enthalpy (energy content) produced by the combined process in the gas turbine, the gas turbine Brayton cycle, and the steam turbine Rankine cycle, thus the term combined-cycle power plant. A typical conventional fossil fuel plant has an efficiency of 30% to 40%, while a combined cycle power plant can have efficiencies of up to 58%, provided the waste heat from the steam turbine is recovered. Figure 2.14 shows the combination of the two cycles [1].

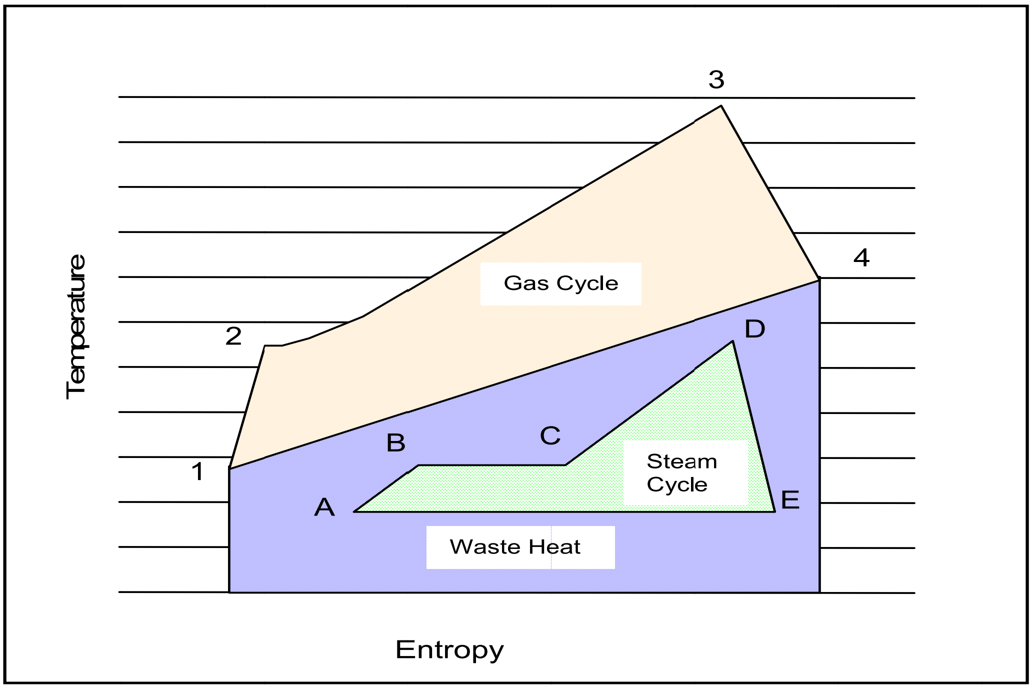


Figure 2.14: Combined-Cycle diagram in temperature / entropy coordinates [1].

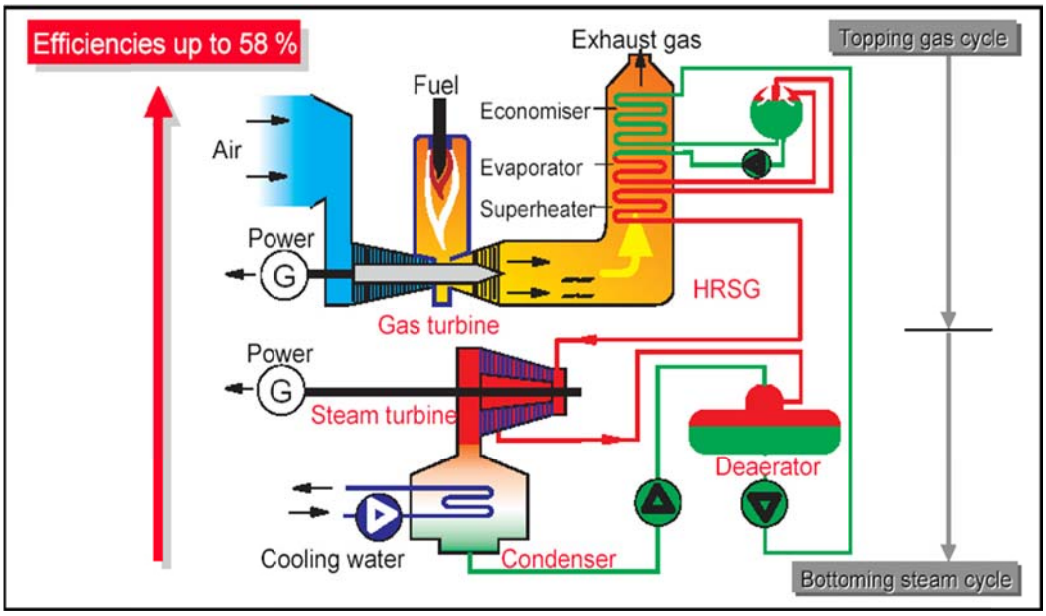


Figure 2.15: Typical Gas-Steam Combined Cycle Power Plant [1].

Referring to Figure 2.14 Point 1 represents ambient conditions. Air is drawn into the compressor where the pressure and temperature of the air is raised (point 2). Point 2 to 3 represents the addition of heat (combustion of fuel), where the pressure and temperature of the air is raised further. Point 3 to 4 represents the transferring of energy from the hot pressurised air to the turbine blades and shaft of the connected electric generator for conversion into electrical energy [1].

The advantage of a combined cycle power plant is in its ability to use the remaining heat in the gas turbine exhaust gas. This heat is extracted by the heat recovery steam generator (see Figure 2.15) and steam is generated. Point 4 to 1 (Figure 2.14) represents the extraction that includes the energy for the steam turbine and waste energy. The steam turbine uses the Rankine cycle. The heat from the gas turbine exhaust is used to increase the water temperature (point A to B). The drum boiler produces steam (point B to C) and additional heat is transferred to the steam in the super-heater (point C to D). At this point, the steam is at high pressure and high temperature. Point D to E represents the expansion of the steam through the steam turbine, transferring energy to the blades of the steam turbine and the shaft of the electric generator. The steam is then condensed (point E to A) and pressurised by a pump (point A to B) to start the cycle again. The development of gas turbines with higher turbine inlet temperatures has improved the efficiency and this made the combined cycle power plant a viable alternative to conventional steam power plants [1].

2.5 MAIN COMPONENTS OF COMBINED CYCLE POWER PLANTS

The previous sections gave an overview of combined cycle power plants. In this section, more details on the major components of combined cycle power plant are provided [1].

2.5.1 GAS TURBINES

Refer to Section 2.3 for a description of a gas turbine.

2.5.2 HEAT RECOVERY STEAM GENERATOR (HRSG)

The heat recovery steam generator is the link between the gas turbine and the steam turbine. The heat recovery steam generators are categorised in three main types:

- Heat recovery steam generator without supplemental firing - This type of heat recovery steam generator is essentially an entirely convective heat exchanger. No additional fuel is used to increase the exhaust gas temperature from the gas turbine. The majority of combined cycle power plants built today makes use of this type of heat recovery steam generator [1].
- Heat recovery steam generator with supplemental firing - Additional fuel is burned in the exhaust duct of the gas turbine to increase the steam generation from the heat recovery steam generator. Supplemental firing is often applied in combined cycle co-generation plants where the amount of process steam must be varied independently of the electric power generated [1], [24].
- A steam generator with maximum supplemental firing - Application of this type is mainly used for the re-powering of existing power plant. The gas turbine replaces the forced draught air blower, feeding the hot combustion air into the boiler [1].

The function of the heat recovery steam generator is to convert the heat energy of the exhaust gas from the gas turbine into steam. Generally, the temperature of the gas turbine exhaust is typically around 535ºC, by adding a steam turbine cycle below the gas turbine cycle, the otherwise wasted heat from the gas turbine is utilised and the overall efficiency of the plant can exceed 60% [24]. The generated steam is used to drive the steam turbine. The heat exchange can take place on up to three pressure levels, depending on the required amount of energy to be recovered. Two or three pressure levels of steam generation are in use today [1]. A further classification is vertical or horizontal, referring to the flow of the gas through the heat recovery steam generator.

2.5.3 STEAM TURBINES

Refer to Section 2.2 for a description of a steam turbine.

2.5.4 ELECTRICAL GENERATORS

Generators for combined cycle power plants are essentially the same as any high-speed generators used in conventional fossil fuel plants. The electrical controls and protection associated with the generator are no different in combined cycle power plants from conventional fossil fuel plants.

2.5.5 CONFIGURATION OF COMBINED CYCLE POWER PLANTS

Combined cycle power plants can be configured in a number of configurations [1], [24], i.e.

- Single-shaft units
- Multi-shaft units

2.5.5.1 SINGLE SHAFT UNITS

In a single-shaft unit, the generator is driven by the gas turbine and the steam turbine, all connected to the same shaft. The advantages of this configuration are [24]:

- Lower capital cost compared to multi-shaft units,
- Only one generator needed,
- Less complex electrical connections,
- Less complex controls,
- Smaller footprint,
- Cleaner than coal fired power stations, with no SO_2 emissions,
- Less staff compared to a coal fired power station,
- Easier material handling (i.e. gas versus coal and ash handling)

Two common designs for single-shaft units are in use today [1]. The one design has the steam turbine and the gas turbine placed on the one side of the generator. In this design, the steam turbine is fixed to the shaft. Figure 2.16 shows the configuration.

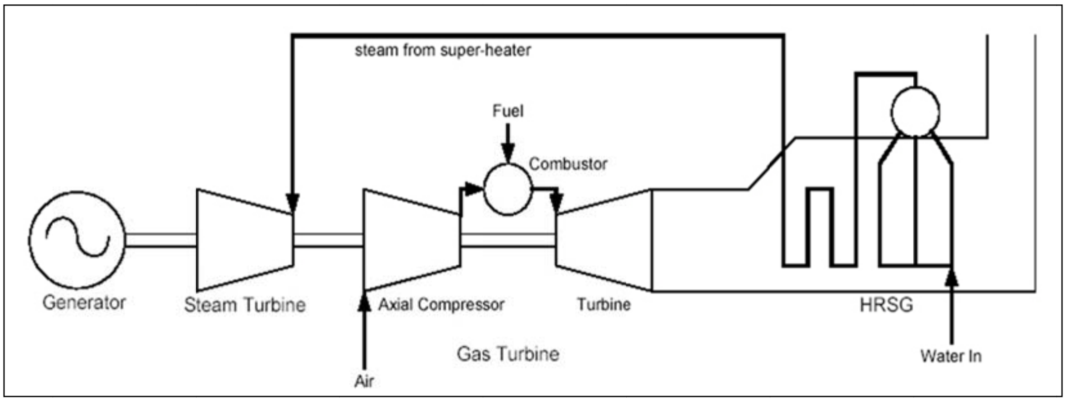


Figure 2.16: Single-shaft unit with generator on end of shaft [1].

The other design has the generator placed between the steam turbine and the gas turbine. In this design, the steam turbine can be fixed to the shaft or can be connected to the shaft by means of a clutch. Figure 2.17 shows the configuration.

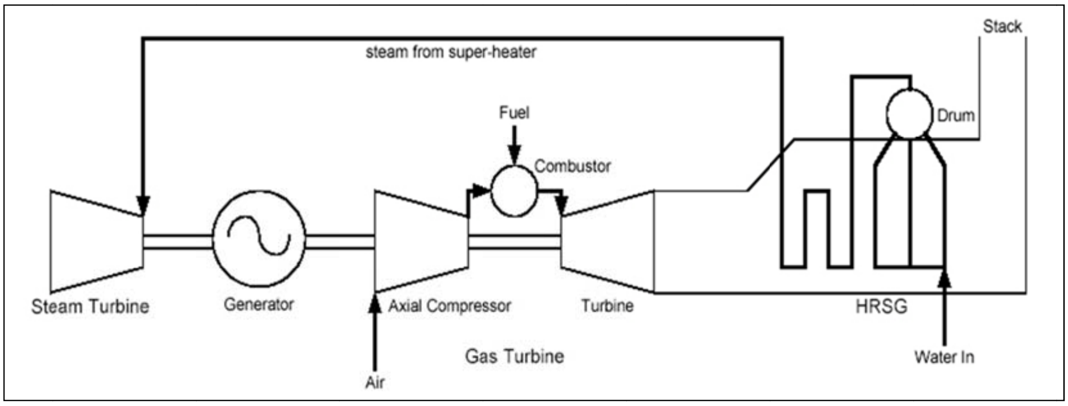


Figure 2.17: Single-shaft unit with generator between turbines [1].

2.5.5.2 MULTI-SHAFT UNITS

In multi-shaft units, the gas turbine and the steam turbine are placed on separate shafts, with one generator connected to each turbine in the system [1], [24].

In multi-shaft units, one or more gas turbines are installed, each with its own heat recovery steam generator, feeding the steam to the steam turbine. The steam turbine efficiency is increased by the combination of the steam generation from all the heat recovery steam generators, as a larger steam volume enters the steam turbine. For smaller units, the exhaust gases from all the gas turbines are combined, feeding one heat recovery steam generator [1], [24].

Different configurations are possible and Figure 2.18 and Figure 2.19 show the configurations:

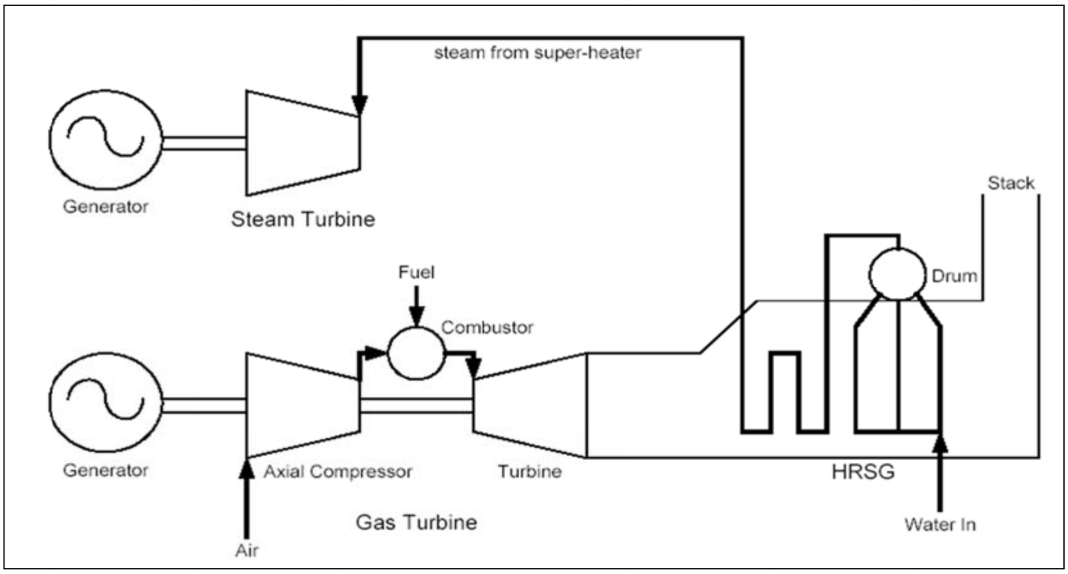


Figure 2.18: Multi-shaft combined cycle power plant with a single gas turbine [1].

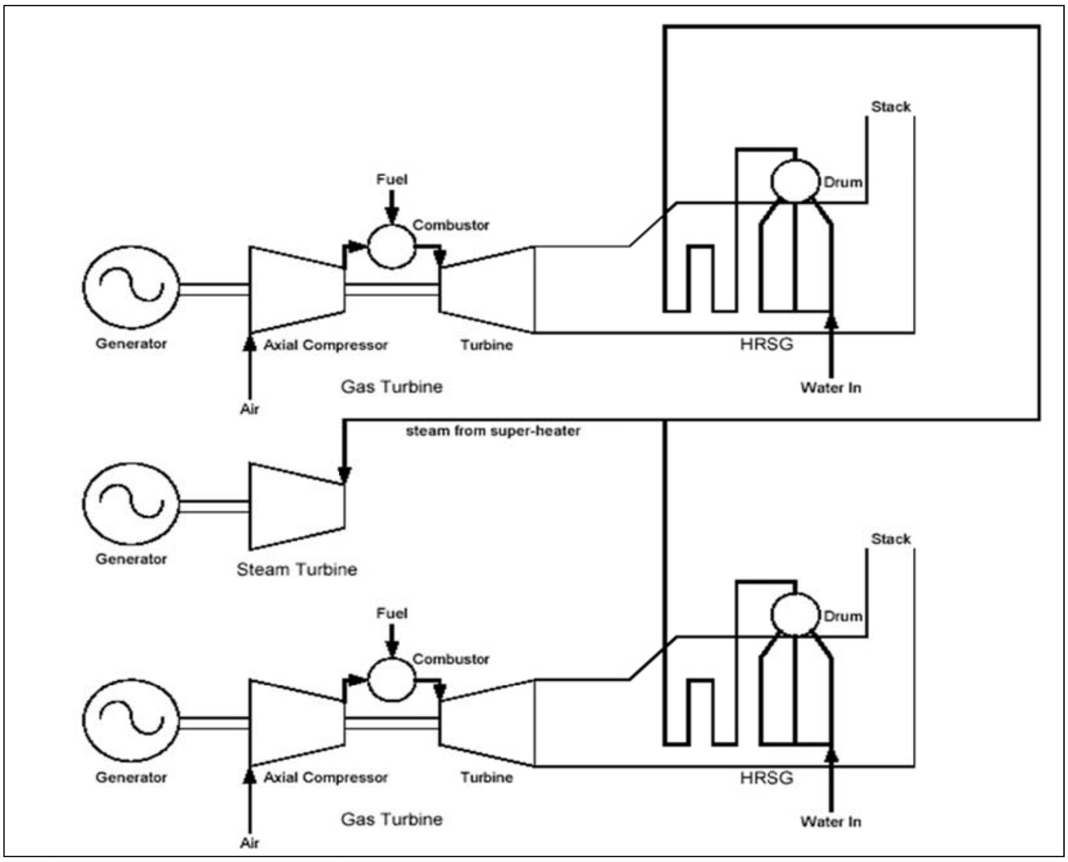


Figure 2.19: Multi-shaft combined cycle power plant with two gas turbines [1].

2.5.6 CONTROLS OF COMBINED CYCLE POWER PLANTS

2.5.6.1 PLANT CONTROLS

The controls of combined cycle power plants are very complex [1]. For the purpose of this dissertation, only the control loops that directly affect the response of the power plant to electrical system disturbances will be discussed.

The main plant control system is responsible for the load control and frequency response of the combined cycle power plant [1]. A load set point is set for the overall plant load control system and the control system determines the loading of the gas turbine. The steam turbine is normally operated in sliding pressure mode, i.e. with steam pressure down to 50% of rated value and the steam turbine valves fully open. With no supplemental firing, the gas turbine controls the total power output of the plant. The steam turbine follows the gas turbine and will generate power with whatever amount of steam is available from the heat recovery steam generator.

After a load change on the gas turbine, the amount of steam available from the heat recovery steam generator will change (with a few minutes delay) and the steam turbine will follow. When combined cycle power plants operate near full load capability, it is normal practice to operate the steam turbine in sliding pressure mode. For smaller loads (i.e. 30% to 50% of full load capability), the valves can be controlled in the partially open position to maintain steam pressure [1].

An important aspect of the load and frequency control of a combined cycle power plant is the ability to react to rapid fluctuations in the network frequency. Load change response of the combined cycle power plant usually takes place over several minutes, while frequency response must occur within seconds. It is important to control the system frequency within limits, as over- and under-frequency conditions could damage equipment. The balance between supply and demand directly influences the system frequency. Speed governors are used to control the system frequency.

To maintain stable operation and to extend the life of the gas turbines, a frequency dead-band in the control system can be set [1]. Within this dead-band, the plant will not respond to small frequency changes. Outside the dead-band, a droop setting is used. The droop setting is in the range of 3% to 8%, with a typical setting of 4% to 5% [1]. Combined cycle power plants can be operated to supply frequency support, i.e. having spinning reserve. To achieve this, the gas turbines are operated at between 40% and 95% of rated load, resulting in the proportional partial loading of the steam turbine.

2.5.6.2 GAS TURBINE CONTROLS

The typical controls of gas turbines in combined cycle power plants are shown in Figure 2.20. Separate start-up and shutdown control loops ramp the unit up and down during start-up and shutdown [1]. The controls ensure proper purging (cleaning) of the gas paths, establishing the flame, controlling the acceleration of gas turbine and the warming of the hot gas paths before allowing the loading of the gas turbine. These control loops are not pertinent to power system analysis, and are therefore not considered in the models applied in this dissertation.

The acceleration control is active during start-up and the shutdown sequence of the turbine. With the unit on-line, the acceleration set point is typically set around $1\%/s^2$. This amount of acceleration is unlikely in large inter-connected systems, but possible in small or islanded systems. For power system studies, the acceleration control may be ignored. However, it may be considered for islanding studies, in smaller systems, and specifically in the case of aero-derivative units [1].

Combustion within the gas turbine is complex and falls outside the scope of this dissertation. Only the major concepts will be discussed here. The key challenges are [1]:

- To maintain a stable flame over a wide range of fuel / air ratios from no-load to full-load conditions, at full speed (or system rated frequency),
- Emission control – CO, NO_X, SO_X, unburnt hydro-carbons and smoke,
- To maintain structural integrity of the combustion chamber and components over the expected life of the unit,
- To maintain temperature within design ranges and to prevent the thermal over-stressing of the turbine materials.

Flame stability in the combustion chambers is an important design goal. However, it is too complex to model and is not relevant to power system analysis [1]. Problems that were experienced with earlier designs were a sudden flameout due to sudden abrupt control commands to decrease fuel flow.

Control of emissions is also important [1]. Various processes can be applied that include steam injection, water injection, selective catalytic reduction or dry-low NO_X combustion. Dry-low NO_X combustion involves moving through multiple modes of combustion until a pre-mix mode is reached. In pre-mix mode, most of the fuel is pre-mixed with air to obtain a lean fuel / air mixture with the consequence that peak flame temperatures are lower, providing a reduction in the formation of NO_X. The modelling of these processes is not important at a power system analysis level, but it is important to remember that the injection of water and steam tend to increase turbine output, but decrease thermal efficiency.

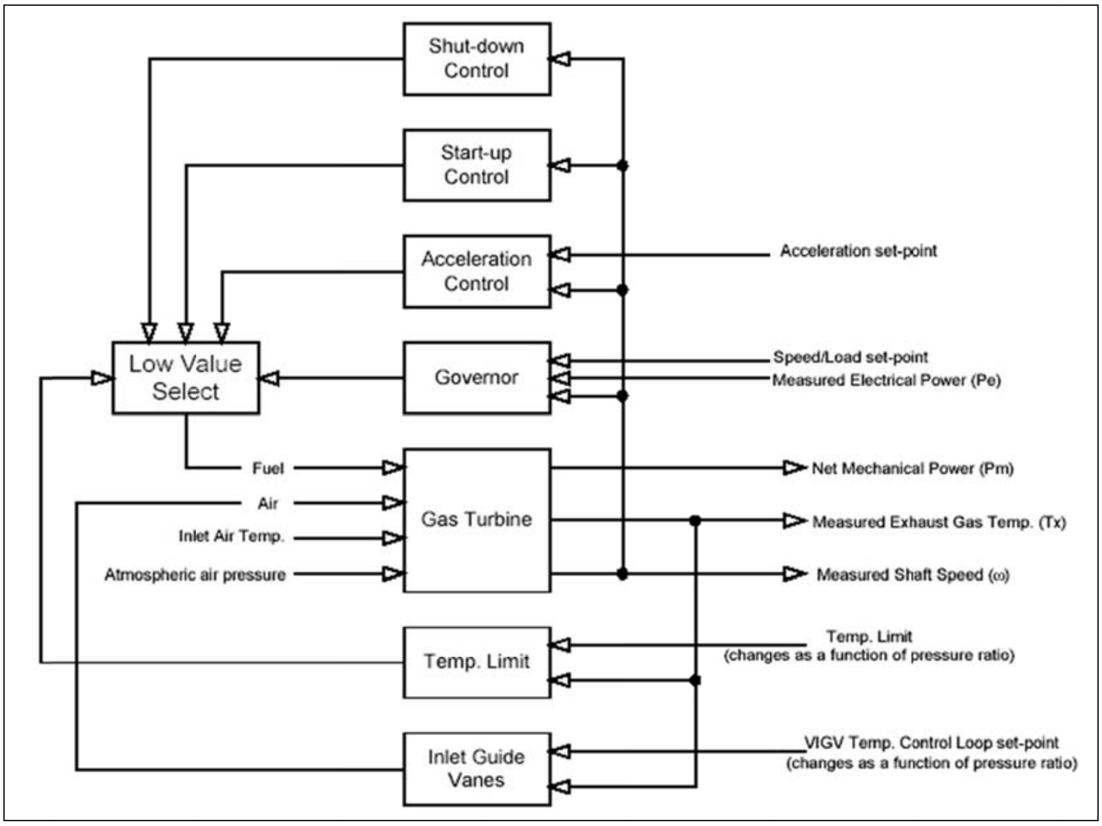


Figure 2.20: Diagrammatic gas turbine control diagram [1].

The speed, loading and temperature limit controls of the unit are of particular importance to power system analysis. The maximum output power of the gas turbine is very dependent on the deviation of its operating frequency from the rated frequency (or speed of the gas turbine), and the ambient conditions in which the gas turbine operates. The gas turbine's maximum operating temperature also determines the maximum output power of the gas turbine. In addition, the inlet guide vane control's purpose is to maintain good steam conditions over the load range by maintaining the gas turbine exhaust temperature [1].

2.5.6.3 STEAM TURBINE CONTROLS

As mentioned, the steam turbine in combined cycle power plants is normally operated in sliding pressure control mode [1]. When the control valves are fully open, the output power from the steam turbine is a function of the steam generated from the heat recovery steam generator, which is a function of the gas turbine loading. At reduced loads, (30% to 50%), the control valves could be partially closed to maintain steam pressure.

In general, the steam turbine does not respond fast to system disturbances, such as a drop in frequency [1]. Governor action will try to maintain system frequency, the result will be the opening or closing of control valves. To prevent operation for small system frequency changes, a dead-band can be set.

Operating the steam turbine in sliding pressure mode (control valves fully open), results in a better overall plant efficiency for the combined cycle power plant, compared to other modes of operation whereby the control valves are regulated [1].

The control of the steam system involves many control loops similar to conventional steam plants [1]. Feed-water levels and temperature need to be controlled. The boiler controls are simpler as the heat source is not controlled, but the steam generated is a function of the gas turbine output.

Additional controls are required for units that have supplemental firing. The controls are used to control the amount of additional heat input.

Another control mode is the speed / load control mode [13]. A speed / load control system should be capable of controlling the speed (or frequency) of the turbine from zero to maximum power output, when the generator is operated in isolation or in parallel with other generators. The control must also be stable when controlling the turbine. In order to be stable, the variation of speed must be below 0.12% for generators below 2 MW, below 0.10% for generators from 2 MW to 5 MW and below 0.07% for generators above 5 MW [13].

Due to the characteristics of steam turbines, the reaction time of the turbine to changes / disturbances in the power system is relative slow. However, governor action to control the speed (or frequency) is fast in order to maintain system frequency. Changes in the input of heat into the boiler do not immediately change the output of the steam turbine. This process usually has a very long time constant.

2.6 SUMMARY

In this chapter, a description of the characteristics of combined cycle power plant equipment was provided. This gives insight into the characteristics of the equipment to allow a better understanding of the dynamic modelling of this equipment in power system simulation software.

Steam and gas turbines and combined cycle power plants were discussed in this chapter. Various generation sources are available, but in this chapter only the steam turbine, the gas turbine and the heat recovery steam generator were studied.

References to the types of gas turbines were made. Mainly three classifications of gas turbines exist, namely the aero-derivative, the light industrial and the heavy-duty industrial gas turbine. They differ mainly in size, but other differences also occur. It is important to understand that each component plays a role in power system studies. Components of the gas turbine that are not important to power system studies were excluded.

Each of the main components of the combined cycle power plants was studied to understand their unique characteristics before the characteristics of combined cycle power plants could be understood.

- Chapter 3 -

3 THEORY OF MODELS

3.1 BACKGROUND

The previous chapter discussed the main components of combined cycle power plants, i.e. the gas turbine, the steam turbine and the heat recovery steam generator. The various configurations, i.e. single-shaft and multi-shaft systems were discussed. A short description of the controls of the combined cycle power plant was included.

In this chapter, the building blocks of the main components of the power plant are discussed and their performance assessed. The correct combination of the building blocks forms dynamic models for various power plants.

3.2 INTRODUCTION TO MODELS

Dynamic models are designed from various building blocks, or functions, each with specific characteristics. By combining the building blocks in a single model, the various characteristics are combined, and the influences from the blocks on the other blocks are modelled. The building blocks represent the actual hardware as accurately as possible.

In order to understand the characteristics of a dynamic model, this chapter provides background on the Laplace transform, as well as some of the available building blocks used to build a dynamic model. Only some of the available building blocks are discussed, as it is not possible to include all the possible variations of each block.

3.3 LAPLACE TRANSFORMATION

This section provides a definition of the Laplace transformation. The Laplace transform is used to represent (or transform) a time function in the frequency domain.

3.3.1 DEFINITION

The Laplace transform pair is defined [14] as:

$$\mathscr{L}\{f(t)\} = F(s) = \int_{-\infty}^{\infty} f(t)e^{-st}dt \tag{3.2.1}$$

$$\mathscr{L}^{-1}\{F(s)\} = f(t) = \frac{1}{2\pi j}\int_{\sigma - j\omega}^{\sigma + j\omega} F(s)e^{st}ds \tag{3.2.2}$$

where $\mathscr{L}\{f(t)\}$ denotes the Laplace transform of the time function $f(t)$, and $\mathscr{L}^{-1}\{F(s)\}$ denotes the inverse Laplace transform, s is a complex variable whose real part is σ, and imaginary part ω, that is $s = \sigma + j\omega$.

In most problems we are concerned with values of time t greater than a reference point, for example $t \geq t_0 = 0$. Since the initial conditions are known, the Laplace transform simplifies to:

$$\mathscr{L}\{f(t)\} = F(s) = \int_{t_0}^{\infty} f(t)e^{-st}dt = \int_{0}^{\infty} f(t)e^{-st}dt \qquad (3.2.3)$$

The Laplace transform only has meaning if it converges, that is if:

$$\left| \int_{0}^{\infty} f(t)e^{-st}dt \right| < \infty \qquad (3.2.4)$$

3.4 BUILDING BLOCK DESCRIPTION

This section gives short descriptions of the building blocks that make up a dynamic model. Refer to Appendix B for the description on the basic building blocks.

3.4.1 FIRST ORDER LAG

Many transducers and input signals pass through first order lag functions to reduce the noise and condition signals before they are used as inputs in controllers and feedback systems. They are therefore an integral part of a dynamic model.

A first order lag function is a low pass filter. A low pass filter removes the high frequency components from the input signal. The output will therefore only contain the low frequency components of the input signal. The first order lag function causes a phase lag between the output and the input for the higher frequencies. For the lower frequencies, no significant phase lag between the output and the input exists. Figure 3.1 shows the function.

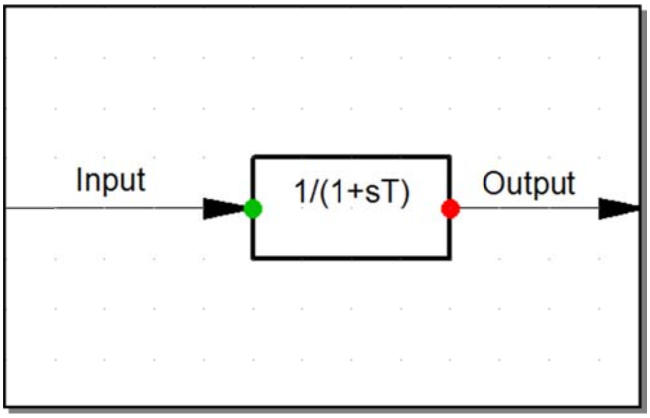
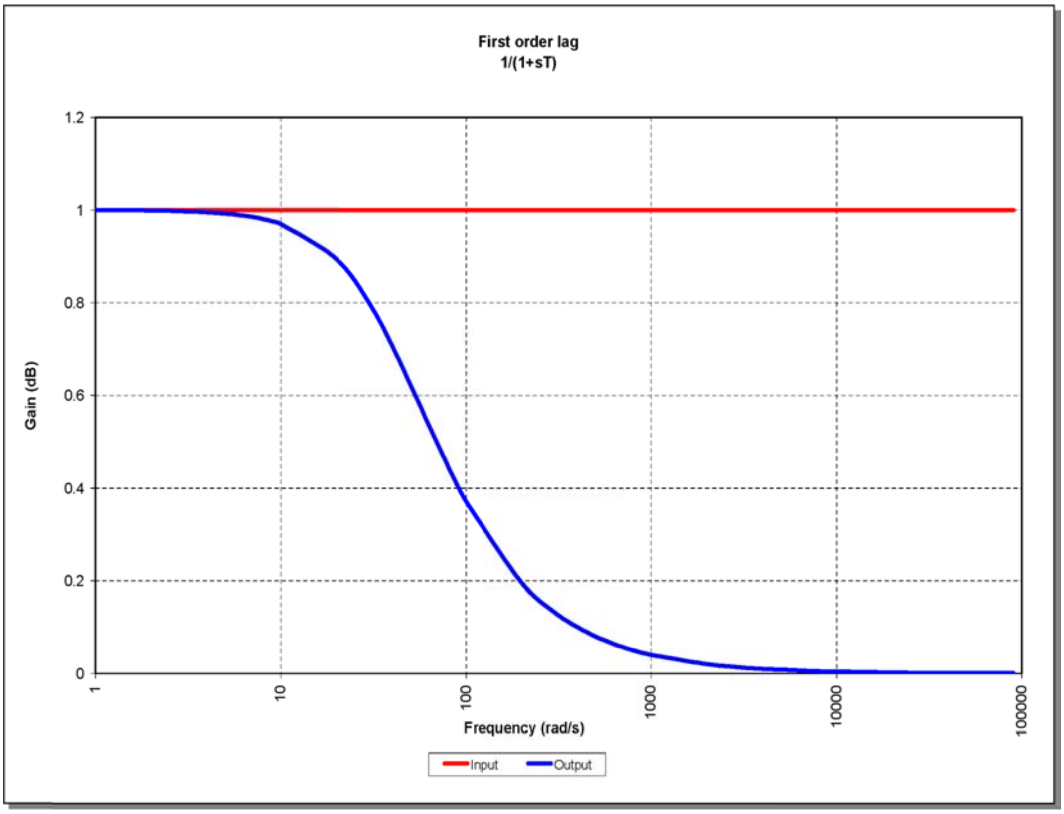


Figure 3.1: First order lag function block

Figure 3.2 shows a graphical representation of the relationship between the input and output signal and the phase shift of the output signal. The first graph shows the low pass filter characteristic. For low frequencies, the output signal is equal to the input signal. For higher frequencies, the output drops to zero. The second graph shows the phase shift of the output signal. For low frequencies, the phase shift is very small but change to -90 ° for higher frequencies.



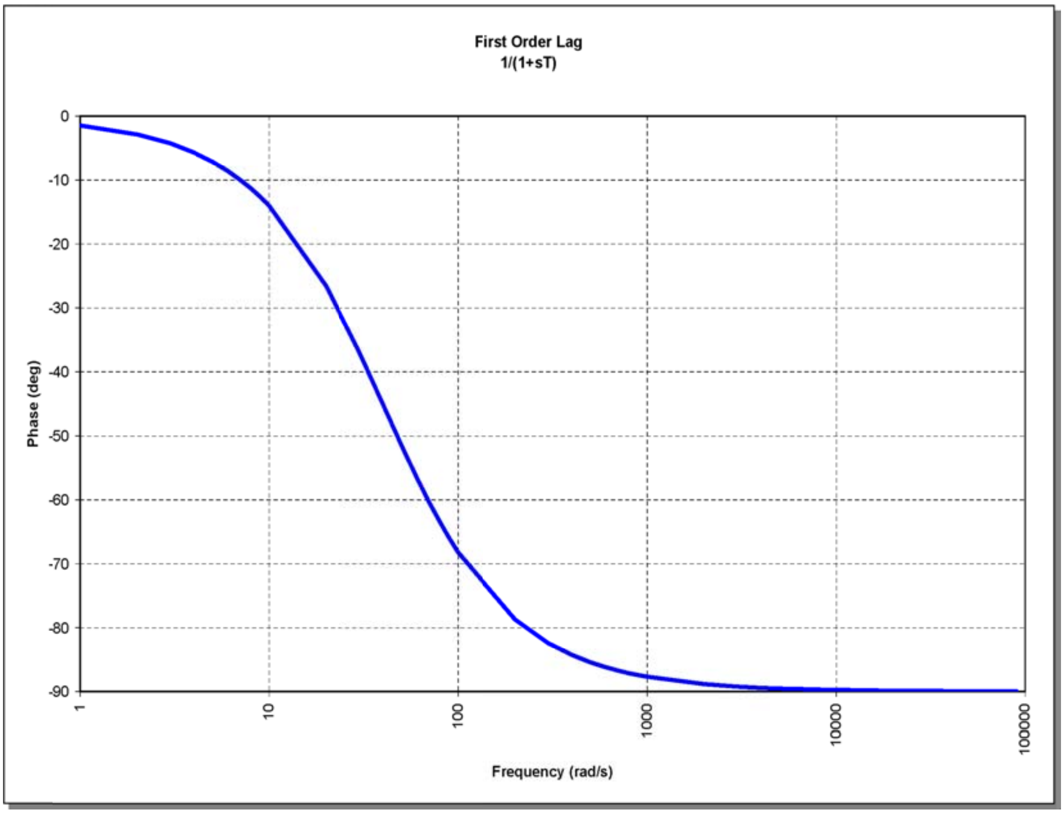


Figure 3.2: First-order lag function block input / output and phase shift relationship

3.4.2 SELECTION GATE

Selection gates are used to determine which controller controls a system with multiple possible control loops. A selection gate is used to select between the input signals. Two possible variations are available. They are high value and low value selection gates. The high value selection gate will select the input signal that has the highest value and marshal it to the output. A low value selection gate will produce the output from the input signal with the lowest value.

The output of the high value selection gate is given by the following expression:

Output of high value selection gate = maximum (Input1, Input 2).

Figure 3.3 shows a low value selection gate. The output of the low value selection gate is given by the following expression:

Output of low value selection gate = minimum (Input1, Input 2).

For both the high value and the low value selection gates, more inputs can be connected.

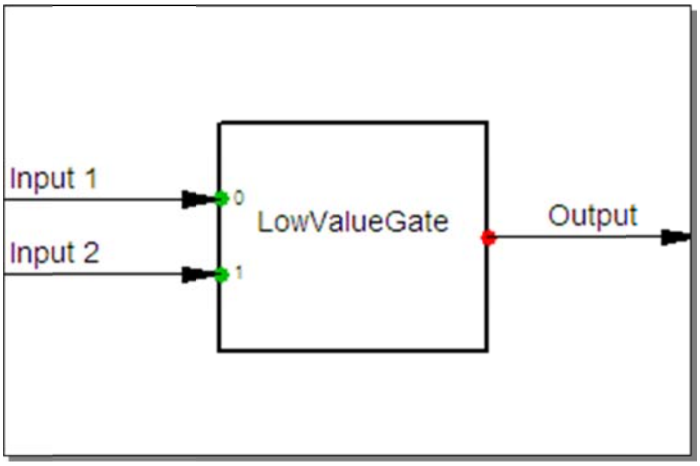


Figure 3.3: Low Value Selection Gate

3.5 PID CONTROLLER

The PID (Proportional, Integral and Derivative) controller algorithm is the most commonly used control algorithm in industrial applications. It consists of three separate controllers [4], [16], [20]:

- Proportional
- Integral
- Derivative

The PID control algorithm does not “know” what the correct output is; it merely moves the process towards the set point, until the process reaches the set point. The algorithm therefore has a feedback loop, i.e. it measures the process. If the loop is not closed, that is if the feedback path between the output and the input is broken or limited, the algorithm does not “know” what the output should be. Under these conditions, the output of the algorithm is meaningless. In order to setup the PID controller correctly, it is important to understand each of the three equations / functions of the PID controller. Two actions are possible with a PID controller, the first is a “direct action” and the second is a “reverse action”. If a controller is direct acting, an increase in its input will result in an increase in its output. With reverse action, an increase in input will result in a decrease in output. The controller action is always the opposite of the process action [4], [16], [20].

In the next sections, each of the three equations is discussed separately with advantages and disadvantages. The three equations use a value called the “error”. The error is the difference between the set point and the feedback signal from the process. If the controller is direct acting, the measurement is subtracted from the set point, while for the reverse acting controller the set point is subtracted from the measurement [4].

Error = set point – measurement (direct action)
Error = measurement – set point (reverse action)

3.5.1 PROPORTIONAL CONTROLLER

The first part of the PID controller is the proportional controller. In its basic form, the output of the controller is the error times the gain. Figure 3.4 shows the proportional control loop. An example of a proportional or gain controller is an amplifier. Another example of a proportional controller is an actuator.

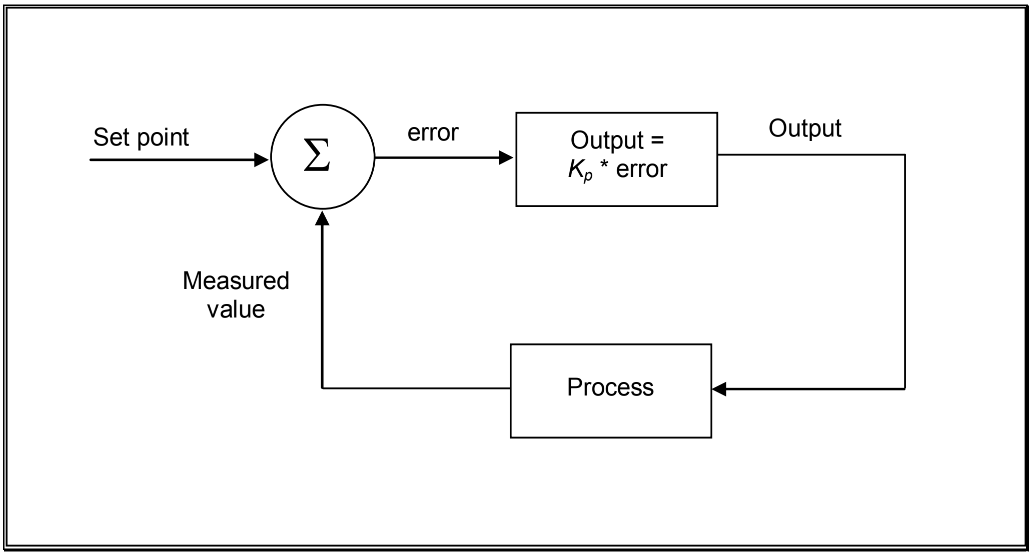


Figure 3.4: Proportional control loop

The output is equal to the error multiplied by the proportional gain (K_p). A change in the process measurement or the set point will cause the output to change. The change of the output is proportional to the current error value, i.e. the error is multiplied by a constant K_p that is the proportional gain [4], [16], [20].

The equation describing the action of the proportional controller is given by:

$$P_{out} = K_p \cdot e(t) \tag{3.5.1}$$

With:

- P_{out}: Output value of the proportional controller
- K_p: Proportional gain, a settable parameter
- $e(t)$: Error function = Set point – measured value

If the proportional gain is a high value, it can result in a large change in the output for a given change in the error. If the proportional gain is too high, the system can become unstable. In contrast, a small gain results in a small output response to a large input error, but results in a less responsive (or sensitive) controller. If the proportional gain is too low, the control action may be too small when responding to system disturbances.

In the absence of disturbances, pure proportional control will not settle at its set point value, but will retain a steady state error that is a function of the proportional gain. Despite the steady-state offset, both the tuning theory and industrial practice indicate that the proportional controller contributes to the bulk of the output change in a PID controller. It is therefore important to set this proportional gain as accurately as possible. Larger values typically mean faster response of the controller, since the larger the error, the larger the proportional controller compensation. An excessively large proportional gain will lead to process instability and oscillation of the controller output.

An additional expression to the proportional gain is often used to describe the proportional action of the controller. This expression is called the "Proportional Band" (PB) [16]. The expression is

$$PB[\%] = \frac{100\,\%}{K_p} \qquad (3.5.2)$$

The proportional band is defined as the error required (as a percentage of full scale) to give a 100% change in the controller's output [16].

3.5.2 INTEGRAL CONTROLLER

The second part of the PID controller is the integral controller. The integral controller gives an output that is proportional to the accumulated error. This implies that this is a slow reaction control mode. This characteristic is evident in its low pass frequency response. The integral controller plays a fundamental role in achieving perfect plant inversion at $\omega = 0$. This forces the steady state error to zero in the presence of a step reference and disturbance. The integral controller (when viewed in isolation) has two shortcomings: its pole at the origin is detrimental to loop stability and it gives rise to an undesirable effect called wind-up (in the presence of actuator saturation). Wind-up occurs when the integrator continues to integrate while the input is constrained. The integrator can reach an unacceptably high value that can lead to poor transient response [16].

The equation describing the action of the integral controller is given by:

$$I_{out} = K_i \int_0^t e(t) \cdot dt \qquad (3.5.3)$$

With:

- I_{out}: Output value of the Integral controller
- K_i: Integral gain, a settable parameter
- $e(t)$: Error function = Set point – measured value

The purpose of the integral controller (when added to the proportional controller) is to accelerate the process output value towards the set point and eliminate the residual steady-state error that occurs with a proportional controller. However, since the integral controller is responding to accumulated errors from the past, it also has the characteristic that the present value can overshoot the set point value. Larger values of K_i imply that steady state errors are eliminated more quickly. However, the trade-off is a larger overshoot: any negative error integrated during transient response must be integrated away by a positive error before the controller reaches steady state [4], [16], [20].

3.5.3 DERIVATIVE CONTROLLER

The third part of the PID controller is the derivative controller. The derivative controller gives an output that is proportional to the rate of change of the error. This implies that this is a fast reaction control mode, which ultimately disappears in the presence of constant errors. It is also referred to as a predictive mode because of its dependence on the trend of the error. The main limitation of the derivative controller (when viewed in isolation), is its tendency to yield large control signals in response to high frequency errors, such as errors induced by set point changes or noise induced by measurements [4], [16], [20].

The equation describing the action of the derivative controller is given by:

$$D_{out} = K_d \cdot \frac{de(t)}{dt} \tag{3.5.4}$$

With:

- D_{out}: Output value of derivative controller
- K_d: Derivative gain, a settable parameter
- $e(t)$: Error function = Set point – measured value

The purpose of the derivative controller is to slowdown the rate of change of the PID controller output and this effect is most noticeable close to the PID controller set point. Therefore, derivative control is used to reduce the magnitude of the overshoot produced by the integral controller. This improves the PID controller stability. However, differentiation of a signal amplifies noise and thus the derivative controller is highly sensitive to noise in the error, and can cause a process to become unstable if the noise and the derivative gain are sufficiently large. Careful setting of K_d is required. Larger values decrease overshoot, but slow down the transient response of the controller. This may even lead to instability due to signal noise amplification in the differentiation of the error [4], [16], [20].

Due to the sensitivity of the derivative controller to noise, a low-pass filter is introduced to filter the measurements, in order to remove higher frequency noise components. However, low-pass filtering and derivative control can cancel each other out, so reducing noise by instrumentation provides a better choice. Alternatively, the derivative controller of the PID controller can be turned off with little loss in overall control in many systems. This is equivalent to using the PID controller as a PI controller [4], [16], [20].

3.5.4 COMPLETE PID CONTROLLER

Following the discussion of the separate controllers of a PID controller, this section discusses the complete PID controller. The mathematical expression that describes a PID controller is [4], [16]:

$$\begin{aligned} PID_{out} &= P_{out} + I_{out} + D_{out} \\ &= K_p \cdot e(t) + K_i \int_0^t e(t) \cdot dt + K_d \frac{de(t)}{dt} \end{aligned} \tag{3.5.5}$$

The Laplace transform for Equation 3.5.5 is:

$$F(s) = K_p + \frac{K_i}{s} + K_d s \tag{3.5.6}$$

Figure 3.5 shows a block diagram of a PID controller.

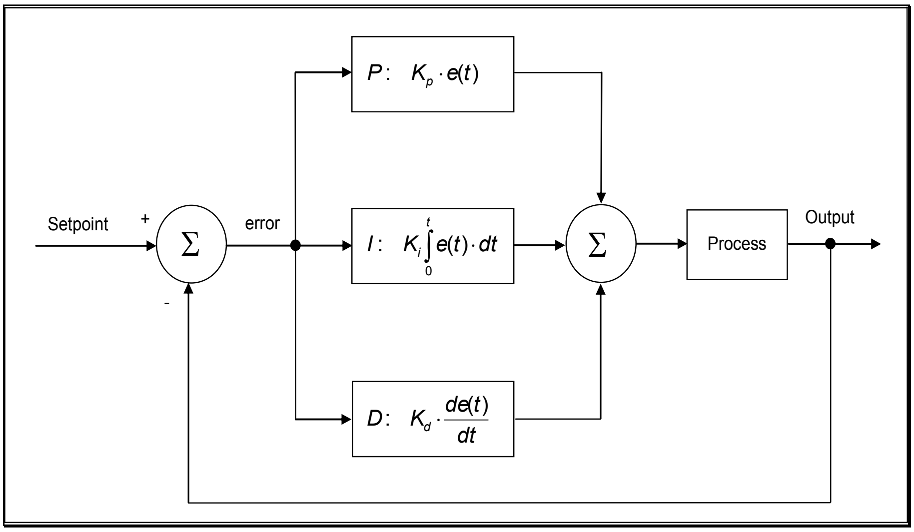


Figure 3.5: PID Controller

The proportional controller (P) determines the reaction to the current error, the integral controller (I) determines the reaction based on the sum of the recent errors and the derivative controller (D) determines the reaction based on the rate at which the error has been changing. The weighted sum of these three controllers is then used to adjust the process via a control element, for example the position of a control valve or the power supply of a heating element [4].

By setting the three constants (K_p, K_i and K_d) of the PID controller, the controller can provide a controlled action designed for specific process requirements. The response of the controller is described in terms of the responsiveness of the controller to the error, the overshoot reaction of the controller to the set point and the degree of system oscillation. It is important to note that a PID controller does not guarantee optimal control of a system or system stability [4].

3.5.5 PID LIMITATIONS

Proportional control is the main method of control in the PID controller. It calculates the control action proportional to the error, and can be used on its own, but proportional control cannot eliminate the error completely [21].

Integral control is the means to eliminate the remaining error, which is left from the proportional action. However, this may result in reduced stability in the control action [21].

Derivative control is sometimes added to introduce dynamic stability to the control loop. Derivative control has no functionality on its own [21].

The available combinations for the Proportional (P), the Integral (I) and the Derivative (D) controllers are as follows [21]:

- P: For use a basic controller,
- PI: The Integral controller action eliminate the remaining error of the Proportional controller,
- PID: To remove instability problems that can occur with PI controllers,
- PD: Used in cascade control, a special application,
- I: Used in the primary controller of cascaded systems.

3.5.6 PID CONTROLLER RESPONSE

The following graphs (Figure 3.6 to 3.8) shows the response of the PID controller for various values of the separate controller gains K_p (proportional), K_i (integral) and K_d (derivative). The graphs are shown for different values of the each specific controller gain, while the other controller gains are kept constant.

Figure 3.6 shows a PID controller with the three gains K_p (proportional), K_i (integral) and K_d (derivative). K_p is varied, while K_i and K_d are kept constant. For smaller values of K_p, larger overshoot occurs, the time to the peak of the overshoot takes longer and the settling time is longer, than for larger values of K_p.

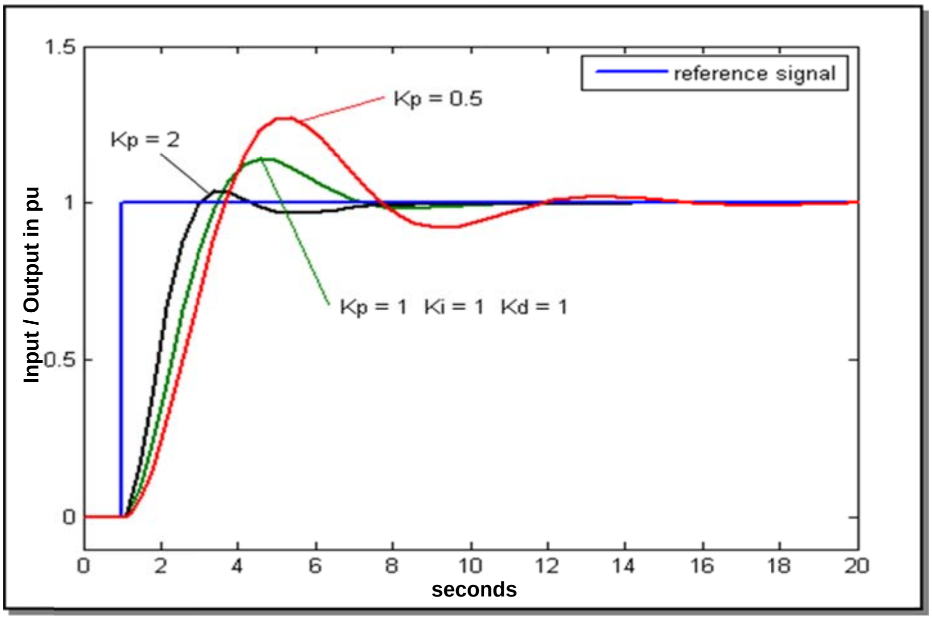


Figure 3.6: PID Controller – Input and output vs. time (adjustments to the proportional gain)

Figure 3.7 shows a PID controller with the three gains K_p (proportional), K_i (integral) and K_d (derivative). K_i is varied, while K_p and K_d are kept constant. For larger values of K_i, larger overshoot occurs (but for small values no overshoot occurs), the time to the peak of the overshoot is slightly faster and the settling time is longer, than for smaller values of K_i.

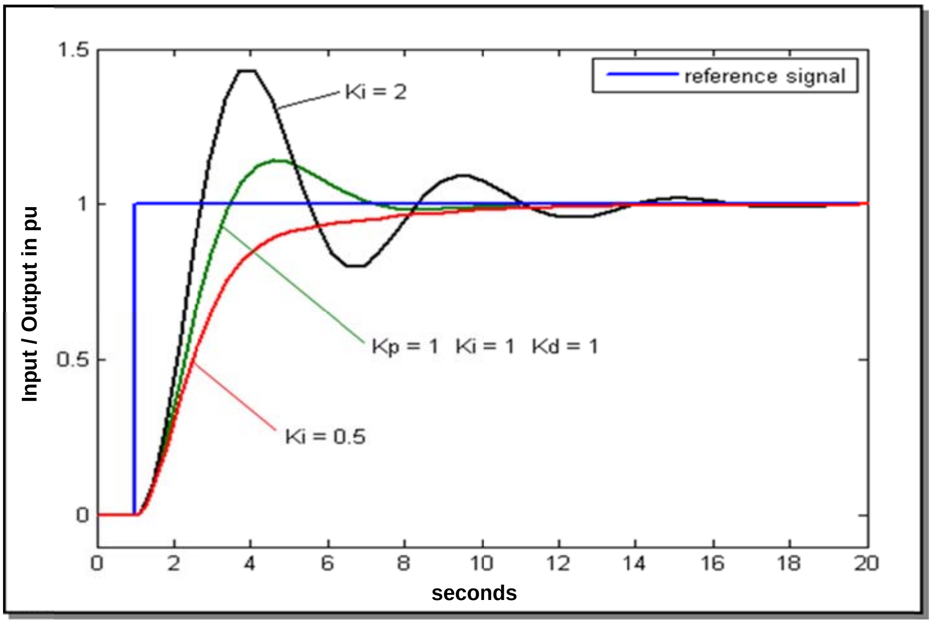


Figure 3.7: PID Controller – Input and output vs. time (adjustments to the integral gain)

Figure 3.8 shows a PID controller with the three gains K_p (proportional), K_i (integral) and K_d (derivative). K_d is varied, while K_p and K_i are kept constant. For different values of K_d, the overshoot are approximately the same, the time to the peak of the overshoot is faster and the settling time is shorter, than for larger values of K_d.

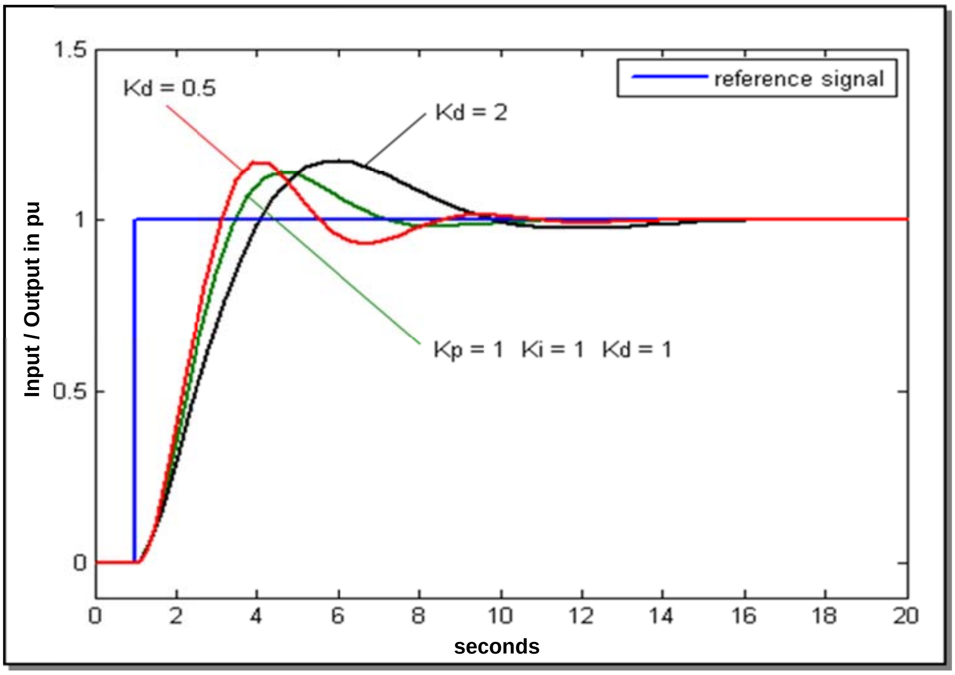


Figure 3.8: PID Controller – Input and output vs. time (adjustments to the derivative gain)

3.6 LEAD-LAG COMPENSATOR

3.6.1 INTRODUCTION

The lead-lag compensator is included here as it is a very commonly used component in control systems for industrial applications.

3.6.2 LEAD-LAG COMPENSATOR

A lead-lag compensator is a component in a control system that improves the frequency response of the system by adjusting the response in a particular frequency band. It is a fundamental block in control theory [16], [17]. The transfer function of the lead-lag compensator is of the form:

$$C(s) = \frac{\tau_1 s + 1}{\tau_2 s + 1} \qquad (3.6.1)$$

When $\tau_1 > \tau_2$, this is a lead network, and when $\tau_1 < \tau_2$, this is a lag network [16].

The lead compensator acts like an approximate derivation [16], [17]. The straight-line approximations to the Bode diagrams are given in Figure 3.9 and Figure 3.10. (with $\tau_1 = 10 \times \tau_2$, $\tau_2 = 1$). The graphs (Figure 3.9 and Figure 3.10) show that the lead-lag compensator produces an approximately 45° (0.785 rad) of phase advance at $\omega = 1/\tau_1$ (i.e. $\omega = 0.1$) without a significant change in gain. For low frequencies, the gain is lower than for high frequencies. The disadvantage is that an increase in the high frequency gain could amplify high frequency noise [16].

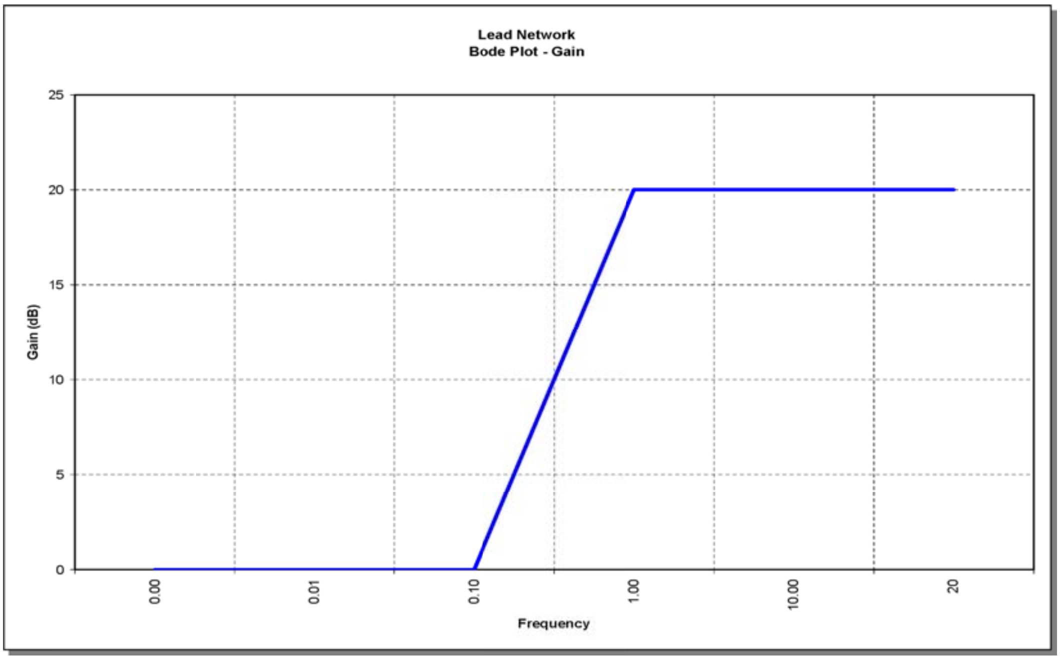


Figure 3.9: Approximate Bode diagrams for lead networks (Gain) [16]

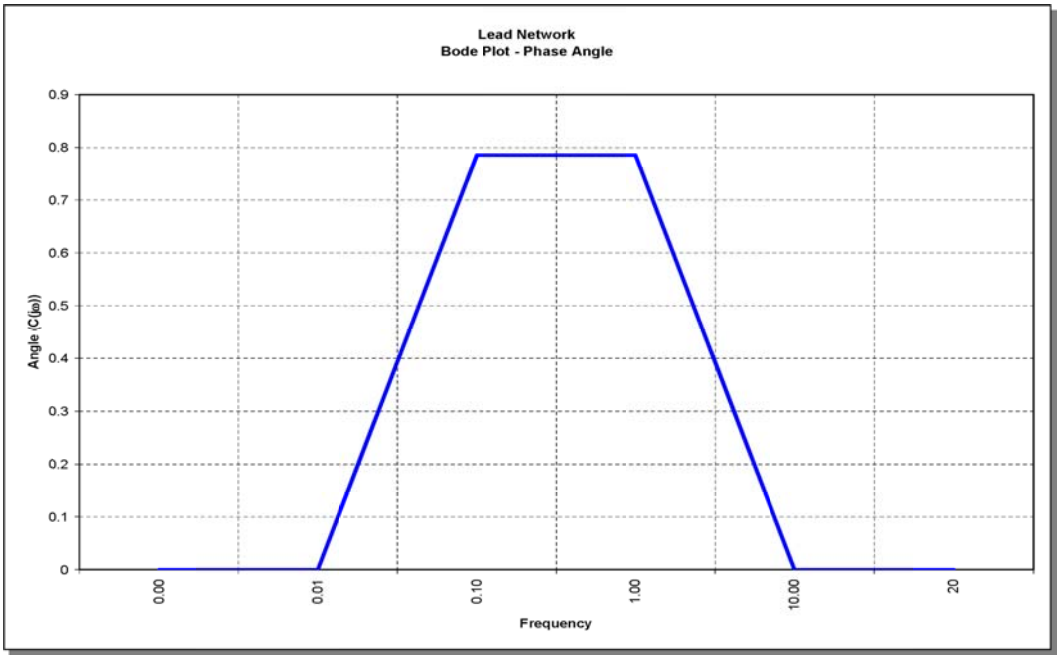


Figure 3.10: Approximate Bode diagram for lead networks (Angle) [16]

An alternative interpretation of a lead network is obtained by considering its pole-zero structure [16]. From (3.6.1) and with $\tau_1 > \tau_2$ we can see that it introduces a pole-zero pair, where the zero (at $s = -1/\tau_1$) is significantly closer to the imaginary axis than the pole (at $s = -1/\tau_2$) [16].

On the other hand, a lag compensator acts like an approximate integrator. For low frequencies, the gain is higher than for high frequencies. This controller will give better low-frequency tracking and disturbance rejection when used in a feedback loop. A disadvantage is the additional phase lag experienced between 1/10 x τ_2 and $10/\tau_1$ [16].

From the point of view of its pole-zero configuration, the lag network introduces a pole (located at s = -1/ τ_2). That is significantly closer to the imaginary axis than the zero (located at s = -1/ τ_1) [16].

The straight-line approximations to the Bode diagrams are given in Figure 3.11 and Figure 3.12. (with $\tau_2 = 10 \times \tau_1, \tau_2 = 1$).

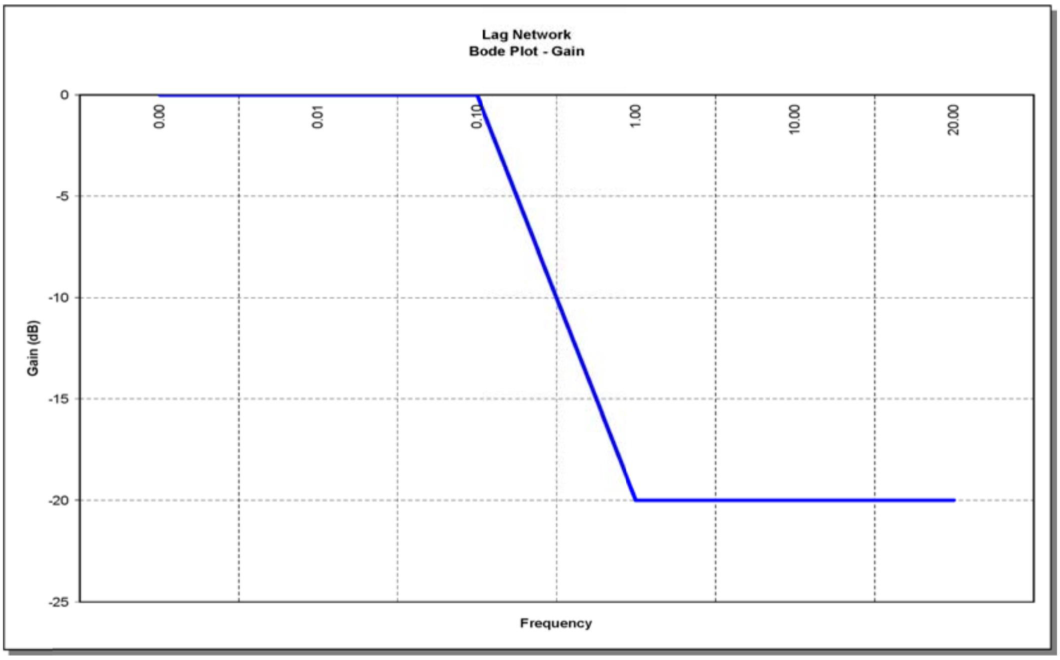


Figure 3.11: Approximate Bode Diagram for lag networks (Gain) [16]

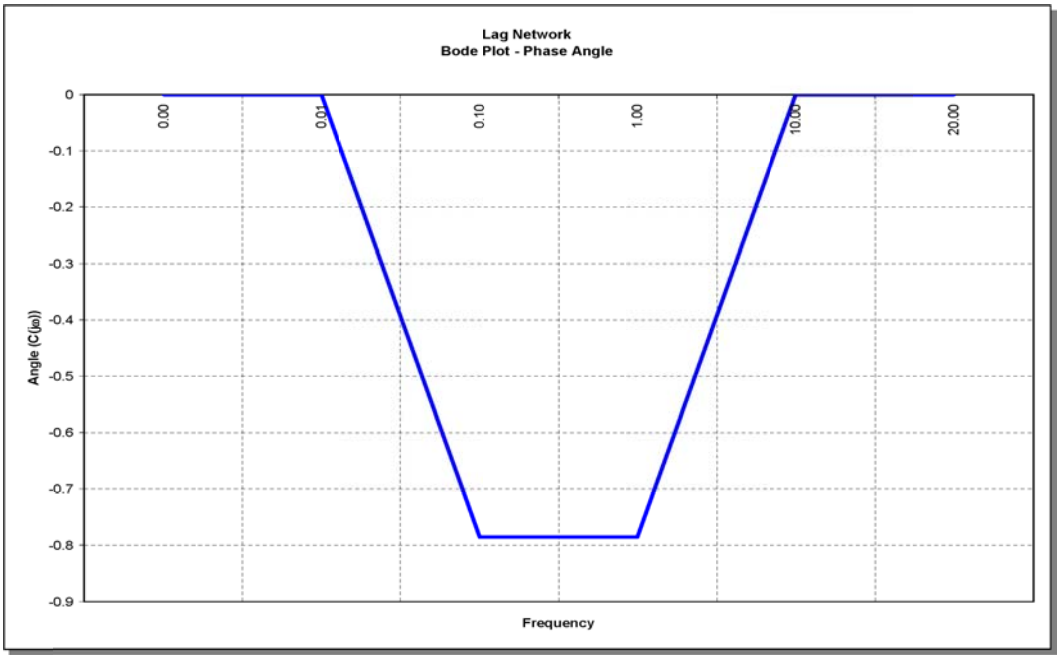


Figure 3.12: Approximate Bode diagram for lag networks (Angle) [16]

3.7 PID CONTROLLER VERSUS LEAD-LAG COMPENSATOR

3.7.1 PID CONTROLLER

The PID controller uses three separate functions in its control strategy to control a process. They are proportional, integral and derivative control. All three the functions together aim to minimise / eliminate the difference between the output signal and the set point. All three functions therefore act on the error signal (error is the difference between the output and the set point). However, the proportional function only amplifies the error; the integral function integrates the accumulated error over time, while the derivative function determines the rate of change over time of the error.

3.7.2 LEAD-LAG COMPENSATOR

The feature of a lead component of a controller that is used in control design is its phase-advance characteristic, while the feature of a lag component of a controller that is used in control design, is its gain characteristic at low frequencies. The lead-lag components of a controller are used to improve the frequency response of the system in key areas.

3.8 SIMULATION OF POWER SYSTEMS

The previous sections discussed the theory of models and included two of the most commonly used control algorithms in control systems namely the PID controller and the lead-lag compensator. This section describes the simulation of power systems, and for dynamic studies, it is important to model the control systems.

The purpose of simulating power systems is to model the real system without having the real system. It is important to obtain the relevant data for the real equipment, in order to model the real power system accurately. The methodology is to build the power system network in the simulation software and to define each element correctly. When the model is built and validated, system studies can be performed on the model. Various studies can be performed, including load flow, fault level, harmonic, dynamic stability and a variety of other studies. For the purpose of this dissertation, the emphasis is on dynamic stability studies. Before the dynamic stability studies can be performed, it is important to determine if the power system is stable, before the application of any disturbance. This is done by the correct initialisation of the system in steady state, and analysing the stability of the system. The type of the disturbance then determines if small signal stability or large signal stability studies are performed.

3.8.1 INITIAL CONDITIONS

The purpose of dynamic models is to model the characteristics of power system elements during dynamic conditions. Dynamic conditions or disturbances include fault conditions on the power system, switching equipment in or out, or changes in steady state conditions, e.g. an increase in the load, and the study of the power system’s behaviour after changes in the condition or configuration of the network. In other words, dynamic conditions refer to changes in the steady state condition of power systems.

In order to model the dynamic behaviour of power systems, it is important to set the dynamic models up correctly to reflect the steady state condition of the power system before the dynamic event or disturbance. The configuration during steady state conditions, i.e. before any disturbances, of the model is referred to as the initial conditions of the power system.

It is important for the accuracy of the simulations that all models in the power system, to be modelled, are stable during steady state conditions.

3.8.2 STEADY STATE STABILTY

Steady state conditions refer to the situation where, for example, a generator’s mechanical input power is equal to the electrical output power delivered by the generator. If the input power is different from the output power the power system will become unstable, e.g. if the input power is more than the output power of the generator, the generator rotational speed will increase, and the system frequency will increase. An over-frequency / over-speed condition could cause damage to equipment.

3.8.3 STEADY STATE STABILTY AND INITIALISATION

The steady state stability and the initialisation of the power system model go hand in hand. Correct initialisation and steady state stability is required before dynamic disturbances are modelled.

If the system (for a given set of parameters and operating point) is stable, but a controller in the network model is not initialised correctly, the system could settle at a new stable operating point. However, this is an incorrect operating point. In this case, the system is stable, but incorrectly initialised. If the mismatch is very large, the system could end at a completely unstable operating point.

If the system (for a given set of parameters and operating point) is unstable, it will be unstable, no matter how accurate the initialisation of the system is done.

3.8.4 GENERAL DYNAMIC MODELLING METHODS

After calculating the initial conditions of the network, and determining that the network is stable before the disturbance, the disturbance is modelled and the response of the system is analysed. The controllers in the system can be set in various ways to control the system. As an example, a governor is set or tuned to give an adequate droop control, or primary frequency control response of the grid. Other plant controls such as the automatic voltage regulator (AVR) and the power system stabiliser (PSS) are set or tuned to ensure adequate transient and small-signal performance or response of the plant or system. There can be instances in small, isolated power systems that the system will experience relatively large cycling loads. In this case, the governor and turbine controls may interact with the load dynamics resulting in very low frequency power oscillations on the system. Power system stabilisers might not resolve the problem adequately, and a closer investigation in the turbine controls and their settings is required. Recommended supplemental controls are then required, to mitigate the interactions [1].

3.8.5 SMALL SIGNAL MODELLING

Small signal stability is the ability of the power system to maintain synchronism when subjected to small disturbances [5]. In this context, a disturbance is considered small if the equations that describe the resulting response of the system can be linearised for the purpose of the system analysis. Power system instability can be of two forms: (1) steady increase of the generator angle due to lack of synchronising torque, and (2) rotor oscillations of increasing amplitude due to insufficient damping torque. In today's power systems, it is found that insufficient damping of oscillations causes small signal stability problems in the system.

The nature of the system response to small disturbances depends on a number of factors that include the initial operating point, the strength of the transmission system and the type of generator excitation control used. For a generator connected radially to a large power system without any voltage regulators (i.e. with a constant field voltage), the instability is due to the lack of sufficient synchronising torque. This results in instability of the system through a non-oscillatory mode as shown in Figure 3.13. (T_S is the synchronising torque, T_D is the damping torque, T_e is the electrical torque, δ is the rotor angle and ω is the speed.) [5].

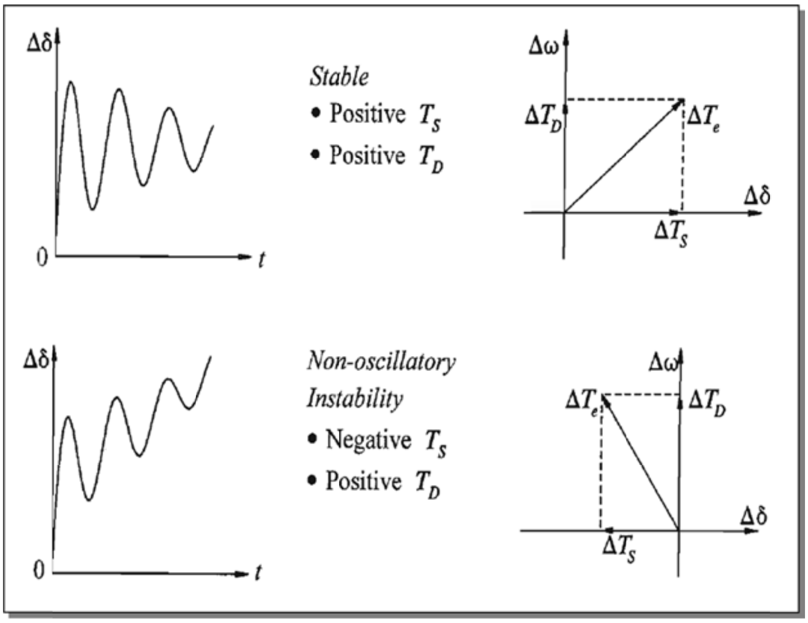


Figure 3.13: Small signal stability and instability with constant field voltage [5]

With continuous acting voltage regulators, the small signal stability problem is one of ensuring sufficient damping of the oscillations in the system. Instability of the system is usually through oscillations with increasing amplitude. Figure 3.14 illustrates the response of generators with automatic voltage controllers [5].

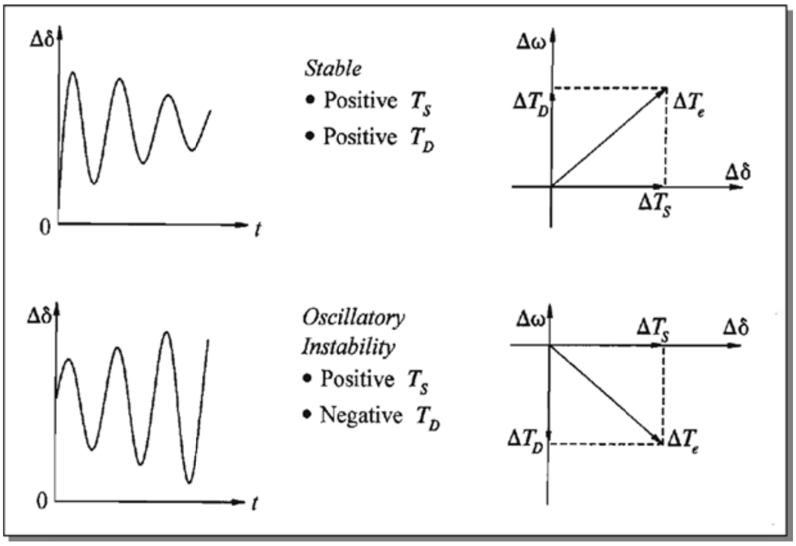


Figure 3.14: Small signal stability and instability with excitation control [5]

In today's power systems, small signal stability is mainly a problem of insufficient damping of oscillations. The following types of oscillations and their stability is of concern [5]:

- *Local modes or machine-system modes* are associated with the swinging of units at a power generating station with the rest of the power system. The term *local* is used to describe that the oscillations occur locally at one station or a small part of the power system.
- *Inter-area modes* are associated with the swinging of several machines in a part of the power system against machines in another part of the system. Two or more groups of closely coupled machines being interconnected by weak ties cause this.
- *Control modes* are associated with generating units. Poorly tuned exciters, speed governors, high voltage direct current (HVDC) convertors and static var compensators (SVCs) are usually cause instability of power systems.
- *Torsional modes* are associated with the turbine-generator shaft system rotational components. Instability of torsional modes may be caused by the interaction of excitation controls, speed governors, high voltage direct current (HVDC) controls and series-capacitor-compensated lines.

Examples of small signal disturbances are:

- a small change (only a few per cent) in loading of the network,
- a small change (a few per cent) in the output power of a generator,
- the switching of a power line in a large power system (important to note that the result of this must be a small change in the steady state conditions of the network).

3.8.6 LARGE SIGNAL MODELLING

Large signal stability is the ability of the power system to maintain synchronism when the system is subjected to severe transient disturbances. The resulting system response involves large deviations of generator rotor angles and is influenced by the non-linear power-angle relationship. The power system stability depends on both the initial operating point of the system and the severity of the disturbance. The power system's post-disturbance steady-state operating point usually differs from the pre-disturbance operating point [5].

A wide variety of disturbances can occur in the power system. These disturbances vary in severity and probability of occurrence. The power system is designed and operated to be stable for a selected set of contingencies. The contingencies that are usually considered are short-circuits of different types, i.e. phase-to-ground, phase-to-phase-to-ground and three-phase faults. These short circuits are usually assumed to occur on transmission lines, but busbar and transformer faults are also considered. It is assumed that the faults are cleared by the appropriate breaker(s) to isolate the faulted equipment. In some cases, high-speed reclosure may occur. Figure 3.15 illustrates the response of a synchronous machine for stable and unstable cases. In the stable case (Case 1), the rotor angle increases to a maximum, then decreases and oscillates with decreasing amplitude until it reaches a steady state. Case 2 shows the rotor angle to increase continuously until synchronism is lost. This form of instability is referred to as *first-swing* instability and is caused by insufficient synchronising torque. In Case 3, the system is stable for a while, but become unstable because of growing oscillations. This form of instability generally occurs when a post-fault steady-state condition is “small-signal” unstable, and is not necessarily a result of the transient disturbance [5].

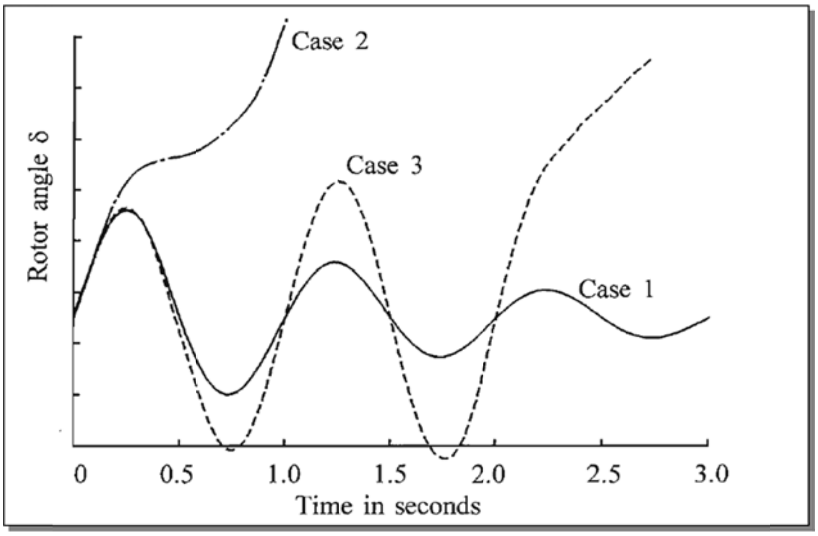


Figure 3.15: Rotor angle response to a transient disturbance [5]

In large power systems, transient instability may not always occurs as first-swing instability as it could be the result of superposition of several modes of oscillations causing large deviations of rotor angles beyond the first swing [5].

In transient stability studies, the study period is usually limited to 3 to 5 seconds following the disturbance. However, in some cases it may be extended to about ten seconds for very large systems with dominant inter-area modes of oscillations [5].

Examples of large signal disturbances are faults on transmission lines, loss of generation units, or the loss of a large load [5]. The system will respond to these disturbances with large changes in generator rotor angles, relatively large changes in load flows and bus voltages. Other power systems variables will also respond with relatively large deviations from their steady state values after the disturbance has occurred. If the resulting angular separation between the power system and the generators remain within certain boundaries, the system will remain in synchronism.

However, if the angular separation is large enough, the generators will lose synchronism with the system. Typical loss of synchronism occurs within 2 to 3 seconds of the disturbance [5].

The transient phenomena can be divided in two main groups, the first swing (transient stability) and the extended term (or long-term phenomena). For the transient stability period, it is generally accepted that the steam turbines (in combined cycle power plants) will have a constant mechanical power output during the first few seconds. This is justified because of the long time constant associated with the heat recovery steam generator (i.e. the time constant of the boiler drum), and due to the sliding pressure control mode that the heat recovery steam generator is operated in. The long time constant of the boiler drum recognises the thermal and pressure energy storage of the drum. Any sudden change to the input of the boiler (i.e. change in heat input), will result in a small change in the boiler drum's output pressure over a relatively short period. For transient stability study purposes, this change is small enough to ignore. The assumption that the steam turbine will have a constant mechanical power output is not correct when fast valving or special protection schemes are used to limit the output of the steam turbine during transients.

In an extended or long-term transient stability study, the focus is on events where the response goes beyond the first few seconds. An example is the response of the heat recovery steam generator with supplemental firing, as the boiler (or drum) has a long heating time constant.

3.8.7 DIFFERENCES BETWEEN SMALL AND LARGE SIGNAL MODELLING

The differences between large signal modelling and small signal modelling is the severity of the disturbance that is modelled.

In small signal modelling, this means that a system or a component can be reduced to a linearised equivalent circuit around its operating point (bias), while still modelling the systems or component with sufficient accuracy. Small signal stability is maintained when sufficient synchronising torque and damping torque is available in the system. Small signal instability is caused by a lack of either synchronising or damping torque in the system.

In large signal modelling, the resulting system response involves large deviations of generator rotor angles and is influenced by the non-linear power-angle relationship. The power system stability depends on both the initial operating point of the system and the severity of the disturbance. The power system's post-disturbance steady-state operating point usually differs from the pre-disturbance operating point.

3.9 SOFTWARE MODEL

3.9.1 INTRODUCTION

In Chapters 2 and 3, background on the various equipment (gas turbines, steam turbines and combined cycle power plants) and the various controllers were provided. Against this background, in order to model the dynamic characteristics of gas turbines and combined cycle power plants accurately for representation in power system simulation software, models were developed by a Cigre Taskgroup [1].

However, these models were not developed for specific simulation software packages. The purpose of this dissertation was therefore to adapt the models to be used in specific power system simulation software. For this dissertation, the models were adapted for the DigSilent PowerFactory Power Simulation Software package [3].

3.9.2 GAS TURBINE MODEL

The gas turbine model is intended for modelling gas turbines in a power system study. If studies with the focus on the performance of specific plant is required, a more detailed model would be required, and even the advice from the manufacturer could be required.

The following diagram (Figure 3.16) shows the open cycle gas turbine model.

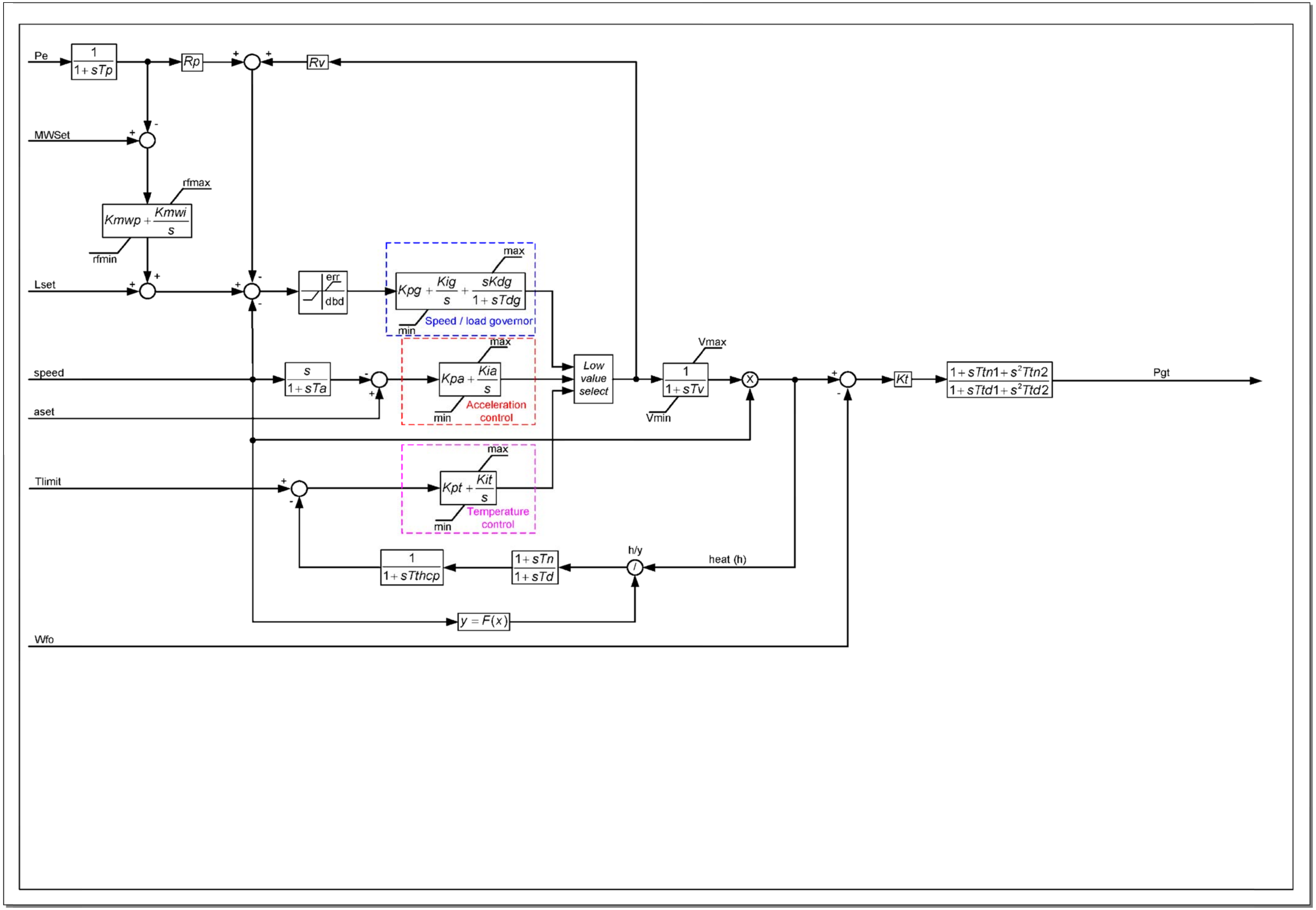


Figure 3.16: Open cycle gas turbine model [1]

3.9.3 COMBINED CYCLE POWER PLANT MODEL

The combined cycle plant model is intended for modelling combined cycle power plants in a power system study. A combined cycle power plant consists of gas turbines and steam turbines in a variety of combinations as discussed in earlier sections of this dissertation. If studies with the focus on the performance of specific plant is required, a more detailed model would be required, and even the advice from the manufacturer could be required [1].

The following points are important for modelling combined cycle power plants in power system studies:

- The major control loops associated with the dynamic response of a gas turbine have to be represented in the model;
- The dependence of the maximum power output varies with system frequency;
- An appropriate representation of the heat recovery steam generator’s dynamic response;
- The inclusion of the dynamics of the steam turbine for long term dynamic studies,
- The combined cycle power plant’s electrical output is controlled by the gas turbine only if no supplemental firing is applied and it is controlled in sliding pressure control mode;
- The available steam will determine the generating power of the steam turbine, the steam turbine will therefore follow the gas turbine;
- The response time of the heat recovery steam generator (typically minutes) determines the steam turbine’s output power response if there is a change in the gas turbine power;
- Two modes of operation are generally in use in combined cycle power plant, the one being sliding pressure (control valves fully open) and the other being fixed pressure control (control valves are throttled);

Figure 3.17 shows the combined cycle power plant model.

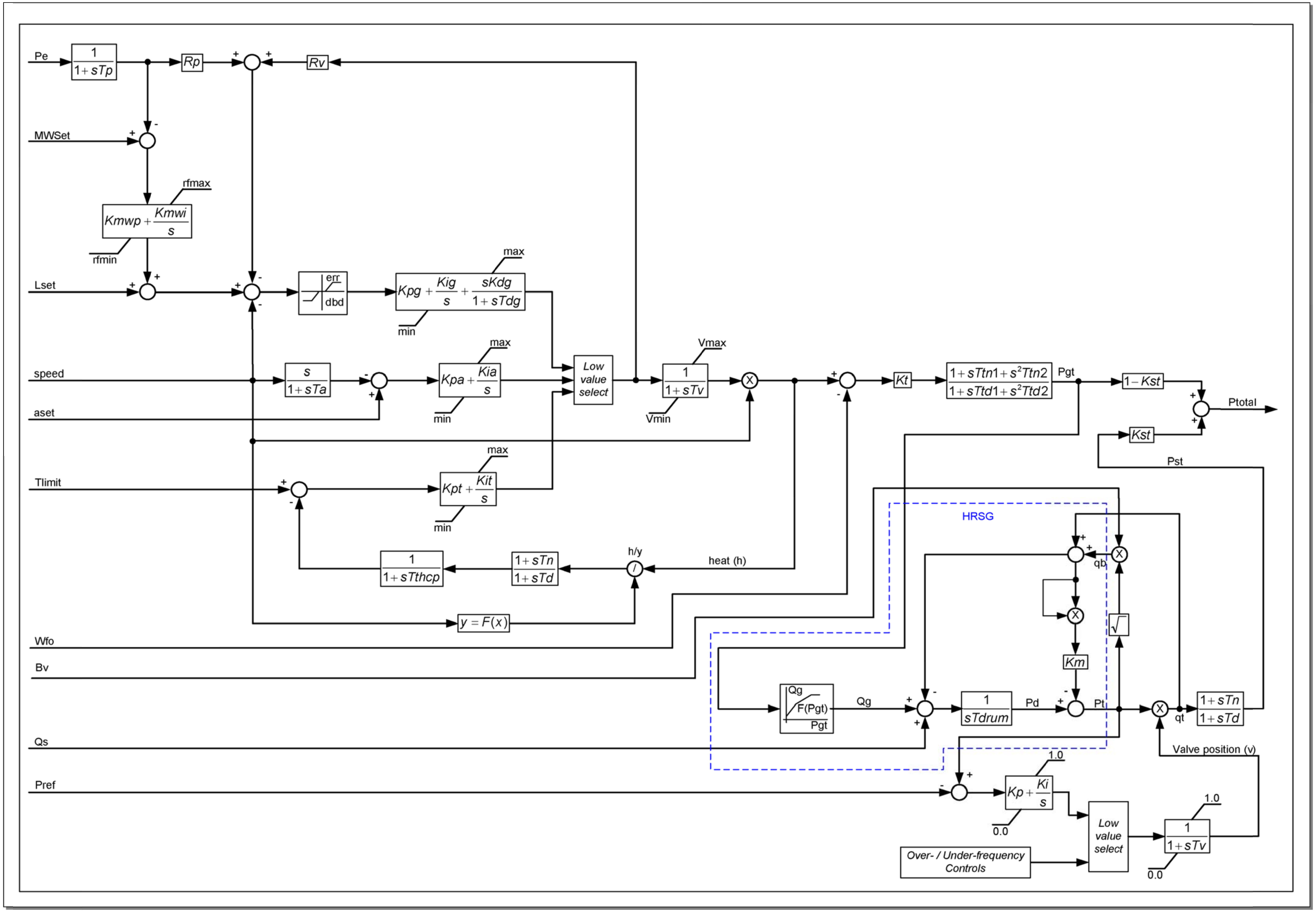


Figure 3.17: Combined cycle power plant model [1]

3.9.4 DESCRIPTION OF MODELS

The purpose of a model is to represent the actual equipment in a simulation. The model must represent the characteristics of the equipment that is required for power system studies, as accurately as possible.

3.9.4.1 GENERAL DESCRIPTION

The gas turbine and combined cycle power plant models [1] are applied in the per-unit system, and the nameplate MW rating of the gas turbine is used as the basis. The models are based on the principle that a gas turbine delivers mechanical power on its shaft. The unit of mechanical power is watt. However, for generators, apparent power (MVA) is used as its base. In the combination of a gas turbine model and a generator model, the mechanical power output from the gas turbine model is the input to the generator model. In the generator model, the mechanical power is converted into electrical power (active power). The automatic voltage regulator (AVR) of the generator is responsible for the generation of reactive power and the control of the generator terminal voltage, while operating within the apparent power (MVA) limits of the generator. To align the gas turbine mechanical power with the generator's active power, the generator is defined on its apparent power rating (MVA) and its rated power factor. From these two values, the generator's active power is calculated, (i.e. active power = apparent power x power factor) and used as a common base for the gas turbine.

The gas turbine model consists of three major control loops. The control loops are:

- The speed or load governor (with parameters Kpg, Kig, Kdg and Tdg)
- The acceleration control loop (with parameters Kpa and Kia)
- The temperature control loop (with parameters Kpt and Kit).

Refer to Figure 3.16 for a diagram of the control loops.

3.9.4.2 PARAMETERS OF THE MODELS

The parameters of the models (for OCGT and CCPP, Figure 3.16 and Figure 3.17) are:

• Lset	Turbine speed / load set-point,
• Kmwp / Kmwi	Reset controller, acting on the set point MWset, representing the plant outer loop control, can be in service, maintaining the unit's output power at a preset value as set by the plant operator,
• dbd	Represents actual programmed dead-band of the modern digital governors of gas turbines,
• Rv, Rp, Tp, Lset, Kpg, Kdg, Kig:	Select the mode of governor action. By setting Rv and Kdg to zero and Rp, Tp, Lset, Kpg and Kig to the appropriate none-zero values, this models a reset-controller with a droop of Rp that acts on speed and electrical power feedback. In contrast, setting Rv, Kdg, Kig and Rp to zero and Lset and Kpg to the

	appropriate non-zero values, this models a standard droop governor with speed feedback only, with droop equal to 1 / Kpg,
max, min	Found on all the control loops. It represents maximum and minimum fuel flow,
Wfo	Fuel flow at full speed and no-load. It represents the fuel flow to allow the gas turbine to run at full speed, but at no-load, as the gas turbine requires fuel to drive the compressor, and therefore is never zero,
Vmax, Vmin	Maximum and minimum valve opening, set to 1.0 and 0.0 respectively,
Kt	Turbine gain, scaling factor of the turbine, that is gross turbine power minus power consumed by the compressor,
Tv	Time constant representing the response of the fuel system,
Tthcp, Tngt, Tdgt	Time constant representing the exhaust (or inlet) temperature measurement system response,
Ta	Acceleration control differentiator time constant,
Ttn1, Ttn2, Ttd1, Ttd2	Turbine dynamics time constants,
Pe	Connected generator’s electrical power in per unit,
Speed	Unit’s speed in per unit,
Tlimit	Gas turbine temperature limit,
F(x)	Function to describe relationship between maximum power output and frequency,
aset	Acceleration set point (pu / s2)
Pgt	Gas turbine output power,
Qg	Heat generated by gas turbine,
Qs	Supplemental firing,
Tdrum	Time constant for the variation of drum pressure, due to variation in steam production and consumption,
Km	Pressure loss / flow friction coefficient,
Pt	Steam pressure at turbine admission valve,
Pst	Steam turbine power,
qt	Steam mass flow,
Td, Tn	Steam turbine time constants,
Bv	Bypass valve,
qb	Steam flow through bypass valve,
Kst	Fraction of total delivered power delivered by the steam turbine,

3.9.4.3 DETAIL DESCRIPTION OF MODELS

In the previous section, the various parameters were discussed to introduce the models. In this section, a detail description of the models is provided. This section starts with a description of the characteristics of the gas turbine, and continues with a detailed description of the characteristics of the gas turbine, the heat recovery steam generator and the steam turbine that is important for power system modelling.

3.9.4.3.1 MAXIMUM POWER OUTPUT OF GAS TURBINES

When the turbine is delivering maximum power output, it is operating at its temperature limit, thus the gas turbine is under temperature control [1]. With changes in ambient air conditions and/or changes in the turbine speed, there will be a change in the airflow through the compressor, resulting in a change in output. Because of a change in fuel demand, as determined by the temperature control loop, this loop is trying to observe the temperature limit. If the turbine is operating below full-load (and also below the temperature limit), for a deviation in the frequency, there will be a transient overshoot in power output as the governor acts to increase the power output of the gas turbine. Within a short period (typically a few seconds – the time constant of the casing and thermocouples measuring the exhaust temperature), the temperature control loop will take control and start to reduce the turbine power output in order to maintain the exhaust temperature limit. The temperature control loop will limit the power output (and therefore the temperature) by limiting the fuel supply if the governor attempts to push it beyond the temperature limit [1].

From the above it is clear that:

- The maximum power output of the turbine is approximately proportional to changes in ambient atmospheric air pressure [1].
- The maximum power output of the turbine is a complex function of ambient temperature, and is non-linear. This dependence is governed by the airflow characteristic of the compressor [1].
- The maximum power output of the turbine is approximately proportional to the variation in frequency (and hence the speed) of the gas turbine. If the turbine operates in a very narrow band around the nominal frequency, the variation of the maximum power output is also very small. For power system study purposes, the parameter Tlimit and the function F(x) represent this relationship. Tlimit is the temperature limit of the gas turbine, which in turn will limit the maximum power output of the gas turbine. Seasonal variations in ambient temperature can be accounted for, using Tlimit. F(x) represents the maximum power output relationship with frequency variations. In the model it is implemented by either a look-up table or a piece-wise linear function [1].

In order to increase the gas turbine power capability, inlet air cooling systems are installed to reduce the air temperature to the intake of the gas turbine. The installation of the cooling systems proves very useful in areas with high average ambient temperatures [1].

3.9.4.3.2 ACCELERATION CONTROL OF GAS TURBINES

The parameter, aset, in the acceleration control loop is set to limit the acceleration of the turbine, as the loop will limit the fuel flow to the turbine.

3.9.4.3.3 SELECTION OF CONTROL LOOPS OF GAS TURBINES

The low value select gate selects between the three major control loops, the loop with the lowest value. In practice, an appropriate anti-wind-up logic is implemented to ensure smooth and proper transitioning between the various control loops connected to the select gate. The detail of this implementation varies between the various manufacturers, and when this becomes important in simulation studies, the user has to consult the manufacturer for the detailed implementation [1].

3.9.4.3.4 GENERAL DESCRIPTION OF COMBINED CYCLE POWER PLANTS

Figure 3.17 shows a model for combined cycle power plant that includes the gas turbine, the heat recovery steam generator and the steam turbine, developed by a Cigre Task Group [1]. The model is per-unitised based on the turbine nameplate data, i.e. the power rating in MW. The heat recovery steam generator and steam turbine arrangements found in many combined cycle power plants could generate steam at various pressures, have multiple steam drums and even have steam admissions at multiple points in the turbine. For the purpose of power system simulations, it is sufficient to simplify this complex steam system by treating it as a single drum and a single turbine admission valve system [1].

3.9.4.3.5 GENERATED HEAT IN HRSG

The model in Figure 3.17 includes, in principle, the thermal and energy storage capability of the boiler drums, the presence of friction and throttling losses in the steam paths. The admission valves are used for controlling turbine inlet pressure [1].

The function, Pgt, describes the amount of heat, Qg, generated by the gas turbine. In this function, the production of steam is taken to be proportional to the amount of heat provided by the gas turbine. If supplemental firing is used, the sum of Qs (amount of heat from the supplemental firing) and Qg determines the amount of steam that is produced. Steam production therefore is proportional to the sum of generated heat (Qg) and the amount of heat from supplemental firing (Qs).

3.9.4.3.6 STEAM PRESSURE AND LOSSES IN HRSG

The time constant, Tdrum, describes the variation of drum pressure, because of steam production and consumption of the steam. Flow friction (and consequent pressure loss in the boiler tubes) and the throttling losses upstream of the steam turbine inlet valves is represented by the coefficient, Km. The value Pd represents the drum steam pressure, losses are represented by the factor Km and the available steam pressure (i.e. drum steam pressure minus losses) is represented by Pt. This steam pressure is available to drive the steam turbine [1].

3.9.4.3.7 STEAM TURBINE POWER

In the steady state, the steam turbine power, Pst, and the steam mass flow, qt, vary linearly with the turbine inlet pressure, Pt. A lead-lag transfer function, with time constants Tn and Td representing the transient characteristics of the steam turbine. As a significant part of the steam turbine power is developed in the intermediate- and low-pressure steam turbine sections, this lead-lag transfer function recognises the fact that this flow lags the flow through the high-pressure section of the turbine [1].

3.9.4.3.8 MULTI SHAFT CCPP's

The model in Figure 3.17 could be further expanded to model multi-shaft combined cycle power plants. The heat recovery steam generator part of the model is replicated, to represent each gas turbine in the plant. However, this did not form part of this dissertation.

3.9.4.3.9 CONTROL MODES OF HRSG STEAM TURBINES

Combined cycle power plants (and specifically the steam turbine) are mainly controlled in two different modes, namely sliding pressure mode (control valves fully open) or fixed pressure mode (control valves partially open or throttled). This is represented in the model by a simple proportional-integral controller (represented by the parameters Kp and Ki) to regulate the inlet steam pressure of the steam turbine. The pressure reference, Pref, can be set to a low value to make the controller inactive [1]. Over- and under-frequency control can be added to the control modes. For this dissertation, the over- and under-frequency control was excluded.

3.9.4.3.10 TOTAL POWER OUTPUT

The model presented in Figure 3.17 represents a single shaft combined cycle power plant. The gas turbine and steam turbine mechanical power outputs are summated as shown in Figure 3.17 to produce the total mechanical power (Ptotal) delivered to the electrical generator. The gas turbine delivers mechanical power, Pgt, while the steam turbine delivers Pst. The constant Kst represents the fraction of power delivered by the steam turbine, in per unit, of the total mechanical power delivered to the generator.

3.9.5 TYPICAL MODEL PARAMETERS

The following list of values for the parameters (see Table 3.1) represents typical behaviour of a heavy-duty gas turbine in simple-cycle or combined cycle power plants [1]. The typical values represent the most common types of gas turbines. Certain values are set to zero to set the characteristic inactive and to represent the most common gas turbines. When the specific characteristic is required in the modelling, the appropriate values are set to model the specific characteristics. For the purpose of this dissertation, some values are set to zero in order to compare the various gas turbine models (existing i.e. GAST, GAST2A and GASTWD models and the new gas turbine (OCGT) model). This was done to show that the new gas turbine model could be set up to obtain similar results as the existing models for the same disturbances in the system. The aim was to show that the new gas turbine model does not replace the existing gas turbine models, but complement them. In Chapter 5 a detailed comparison is made between the proven existing models (GAST, GAST2A, GASTWD) and the new gas turbine (OCGT) model.

Table 3.1. Parameters of the OCGT 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 power control is in-service (i.e. when Kmwp and / or Kmwi ≠ zero)
Lset	(3)	Load / speed reference set-point
Dbd	0.0003	Intentional dead-band
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. The plant outer control loop is modelled by parameters Kmwp, Kmwi, rfmax, rfmin and MWset. If this loop is in-service, it will greatly change the unit's response. 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 resulted 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 re-adjusted 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 is set equal to the initial steady-state value of turbine output during initialization of the model, i.e. the loading of the generator in MW during steady state conditions (before dynamic analysis of the model).
3. The parameter is defined by the user and/or by simulation program during initialization of the model.
4. The value of Tlimit as given in the table ensures that the gas turbine reaches its temperature limit when the turbine output reaches the turbine nameplate rating, i.e. 1 pu mechanical power output. However, if the user wishes to simulate a different ambient condition, where 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. The model requires that a value of F(0.96) is selected to yield a steady-state GT output of 96% of its peak output. Some basic algebra can be done to show F(0.96) = (0.97 * (Tlimit – Wfo) + Wfo) / Tlimit.

The following list of values for the parameters represents typical behaviour of a steam turbine in combined cycle power plants.

Table 3.2. Parameters of the CCPP 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 the manufacturer must be consulted for this information.
2. This is the amount of supplemental firing, in per unit, applied to the boiler. The user defines this value, or it is obtained 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.

3.10 SUMMARY

In this chapter, the theory of the Cigre models was discussed. The chapter started with the theory and ended with simulation models for the prime movers (i.e. the gas turbine model (OCGT) and the combined cycle power plant model (CCPP)).

The Laplace Transform represents a time function in the frequency domain. The building blocks for a complete simulation model were presented and discussed. Various types of controllers (for example a PID controller) are represented by various building blocks. Combining the controllers and building blocks, a complete simulation model can be developed that represents the prime mover or any other controller. Various combinations of open cycle and combined cycle power plants can be modelled by setting up the simulation models to represent the various plants correctly. For this dissertation, only the single shaft combined cycle power plant was modelled.

Cigre developed models for open cycle gas turbines and combined cycle power plants [1]. The aim of this dissertation was to adapt these models to be available in a specific power system simulation package. The models are called OCGT (Fig 3.16) (for the open cycle gas turbine) and CCPP (Fig 3.17) (for the combined cycle power plant) and were adapted for the DigSilent PowerFactory power system simulation package.

- Chapter 4 -

4 EXISTING MODELS

4.1 INTRODUCTION

The previous chapter discussed the theory of models, their building blocks and controllers. The previous chapter included a discussion on the modelling of prime movers and their controls. The focus was on the gas turbine model (OCGT) and the combined cycle power plant model (CCPP) that included the heat recovery steam generator model, developed by Cigre [1]. For this dissertation, the models (OCGT and CCPP) were adapted to be available in DigSilent PowerFactory.

This chapter contains a study of the existing gas turbine models used in power system simulation software. The existing gas turbine models are called GAST, GAST2A and GASTWD. In this chapter, each of the existing models will be discussed to show their respective characteristics. The open cycle gas turbine (OCGT) model is compared to the existing models to show the similarities as well as the differences between the models. The existing models from two power simulation packages, namely PSS/E [2] and DigSilent PowerFactory [3] were included to show that the existing gas turbine models are not limited to one analysis package and to make the comparisons. The comparison shows the similarities and differences between the models. This is done to show where the three existing models' limitations are if combined cycle power plant is to be modelled.

4.2 EXISTING MODELS

The existing models discussed in this chapter were obtained from the PSS/E [2] and DigSilent PowerFactory [3] manuals. The available models include various gas turbine models.

4.3 PSS/E MODELS

The following gas turbine models are available in PSS/E [2]:

4.3.1 GAST – GAS TURBINE MODEL

This model is a representation of the dynamic characteristics of industrial gas turbines. The model is accurate for small speed variations, i.e. for variations within 5% of nominal rating. The model (see Figure 4.1) includes a forward path, with the governor time constant T_1, and a combustion chamber time constant T_2, together with a load limiting feedback path, as the load limit is sensitive to the turbine exhaust temperature. The exhaust temperature measuring system time constant is represented by T_3 [2].

The ambient temperature load limit parameter [CON (J+5)] (Load Limit in Figure 4.1) is set at unity when the turbine is operating at the design ambient temperature. As the ambient temperature rises, this parameter must be lowered, as the turbine dynamic behaviour is dependent on the ambient temperature. The supplier / manufacturer of the turbine can provide this data. The constant K_T adjusts the gain of the load-limited feedback path [2].

During initialisation of the model, the load reference [VAR(L)] is set equal to P_{MECH}, the shaft power of the gas turbine. The load limited feedback path controls the fuel-flow to the gas turbine through the low value select gate. This occurs when its output is lower than the original load reference that is decremented by the droop gain, 1/R. The damping coefficient, D_{TURB}, represents the speed damping of the turbine rotor [2].

There is a distinction between the maximum fuel valve opening, V_{MAX} [CON(J+6)], and the ambient temperature load limiter. The maximum fuel valve opening is an operational controller. The plant control operator can adjust it to allocate load in a plant. The ambient temperature load limit is a turbine design parameter. The load limiting feedback serves a protective function [2].

The turbine damping factor D_{TURB}, is equal to $\partial P_{TURBINE} / \partial_{SPEED}$. The value is in per unit, on a base that is equal to unit rating. For steam, nuclear and gas turbines the value for D_{TURB} can be taken as zero, while for hydro turbines, the value lies between 0 and 0.5 pu [2].

Figure 4.1 shows the PSS/E GAST gas turbine model.

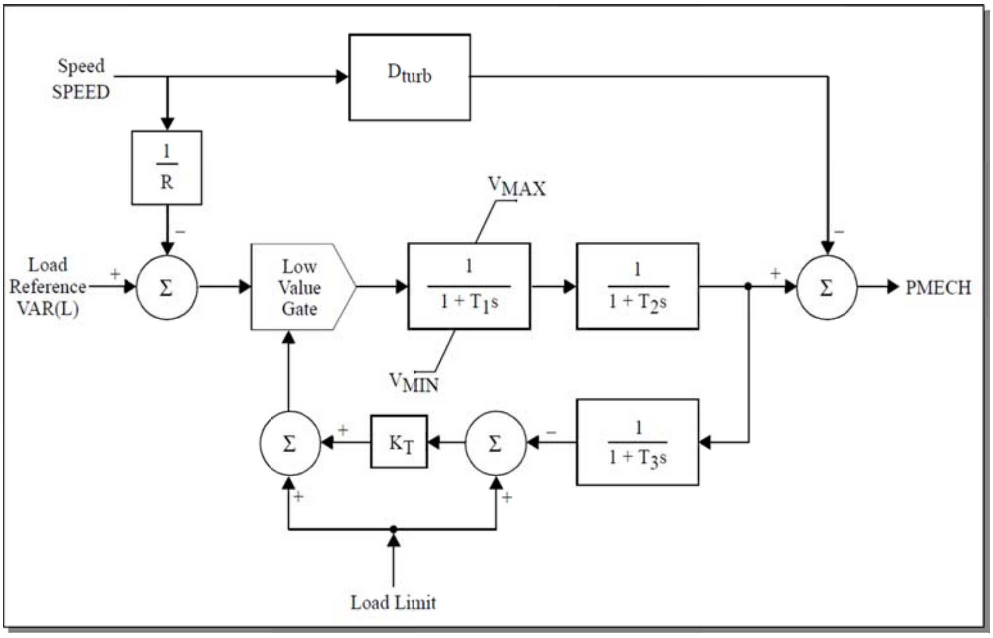


Figure 4.1: PSS/E GAST model [2]

Table 4.1. Model GAST constants

CONs	Description	Unit
J	R (speed droop)	pu
J+1	T_1 (>0)	s
J+2	T_2 (>0)	s
J+3	T_3 (>0)	s
J+4	Ambient temperature load limit, AT	
J+5	K_T	

CONs	Description	Unit
J+6	V_{MAX}	pu
J+7	V_{MIN}	pu
J+8	D_{TURB}	

Table 4.2. Model GAST states

STATEs	Description
K	Fuel valve
K+1	Fuel flow
K+2	Exhaust temperature

Table 4.3. Model GAST variables

VAR	Description
L	Load reference

Typical values for the parameters of the GAST model are:

Table 4.4. Model GAST typical values

Parameter	Value	Unit
R	0.05	pu
T_1	0.4	s
T_2	0.1	s
V_{MAX}	1.0	pu
V_{MIN}	-0.05	pu
Ambient temperature load limit	1.0 at 26.7 °C 0.9 at 40.6 °C	
K_T	2.0	
D_{TURB}	0.0	

4.3.2 GAST2A – GAS TURBINE MODEL

GAST2A is a more detailed representation of a gas turbine than the GAST model. This model (see Figure 4.2) is intended for simulations close to rated speed only, similar to the GAST model. The major difference between the GAST and GAST2A models is that the GAST2A model includes a temperature control loop.

The GAST2A model has three control loops:

- Speed control
- Temperature control
- Fuel flow control

The speed governor has two possible configurations. The first configuration is the droop configuration and the second is isochronous mode. By selecting the variable Z [CON(J+3)] in the speed governor block in the model shown in Fig. 4.2 to either 0 or 1, the speed governor is set in the droop mode or the isochronous mode. The speed control loop assumes the lowest value between temperature control and the speed governor.

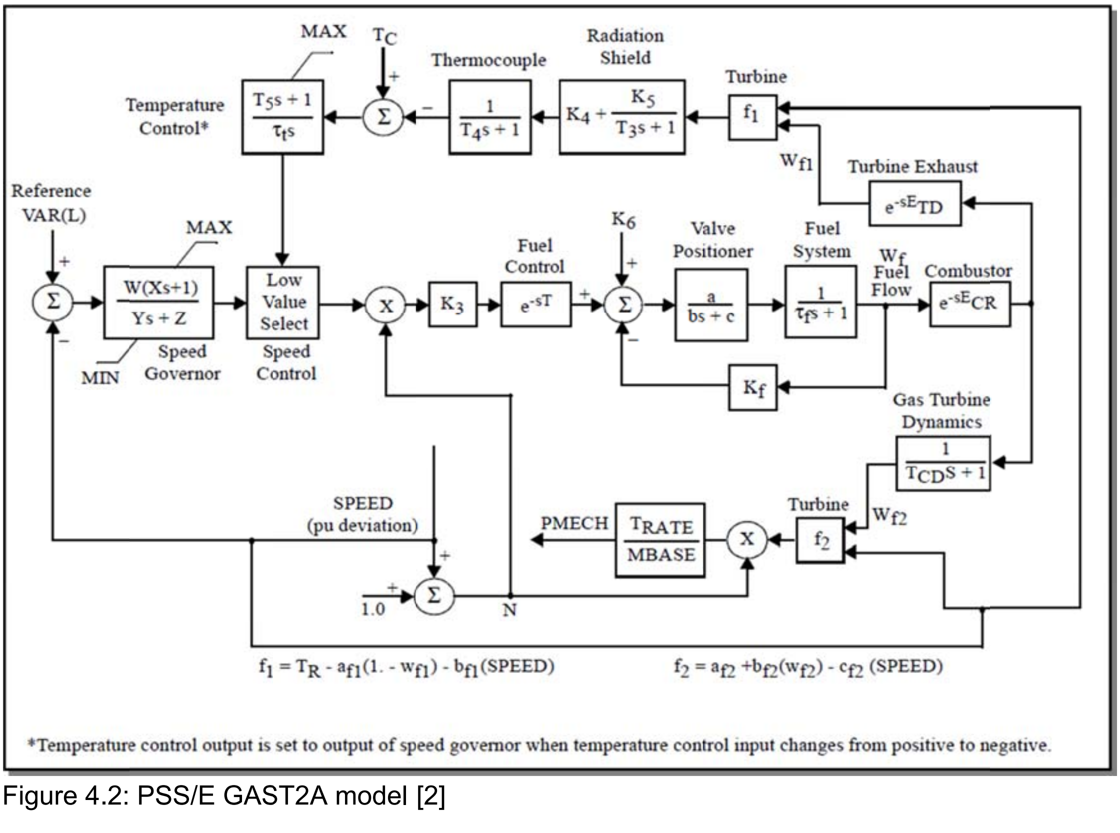
Droop mode: This mode of operation for a turbine will allow the machine to react to the load variation by changing its speed. This mode is utilised when multiple machines run in parallel that share the total load between the machines. For this sharing to be equal, all the machines need to have same droop characteristic or settings. Typically, if a machine has a droop of 4%, it means for a change of 1% in rated speed, the load of the machine will vary by 25% of its rated load. This is also referred to as “load control” mode of operation.

Isochronous mode: In this mode, the turbine is not affected by its load and regardless of the machine’s loading, it will maintain the system frequency. This mode is also referred to as “frequency control” mode of operation. In the case of systems that are not connected to a grid (also known as island operation), it is required to run at least one machine in this mode, to take care of the load variations and to maintain the system frequency. This mode is also known as automatic generator control (AGC).

The fuel flow control loop controls fuel flow. The input K_6 to the summing function represents the minimum fuel flow required for the gas turbine to operate at no-load and rated speed. The speed deviation and the speed control loop determine the action of the fuel flow control system.

The exhaust temperature is determined by the fuel flow in the gas turbine. The temperature control loop limits the mechanical output of the gas turbine, once the maximum exhaust gas temperature has been reached.

Figure 4.2 shows the PSS/E GAST2A gas turbine model.



*Temperature control output is set to output of speed governor when temperature control input changes from positive to negative.

Figure 4.2: PSS/E GAST2A model [2]

Table 4.5. Model GAST2A constants

CONs	Description	Unit
J	W – governor gain (1/droop), based on turbine rating	
J+1	X – governor lead time constant	s
J+2	Y – governor lag time constant	s
J+3	Z – governor mode: 1 – droop or 0 – Isochronous	
J+4	E_{TD}	s
J+5	T_{CD}	s
J+6	T_{RATE} – turbine rating	MW
J+7	T	s
J+8	Max – limit (on turbine rating)	pu
J+9	Min – limit (on turbine rating)	pu
J+10	E_{CR}	s
J+11	K_3	
J+12	a (>0) – valve positioner	
J+13	b (>0) – valve positioner	s
J+14	c – valve positioner	
J+15	τ_f (>0)	s
J+16	K_f	
J+17	K_5	
J+18	K_4	
J+19	T_3 (>0)	s
J+20	T_4 (>0)	s
J+21	τ_t (>0)	s
J+22	T_5 (>0)	s
J+23	a_{f1}	
J+24	b_{f1}	
J+25	a_{f2}	
J+26	b_{f2}	
J+27	c_{f2}	
J+28	T_R – rated temperature	°F / °C (1)
J+29	K_6 – minimum fuel flow	pu
J+30	T_C – temperature control	°F / °C (1)

Note.

1. Units can be in °F or °C depending on the constants a_{f1} and b_{f1}

Table 4.6. Model GAST2A states

STATEs	Description
K	Speed governor
K+1	Valve positioner
K+2	Fuel system
K+3	Radiation shield
K+4	Thermo-coupler
K+5	Temperature control
K+6	Gas turbine dynamics
K+7	Combustor
K+8	Combustor
K+9	Turbine / exhaust
K+10	Turbine / exhaust
K+11	Fuel controller delay
K+12	Fuel controller delay

Table 4.7. Model GAST2A variables

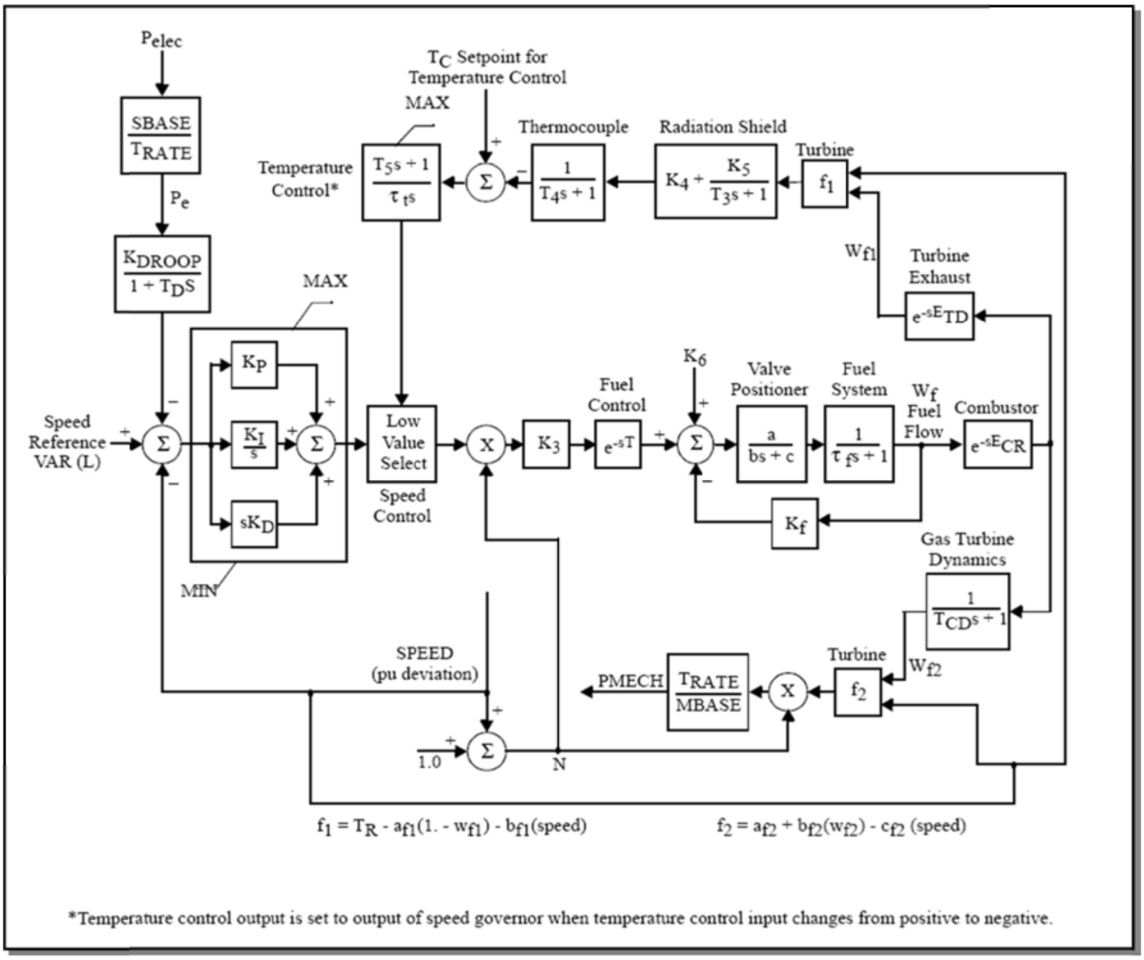
VARs	Description
L	Governor reference
L+1	Temperature reference flag
L+2	Low value select output
L+3	Output of temperature control

4.3.3 GASTWD – WOODWARD GAS TURBINE MODEL

This model (refer to Figure 4.3) represents the same detail as GAST2A for the dynamics of a gas turbine. However, the governor system is based on a Woodward governor, which is a digital speed controller. This governor consists of an electric speed sensor and includes proportional, integral and derivative control or PID control as part of the speed control.

There are two differences between the GAST2A and the GASTWD models. In the GASTWD model, the speed governor is modelled by a PID controller, while in the GAST2A model the speed governor is modelled by a lead-lag compensator. The second difference for the GASTWD model is an additional speed reference feedback loop.

Figure 4.3 shows the PSS/E GASTWD gas turbine model.



*Temperature control output is set to output of speed governor when temperature control input changes from positive to negative.

Figure 4.3: PSS/E GASTWD model [2]

Table 4.8. Model GASTWD constants

CONs	Description	Unit
J	K_{DROOP}, based on turbine rating	
J+1	K_P	
J+2	K_I	
J+3	K_D	
J+4	E_{TD}	s
J+5	T_{CD}	s
J+6	T_{RATE} – turbine rating	MW
J+7	T	s
J+8	Max – limit (on turbine rating)	pu
J+9	Min – limit (on turbine rating)	pu
J+10	E_{CR}	s
J+11	K_3	
J+12	a (>0) – valve positioner	
J+13	b (>0) – valve positioner	
J+14	c – valve positioner	
J+15	τ_f (>0)	s
J+16	K_f	
J+17	K_5	
J+18	K_4	
J+19	T_3 (>0)	s

CONs	Description	Unit
J+20	T_4 (>0)	s
J+21	τ_t (>0)	
J+22	T_5 (>0)	s
J+23	a_{f1}	
J+24	b_{f1}	
J+25	a_{f2}	
J+26	b_{f2}	
J+27	c_{f2}	
J+28	T_R – rated temperature	°F / °C (1)
J+29	K_6 – minimum fuel flow	pu
J+30	T_C – temperature control	°F / °C (1)
J+31	T_D – power transducer (>0)	s

Note.

1. Units can be in °F or °C depending on the constants af1 and bf1

Table 4.9. Model GASTWD states

STATEs	Description
K	Speed governor
K+1	Valve positioner
K+2	Fuel system
K+3	Radiation shield
K+4	Thermo-coupler
K+5	Temperature control
K+6	Gas turbine dynamics
K+7	Combustor
K+8	Combustor
K+9	Turbine / exhaust
K+10	Turbine / exhaust
K+11	Fuel controller delay
K+12	Fuel controller delay
K+13	Power transducer

Table 4.10. Model GASTWD variables

VARs	Description
L	Governor reference
L+1	Temperature reference flag
L+2	Low value select output
L+3	Output of temperature control
L+4	Derivative control

4.4 DIGSILENT POWERFACTORY MODELS

The purpose of including a second power simulation package's gas turbine models here is to show that the three existing models, GAST, GAST2A and GASTWD are not limited to one simulation package only. However, small differences in the models of the two simulation packages (PSS/E and DigSilent PowerFactory) were found.

4.4.1 GAST - GAS TURBINE MODEL

This model (refer to Figure 4.4) is in essence the same model as the PSS/E GAST Gas Turbine model [2], except that the DigSilent GAST model requires the rated power of the turbine as an input variable. However, if the user enters a "0", the value is calculated internally in the "Turb" block in the model from the input signals Pg, sgnn and cosn. Pg is the Electrical Power (not used in the block), sgnn is the Nominal Apparent Power of the generator, and cosn is the generator's rated power factor. By entering a "0" in the model, the rated power output (in MW) of the gas turbine is set equal to the power output (in MW) of the generator. By entering a value for the gas turbine that is larger than the power output of the generator, the effect of the model's performance is changed marginally. However, if the gas turbine has a smaller rated output power than the generator, the model's performance is influenced drastically and the system can become unstable if the output from the generator is larger than the rating of the gas turbine. The influence of disturbances in the network could also cause instability, in comparison to the response if the turbine has an equal or larger rated output than the generator. Some of the blocks in Figure 4.4 contain two values, e.g. K and Kt. The top value (or function) refers to the function of the block, while the bottom value refers to the variable (or variable name) of the function. Variables of the various functions are "K" or "T". "Turb" is a special function. "Const" is a settable constant value. The input values "w" are the actual speed, "wref" is the reference speed, "psetp" is the load set point and "psco" is an input used from the power system stabiliser, while "pt" is the turbine output power.

Refer to Section 4.3.1 for a description of the model.

Figure 4.4 shows the DigSilent PowerFactory GAST gas turbine model.

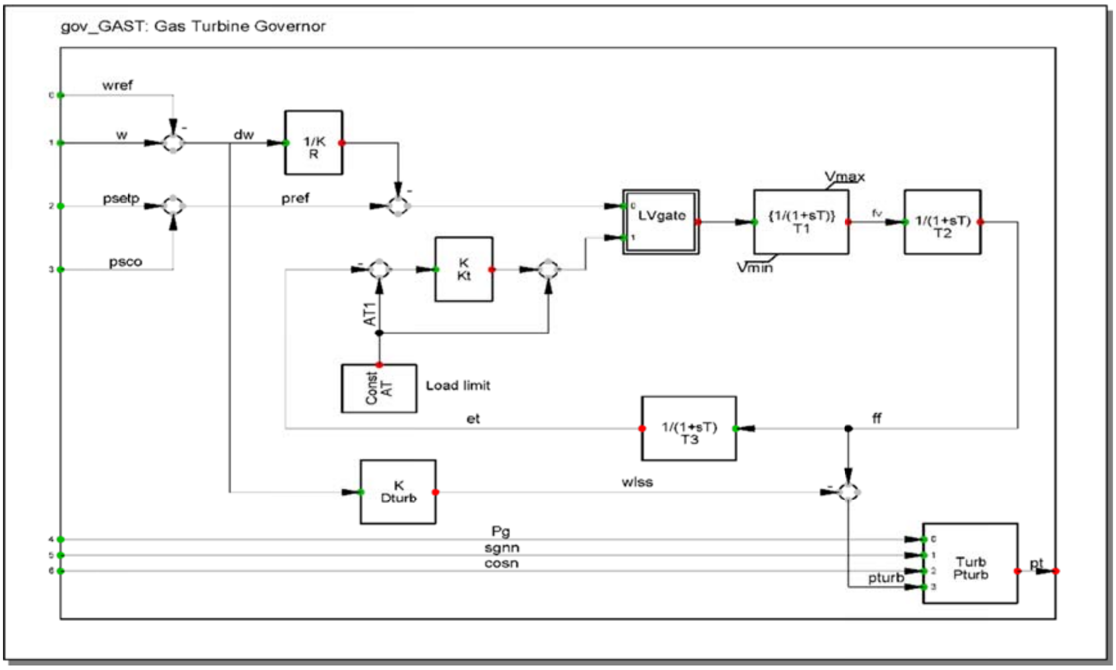


Figure 4.4: DigSilent PowerFactory GAST Model [3]

Table 4.11. DigSilent GAST model variables [3]

Variable	Description	Unit	Typical Value
R	Speed Droop	pu	0.047
T1	Controller Time Constant	s	0.4
T2	Actuator Time Constant	s	0.1
T3	Compressor Time Constant	s	3
AT	Ambient Temperature Load Limit	pu	1
Kt	Turbine Factor	pu	2
Vmin	Controller Minimum Output	pu	0
Vmax	Controller Maximum Output	pu	1
Dturb	Frictional Losses Factor	pu	0.001
Pturb	Turbine Rated Power 0 = Pturb = Pgen	MW	0

4.4.2 GAST2A – GAS TURBINE MODEL

This model (see Figure 4.5) is in essence the same model as the PSS/E GAST2A Gas Turbine model [2], except that the DigSilent GAST2A model requires the rated power of the turbine as an input variable. If the user enters a “0”, the value is calculated internally in the “Trate” block in the model from the input signals sgnn and cosn. The “Trate” block works similar to the “Turb” block in the GAST model. (Refer to Section 4.4.1 for a detail description).

The only other difference between the PSS/E and the DigSilent model is the calculation of the deviation of the speed from unity speed. The PSS/E model receives the deviation value as an input value, while it is calculated in the DigSilent model.

The same philosophy is followed for the descriptions in each of the blocks in Figure 4.5 as for the blocks in Figure 4.4. The top value (or function) describes the function of the block, while the bottom value(s) represent the variable(s) of the function. The functions' variables (top value or function) and the description of the variable (bottom value or variable) may have the same or may have a different description. "F1", "F2", "Delay_no_inc", "GTc", "Valve_pos" and "PtdbPg" are special functions.

Refer to Section 4.3.2 for a description of the model.

Figure 4.5 shows the DigSilent PowerFactory GAST2A gas turbine model.

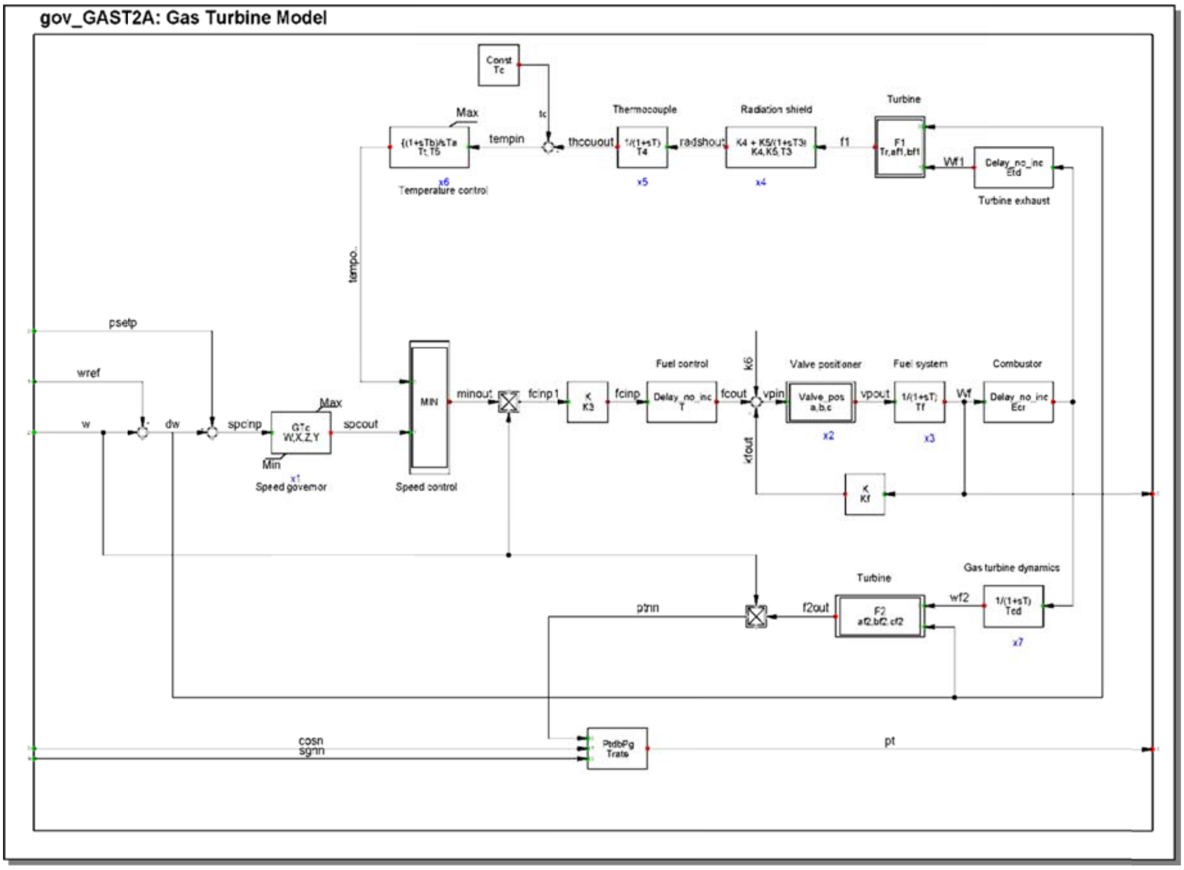


Figure 4.5: DigSilent PowerFactory GAST2A Model [3]

Table 4.12. DigSilent GAST2A model variables

Variable	Description	Unit	Typical Value
W	Speed Controller Gain	pu	25
X	Speed Controller Der. Time Constant	s	0.479
Z	Speed Controller Is/Dr	pu	1
Y	Speed Controller Time Constant	s	0.4
Etd	Temperature Controller Delay	s	0
Tcd	Turbine Time Delay	s	0.1
Trate	Turbine Rated Power	MW	Note 1
T	Fuel Control Delay Time	s	0
Min	VCE Lower Limit	pu	-0.15

Variable	Description	Unit	Typical Value
Max	VCE Upper Limit	pu	1.79
Ecr	Fuel System Delay	s	0.01
K3	Turbine Factor	pu	0.531
A	Fuel System Prop. Characteristic	pu	1
B	Fuel System Time Constant	s	0.05
c	Fuel System I/L Factor (0/1)	pu	1
Tf	Fuel System Delay	s	0.4
Kf	Fuel System Feed Back Factor	pu	0
K4	Radiation Shield Prop. Factor	pu	0.9
K5	Radiation Shield Integr. Factor	pu	0.1
T3	Radiation Shield Time Constant	s	18
T4	Thermocouple Time Constant	s	3
Tt	Temperature Controller Time Constant	s	500
T5	Temperature Controller Gain	pu	3.3
af2	Turbine Characteristic, Constant	pu	-0.885
bf2	Turbine Characteristic, Torque	pu	1.3
cf2	Turbine Characteristic, Speed	pu	0.5
Tr	Rated Exhaust Temperature	°F	1000
Tc	Temperature Control	°F	1300
af1	Turbine 1st Factor	pu	758.3
bf1	Turbine 2nd Factor	pu	1.3
K6	Compressor Factor	pu	0.469

Notes:

1. Value based on actual Gas Turbine Power Rating

4.4.3 GASTWD – GAS TURBINE MODEL

This model (see Figure 4.6) is in essence the same model as the PSS/E GAST2A Gas Turbine model [2], except that the DigSilent GASTWD model requires the rated power of the turbine as an input variable. The “Trate” block works similar in this model as is the case in the GAST2A model. Refer to Section 4.4.2 for a detailed description.

There are two other differences between the PSS/E and the DigSilent model. The first is the calculation of the deviation of the speed from unity speed. The PSS/E model receives the deviation value as an input value, while it is calculated in the DigSilent model. The second difference is in the Digsilent model, “psetp” (Load set point value) can be an input to the model. This is used to change the loading of the turbine. The same philosophy is followed for the descriptions in each of the blocks in Figure 4.6 as for the blocks in Figure 4.4 and Figure 4.5.

Refer to Section 4.3.3 for a description of the model.

Figure 4.6 shows the DigSilent PowerFactory GASTWD gas turbine model.

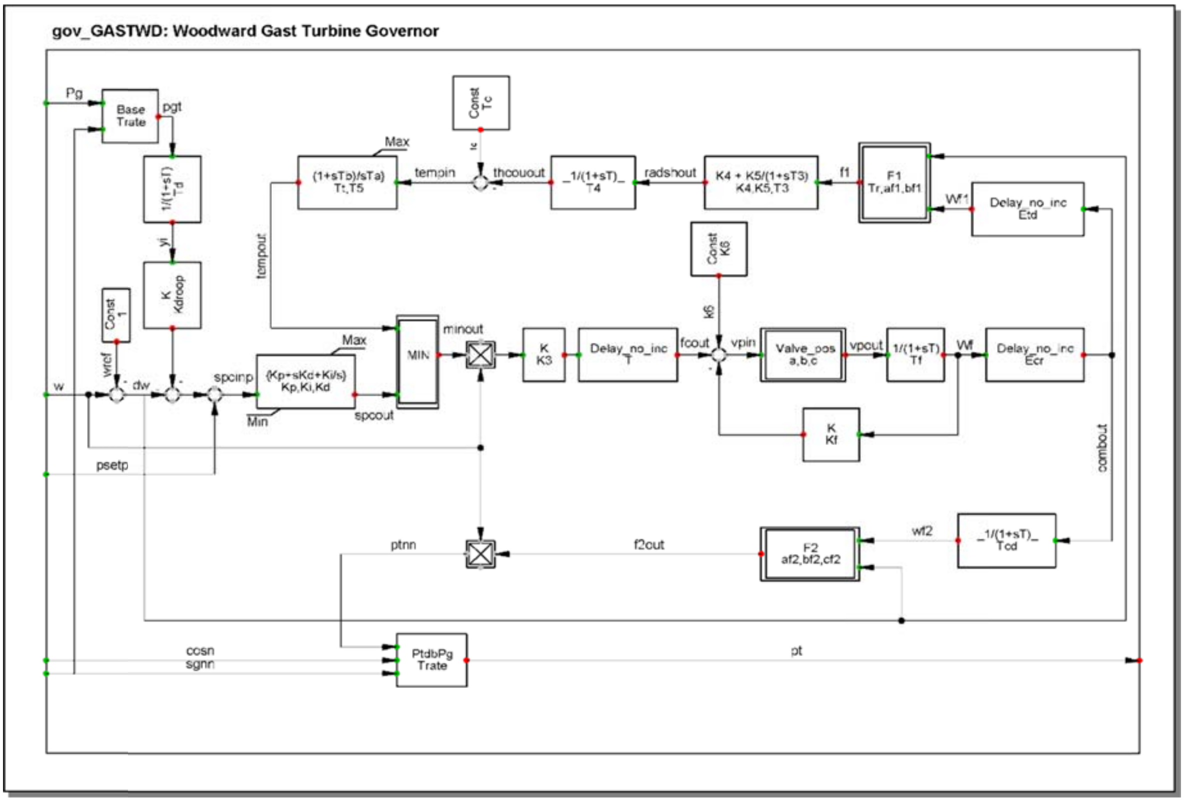


Figure 4.6: DigSilent PowerFactory GASTWD Model [3]

Table 4.13. DigSilent GASTWD model variables

Variable	Description	Unit	Typical Value
Kdroop	Power Controller Droop	pu	0.05
Td	Power Transducer Delay Time	s	6
Kp	Speed Controller Proportional Factor	pu	20
Ki	Speed Controller Integral Factor	1 / s	5
Kd	Speed Controller Derivative Factor	pu	0
Etd	Turbine Exhaust Delay	s	0.04
Tcd	Turbine Dynamics Delay Time	s	0.49
Trate	Turbine Rated Power	MW	Note 1
T	Fuel Control Delay Time	s	0.05
Min	VCE Lower Limit	pu	-0.1
Max	VCE Upper Limit	pu	1
Ecr	Combustor Delay Time	s	0.01
K3	Turbine Factor	pu	0.77
A	Valve Positioner Prop. Characteristic	pu	1
B	Valve Positioner Time Constant	s	0.1
C	Valve Positioner I/L Factor (0/1)	pu	1
Tf	Fuel System Delay	s	0.79
Kf	Fuel System Feed Back Factor	pu	0
K4	Radiation Shield Prop. Factor	pu	0.8
K5	Radiation Shield Integr. Factor	pu	0.2
T3	Radiation Shield Time Constant	s	15
T4	Thermocouple Time Constant	s	2.5

Variable	Description	Unit	Typical Value
Tt	Temperature Controller Time Constant	s	450
T5	Temp. Contr. Derivative Time Constant	s	3.3
af2	Turbine Characteristic, Constant	pu	-0.3
bf2	Turbine Characteristic, Torque	pu	1.3
cf2	Turbine Characteristic, Speed	pu	0.5
Tr	Rated Exhaust Temperature	°F	1001
K6	Compressor Factor (Min. Flow)	pu	0.23
Tc	Temperature Control Set point	°F	1001
af1	Turbine 1st Factor	pu	700
bf1	Turbine 2nd Factor	pu	550

Notes:

1. Value based on actual Gas Turbine Power Rating

4.5 PSS/E AND DIGSILENT MODEL COMPARISON

From the discussion in Section 4.4 it can be seen that the PSS/E and the DigSilent GAST, GAST2A and GASTWD gas turbine models are in essence the same models, except that the DigSilent models require an additional input for the turbine rated power for the GAST, GAST2A and GASTWD models. However, if the user enters a “0” the value is calculated internally from the relevant input signals.

The inclusion of a second power simulation package’s gas turbine models was done to show that implementation of the gas turbine models (GAST, GAST2A and GASTWD) is not limited to one package only as these models are proven models [2], [3].

4.6 EXISTING AND NEW MODELS COMPARISON

From the discussion in Chapter 3 and 4, the following table gives a comparison between the existing models (GAST, GAST2A and GASTWD) and the open cycle gas turbine (OCGT) and combined cycle power plant models (CCPP). Only the main control loops / features of the models are highlighted. The new open cycle gas turbine and combined cycle power plants are the new models as developed by Cigre [1]. For the purpose of this dissertation, the models will be referred to as OCGT (gas turbine model) and CCPP (combined cycle power plant model).

Table 4.14. Existing and New Gas Turbine Model Comparison

Control Loop / Feature		GAST	GAST2A	GASTWD	OCGT	CCPP
Ambient Temperature Limit		Yes	Yes	Yes	Yes	Yes
Speed Governor	Lead-lag Compensator	Yes	Yes	-	-	-
	PID Controller	-	-	Yes	Yes	Yes
Governor Mode	Droop	Yes	Yes	Yes	Yes	Yes
	Isochronous	-	Yes	Yes	Yes	Yes
Temperature Control		-	Yes	Yes	Yes	Yes
Fuel Flow Control		Yes	Yes	Yes	Yes	Yes
Gas Turbine Dynamics		-	Yes	Yes	Yes	Yes
Acceleration Control		-	-	-	Yes	Yes
HRSG Controls	Supplemental Firing	-	-	-	-	Yes
	Turbine Dynamics					
	Turbine Speed Control					
	Bypass Valve					

The table shows that the existing gas turbine models and the new models have several features / control loops in common. The main difference is the governor controller type between the GAST and GAST2A models and the other models. For the GAST and GAST2A models, it is a lead-lag compensator while all the other models make use of a PID controller. The additional feature / control loop for the OCGT model is the acceleration control loop. All the models can be set to be used in isochronous mode, except the GAST model. The CCPP model includes the HRSG in its controls / features.

4.7 SUMMARY

This chapter described the various gas turbine governor models available for the different available gas turbines. The available models were obtained from the PSS/E manuals [2] and the DigSilent PowerFactory [3] manuals.

The available (or existing) models are GAST, GAST2A and GASTWD. These models were developed to model only open cycle gas turbines and their controls. The GAST model is the most commonly used model [8]. The model is simple and WSCC compliant [8]. (A WSCC compliant model is a model that can be directly used in specific commercial power system simulation software). Both PSS/E [2] and DigSilent PowerFactory [3] have the model included. GAST2A was developed by General Electric to represent machines of a specific frame sizes built by them [8].

It is clear that the standard (or existing) dynamic models in the simulation software are not optimally developed for the simulation of certain types of combined cycle power plants. The GAST, GAST2A, and GASTWD models are designed for open cycle gas turbines only. The gas turbine model (OCGT) was developed for an open cycle gas turbine. The combined cycle power plant model (CCPP) was developed for combined cycle power plants, in a variety of configurations. The various configurations can be modelled by adapting the model to represent the various plant configurations [1].

Although the various simulation packages use the same models, it is shown that these models are the standard for traditional gas turbine models, and therefore the adaptation of the new gas turbine and combined cycle power plant models are necessary. The existing models also do not make provision for the combined cycle power plants that include the heat recovery steam generators (HRSG) and its controls. A comparison was made on the functional basis of each model. In the coming chapter each model will be dynamically assessed.

- Chapter 5 -

5 SIMULATIONS / EVALUATION

5.1 INTRODUCTION

Once the dynamic model for the gas turbine (OCGT) and the combined cycle power plant model (CCPP) were adapted to be used in the power system simulation software (DigSilent PowerFactory), an electrical test network was developed for the simulations and evaluation (see section 5.2). Although gas turbine models exist, a new gas turbine model (OCGT) was adapted. As shown in the previous chapter, the new gas turbine model (OCGT) combined all the features of the existing gas turbine models and an added feature, namely acceleration control. The new gas turbine model therefore complements the existing gas turbine models, but it does not replace them. The combined cycle power plant model (CCPP) included the new gas turbine model and the heat recovery steam generator model. There are no existing models for combined cycle power plants.

The simulations were performed using the network in Section 5.2 to show the similarities and differences between the new gas turbine model and the existing gas turbine models. The difference between the new gas turbine model and the existing gas turbine models is highlighted. The differences between the combined cycle power plant model and open cycle gas turbine models are also discussed.

5.2 SIMULATION NETWORK

The basis for this network is the well-known IEEE Nine Bus network (documented in [23]) that is found in handbooks and used for transient stability studies as a test case. The network is shown in Figure 5.1. In this dissertation, the network was modified and two buses were added. Additional loads and an external grid were added. An additional generator with its associated generator transformer was added. This additional generator, with its prime mover and controls were used for the new models, i.e. the OCGT and CCPP models. The same generator was used for modelling the existing models, i.e. GAST, GAST2A and GASTWD. This was done to compare the five models in the same network, with the same conditions before and after disturbances in the network.

Figure 5.1 shows the modelled network.

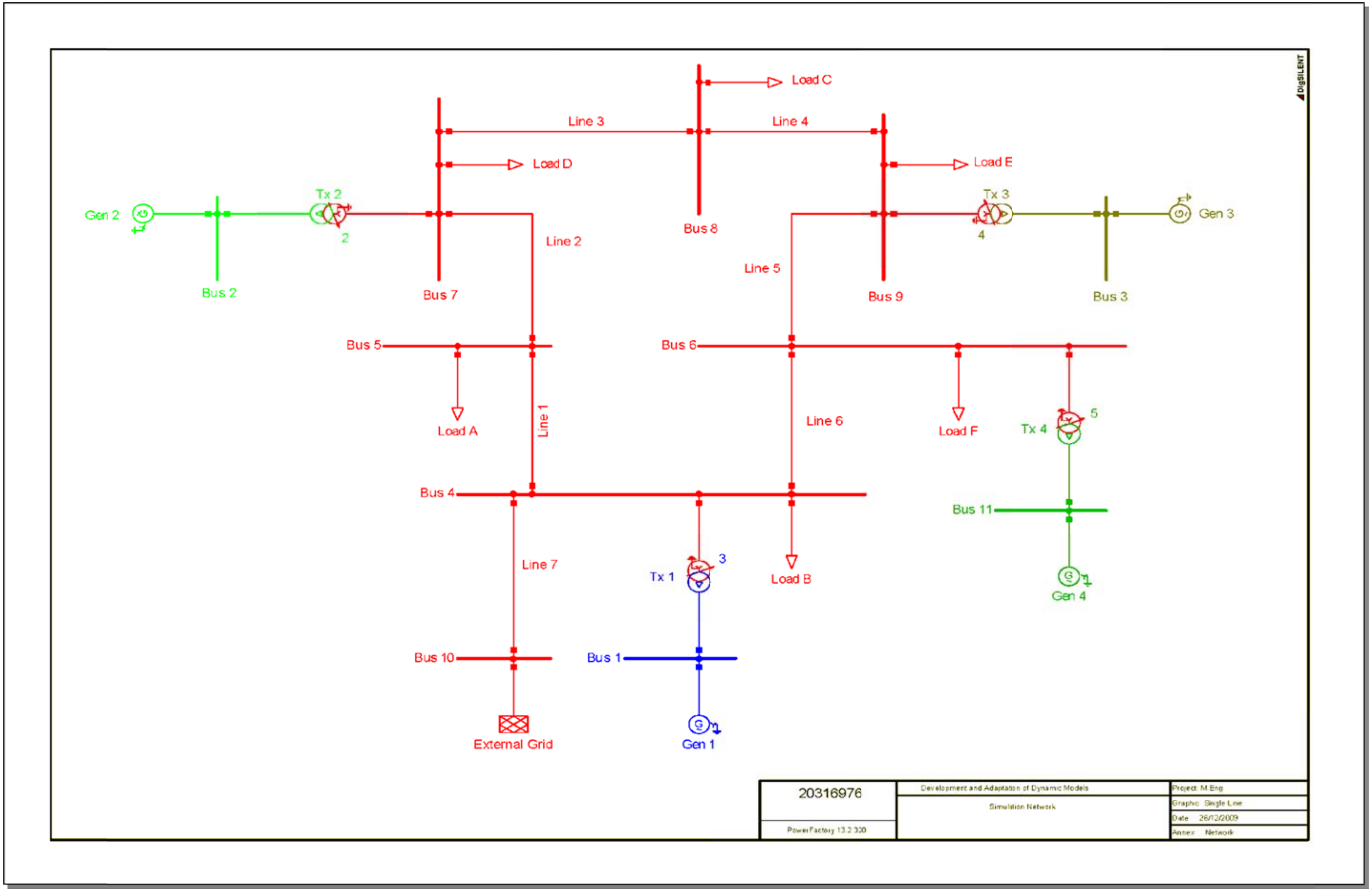


Figure 5.1: Modelled Network Single Line Diagram

The electrical network consists of a ring network (220 kV overhead lines), interconnecting six 220 kV busbars. The interconnectors have various lengths, but for the purpose of the network, the impedance per unit length of the overhead lines was set the same.

The same electrical network was used for all the simulations. DigSilent PowerFactory has the ability to create revisions / variations of the same network, allowing individual changes to each of the revisions / variations of the network. A change in the base network also changes all the revisions / variations of the network, but changes in the revisions / variations do not change the base network. This feature created the opportunity to run simulations for each of the five gas turbine models with the same base network and same disturbance. This made comparing the various gas turbine models easy.

An external grid was added as the “Swing Bus” of the network. The external grid was set to model a large power system that is not influenced by disturbances in a network.

5.3 AIM

The aim of the simulations was to investigate the functionality of the models (OCGT and CCPP) and compare it to the existing gas turbine models’ performance (i.e. GAST, GAST2A and GASTWD). Both the open cycle gas turbine and the combined cycle version of the Cigre models were studied. The reason is to show that the new gas turbine model contains all the characteristics of the existing gas turbine models, but includes an additional control, namely acceleration control. The simulations are set up to show the similarities and differences between the new and existing gas turbine models.

The combined cycle power plant model contains the heat recovery steam generator and the steam turbine. Additional studies were performed to show the differences between combined cycle power plant and open cycle gas turbines.

5.4 METHODOLOGY.

5.4.1 NETWORK MODEL

As described in Section 5.1, a simple electrical network was developed in DigSilent PowerFactory (version 14.0) to model the various gas turbine models. A base network was developed as described in Section 5.1 (with the single line diagram included in the section and equipment data provided in Appendix A).

The external grid was modelled as a power system that is not influenced by disturbances in the network, i.e. its inertia (or acceleration time constant as defined by the software) was set with a very high value. Line 7 has a length of 100 km to put the power system (external grid) an electrical distance away from the network to be studied. The power system simulation software requires that there must be at least one bus that must be set as a “swing bus”. A swing bus is a bus that is used by the simulation software to balance the power requirements of the simulated network. The external grid is also used as a reference in the simulated network.

Generators Gen 1 to Gen 4 are of different sizes. (Refer to Appendix A). The generators were set with different inertias, different power ratings and different voltage ratings. The connected transformers therefore also have different power, impedance and secondary voltage ratings. The same voltage controller type (IEEE Type 3) was applied on all four generators. The same type of steam turbine controller (IEEE Type 1) was applied on generators Gen 1 to Gen 3.

Various size loads (Load A to F) were connected to the 220 kV (red colour) buses. These loads were modelled as voltage dependent loads.

5.4.2 SETTING UP OF GAS TURBINE MODELS

To achieve the first part of the aim (i.e. to show the similarities between the new gas turbine model and the existing gas turbine models), the various gas turbine models were set up with the same values for the same parameter of each model. Typical values were used for the parameters. These values were obtained from the PSS/E documentation [2]. For the new gas turbine model and the combined cycle power plant model, the additional functions were disabled by setting the relevant parameter to the appropriate value. To disable the functions, the relevant parameter was set to zero [1].

To achieve the second part of the aim (i.e. to show the differences between the new gas turbine model and the existing gas turbine models), the functions that were disabled for the first part were enabled by setting the relevant parameter of the specific function to an appropriate non-zero value [1].

5.4.3 SIMULATIONS

The same network was used in the simulations and evaluations of the various gas turbine models. The GAST, GAST2A, GASTWD, and the new Cigre models (OCGT, and CCPP) were included individually in each revision.

The same disturbances were modelled in each variation (revision) and the results were plotted and compared.

The first step was to set up the various gas turbine models (where possible) to achieve the same results. Although the models are different, parameters that represent the same equipment (i.e. control, time delay, time constant or gain) were set the same in the various models. The reason was to evaluate the response of the models to the same disturbance and compare the results of the various models. The aim was to show that the new gas turbine model complements the existing gas turbine models, and that it does not replace the existing models.

The second step was to set up the additional features of the new gas turbine model. The aim was to show the difference between the new gas turbine model and the existing gas turbine models. The new combined cycle power plant model was set up to show the differences between combined cycle power plants and open cycle gas turbines.

The following disturbances were modelled to show that the new gas turbine model contains similar characteristics as the existing gas turbine models. These disturbances were chosen, as they represent most types of disturbances that occur in a power system. The chosen disturbances represent typical disturbances that occur in a power system. The aim of these simulations is to show that the new gas turbine model (OCGT) does not replace the existing gas turbine models, but complements them. The hypothesis is that under these fault conditions the new model should react in a similar fashion as the existing models. The selected disturbances are:

- Three-phase fault on the gas turbine generator terminals, fault cleared in 100 ms, Gen 4 not tripped,
- Three-phase fault on Line 6, close to Bus 6, cleared in 100 ms, Line 6 tripped,
- Line 6 switched out of service,
- Generator Gen 3 switched out of service,
- Generator Gen 4 output decreased by 50% of machine rated output,
- 100% Load shedding of Load D (switched out of service),
- Critical clearing time for Gen 4 (three-phase fault on generator terminals)

For all the above disturbances two systems were modelled, i.e. a high inertia and a low inertia system. For the high inertia system, the external grid was set with a very high inertia (or acceleration time constant). A low value for the external grid was selected for the low inertia system.

The following disturbances in the network and changes to the models were modelled to show the differences between the new gas turbine model (OCGT) and the existing gas turbine models. The characteristics of the combined cycle power plant were modelled to show the differences between the combined cycle power plant and open cycle gas turbines. The disturbances were chosen based on the major differences between the heat recovery steam generator (HRSG) model and existing steam turbine models, to show the differences of the new CCPP model (i.e. to achieve the second aim):

- The acceleration control loop of the OCGT model. A three-phase fault on the gas turbine generator terminals to show the response of the acceleration control,
- Supplemental firing on the HRSG (part of the CCPP model). Two variations of the supplemental firing were modelled.
- Opening the bypass valve on the HRSG (part of the CCPP model).

5.5 RESULTS

The simulations are aimed at the study of the response of Generator 4 (i.e. the gas turbine and combined cycle power plant generator). A comparison between the results of the new and the proven gas turbine models are made.

5.5.1 THREE-PHASE FAULT ON GAS TURBINE TERMINALS

Expected outcome:

The expected outcome of this simulation is that the five gas turbine models will provide the same results. The final steady state operating points after the disturbance are expected to be the same as for the pre-disturbance period.

High and low inertia systems: (Result 5.5.1.a to Result 5.5.1.d)

For this disturbance, a fault on the gas turbine generator (Gen 4) terminals was modelled; the duration of the disturbance was 100 ms, without tripping the generator, or any other breaker in the network.

During the fault, the generator terminal voltage, the active, the reactive and the apparent power dropped to zero, as no impedance was modelled in the fault. The power is zero as the voltage on the generator terminal is zero during the fault period. However, the generator current increases during the fault period. During the fault condition, the speed of the generator increases as the prime mover (the gas turbine) is producing active power while the generator is not delivering any active power to the network. The rotor angle increases during the fault condition.

After the fault condition, the terminal voltage, the reactive power, the current and the power return to the pre-fault values. The speed and rotor angle oscillate around the pre-disturbance value and return to the pre-disturbance value after a while. The speed returns to the rated speed. The system reaches a steady state operating point after a while.

The simulations were performed without tap changing for the period of the simulation.

Conclusion:

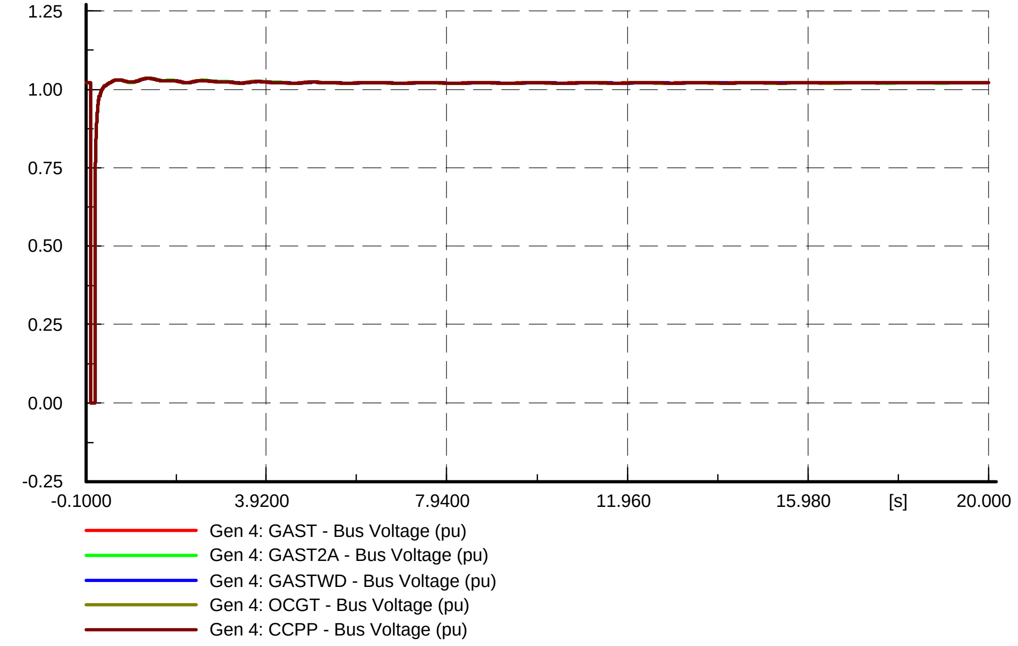
The results show that the new gas turbine model’s response is similar to the existing gas turbine models for the same disturbance. This indicates that the new gas turbine model is settable to achieve the same results as the existing gas turbines. All the gas turbine models return to the pre-disturbance operating points, as there was no change in the network configuration.

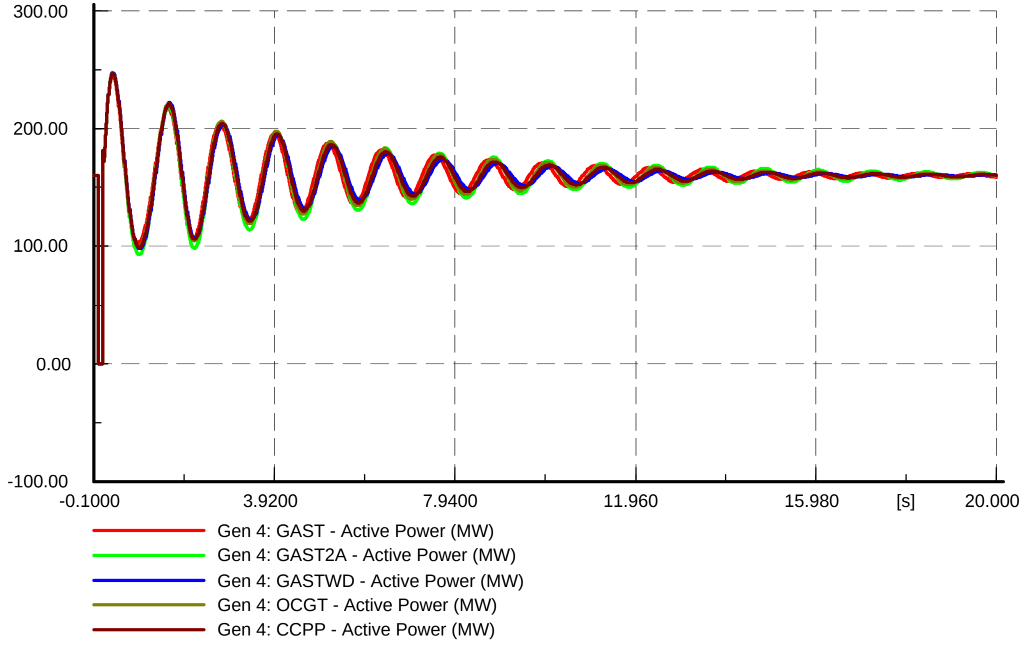
High inertia system: (Result 5.5.1.a and Result 5.5.1.b)
The system is stable during and after this disturbance, as can be seen from the various graphs.

The results show that the various gas turbine models' response to the disturbance is similar with small variations between the models. The five gas turbine models' response is almost identical during the first approximately 1.5 seconds, where after the responses of the models start to deviate from each other. Although there are small variations between the various models' responses, in general the same results are obtained. The differences between the results for the five models are not critical.

The oscillations in the network take a while to disappear, as the inertia of the grid (swing bus of the network) is very high. The high inertia of the grid causes the whole system to respond slower to disturbances than a system with a lower inertia.

Low inertia system: (Result 5.5.1.c and Result 5.5.1.d)
The results show that the oscillations disappear faster for the system with a low inertia. The low inertia of the grid causes the whole system to respond faster than the system with the high inertia. The differences between the results of the five models are small and not critical.





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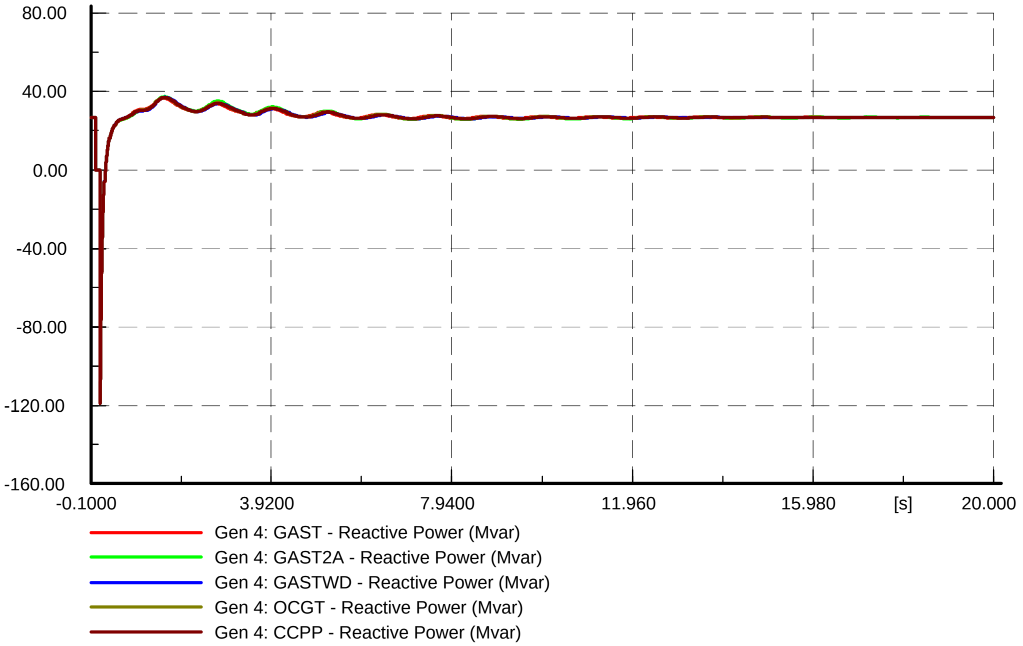
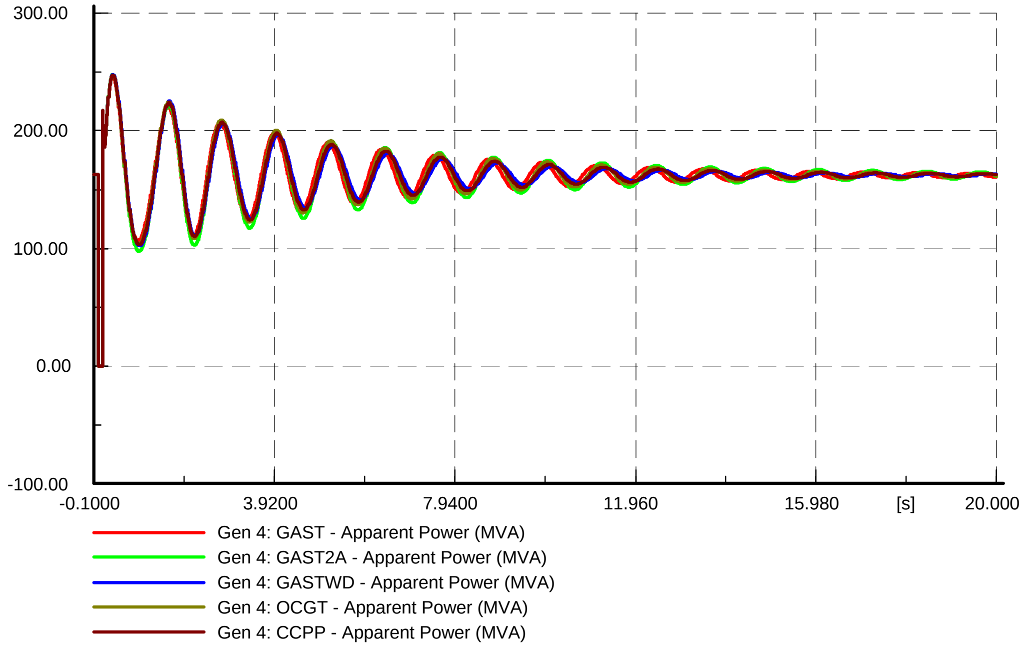
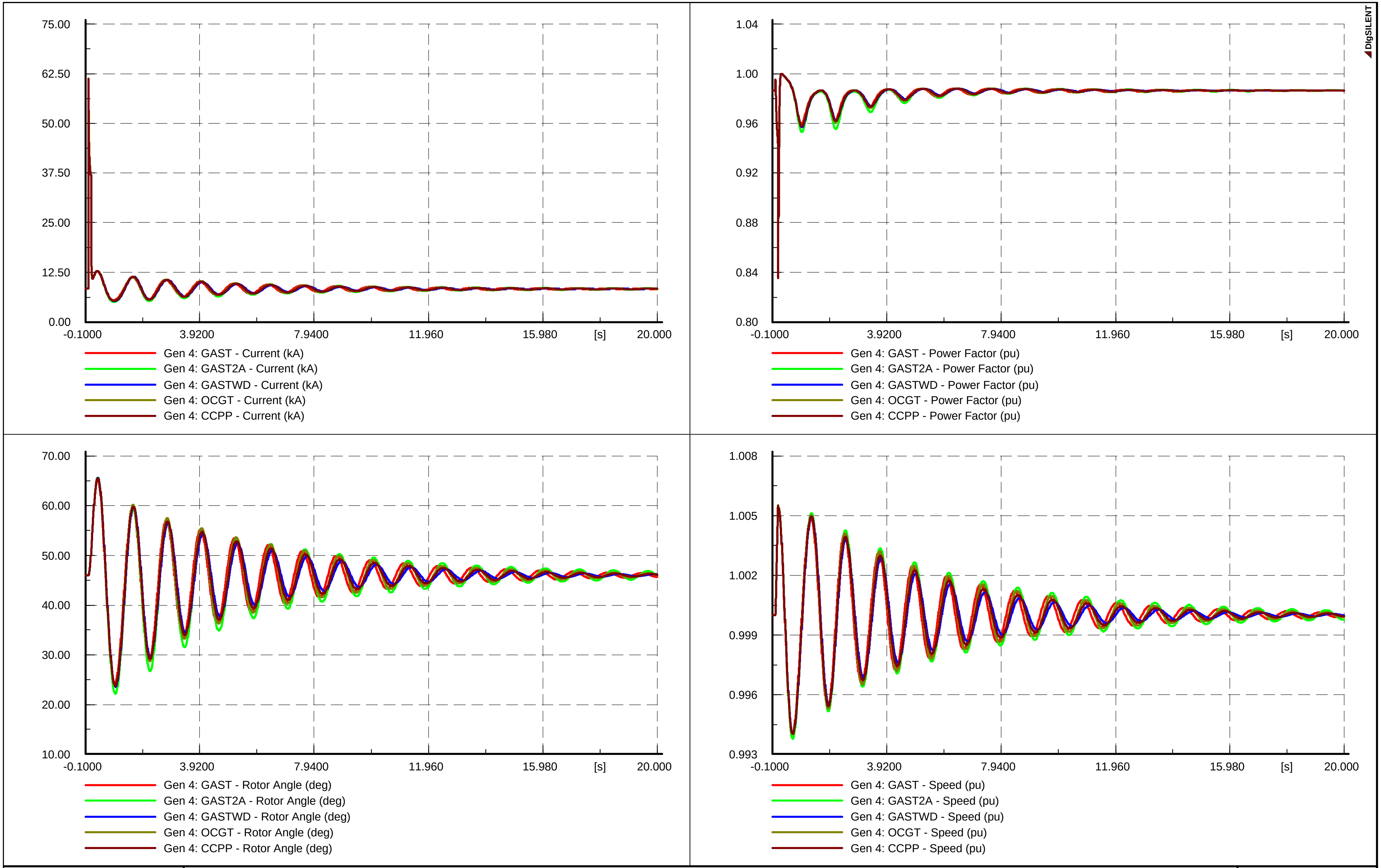




20316976	Development and Adaptation of Dynamic Models	Results 5.5.1.a	Date: 25/04/2011
	3 Phase fault on Generator 4 terminals (100 ms, Generator 4 not tripped)	External grid with high acceleration time constant	Annex: /1



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Development and Adaptation of Dynamic Models

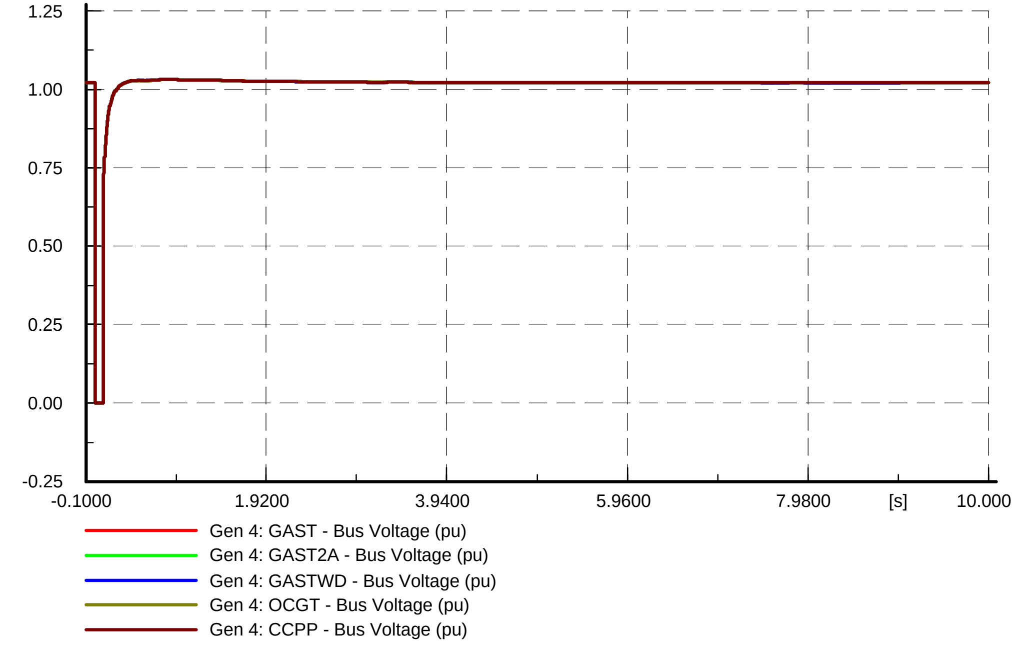
Results 5.5.1.b

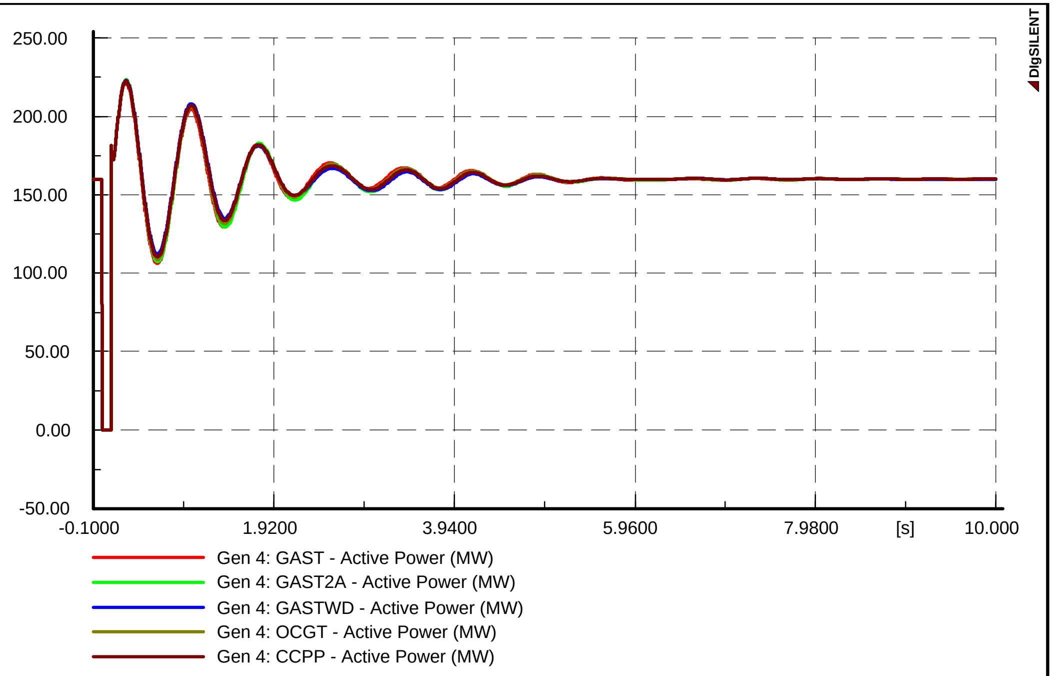
Date: 25/04/2011

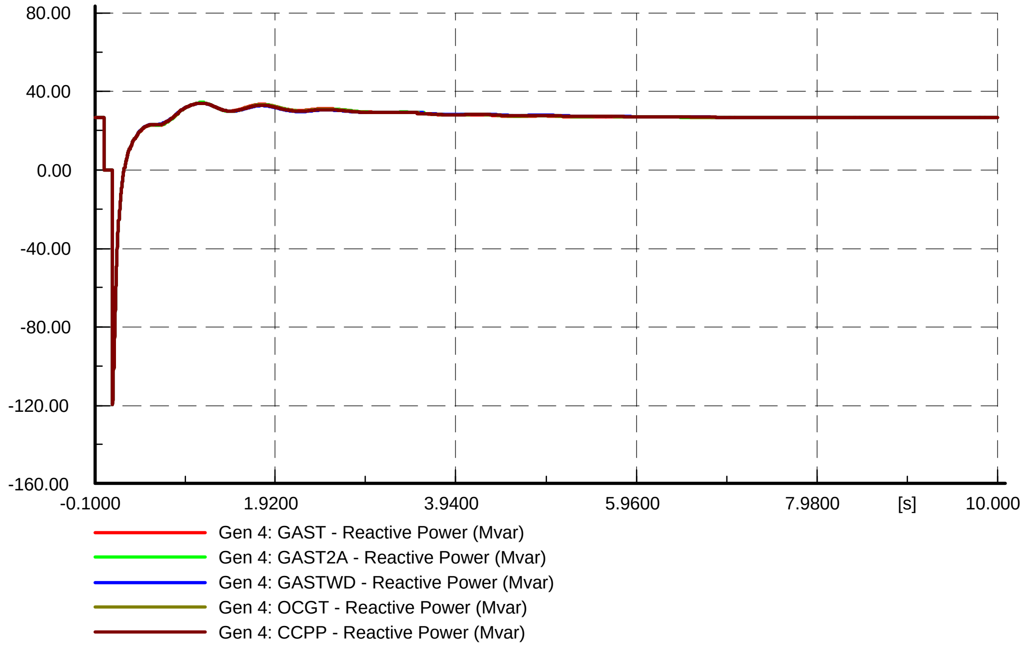
3 Phase fault on Generator 4 terminals (100 ms, Generator 4 not tripped)

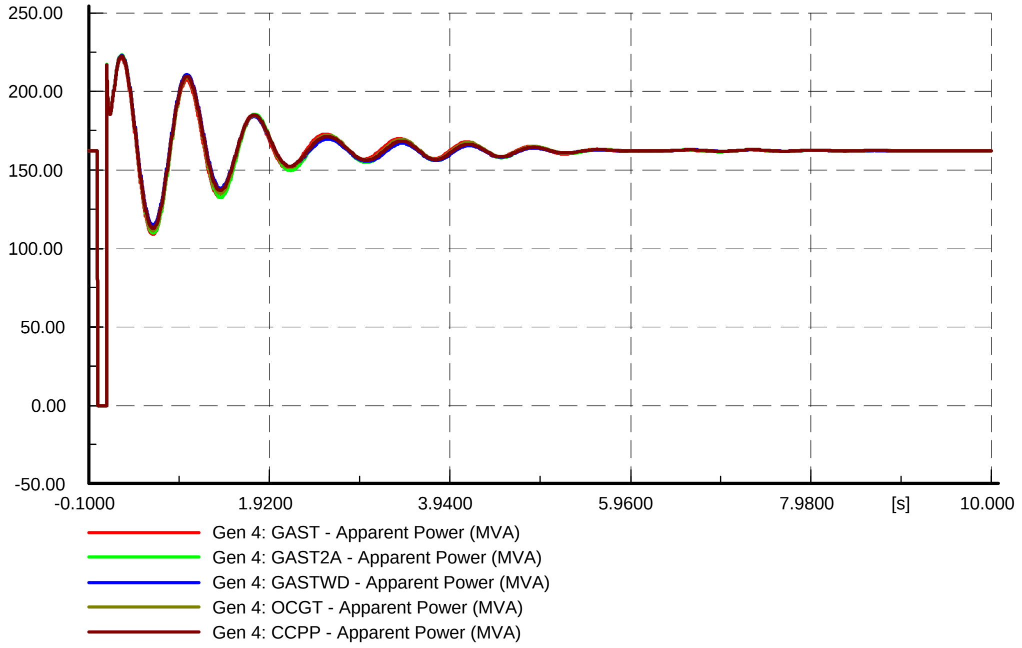
External grid with high acceleration time constant

Annex: /2

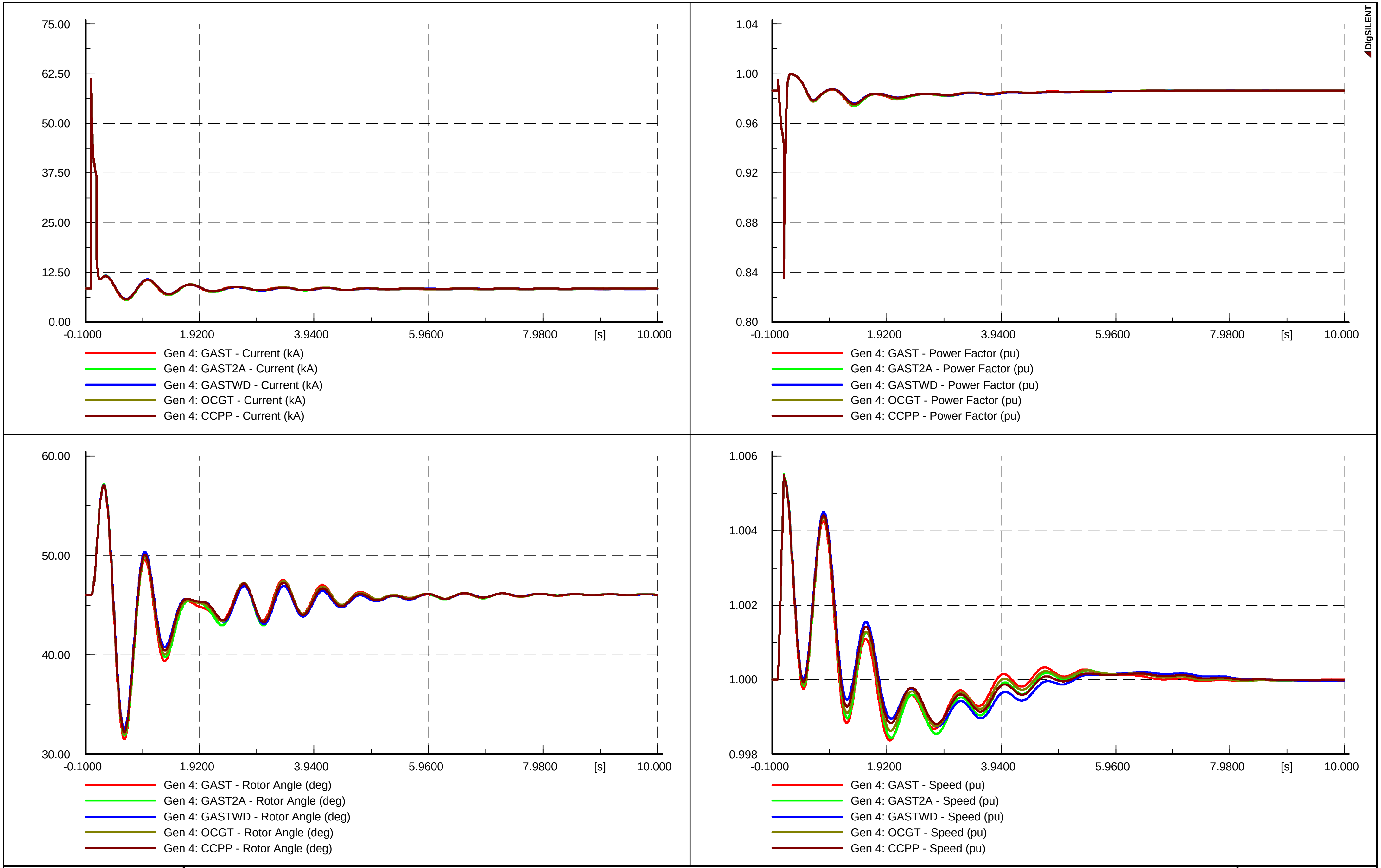








20316976	Development and Adaptation of Dynamic Models 3 Phase fault on Generator 4 terminals (100 ms Generator 4 not tripped)	Results 5.5.1.c External grid with low acceleration time constant	Date: 08/11/2010 Annex: /1



20316976	Development and Adaptation of Dynamic Models 3 Phase fault on Generator 4 terminals (100 ms Generator 4 not tripped)	Results 5.5.1.d External grid with low acceleration time constant	Date: 25/04/2011 Annex: /2

5.5.2 THREE-PHASE FAULT ON LINE 6

This disturbance was modelled as a three-phase fault on Line 6, close to Bus 6 and cleared in 100 ms, by tripping both the breakers of Line 6.

Expected outcome:

The expected outcome of this simulation is that the five gas turbine models will provide similar results, and that the operating points (i.e. active power and speed) for the gas turbine models will be same after the disturbance compared to the pre-disturbance period. Differences are expected for the voltage, the reactive power, the power factor, the apparent power and the rotor angle, as a change in the network configuration was modelled, for all gas turbine models.

High and low inertia system: (Result 5.5.2.a to Result 5.5.2.d)

During the disturbance, the generator terminal voltage dropped to 0.3 pu, as there is impedance between the generator and the fault (i.e. the Gen 4 generator transformer). As a result, the active, the reactive and the apparent power are higher than for a three-phase fault on the generator terminals. The power factor is very close to zero as the path between the generator and the fault is consisting mainly of reactance. During the fault condition, the speed of the generator increases as the source of the generator (the gas turbine) is producing active power while the generator is not delivering the same amount of active power to the network.

After the disturbance, the terminal voltage, the reactive power, the current and the power return to the pre-disturbance values, with relatively small differences in the values compared to the pre-disturbance values. The speed and rotor angle oscillates around the pre-disturbance value and returns to the pre-disturbance value after a while. The speed returns to the rated speed. The system reaches a steady state operating point after a while.

The simulations were performed without tap changing for the period of the simulation.

Conclusion:

> The results show that the new gas turbine model's response is similar to the existing gas turbine models for the same disturbance. This indicates that the new gas turbine model is settable to achieve the same results as the existing gas turbine models. The results show that the gas turbine returns to the pre-disturbance operating points of speed and active power. As the change in the network configuration is small (i.e. a small change in the network impedance as seen from generator 4), the voltage, the reactive power, the power factor, the apparent power and the rotor angle settle at slightly different values compared to the pre-disturbance values.

High inertia system: (Result 5.5.2.a and Result 5.5.2.b)

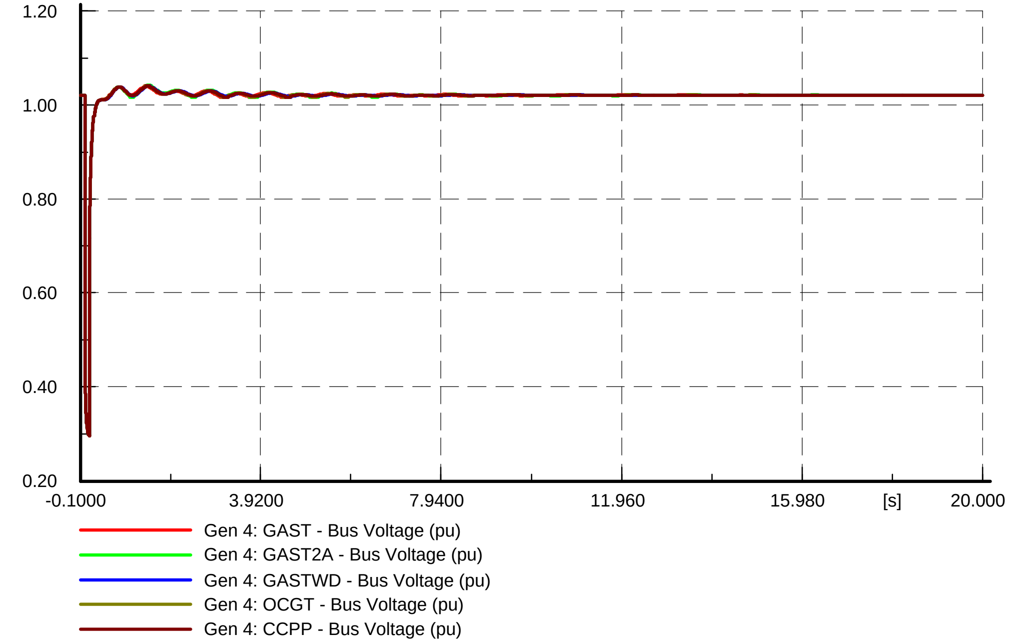
The system is stable during and after this disturbance, as can be seen from the various graphs.

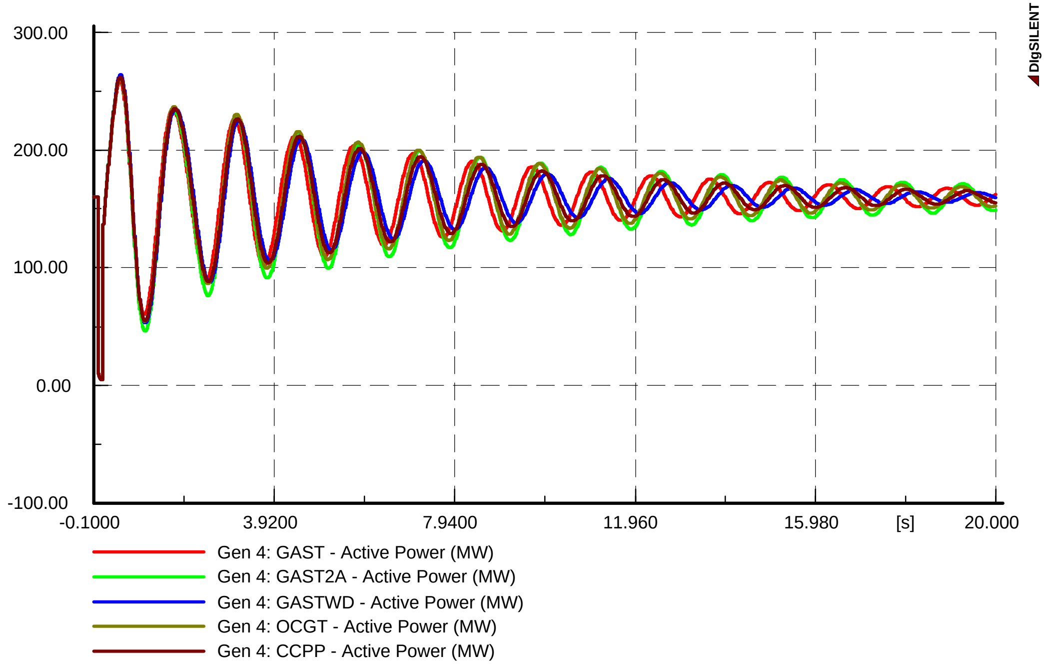
The results show that the various gas turbine models' response to the disturbance is similar with small variations between the various gas turbine models. The gas turbine models' response is almost identical during the first approximately 1.5 seconds, where after the responses of the models start to deviate from each other, but the differences are negligible. Although there are small variations between the various models' responses, in general the same results are obtained.

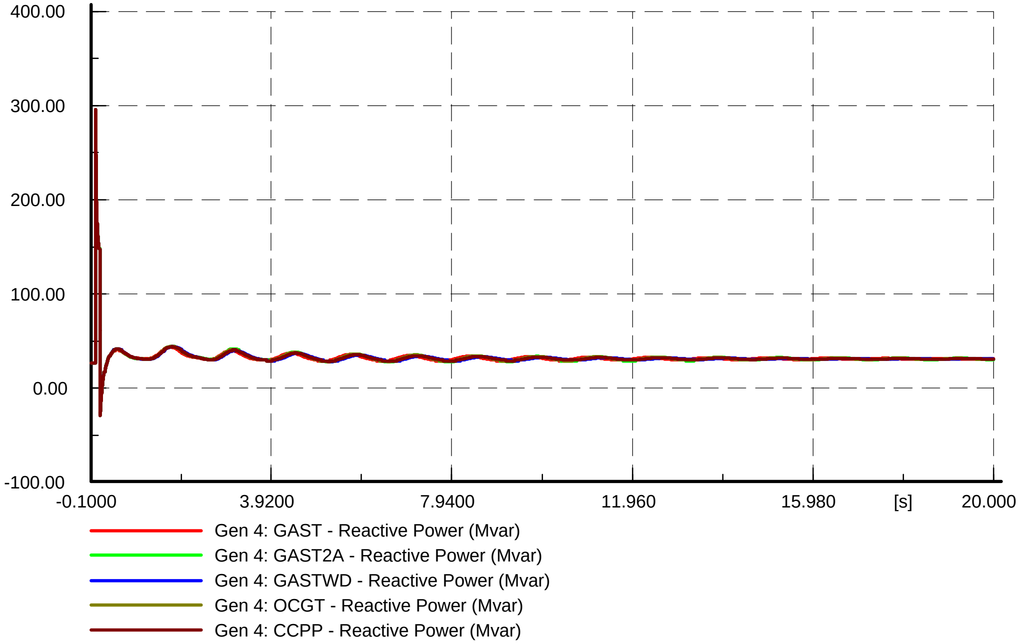
The oscillations in the network take a while to disappear, as the inertia of the grid (swing bus of the network) is very high. The high inertia of the grid causes the whole system to respond slower to disturbances than a system with a lower inertia. The system reaches stability after a while as the oscillations are damped.

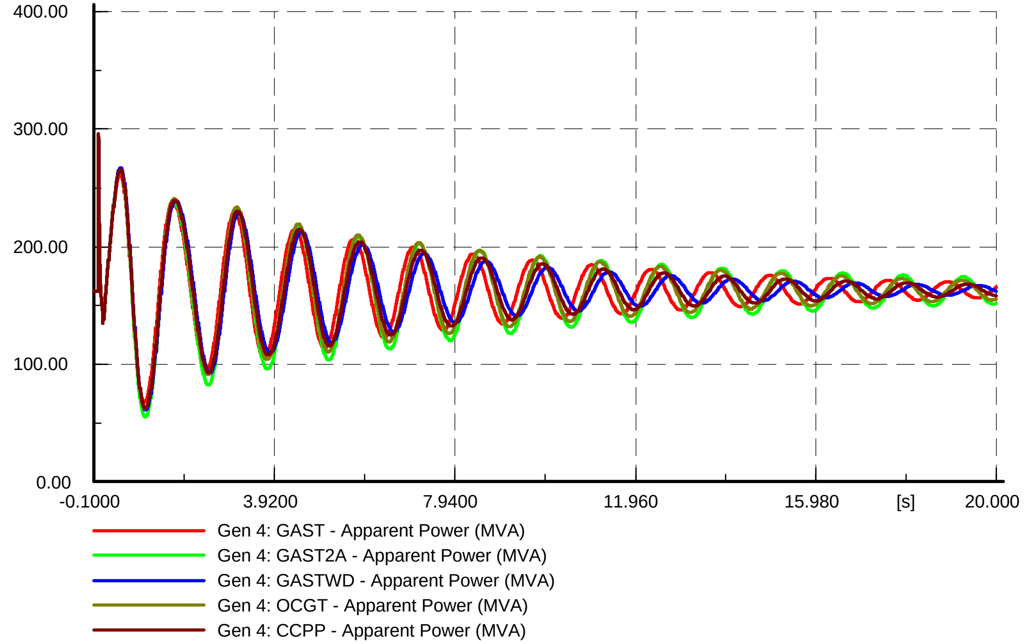
Low inertia system: (Result 5.5.2.c and Result 5.5.2.d)

The results show that the oscillations disappear faster for the system with a low inertia. The low inertia of the grid causes the whole system to respond faster than the system with the high inertia. The low inertia system contains less rotating mass than a high inertia system and therefore the variation in speed is easier to control. The oscillations are more damped in the low inertia system, and therefore they exist for a shorter period before the oscillations disappear completely, compared to the high inertia system.









20316976

Development and Adaptation of Dynamic Models

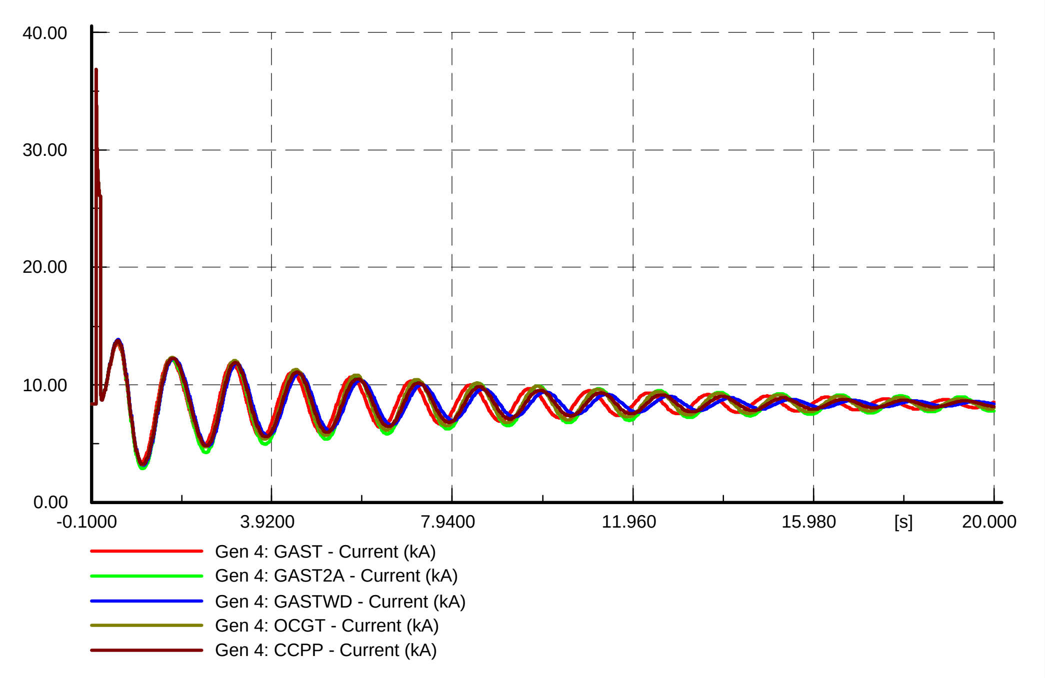
3 Phase fault (100ms) on Line 6, close to Bus 6, Line 6 tripped

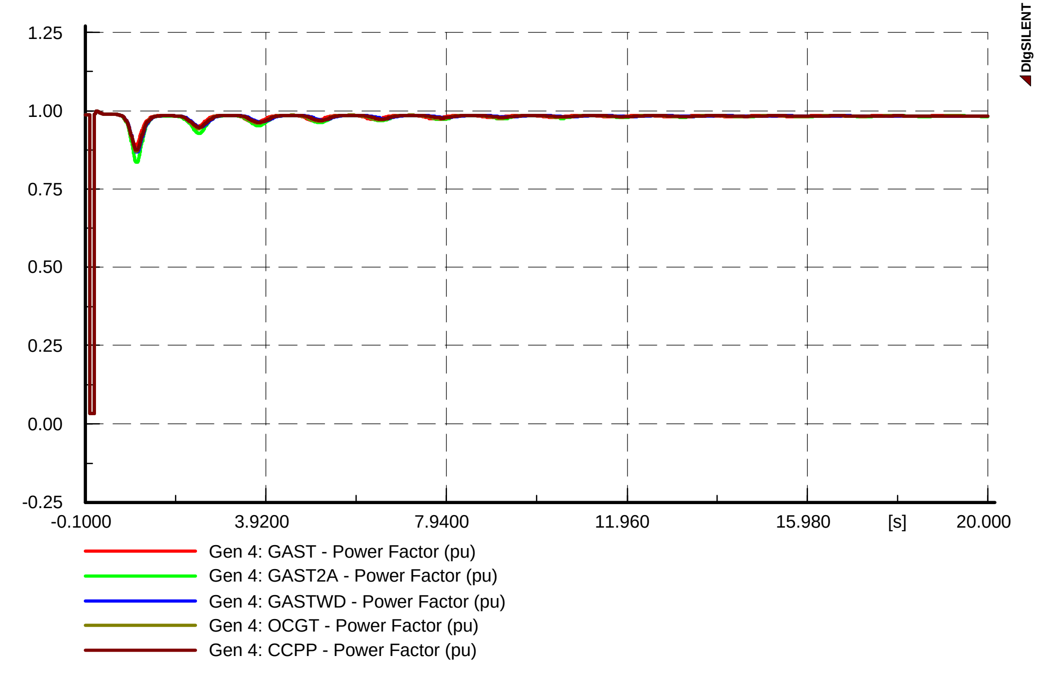
External grid with high acceleration time constant

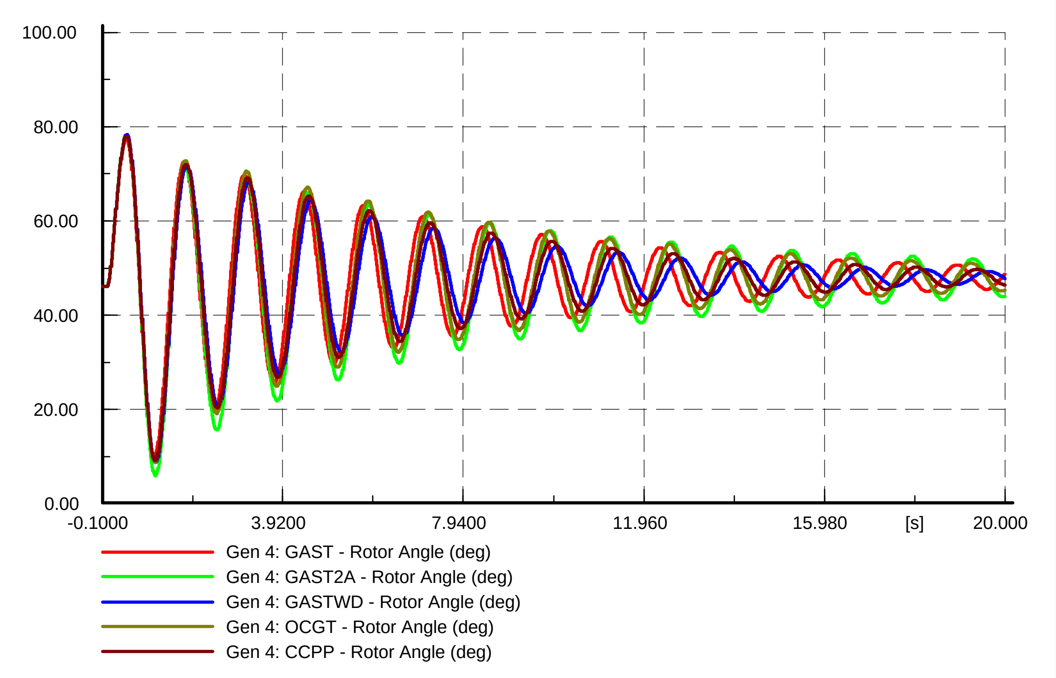
Results 5.5.2.a

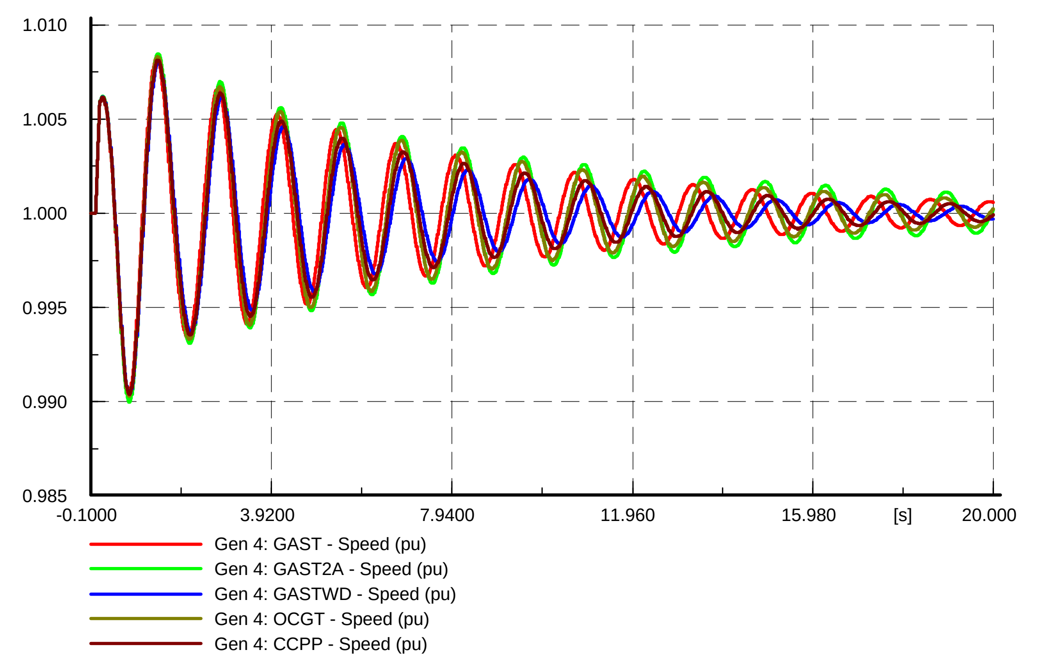
Date: 25/04/2011

Annex: /1









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Development and Adaptation of Dynamic Models

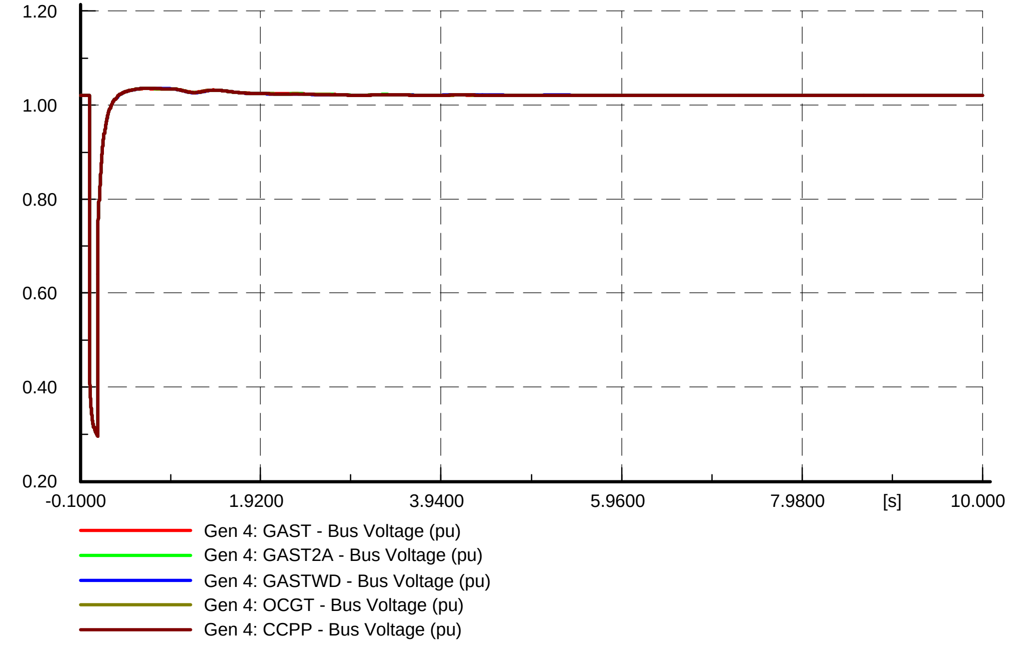
Results 5.5.2.b

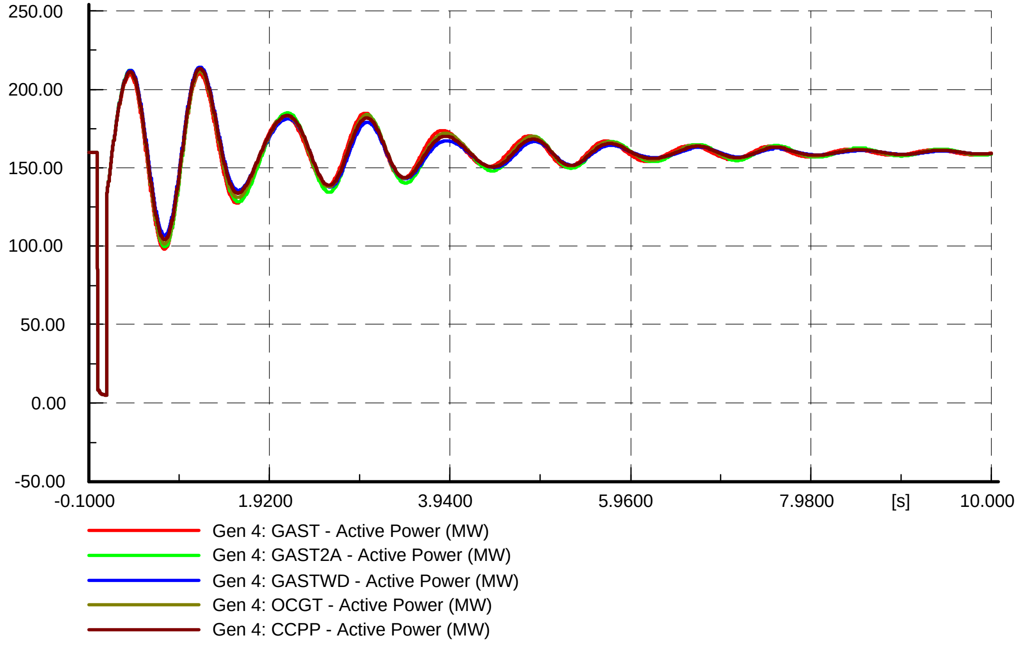
Date: 25/04/2011

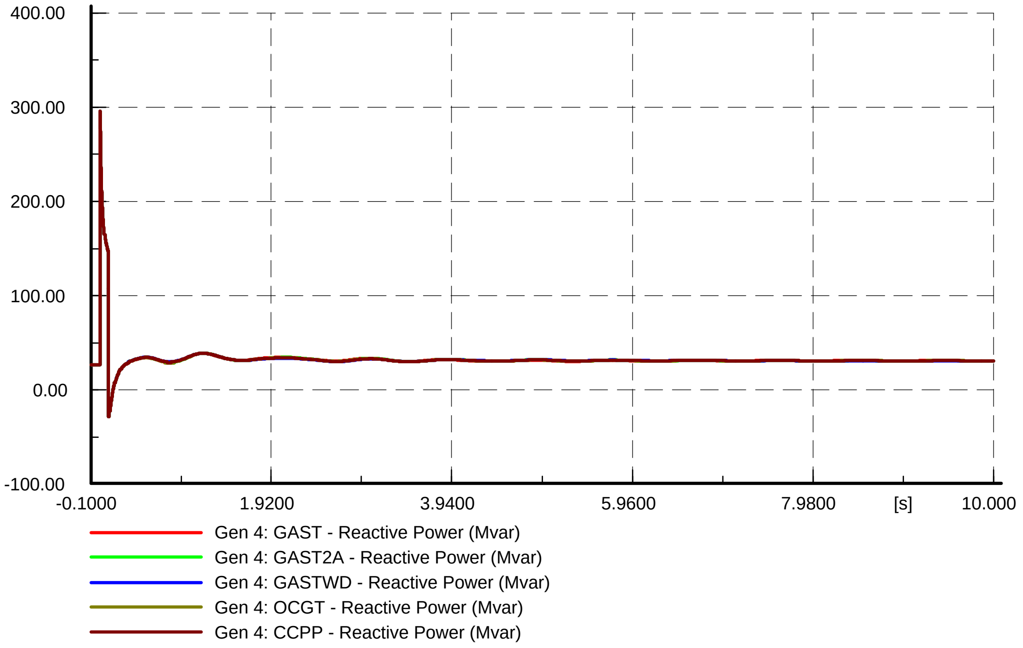
3 Phase fault (100ms) on Line 6, close to Bus 6, Line 6 tripped

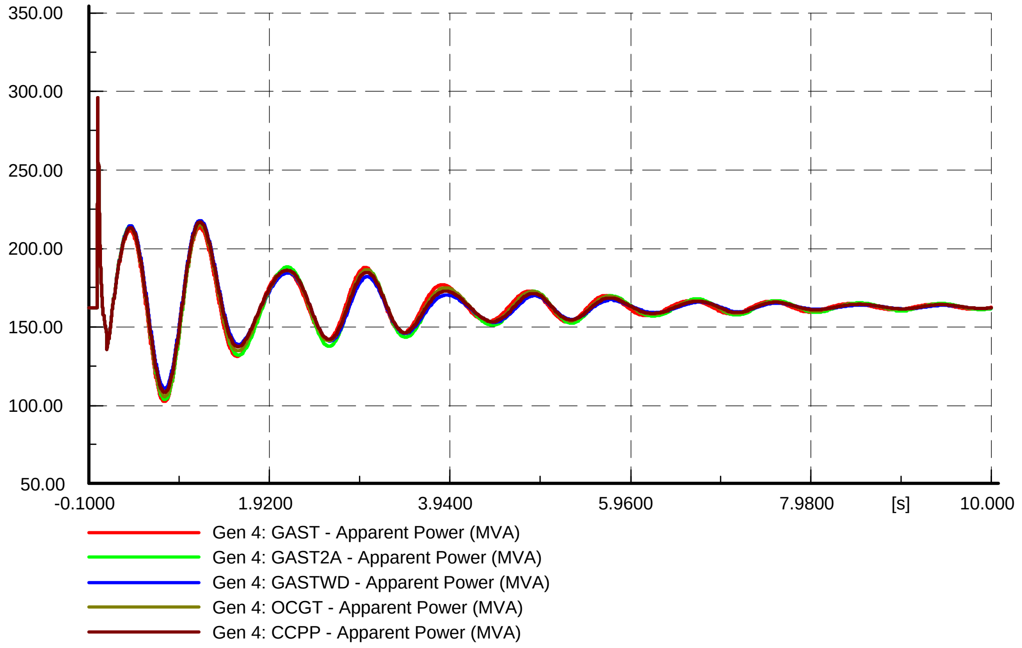
External grid with high acceleration time constant

Annex: /2









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Development and Adaptation of Dynamic Models

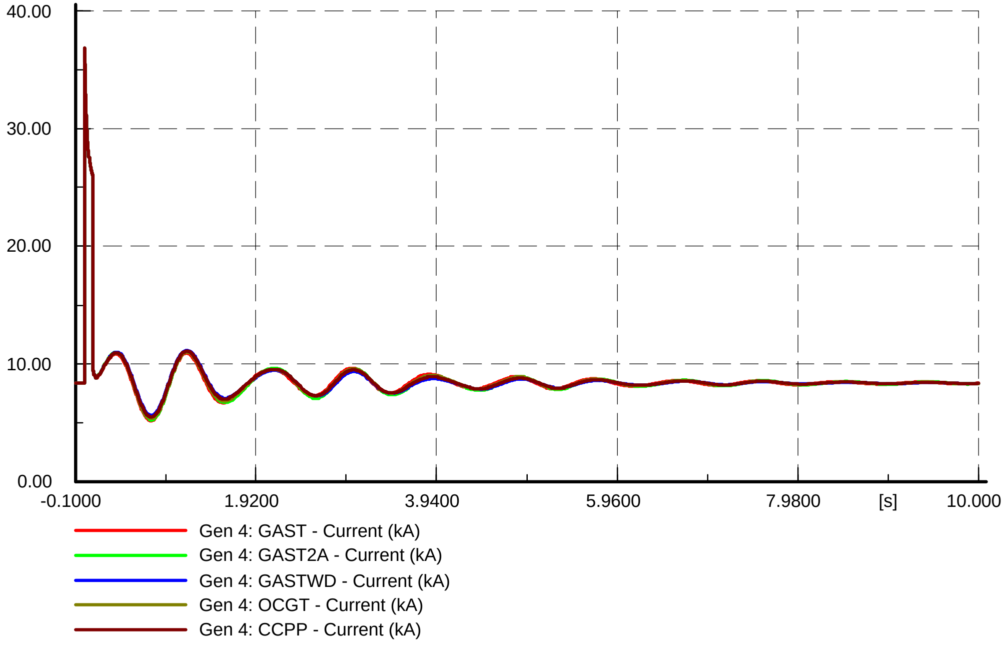
Results 5.5.2.c

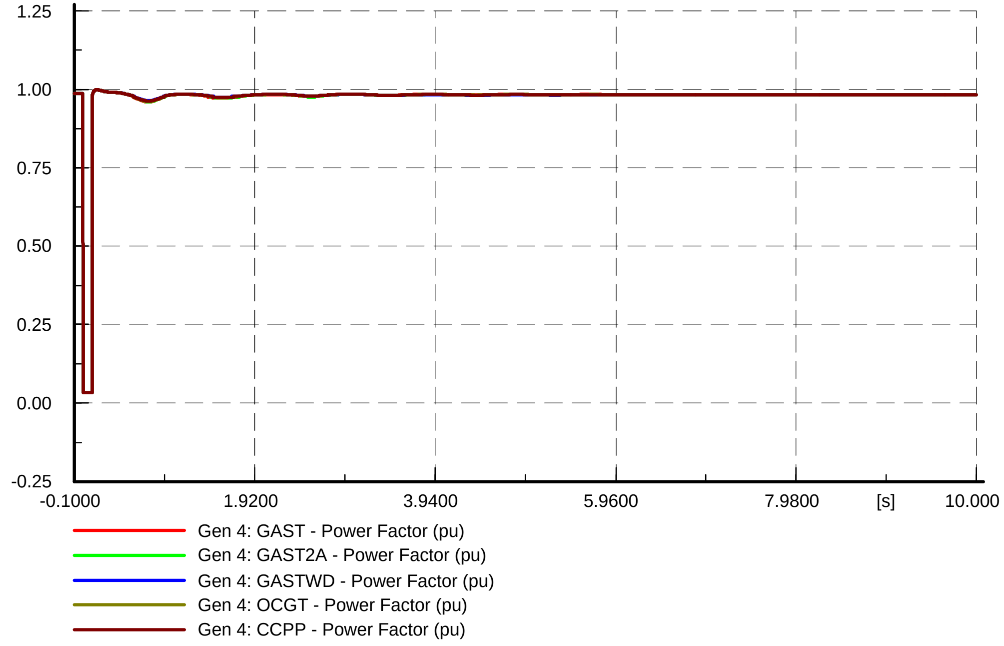
3 Phase fault (100ms) on Line 6, close to Bus 6, Line 6 tripped

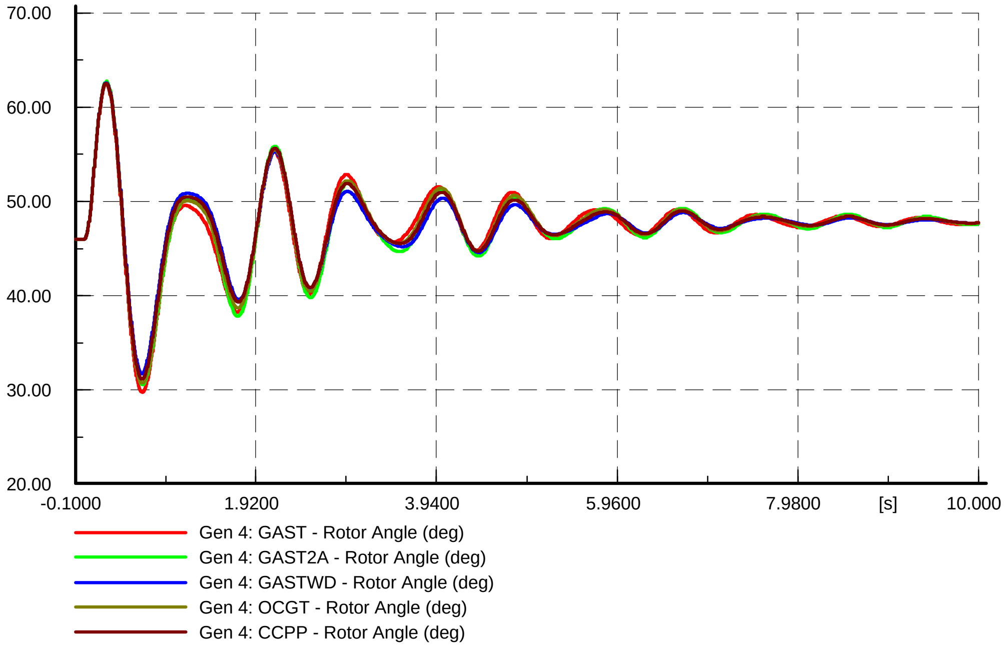
External grid with low acceleration time constant

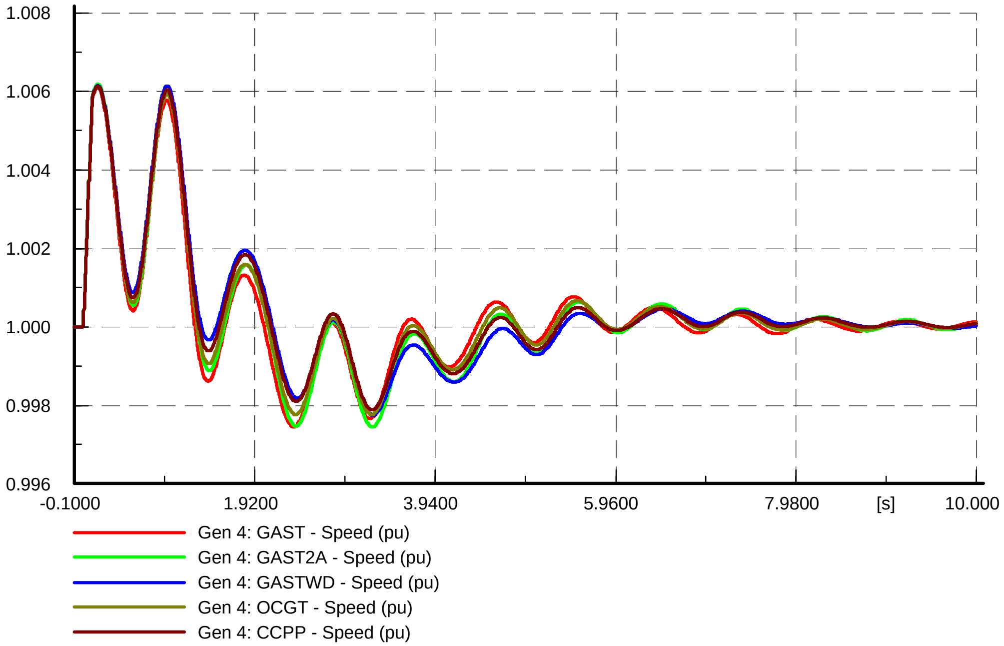
Date: 25/04/2011

Annex: /1









20316976

Development and Adaptation of Dynamic Models

Results 5.5.2.d

Date: 25/04/2011

3 Phase fault (100ms) on Line 6, close to Bus 6, Line 6 tripped

External grid with low acceleration time constant

Annex: /2

5.5.3 LINE 6 SWITCHED OUT OF SERVICE

For this disturbance, Line 6 was switched out of service, by opening both the line's breakers.

Expected outcome:
For this disturbance, the expected outcome from the various gas turbine models should be similar, and that the operating points for active power and speed for all the models are expected to be the same as before the disturbance. Differences are expected for the voltage, the reactive power, the power factor, the apparent power and the rotor angle, as a change in the network configuration was modelled, for all gas turbine models.

High and low inertia system: (Result 5.5.3.a to Result 5.5.3.d)
After the disturbance, the generator terminal voltage decreases marginally, but stabilise after a period. Just after the switching out of Line 6, there is a transient response of the active, the reactive and the apparent power, the current, the power factor, the rotor angle and the speed of the generator. The active power stabilises at the pre-disturbance value, but due to the change in the network configuration, the reactive and apparent power, the current, the power factor, the generator terminal voltage and the rotor angle stabilise at different values compared to pre-disturbance values. The speed of the generator oscillates around the rated value, but stabilises after a while at rated speed. The generator reaches a stable operating point after a while.

The simulations were performed without tap changing for the period of the simulation.

Conclusion:

The results show that the new gas turbine model's response is similar to the existing gas turbine models for the same disturbance. The five gas turbine models provided similar results, indicating that the new gas turbine model is settable to achieve the same results as the existing gas turbine models. All the models' operating points (active power and speed) return to the pre-disturbance operating point. The voltage, the reactive power, the power factor, the apparent power and rotor angle settles at new values, as a change in the network configuration occurred after the disturbance.

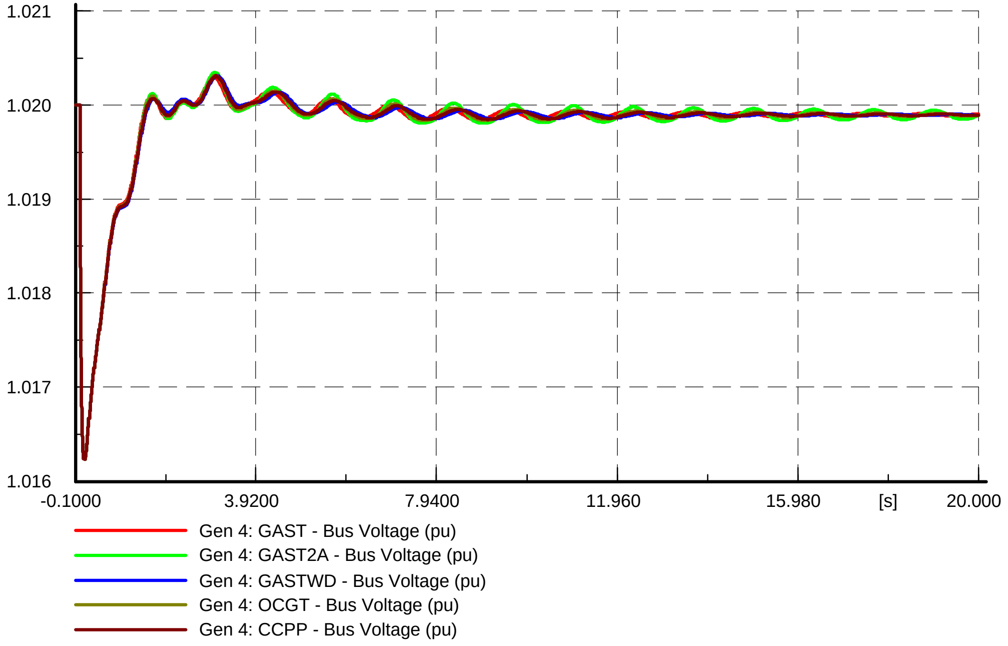
High inertia system: (Result 5.5.3.a and Result 5.5.3.b)
The system is stable for this disturbance, as can be seen from the various graphs.

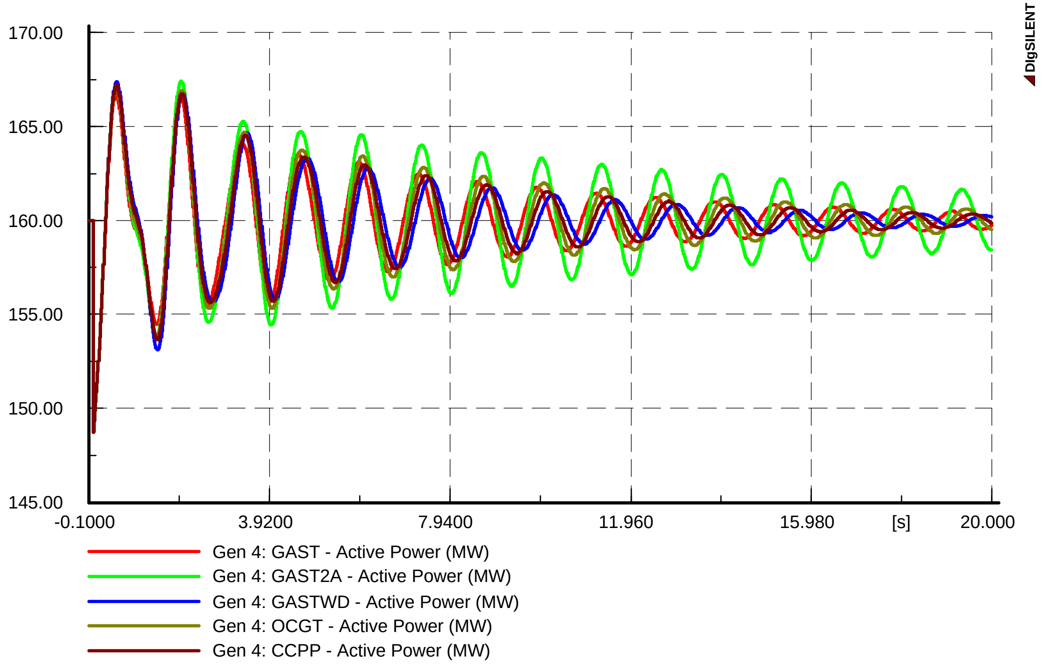
The results show that the various gas turbine models' response to the disturbance is similar with small variations between the models. The five gas turbine models' response is almost identical during the first approximately 1.5 seconds, where after the responses of the models start to deviate from each other. The variations between the models, with the exception of GAST2A, are negligible, although the difference between GAST2A and the other models is less than 1%.

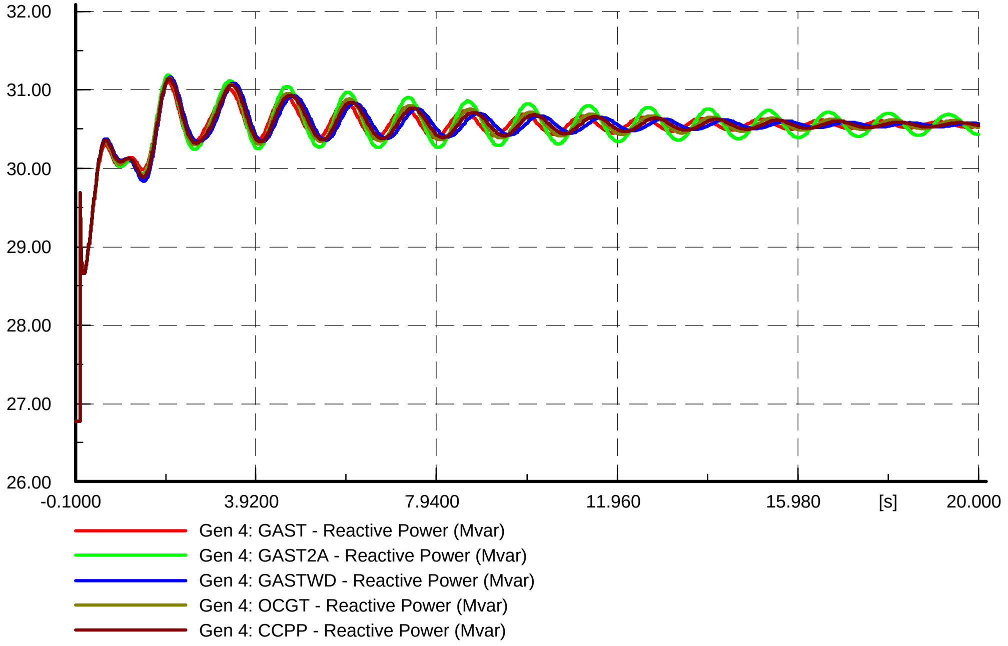
The results show that this disturbance is less severe as the disturbances that were modelled in Section 5.5.1 and 5.5.2. The disturbances of Section 5.5.1 and 5.5.2 were three-phase faults (on the gas turbine generator terminals and Line 6), while the disturbance that was modelled for Section 5.5.3 is the switching out of Line 6. For example, the maximum speed deviation for the disturbances of Section 5.5.1 and 5.5.2 is approximately 1%, while the maximum deviation for the speed for this disturbance (Line 6 switched out of service) is less than 0.1%.

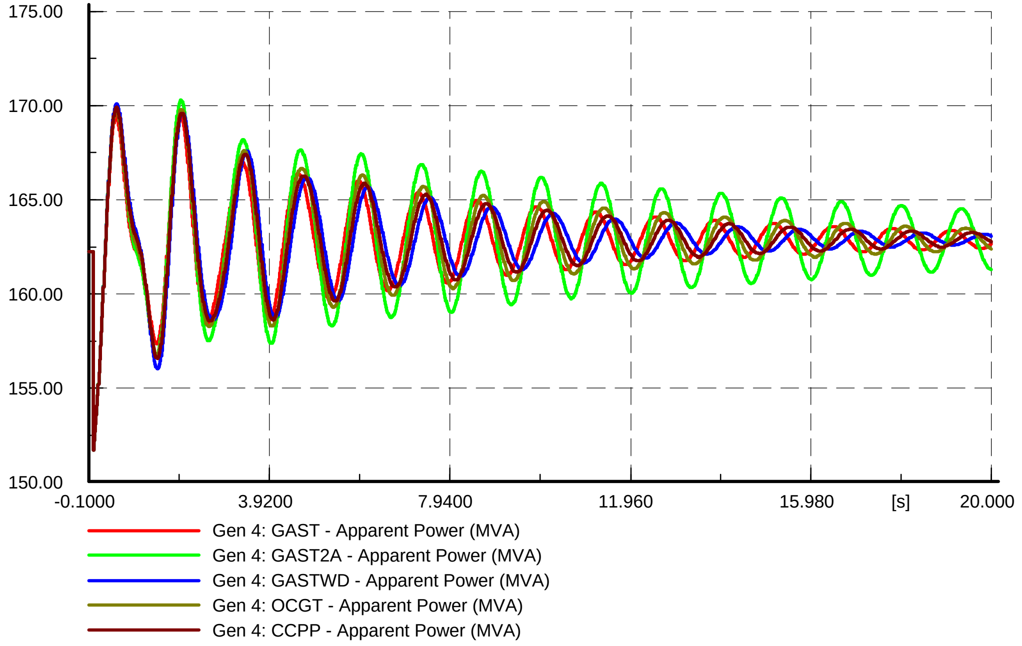
Although this disturbance is less severe than the two disturbances that were modelled for Sections 5.5.1 and 5.5.2, there are still oscillations in the network, and compared to the low inertia system, take longer to disappear. The system reaches steady state after a while as the oscillations are damped.

Low inertia system: (Result 5.5.3.c and Result 5.5.3.d)
The differences between the various gas turbine models' results are smaller than for the high inertia system. All the models responses are almost identical, with negligible differences. Compared to the two disturbances that were modelled for Section 5.5.1 and Section 5.5.2, this disturbance is not as severe. The oscillations are disappearing faster in the low inertia system compared to the high inertia system. The low inertia system contains less rotating mass than a high inertia system and therefore the variation in speed is easier to control.









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Development and Adaptation of Dynamic Models

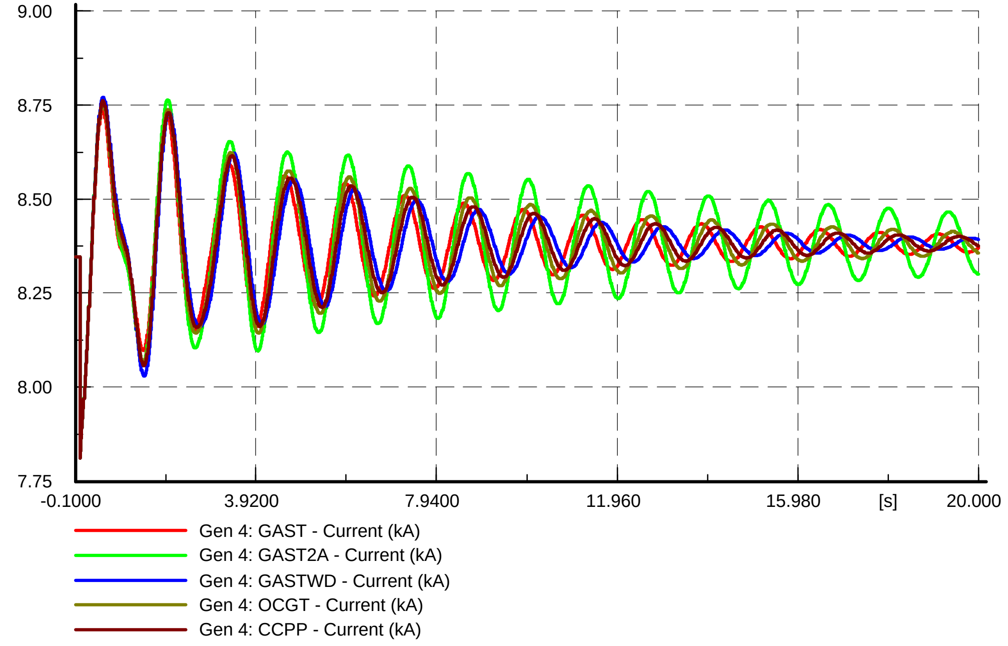
Results 5.5.3.a

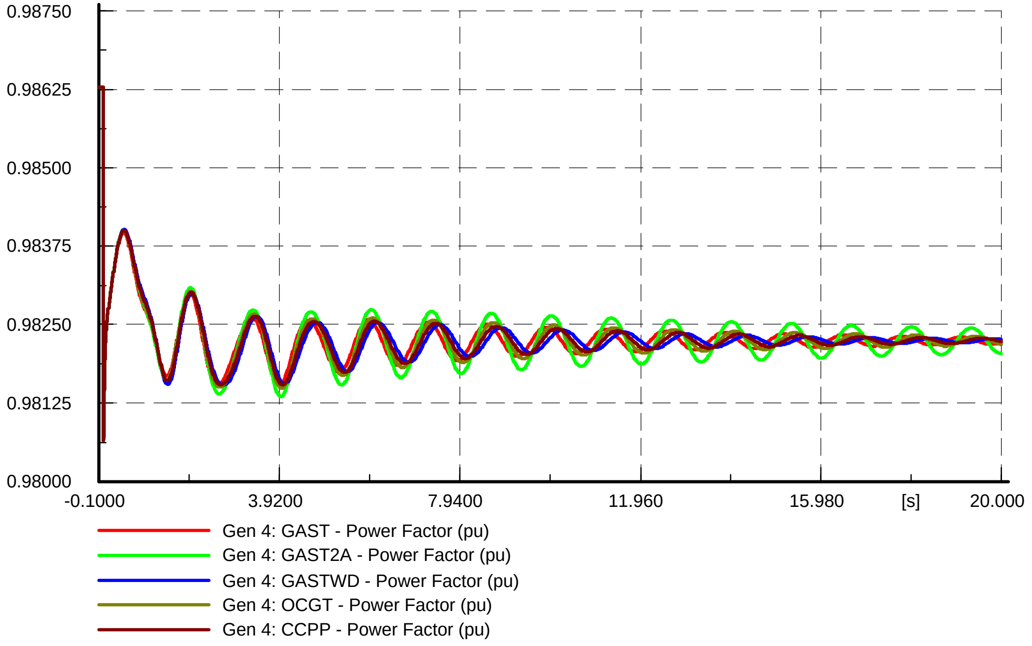
Line 6 switched out of service

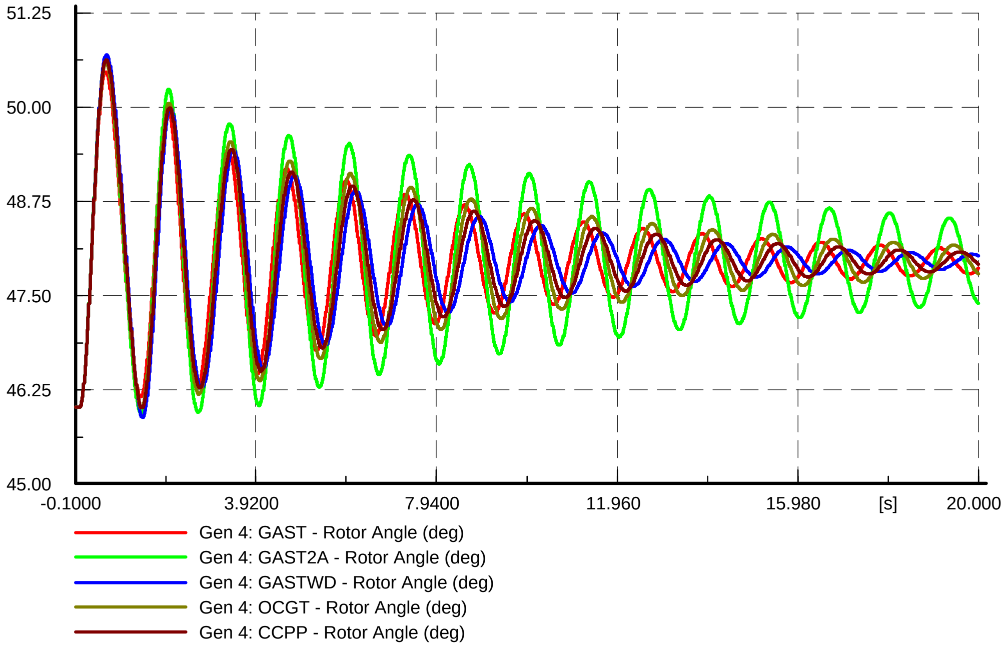
External grid with high acceleration time constant

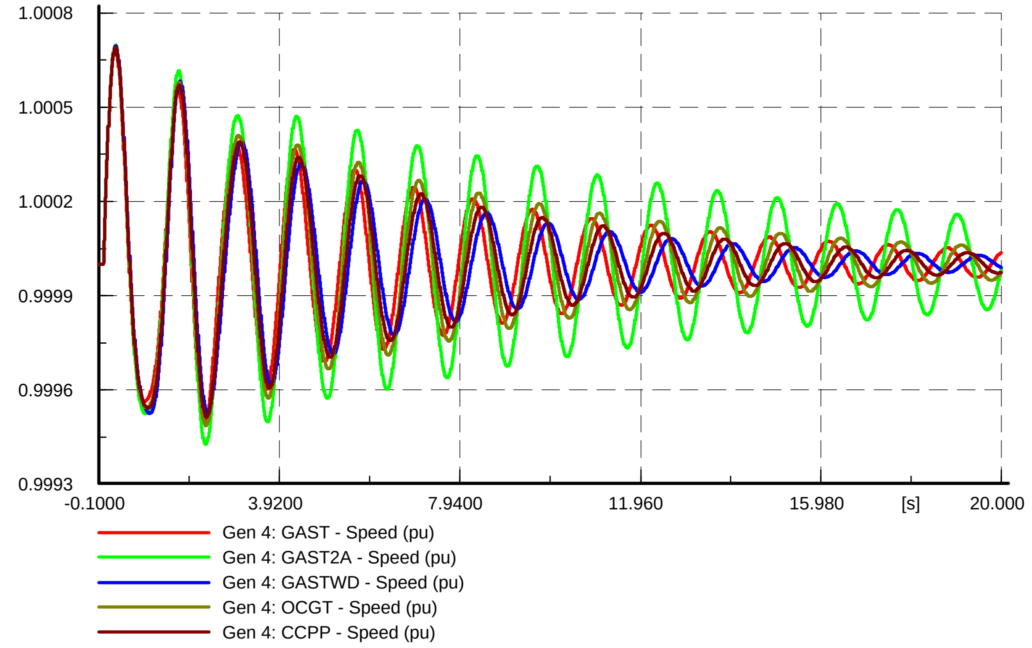
Date: 25/04/2011

Annex: /1









20316976

Development and Adaptation of Dynamic Models

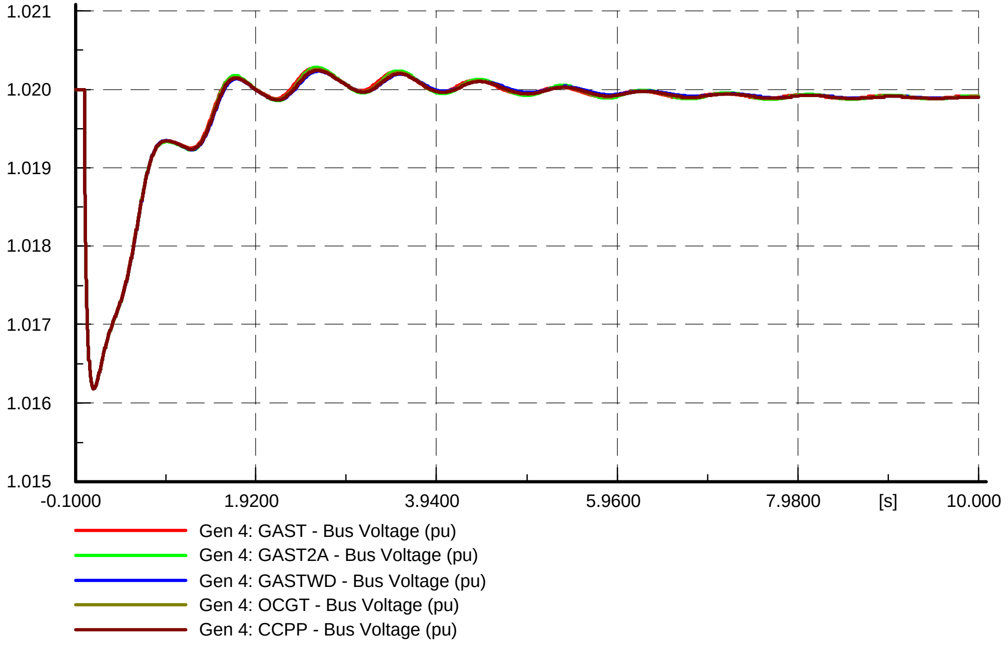
Results 5.5.3.b

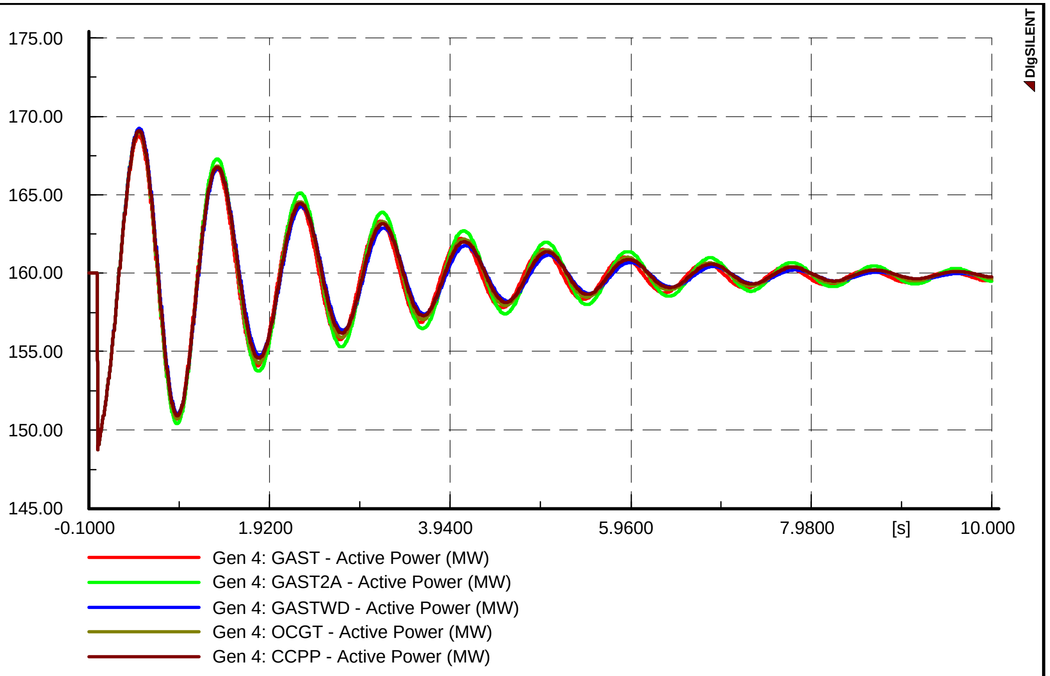
Date: 25/04/2011

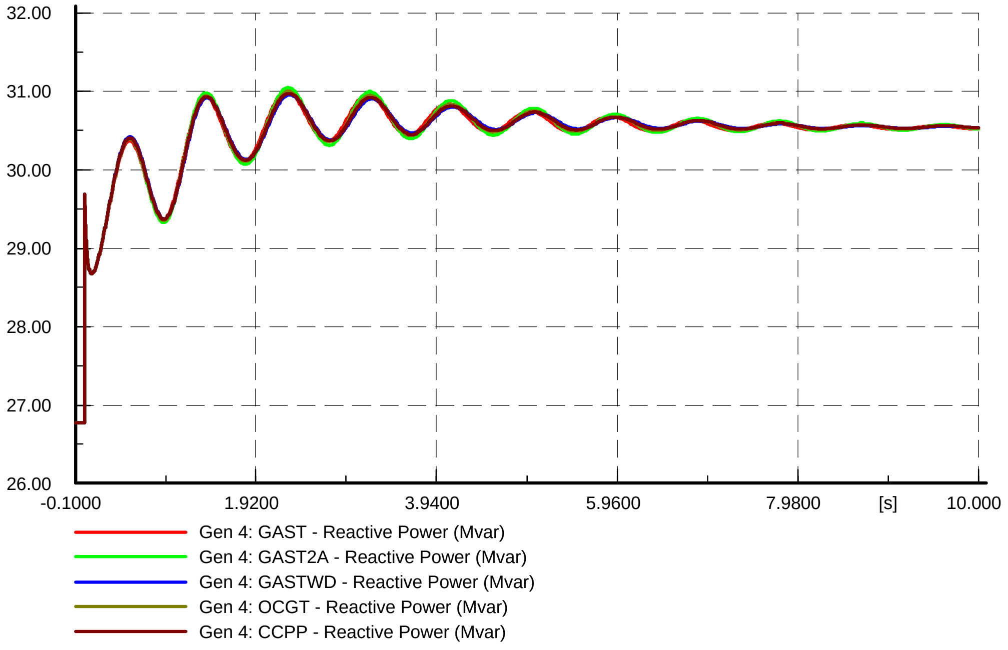
Line 6 switched out of service

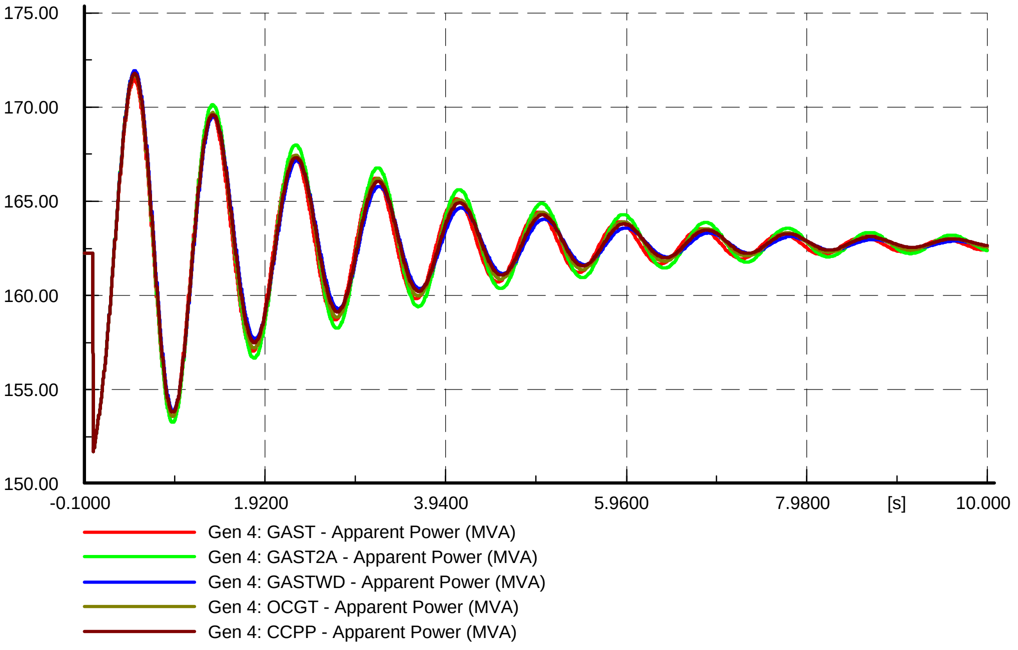
External grid with high acceleration time constant

Annex: /2









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Development and Adaptation of Dynamic Models

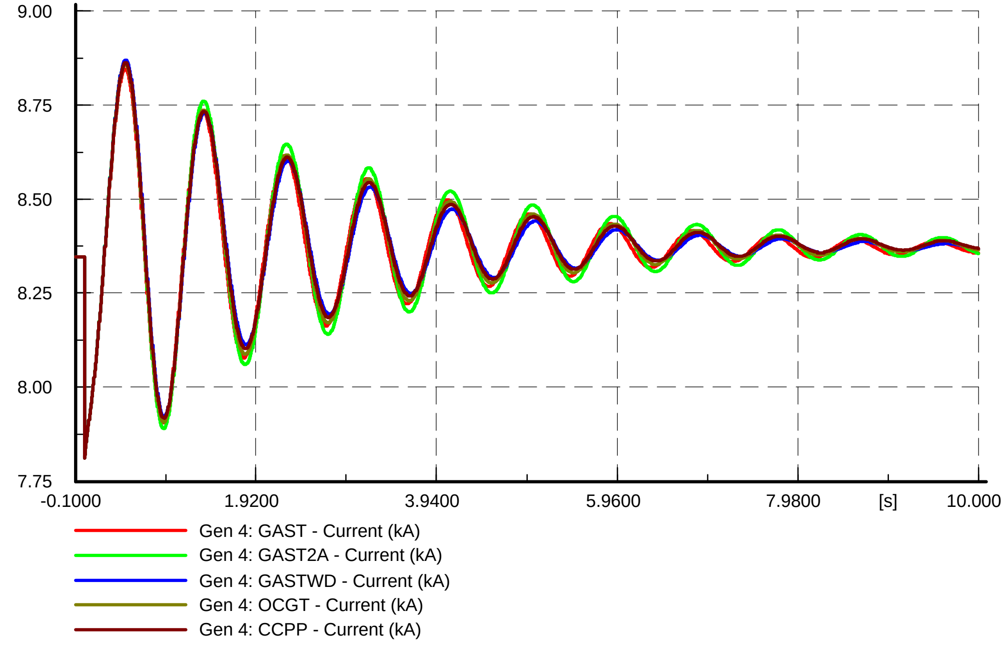
Results 5.5.3.c

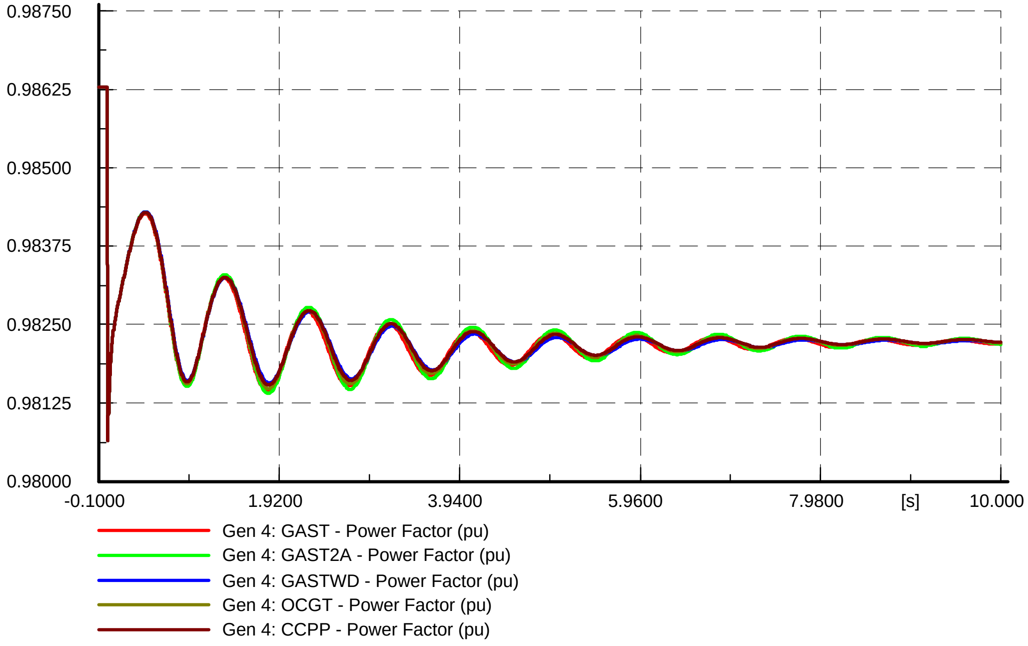
Date: 25/04/2011

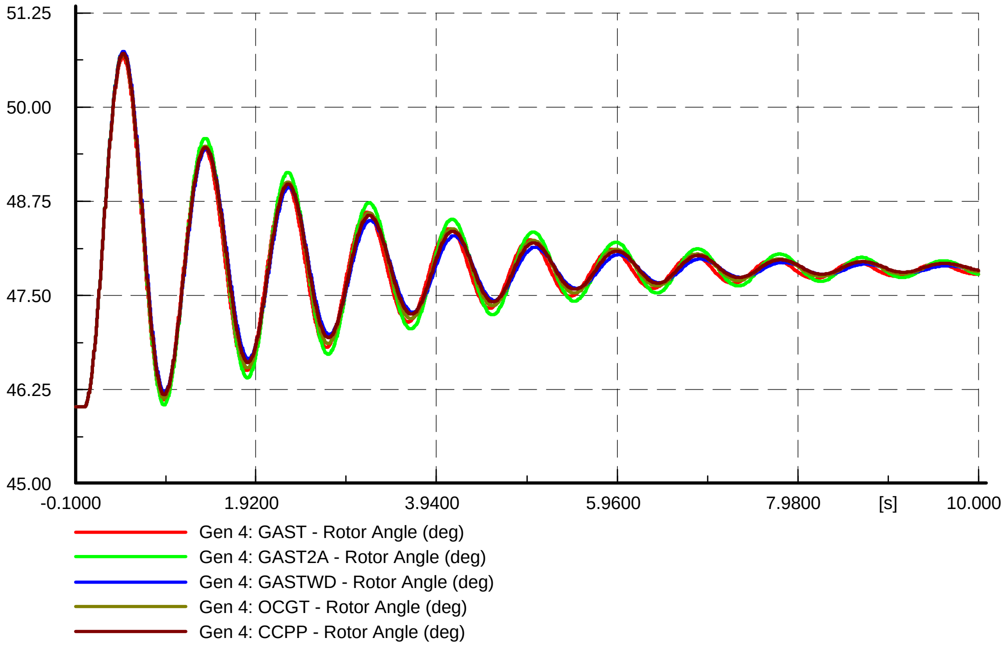
Line 6 switched out of service

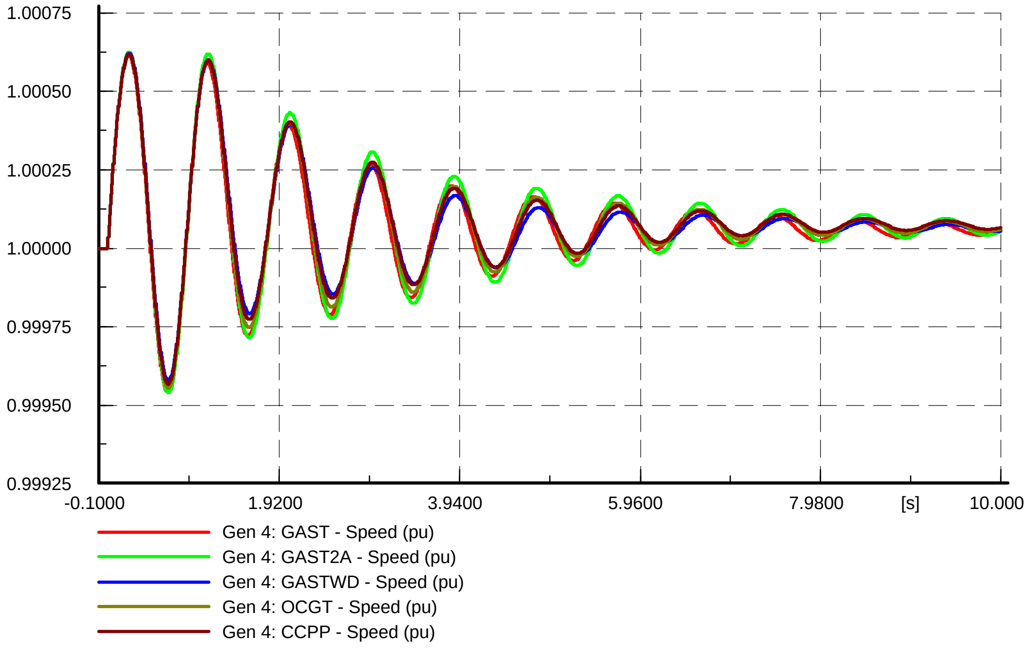
External grid with low acceleration time constant

Annex: /1









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Development and Adaptation of Dynamic Models

Results 5.5.3.d

Line 6 switched out of service

External grid with low acceleration time constant

Date: 25/04/2011

Annex: /2

5.5.4 GENERATOR GEN 3 SWITCHED OUT OF SERVICE

For this disturbance, Generator 3 was switched out of service, by opening the Generator 3 breaker.

Expected outcome:

For this disturbance, the expected outcome from the various gas turbine models should be similar. The operating points for active power and speed for all the models are expected to be the same as before the disturbance. Differences are expected for the voltage, the reactive power, the power factor, the apparent power and the rotor angle (for all gas turbine models), as a change in the network configuration was modelled by switching Generator 3 out of service.

High inertia system: (Result 5.5.4.a and Result 5.5.4.b)

After the disturbance, the generator terminal voltage decreases marginally but stabilises after a period, with oscillations after the disturbance. Just after the switching out of Generator 3, there is a transient response of the active, the reactive and the apparent power, the current, the power factor, the rotor angle and the speed of the generator. The active power stabilises at the pre-fault value, but due to the change in the network configuration, the reactive and apparent power, the current, the power factor, the generator terminal voltage and the rotor angle stabilise at different values compared to pre-disturbance values. The speed of the generator oscillates around the rated value, but stabilise after a while at value below the system rated speed.

Low inertia system: (Result 5.5.4.c and Result 5.5.4.d)

After the disturbance, the system reaches a steady state operating point after a while. There is a small difference between the various gas turbine models, with the exception of the GAST model. The differences in the results that the GAST model delivered are larger than the results of the other models. The GAST model settled at approximately 177 MW, while the other models settled closer to the pre-disturbance value of 160 MW. The results are approximately 155 MW for GAST2A, 158 MW for GASTWD, 161.5 MW for CCPP and 162 MW for OCGT. The GAST2A, GASTWD, OCGT and CCPP settled within 4.5% of each other, while GAST settled at more than 14% away from GAST2A and more than 9% away from OCGT. The same phenomenon is visible for the current, the rotor angle, the apparent power and the speed of the turbine. GAST settled at more than 10% away from the pre-disturbance value of 160 MW. The same applies for the current, the power factor, the rotor angle and the speed.

The voltage and the reactive power for all the models settled at approximately the same value (although there is a small difference in the voltage between GAST and the other models).

The simulations were performed without tap changing for the period of the simulation.

Conclusion:

High inertia system:

The system is stable for this disturbance, as can be seen from the various graphs.

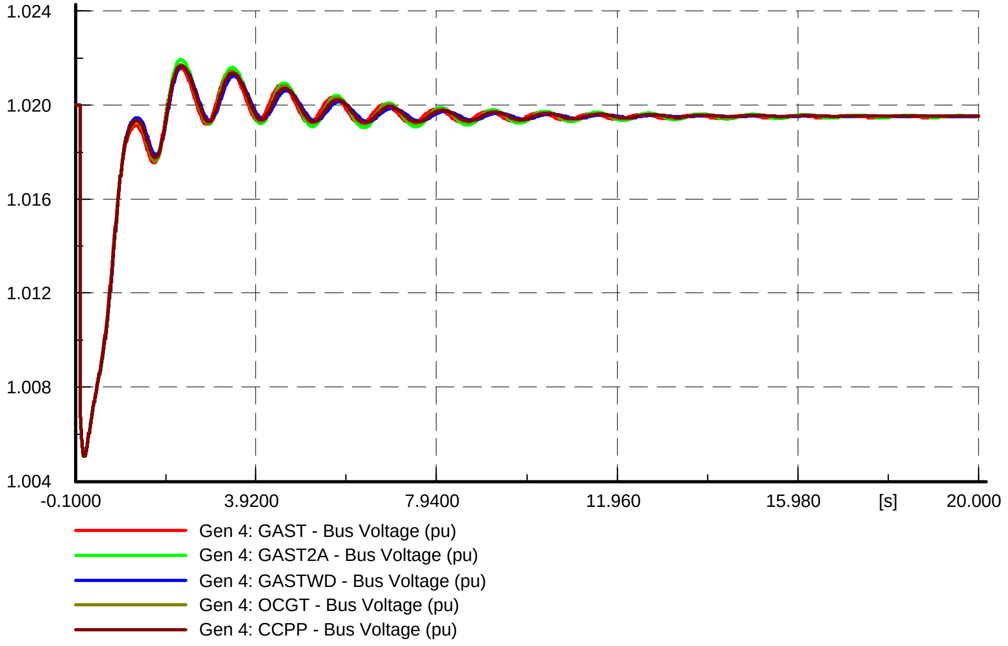
The results show that the various gas turbine models' response to the disturbance is similar, but with small variations between the models. The models' response is almost identical during the first approximately 1.5 seconds, where after the responses of the models start to deviate from each other. The deviations of the models for this disturbance are very small and in general, the same results are obtained.

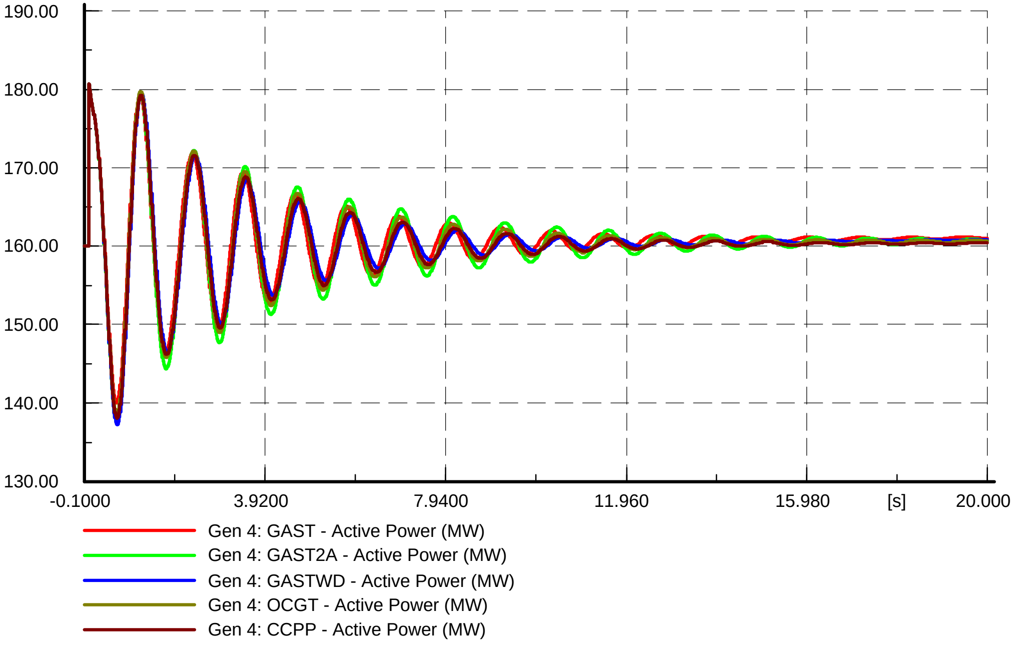
The high inertia system responds slower to the disturbance and experiences larger oscillations for a longer period, compared to the low inertia system. The system reaches a steady state operating point after a while as the oscillations are damped.

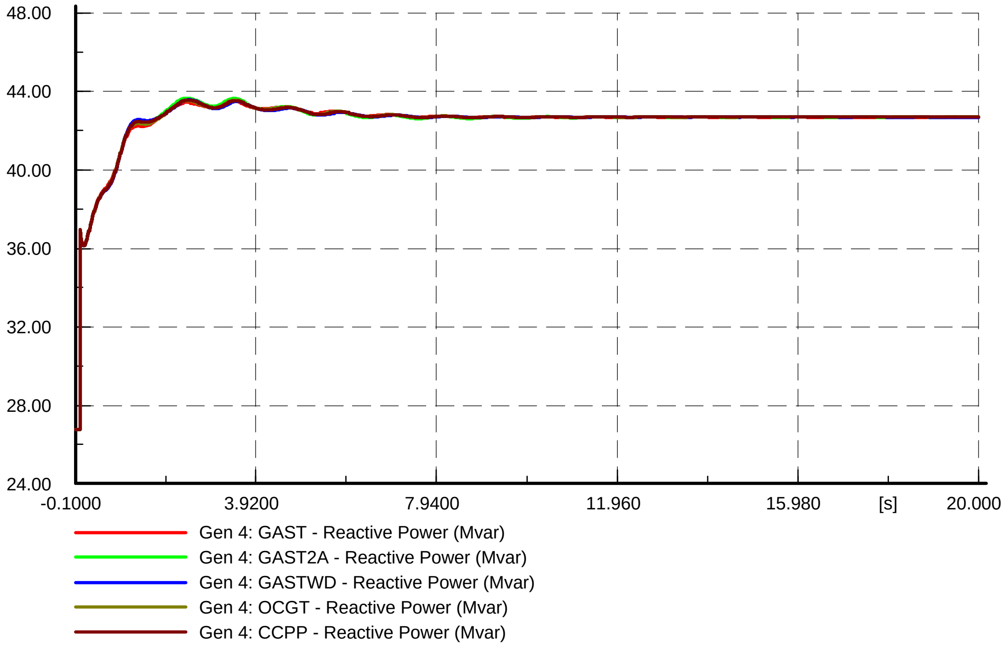
Low inertia system:

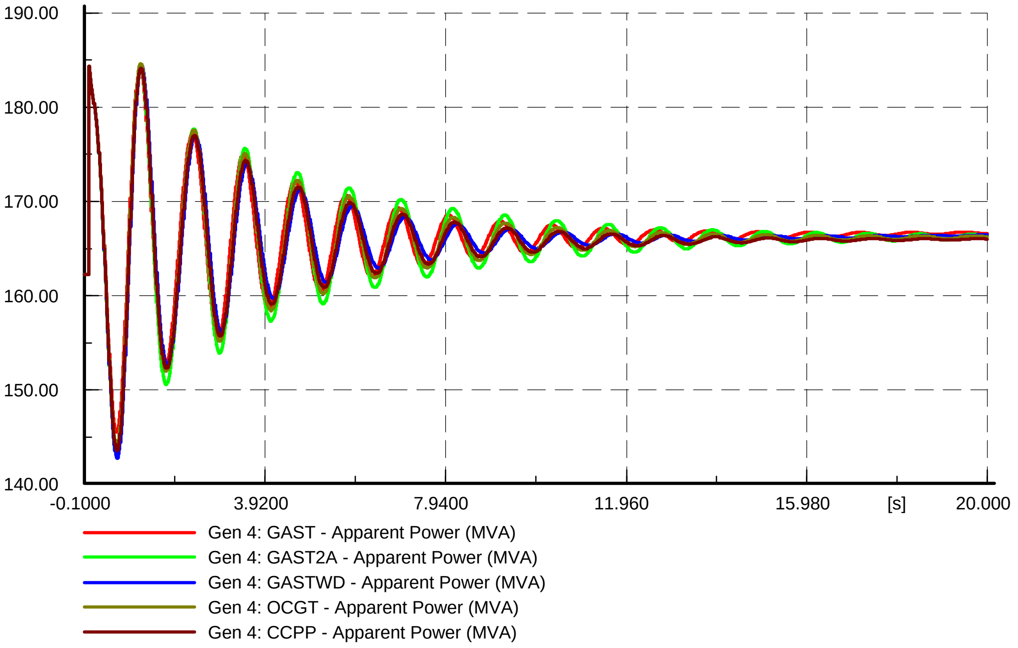
Although the generator voltage and the reactive power settled at close to the pre-disturbance value, the simulations show that the system's frequency deviates from the rated frequency (or rated speed). The governors in the network model were set with non-zero droop settings. No generators were set to operate in isochronous mode. The droop settings cause the speed of the generators to change with a change in load. Due to the change in the network configuration, each generator experienced a change in load and therefore also a change in speed. As no generators were set to control the system frequency / speed at rated values, the whole system's frequency / speed will deviate from the rated values, but will settle at a new value.

The expected outcome was that the various models would provide similar results. However, only four of the five models provided results with differences within 4.5% of each other. The GAST model deviated the most from the other four models. The expected result was that the models should settle at a new frequency / speed, but will settle at the pre-disturbance value for active power. However, only OCGT and CCPP settled within 1.25% of the pre-disturbance value. GASTWD, GAST2A and GAST settled more than 2.5% from the pre-disturbance value. Based on this expectation and the actual results, OCGT and CCPP are the most accurate of the models. It is important to note that the loading of the different models were not changed, and therefore the pre-disturbance and post-disturbance loading of the turbines were expected to stay the same. The main reason for the deviation is the non-zero-droop setting of the governor controllers. The non-zero droop setting cause the speed of the generators to change with a change in the loading of the generators. No generator was set to operate in isochronous mode, and therefore no frequency / speed control was performed in the system.









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Development and Adaptation of Dynamic Models

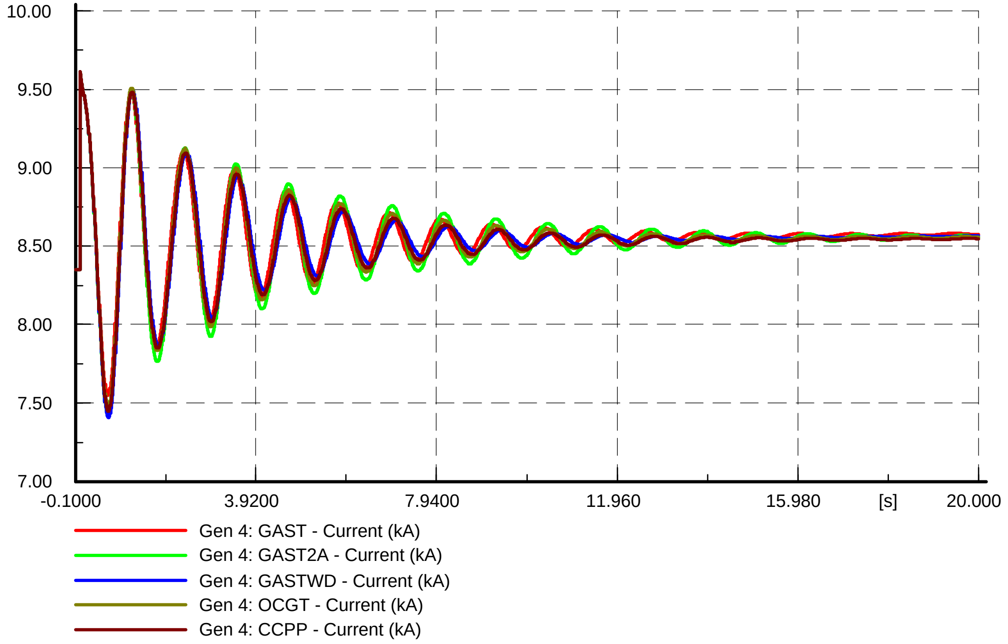
Results 5.5.4.a

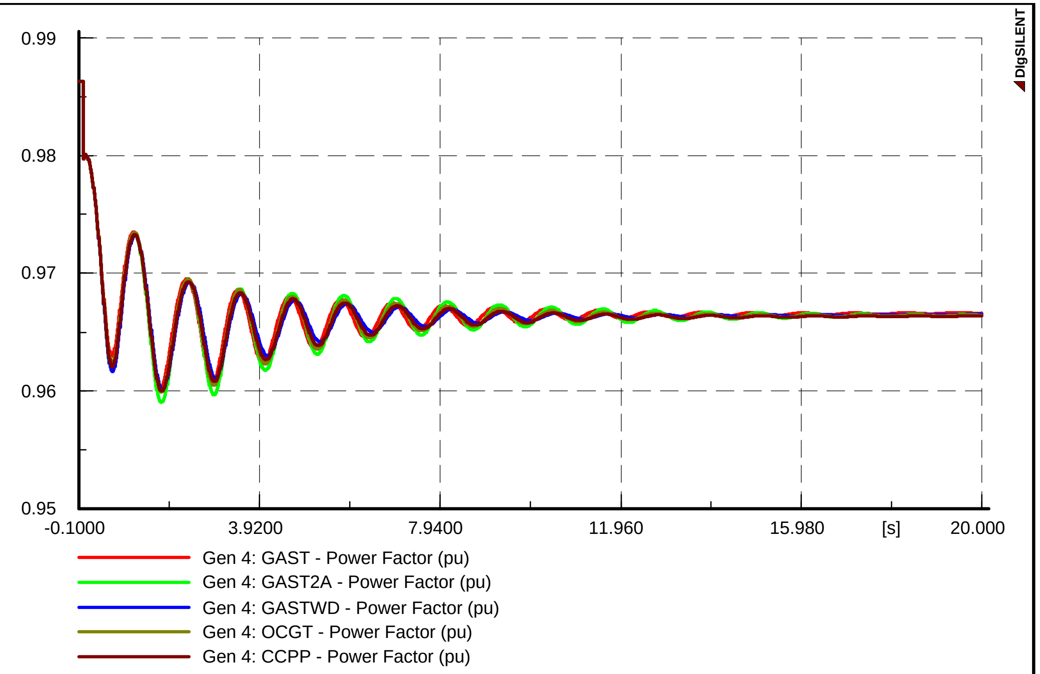
Date: 25/04/2011

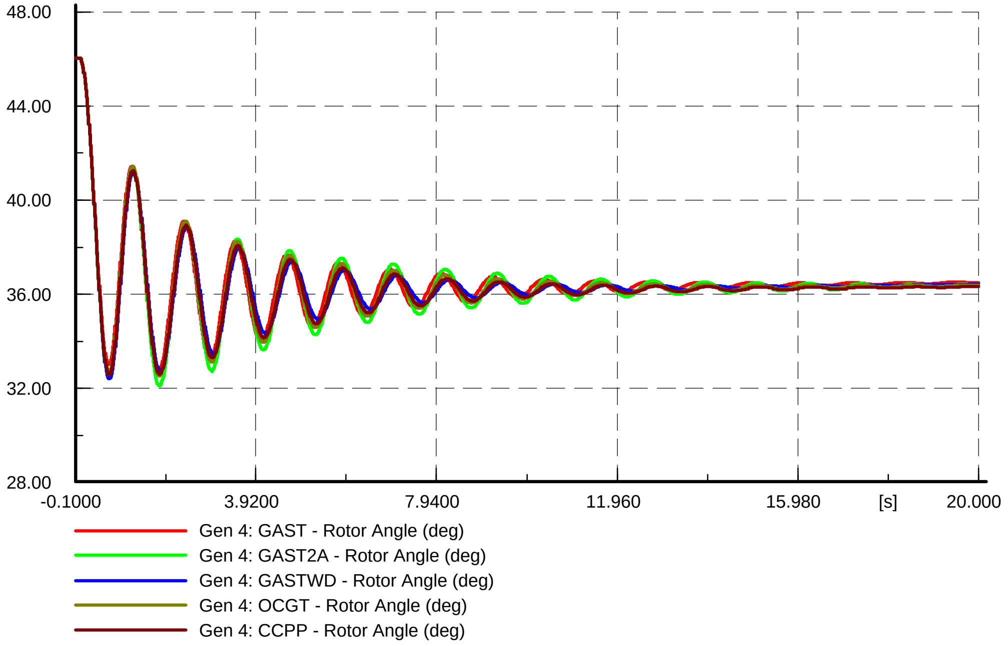
Generator 3 switched out of service

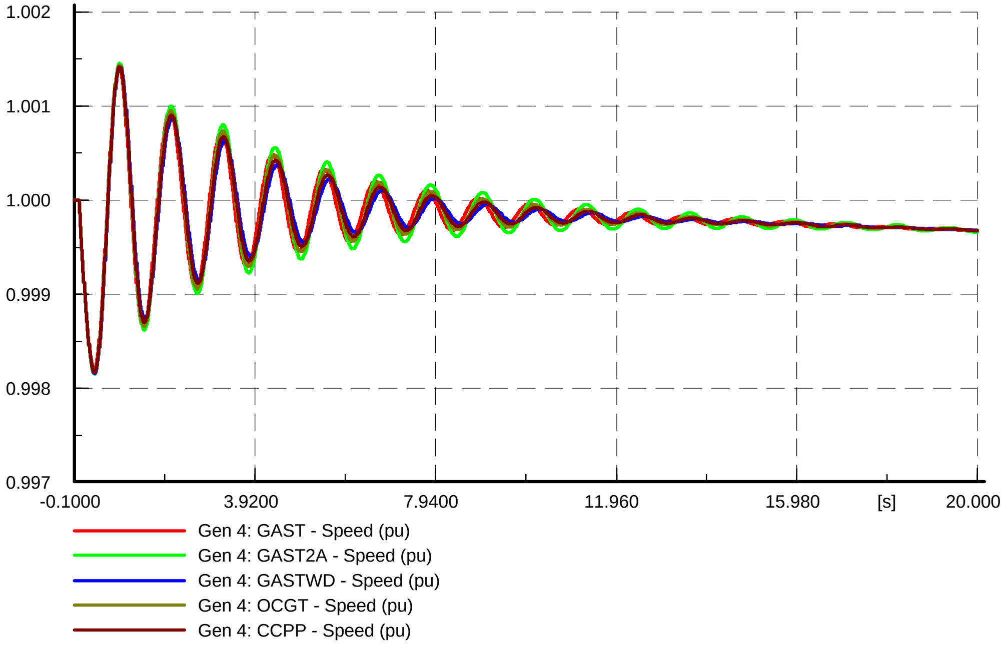
External grid with high acceleration time constant

Annex: /1









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Development and Adaptation of Dynamic Models

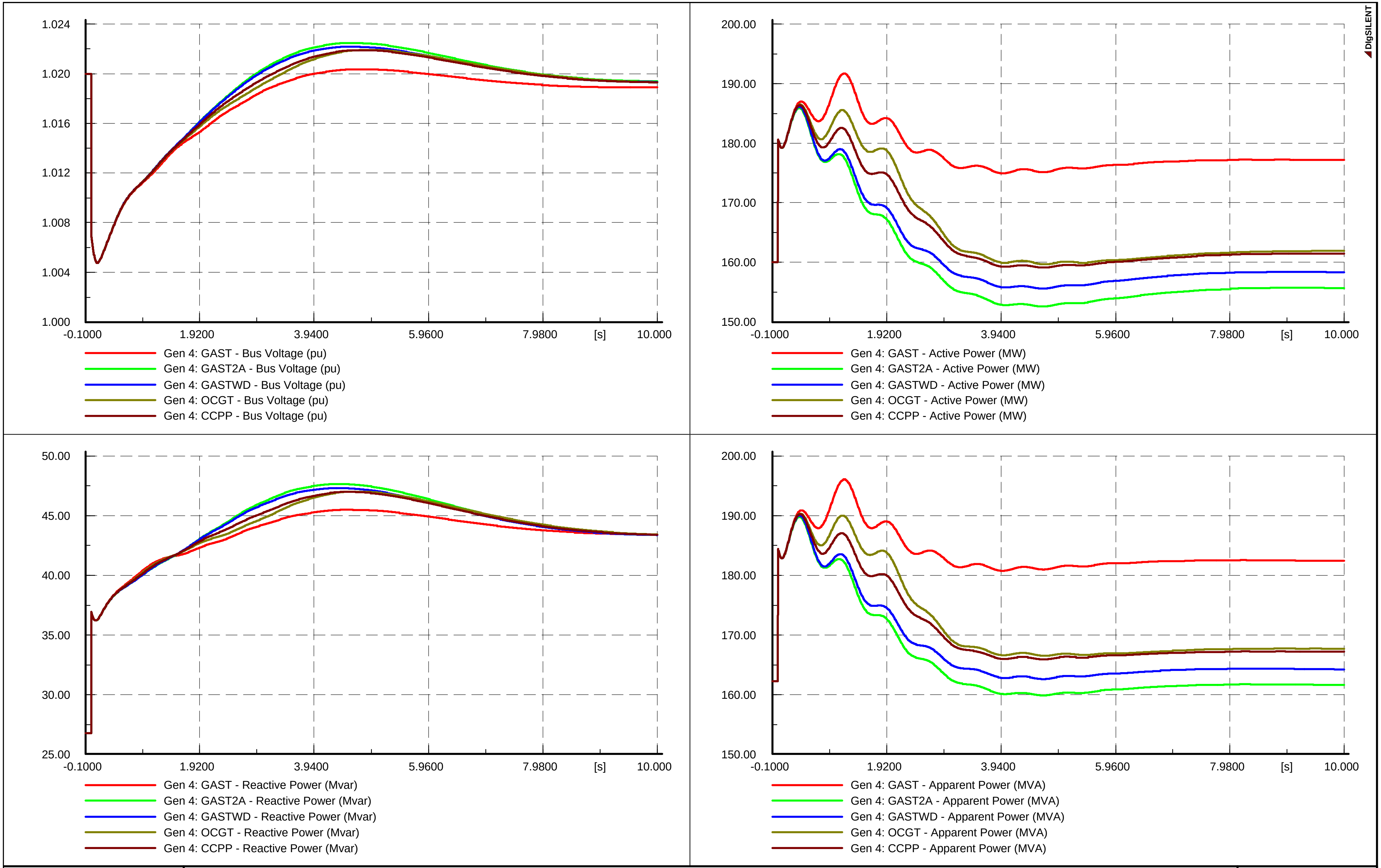
Results 5.5.4.b

Date: 25/04/2011

Generator 3 switched out of service

External grid with high acceleration time constant

Annex: /2



20316976

Development and Adaptation of Dynamic Models

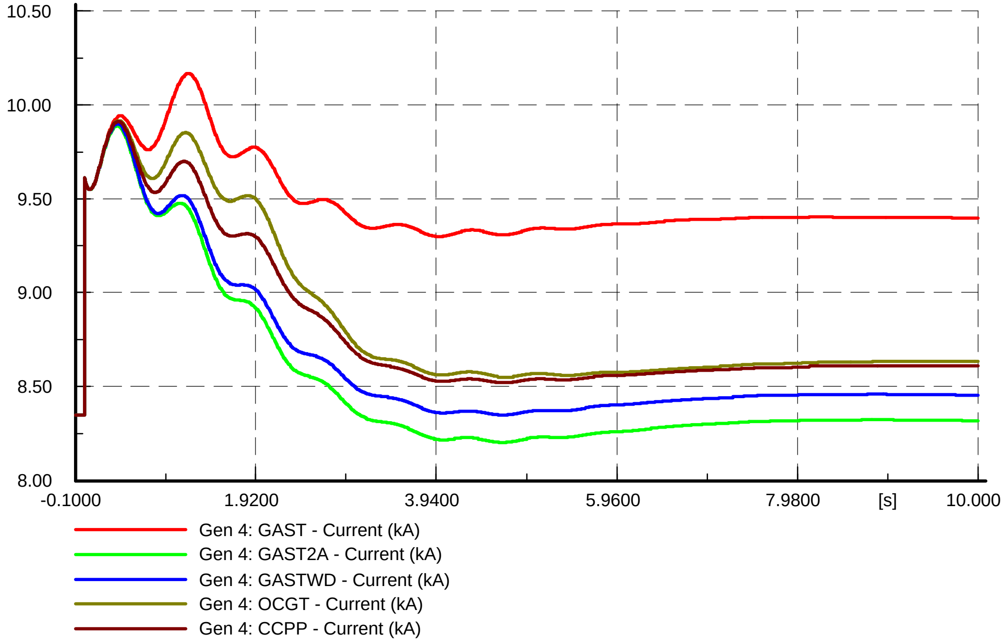
Results 5.5.4.c

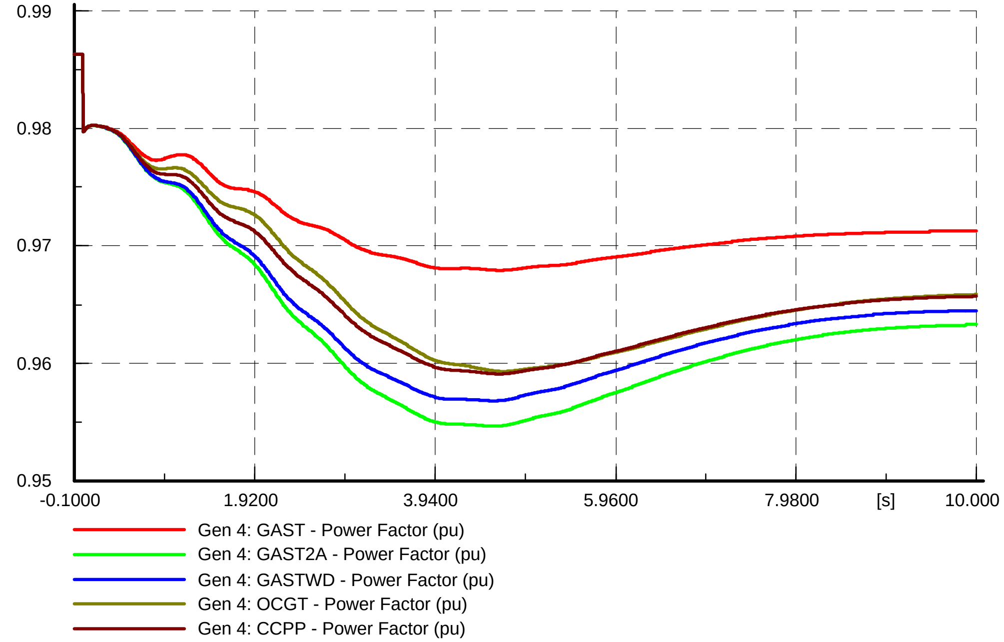
Date: 25/04/2011

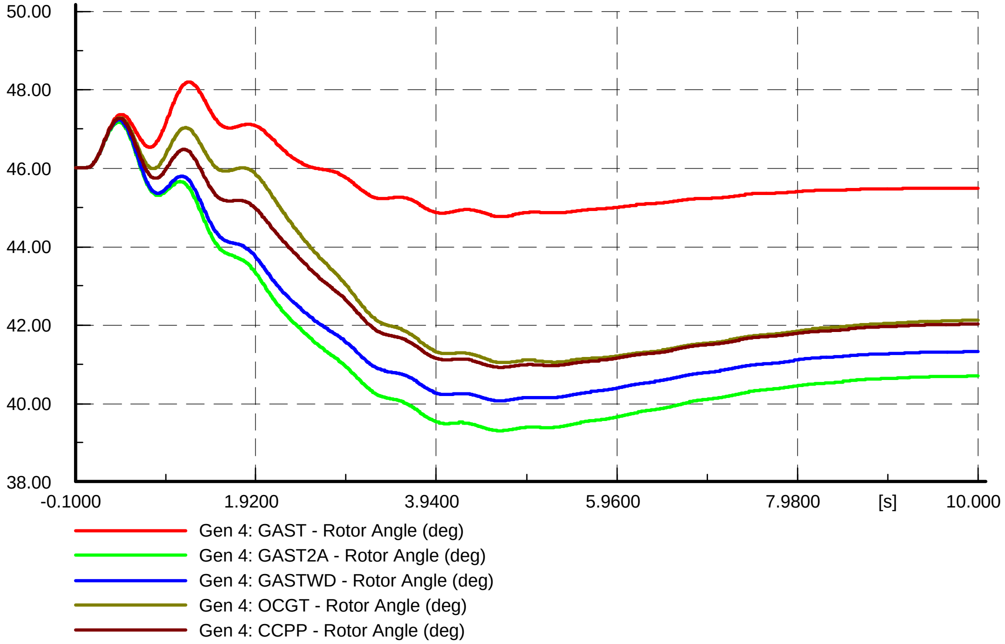
Generator 3 switched out of service

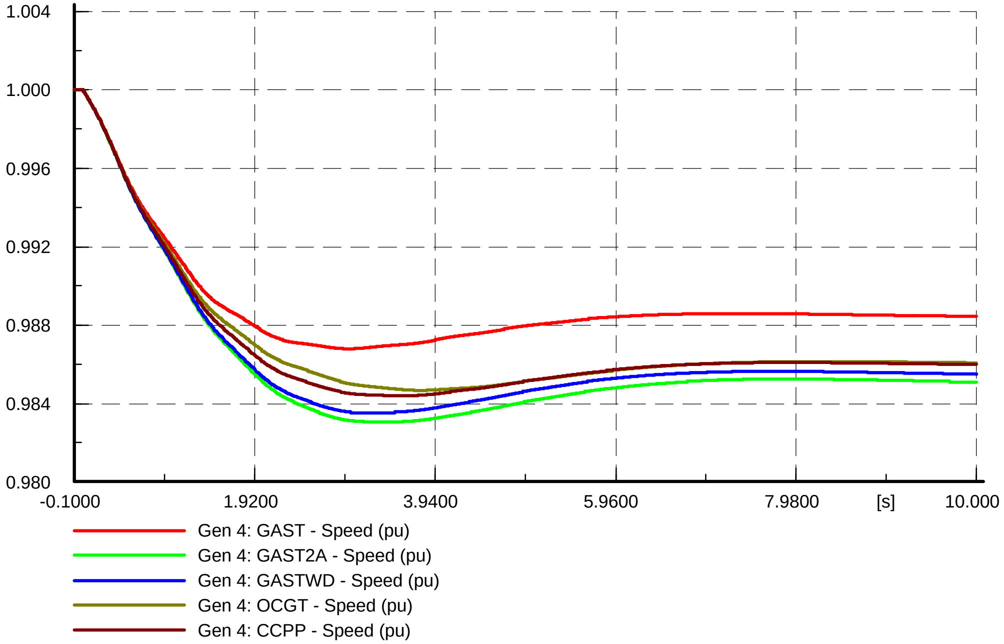
External grid with low acceleration time constant

Annex: /1









20316976

Development and Adaptation of Dynamic Models

Results 5.5.4.d

Date: 25/04/2011

Generator 3 switched out of service

External grid with low acceleration time constant

Annex: /2

5.5.5 GENERATOR GEN 4 OUTPUT DECREASED BY 50%

For this disturbance, the power output of Generator 4 was decreased by 50% of the generator's power rating. The 50% is based on the rated active power of the generator. The rating of the generator is 200 MVA and a power factor of 0.9. This gives the active power rating of the generator as 180 MW.

Expected outcome:

For this disturbance, the expected outcome from the various gas turbine models should be similar. The post-disturbance operating points for active power and speed for all the models are expected to be the same. A 50% decrease in active power is equal to a decrease of 90 MW. The generators should therefore settle at 70 MW, as the pre-disturbance value is 160 MW.

High inertia system: (Result 5.5.5.a and Result 5.5.5.b)

After the disturbance, the generator terminal voltage increases but settles after a period at a slightly higher value than the pre-disturbance value. Just after the decreasing the power output of Generator 4, there is a transient response of the active, the reactive and the apparent power, the generator current, the power factor, the rotor angle and the speed of the generator. All the gas turbine models settle at the approximately the same value of 70 MW for active power. The reactive power increases, but the apparent power, current and power factor decreases. The rotor angle also changes (compared to the pre-disturbance value), but settles after a period at a new value. The speed of the generator oscillates around the rated speed, but settles at a value below rated speed after a period.

Low inertia system: (Result 5.5.5.c and Result 5.5.5.d)

All the models, with the exception of GAST settle at the expected value of 70 MW. GAST settles at approximately 89 MW, an error of approximately 27% (based on the expected value). The other models settle as follows: GAST2A at 69 MW, GASTWD at 71 MW, CCPP at 74 MW and OCGT at 75 MW. The differences are not substantial between GAST2A, GASTWD, OCGT and CCPP as the results are within approximately 7% of the expected value.

The simulations were performed without tap changing for the period of the simulation.

Conclusion:

High inertia system:

The system is stable for this disturbance. The results show that the various gas turbine models' response to the disturbance is similar, but with small variations between the models. The models' response is almost identical during the first approximately 1.5 seconds, where after the responses of the models start to deviate from each other. The deviations of the models for this disturbance are relatively small and in general, the same results are obtained.

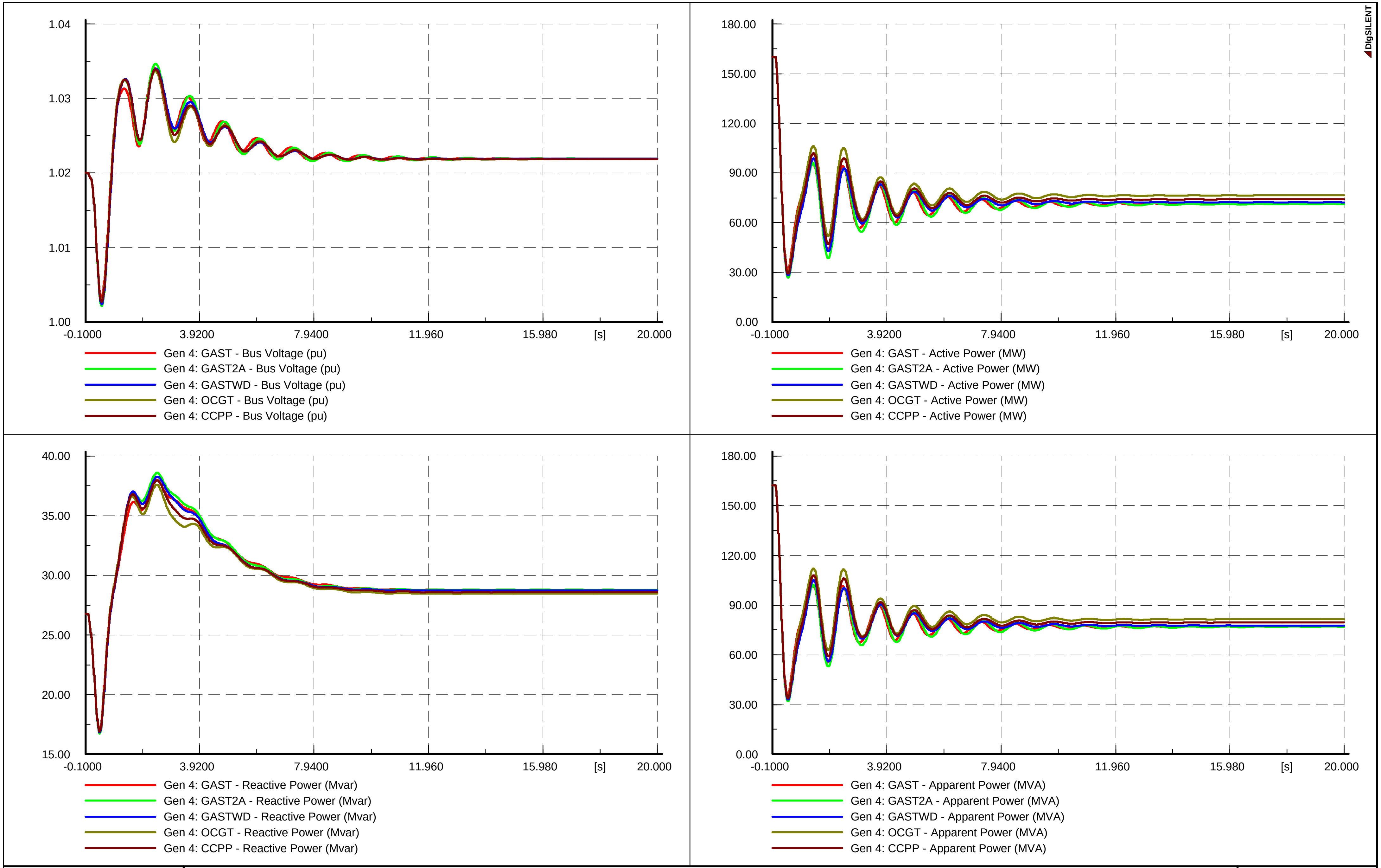
Low inertia system:

The various models settle near the expected value of 70 MW (within 7% of 70 MW), with the exception of GAST that settled at approximately 89 MW (27% difference between actual and expected value).

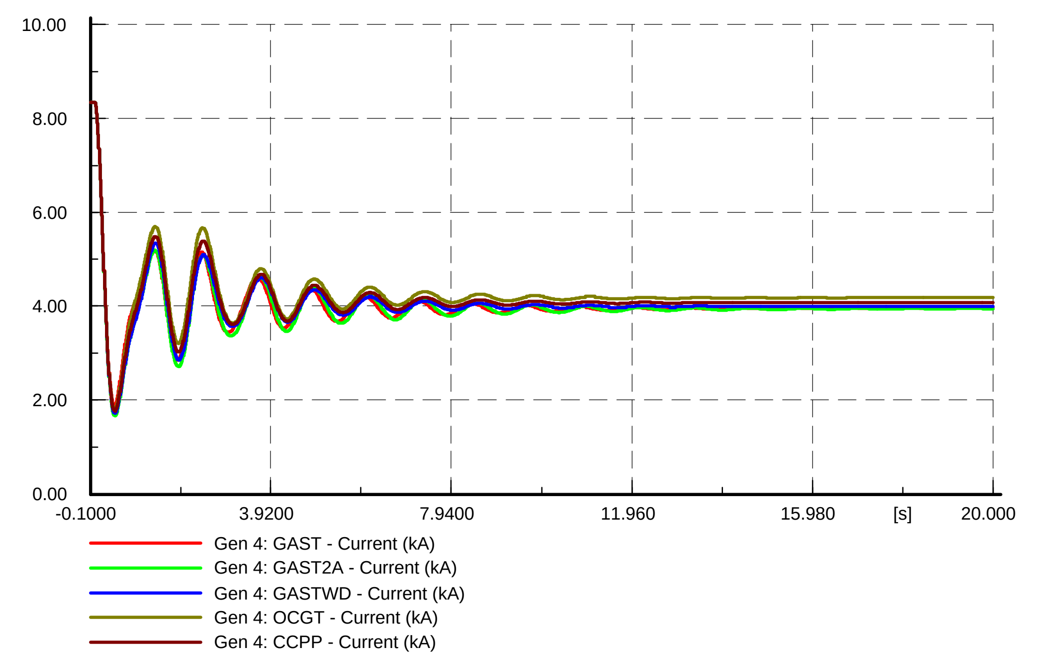
The system does not settle at rated speed (or rated frequency). The reason is the non-zero droop settings of the generator governor controllers. See Section 5.5.4 for a description of the reason. For this disturbance, the controllers are unable to eliminate the error too.

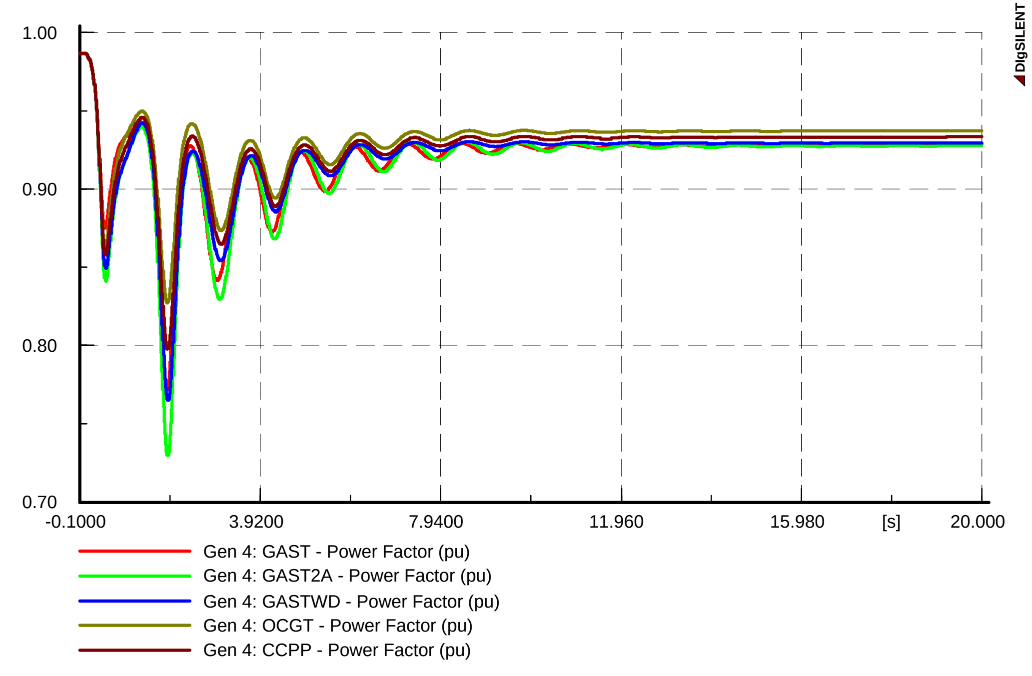
There is also a small difference between the various gas turbine models, except for GAST. The GAST model settles at substantially different values after the disturbance compared to the other models. The other models settle closer to each other and to the expected value. As with the other disturbances, for this disturbance the various models (except GAST) respond the same for the first approximately 2 seconds.

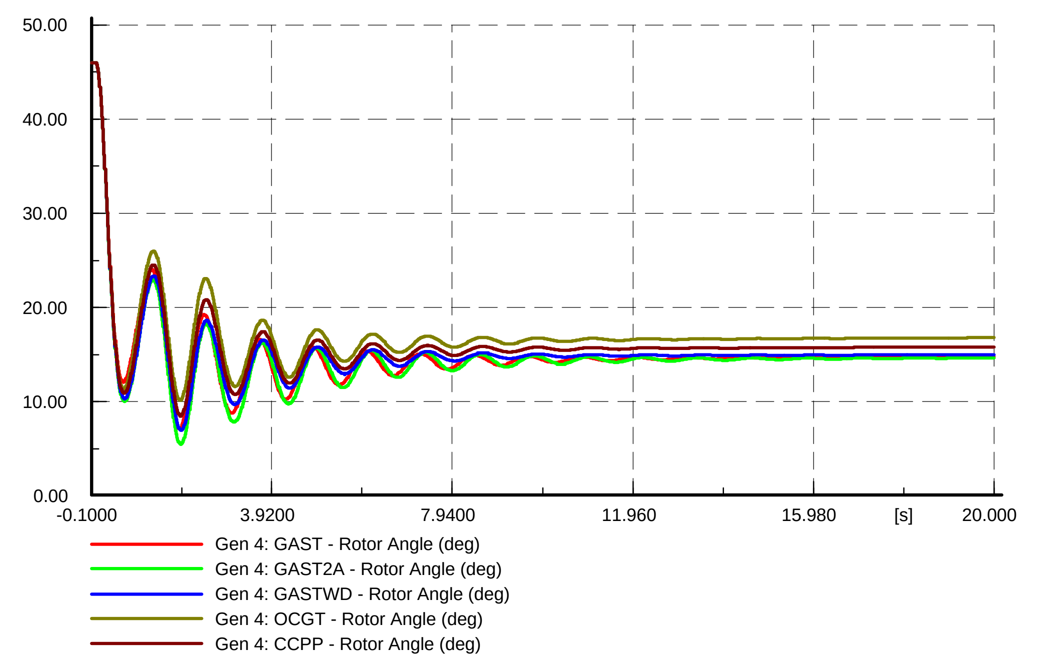
The difference between GAST and the other models is that GAST is the simplest model and does not take into account all the characteristics of the gas turbine. The GAST model does not take into account the gas turbine dynamics.

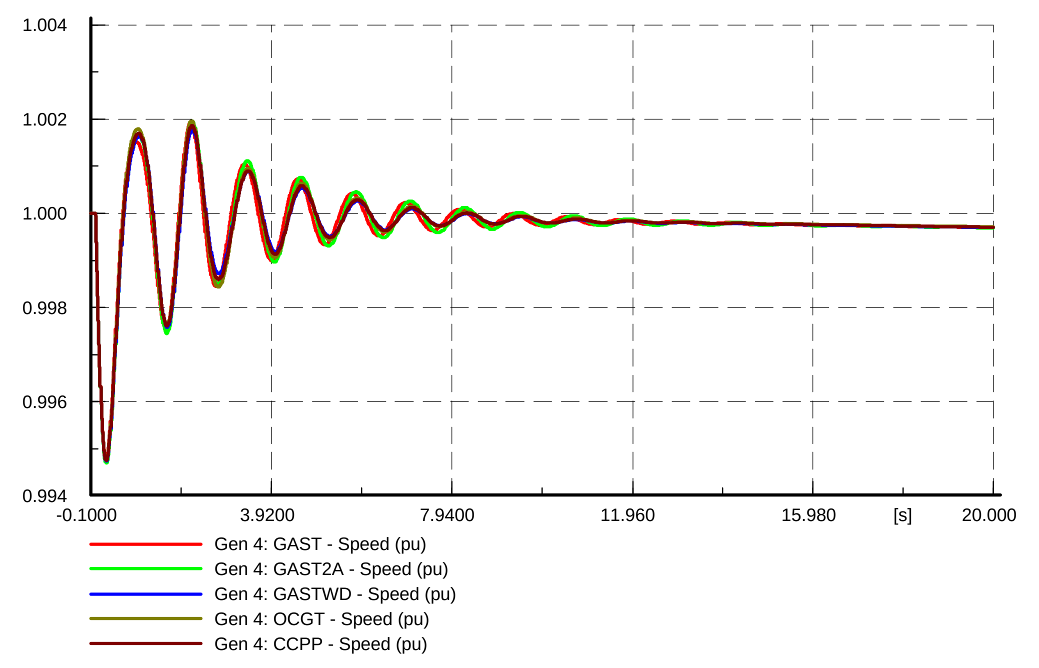


20316976	Development and Adaptation of Dynamic Models	Results 5.5.5.a	Date: 25/04/2011
	Generator 4 output reduced by 50 %	External grid with high acceleration time constant	Annex: /1









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Development and Adaptation of Dynamic Models

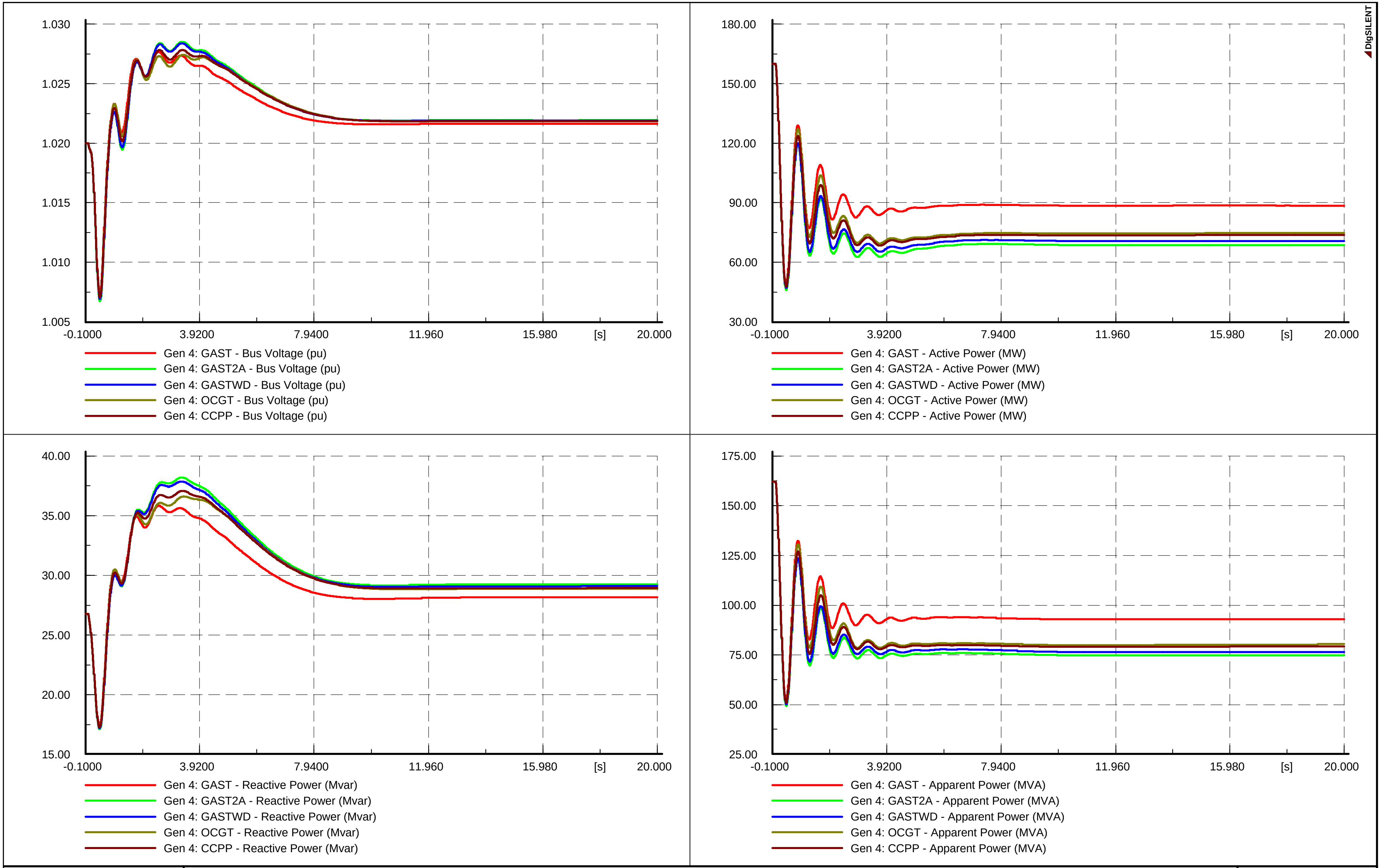
Results 5.5.5.b

Date: 25/04/2011

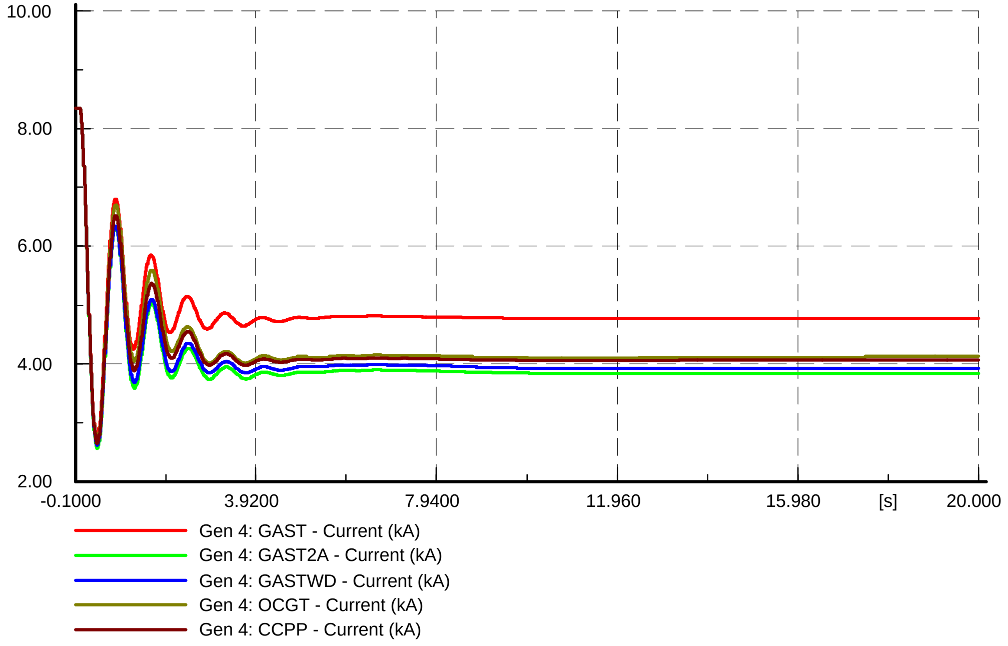
Generator 4 output reduced by 50 %

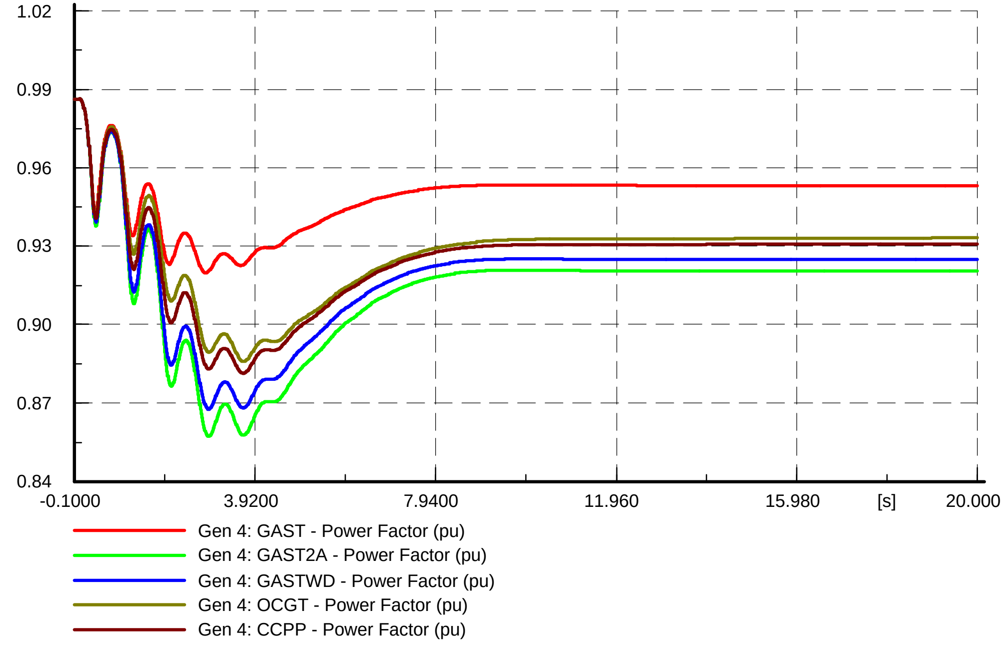
External grid with high acceleration time constant

Annex: /2



20316976	Development and Adaptation of Dynamic Models Generator 4 output reduced by 50 %	Results 5.5.5.c External grid with low acceleration time constant	Date: 25/04/2011 Annex: /1





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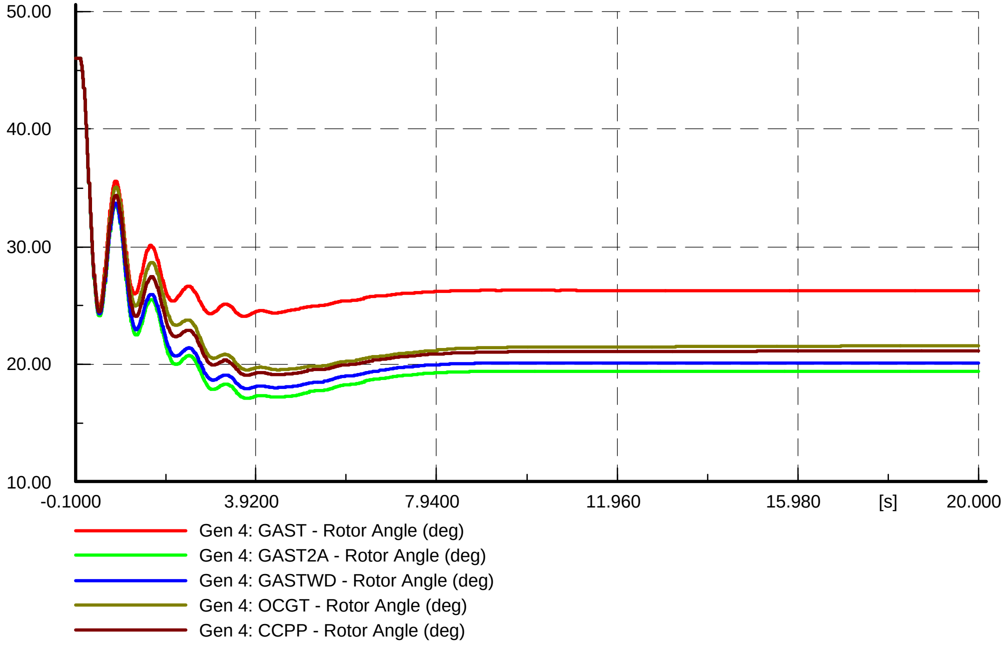

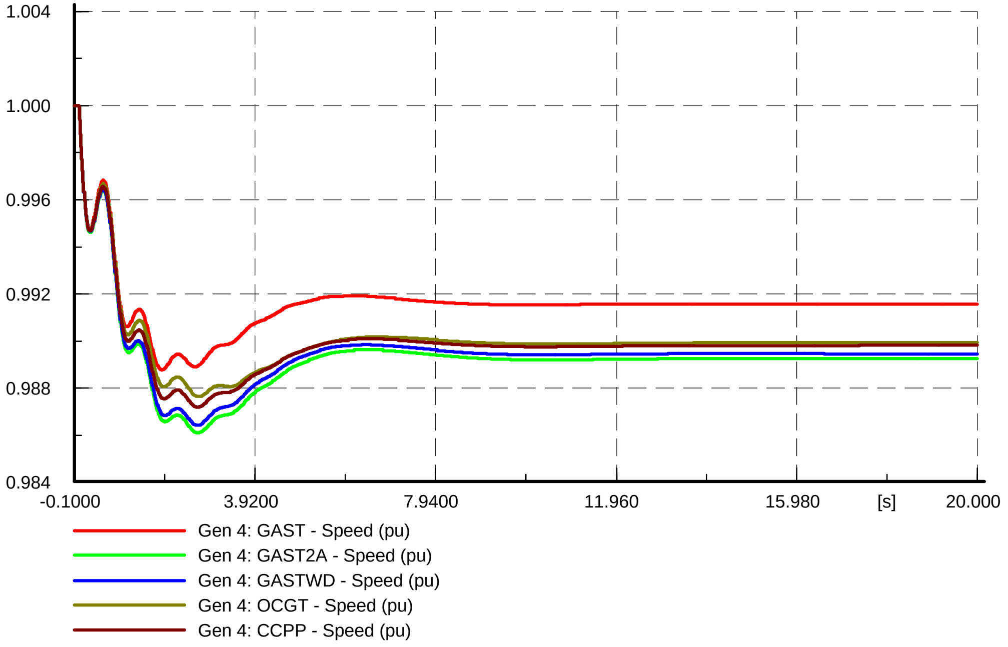

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Development and Adaptation of Dynamic Models

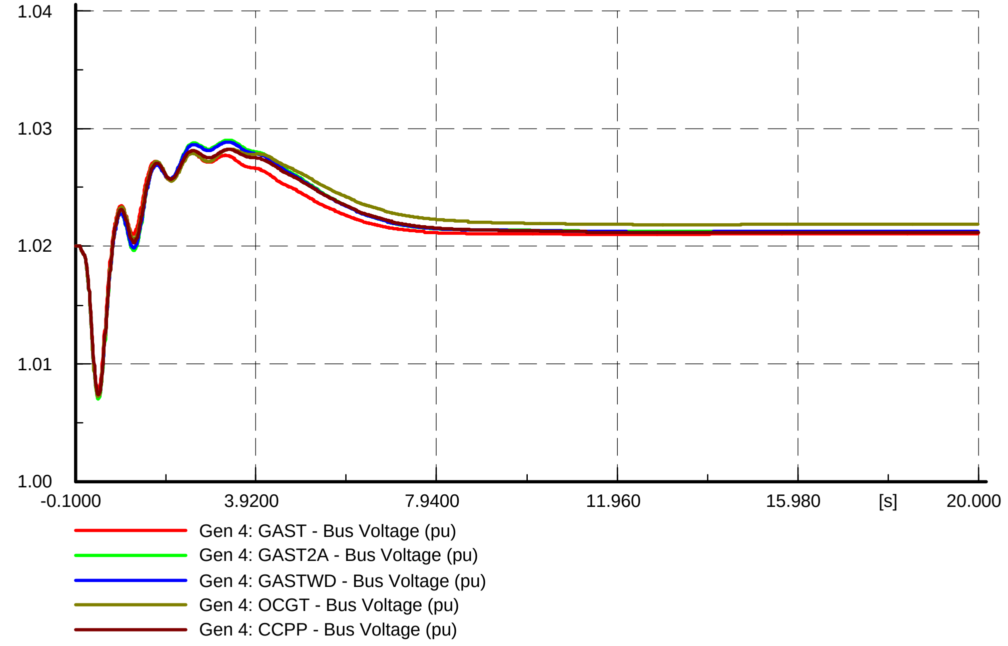
Results 5.5.5.d

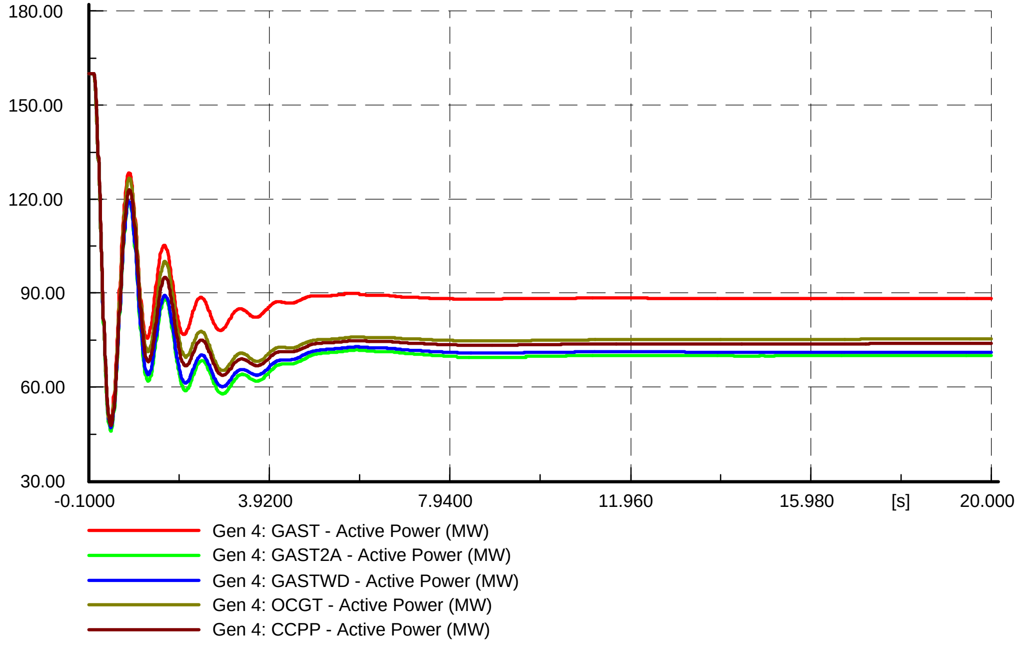
Date: 25/04/2011

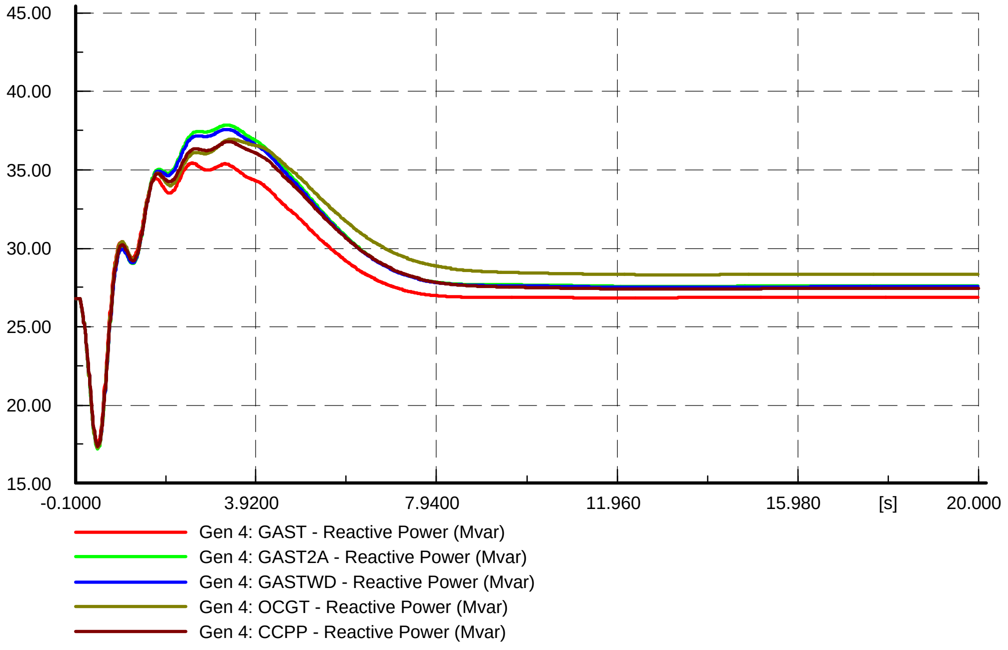
Generator 4 output reduced by 50 %

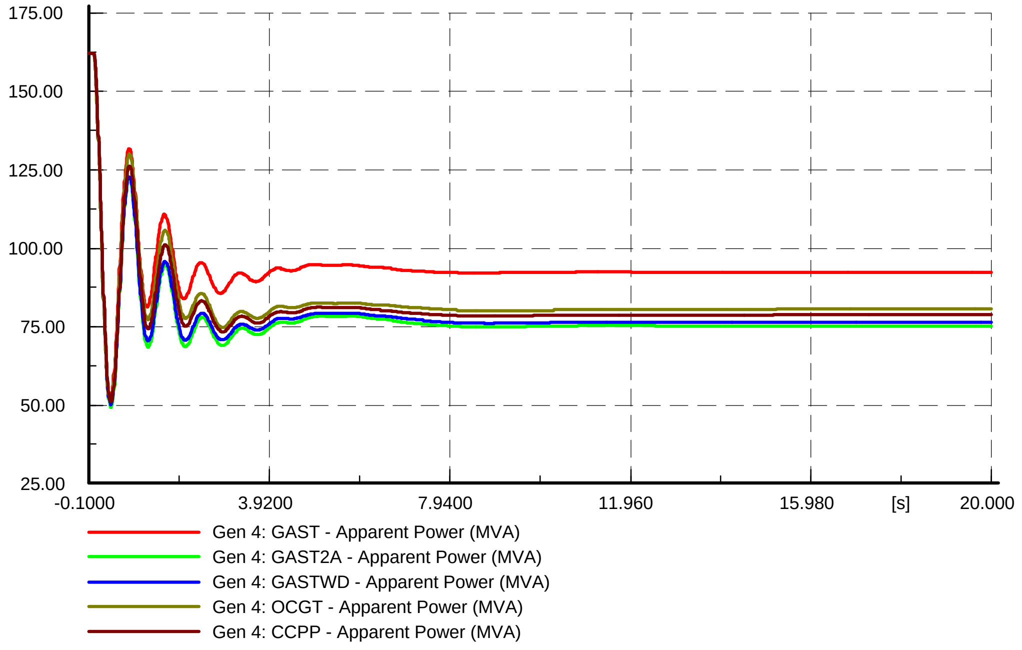
External grid with low acceleration time constant

Annex: /2









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Development and Adaptation of Dynamic Models

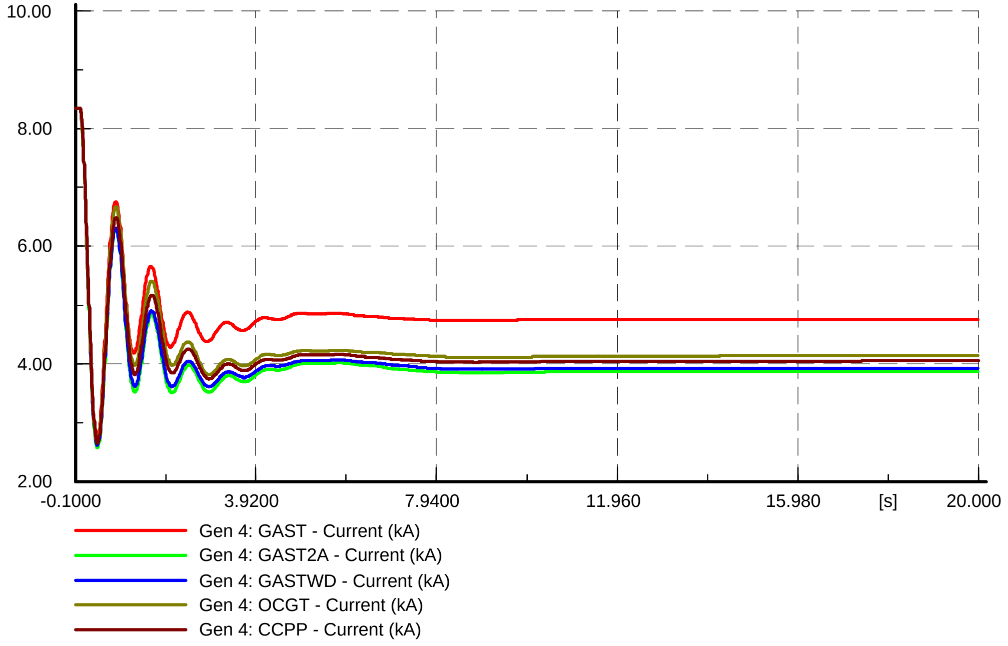
Generator 4 output reduced by 50 % (Higher gains)

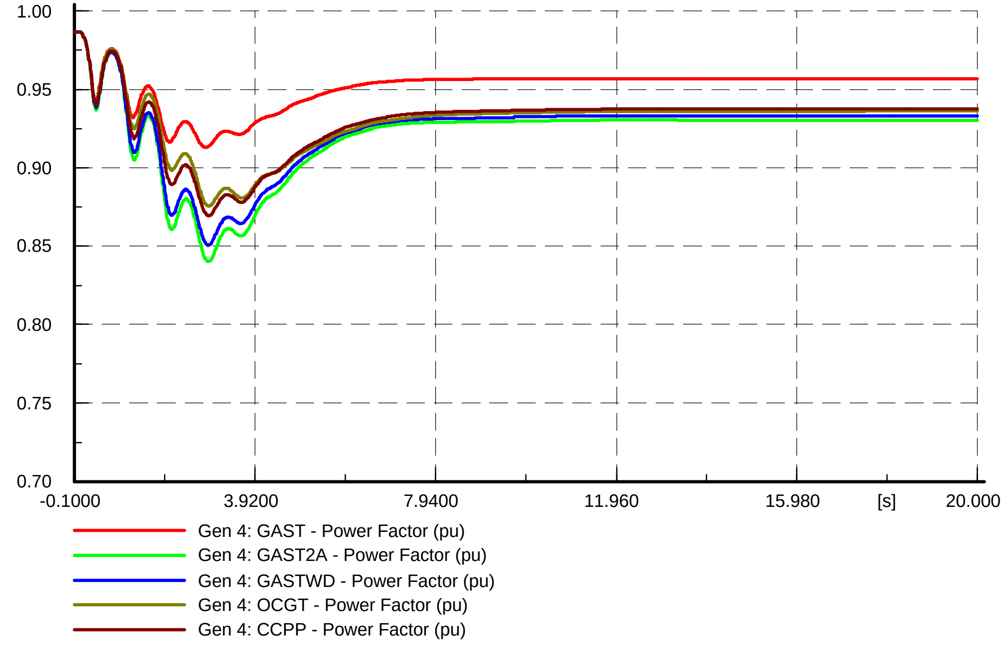
External grid with low acceleration time constant

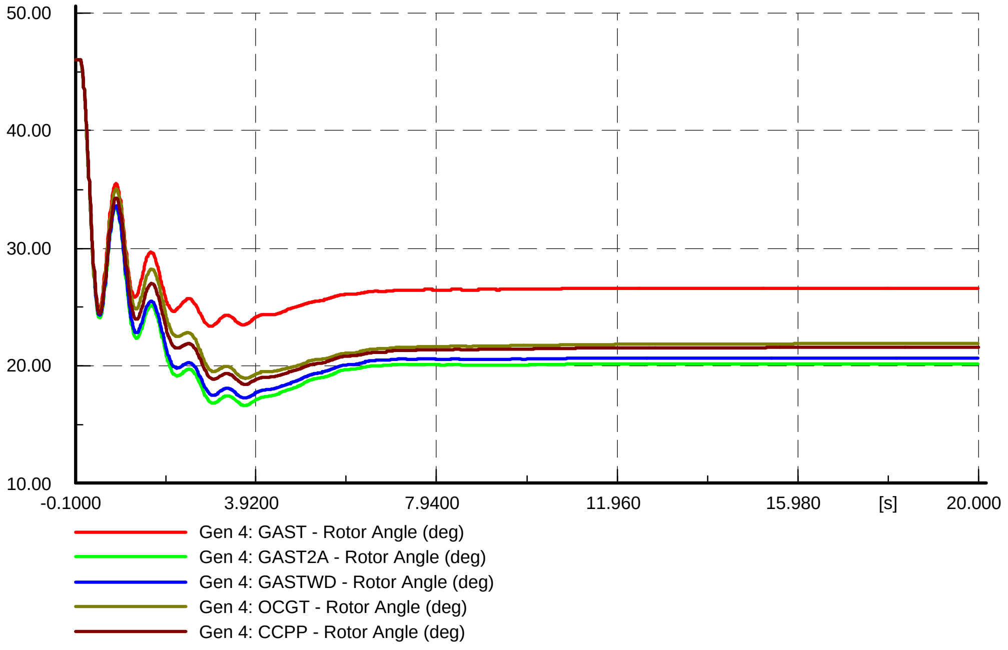
Results 5.5.5.e

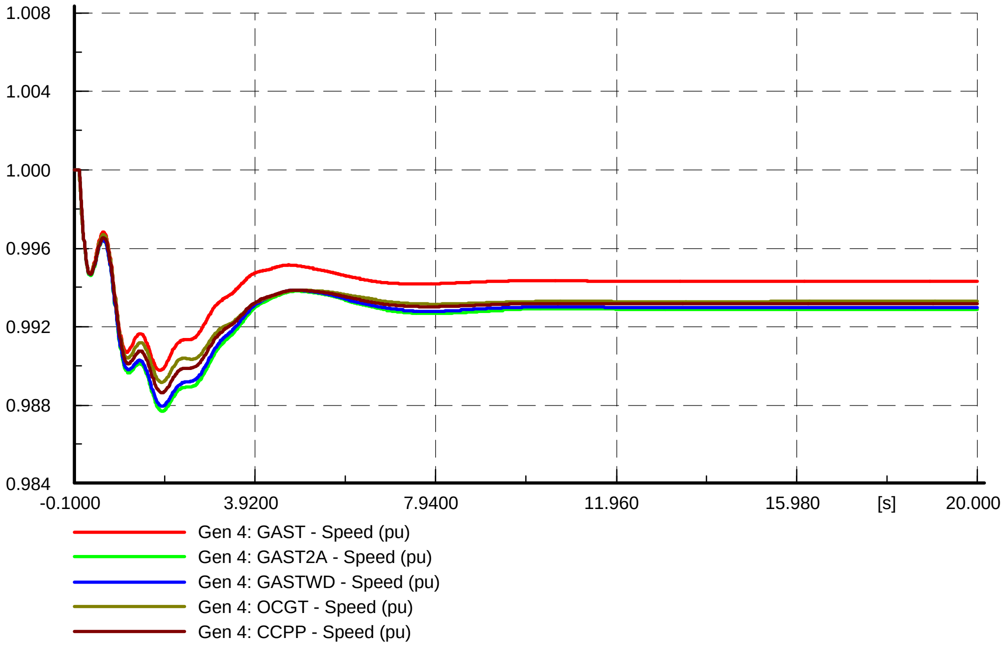
Date: 25/04/2011

Annex: /1









20316976

Development and Adaptation of Dynamic Models

Results 5.5.5.f

Date: 25/04/2011

Generator 4 output reduced by 50 % (Higher gains)

External grid with low acceleration time constant

Annex: /2

5.5.6 SHED LOAD D

For this disturbance, Load D is switched off. Load D is a substantial load of 100 MW with a power factor of 0.95 lagging. The total load of the simulated network is 585 MW with a lagging power factor of 0.95. The 100 MW load is approximately 17% of the total load.

Expected outcome:
The expected outcome of this simulation is that the various models will provide the same results. A change to the operating point of generator 4 is expected, but the post-disturbance operating point for active power and speed should be same as the pre-disturbance operating point. A change is expected in the reactive power, the power factor, the apparent power and the rotor angle for all gas turbine models.

High inertia system: (Result 5.5.6.a and Result 5.5.6.b)
After the disturbance, the system returns to a steady state operating point. Generator 4 returns to its pre-disturbance values for active power. However, the system settles at a value for speed slightly higher than rated speed. This is expected, as the disturbance in the network did not change the operating point (i.e. active power) of Generator 4. The results show that each model settles at the same values for active power, while there are small changes to the apparent power, current, power factor, rotor angle and the speed / frequency of the system.

The external grid is the swing bus for this network. The swing bus is important in power system studies, as this bus is used for the power balance of the network. The swing bus in the modelled network was also set with a very high inertia constant, therefore approaching an infinite bus. The generators will therefore settle at their respective pre-disturbance set values (with specific reference to the active power and speed), while the swing bus will balance the power requirements in the system, due the change in the system's loading.

Low inertia system. (Result 5.5.6.c and Result 5.5.6.d)
Although all the models settles at a steady state operating point after the disturbance, each model settles at different values for active, apparent power, current, power factor and rotor angle. As the disturbance was to shed Load D, it was expected that the various models would settle at the pre-disturbance values. However, each of the models settles at different values for active power (The pre-disturbance value for the active power is 160 MW). The GAST model settles at approximately 132 MW, GAST2A at approximately 140 MW, GASTWD at approximately 132 MW, CCPP at approximately 147 MW and OCGT at approximately 142 MW. This in turn causes the rotor angle, the power factor, the reactive and apparent power to settle at different values. The differences are substantial. The OCGT and CCPP models settles the closest to the pre-disturbance value of 160 MW, while GAST and GASTWD settles the furthest from the pre-disturbance value.

Although all the models are stable (i.e. no change in voltage, power, rotor angle, speed), the steady state speed (or frequency) after the disturbance varies between 1.008 pu and about 1.010 pu. for the models

Conclusion:

High inertia system:

The system is stable during and after this disturbance, as can be seen from the various graphs.

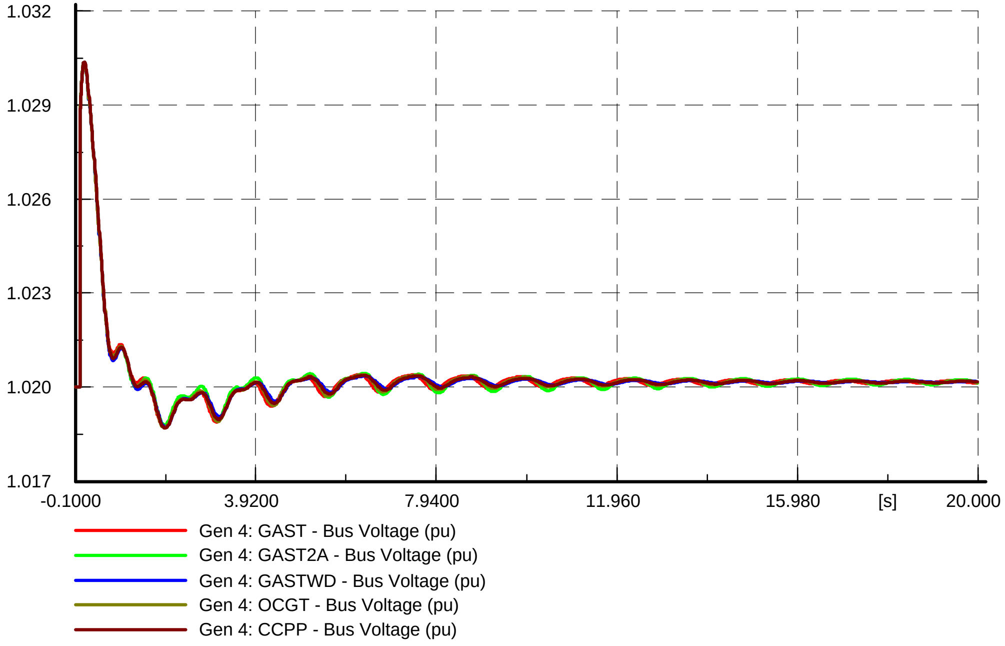
The results show that the various gas turbine models' response to the disturbance is similar with small variations between the models. This disturbance was chosen to provide an additional simulation to prove the similarities between the various gas turbine models. The responses are almost identical for the first 1.5 seconds, where after small, but negligible deviations are observed. The oscillations in the network are damped and the system reaches a constant speed after a while. However, the speed is not rated speed, because of the non-zero droop settings of the generator governor controllers.

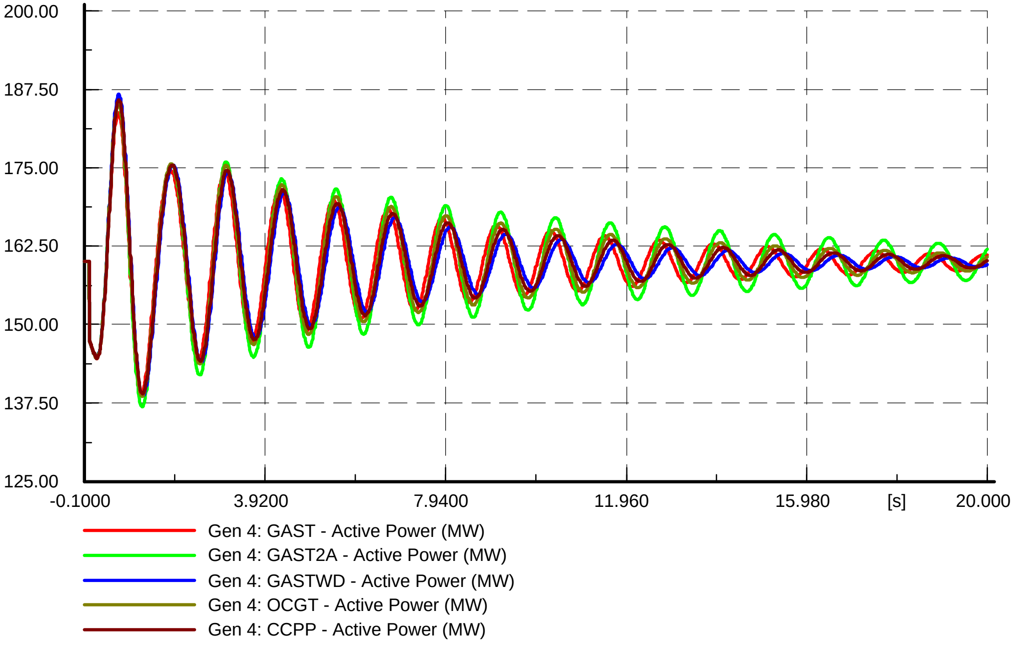
As with all the previous studies, this simulation shows that the controllers (i.e. governors) for the various gas turbine models provide similar results for the same disturbance in the network. It shows that the controllers are settable to achieve the similar results (i.e. to achieve the first part of the aim of the studies).

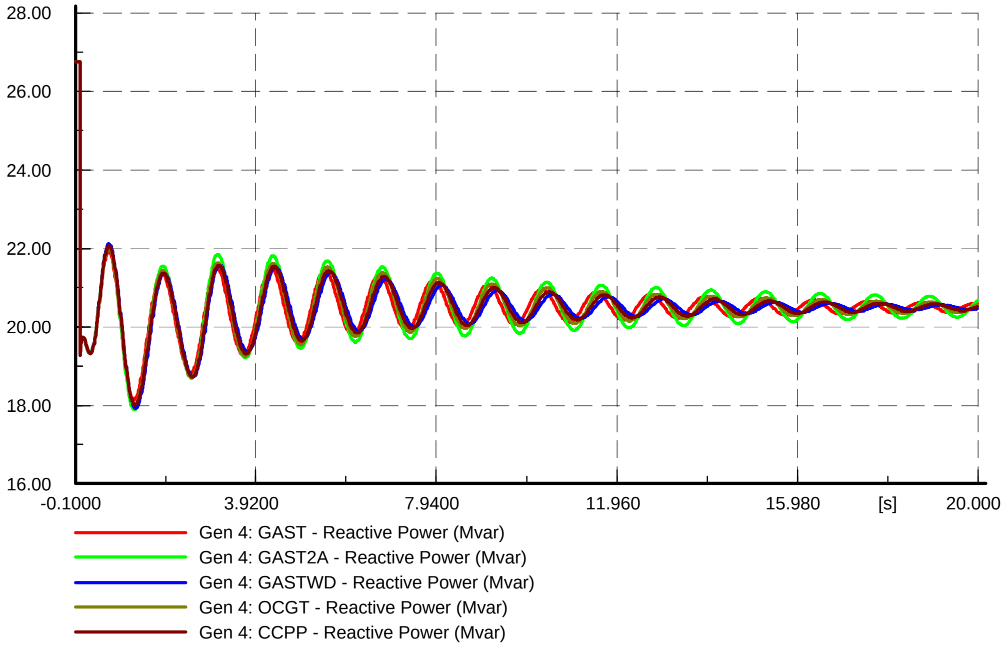
Low inertia system:

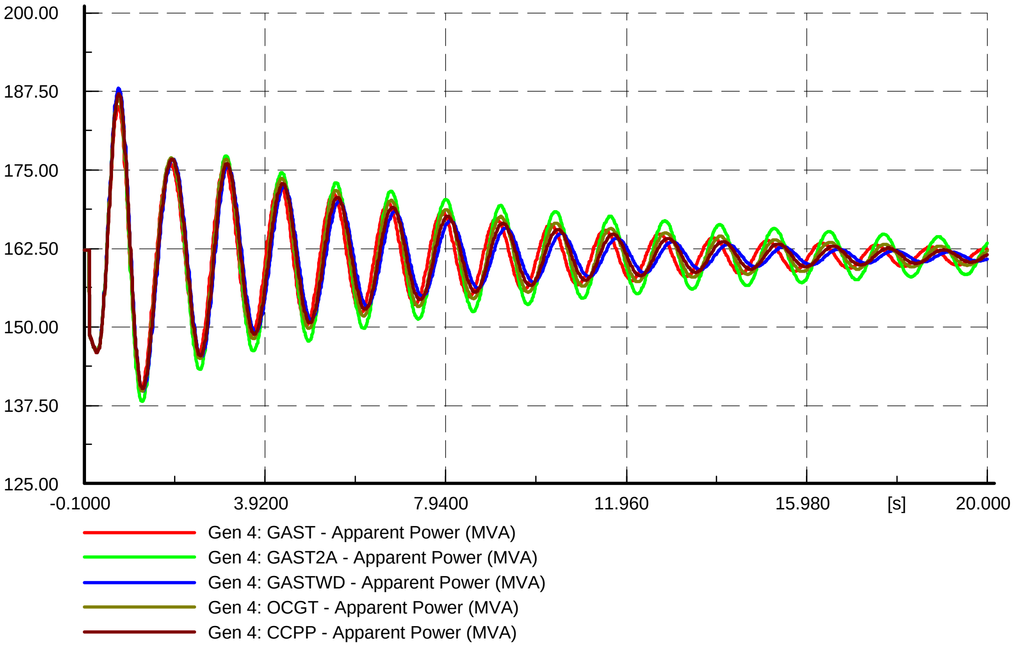
Although the generator voltage and the reactive power settles at close to the pre-disturbance value, the simulations show that the system does not maintain rated speed (or rated frequency). Although the system speed is not rated speed, it does settle at a steady state operating point. The reason is the generator governor controllers' non-zero droop settings. Refer to Section 5.5.5 for an explanation.

There are differences in the results between the various gas turbine models. As indicated above the various models stabilise at various values after the disturbance, but during the first approximately 2 seconds, all the models respond the almost the same to the disturbance.

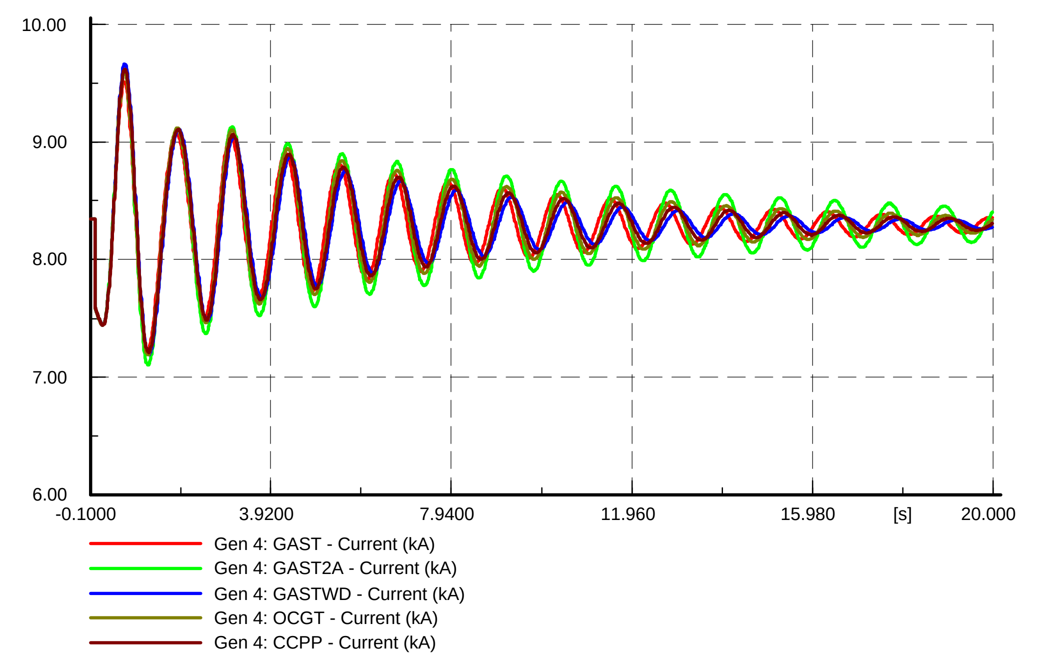


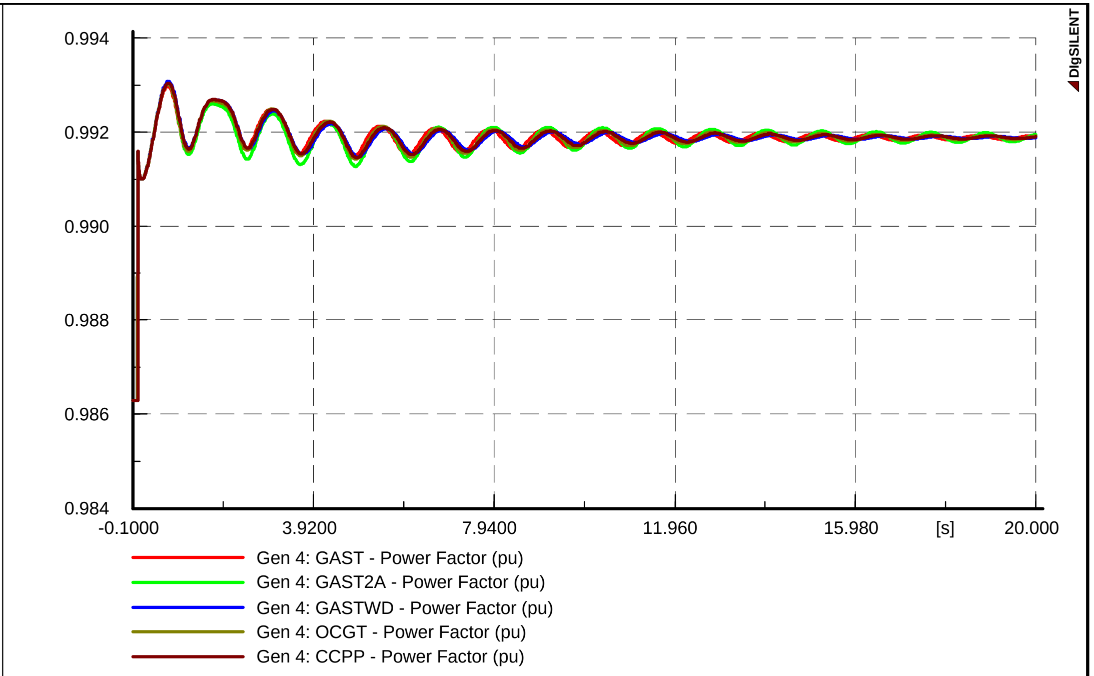


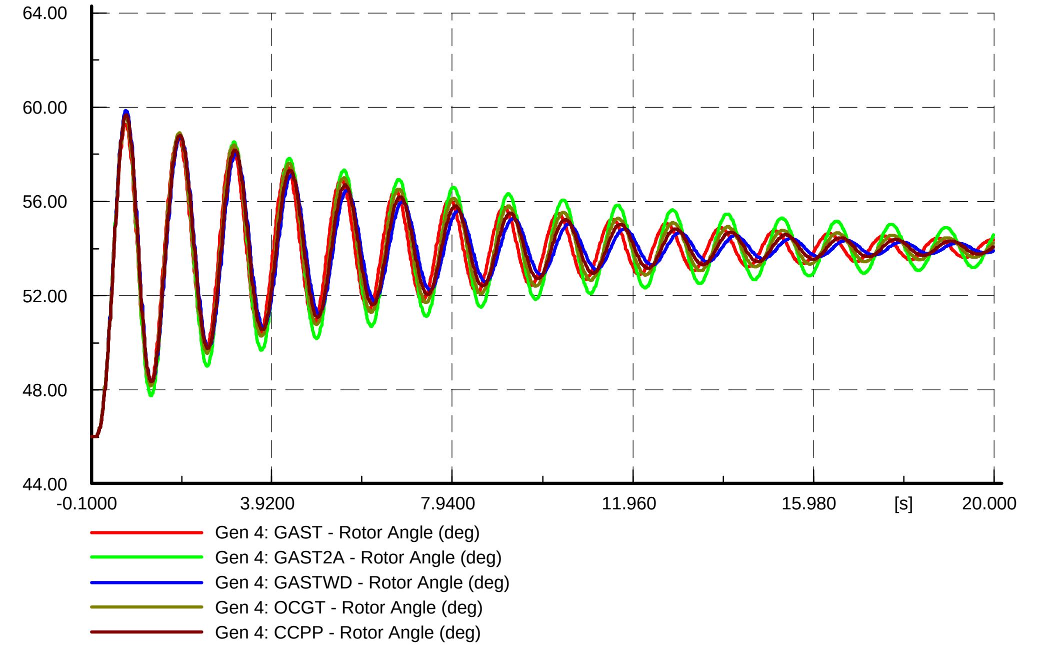


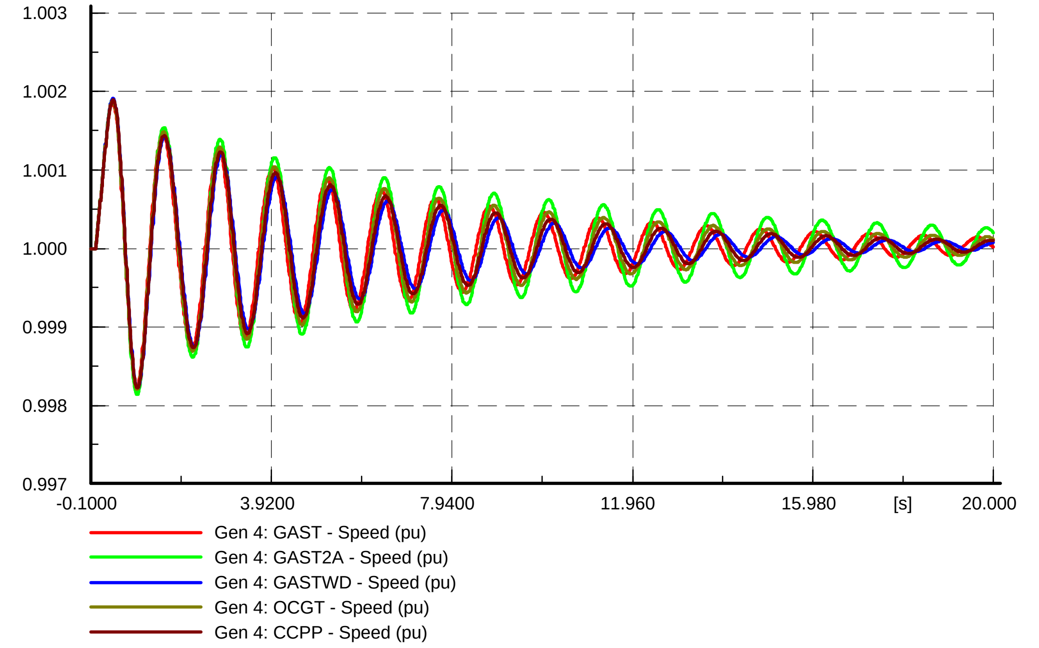


20316976	Development and Adaptation of Dynamic Models 100 % Loadshed of Load D	External grid with high acceleration time constant	Results 5.5.6.a	Date: 25/04/2011 Annex: /1









20316976

Development and Adaptation of Dynamic Models

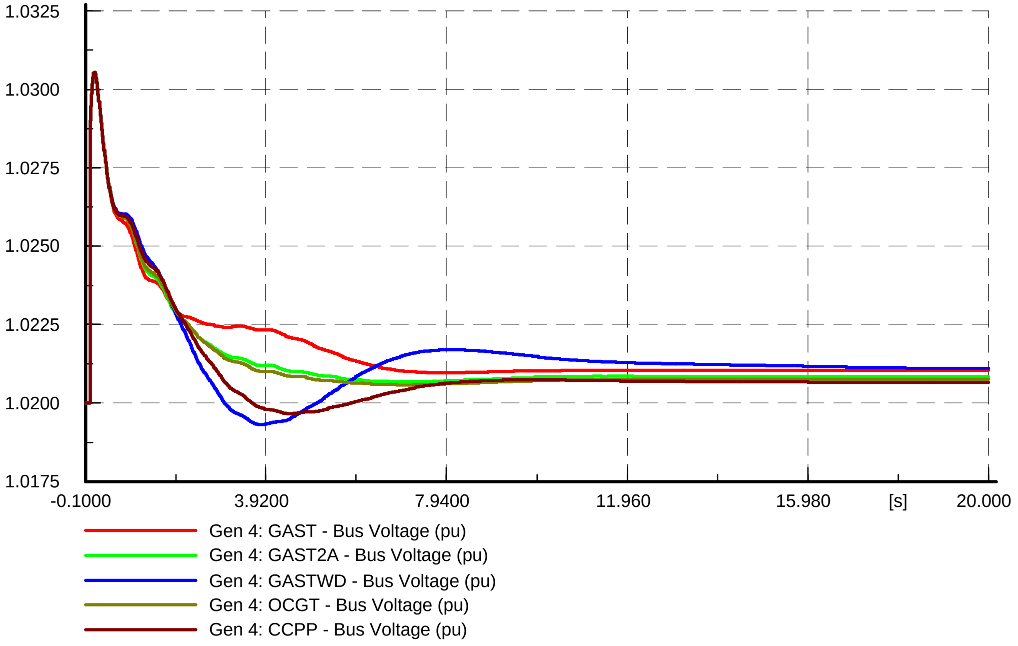
Results 5.5.6.b

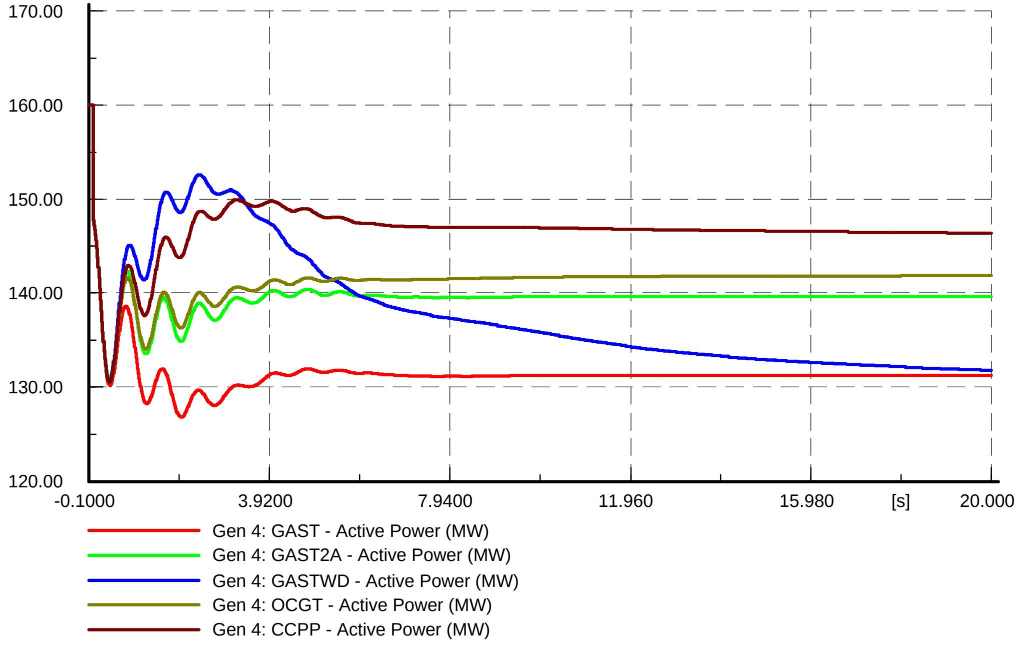
Date: 25/04/2011

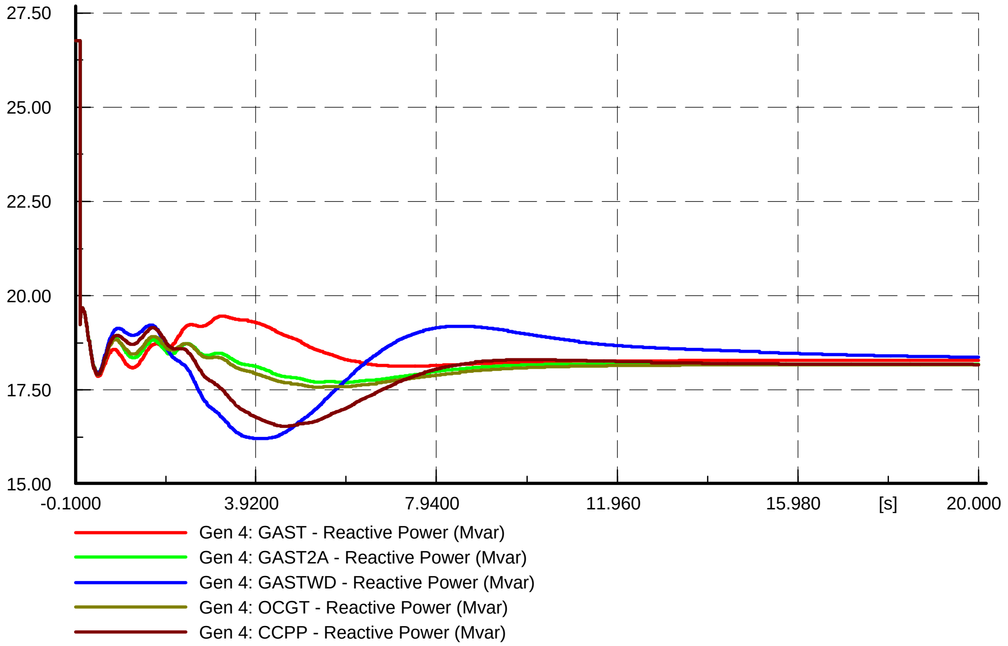
100 % Loadshed of Load D

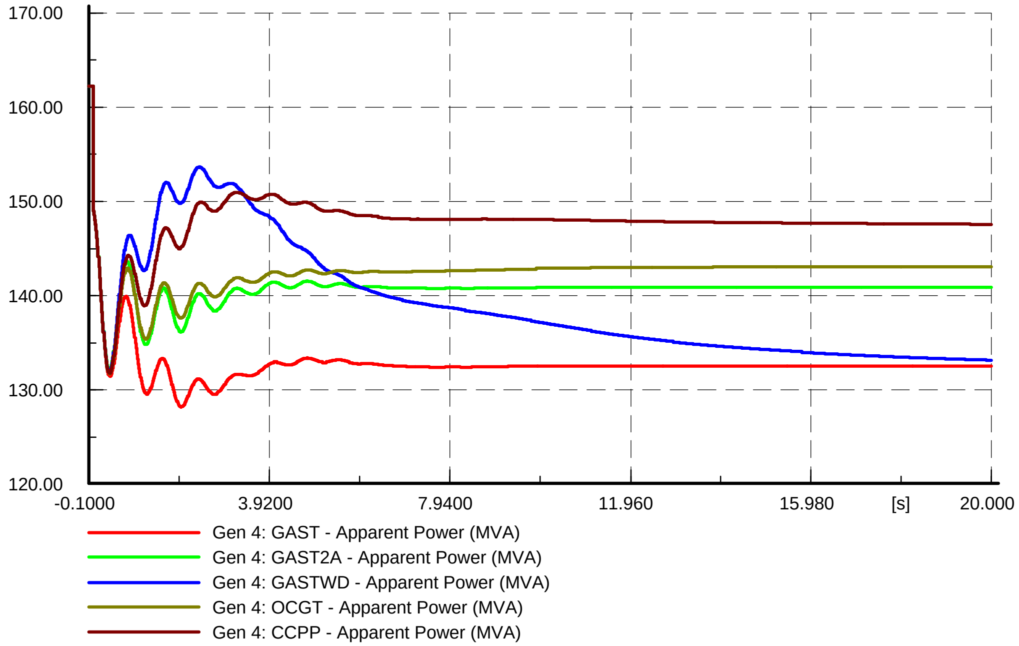
External grid with high acceleration time constant

Annex: /2









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Development and Adaptation of Dynamic Models

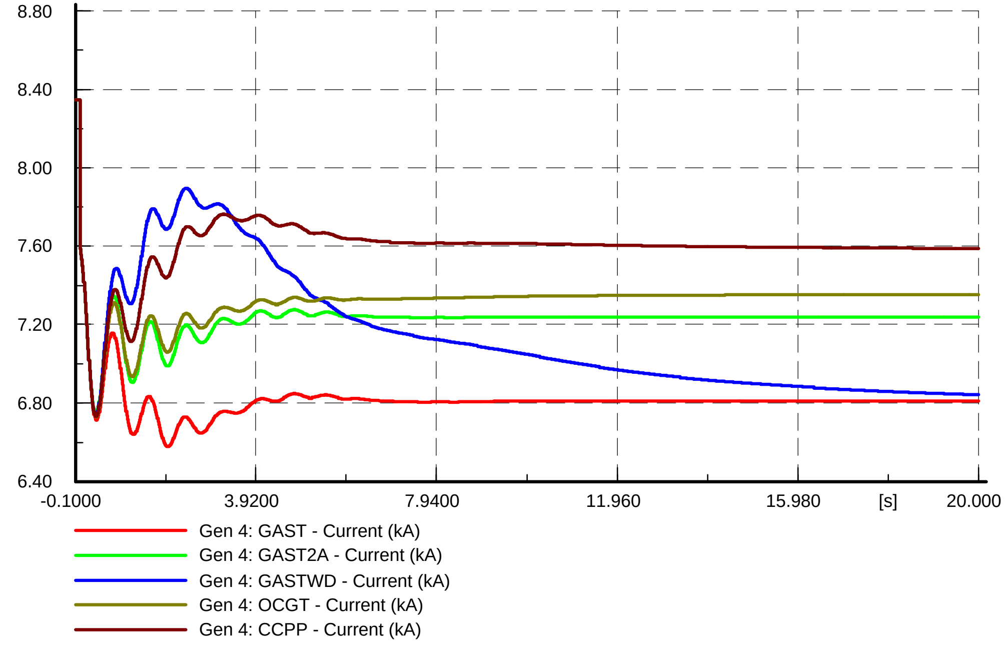
100% Loadshed of Load D

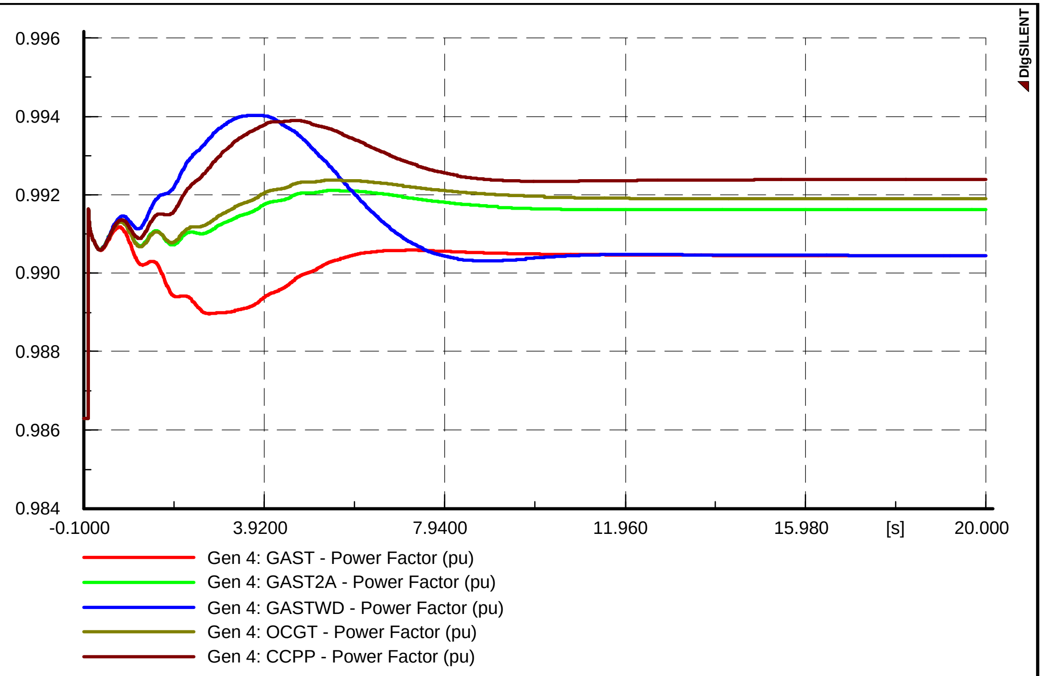
External grid with low acceleration time constant

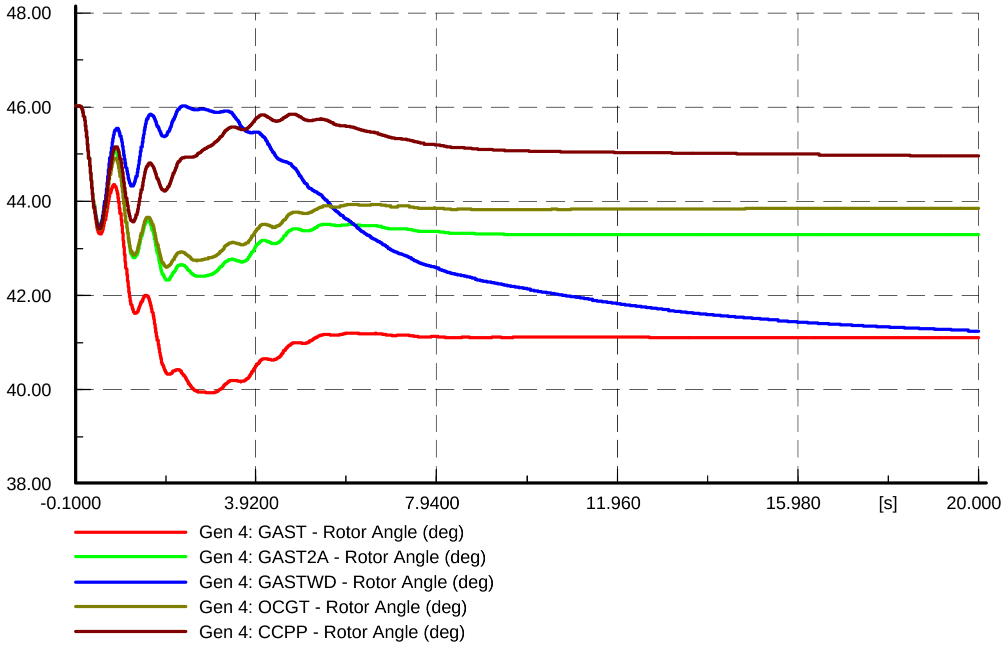
Results 5.5.6.c

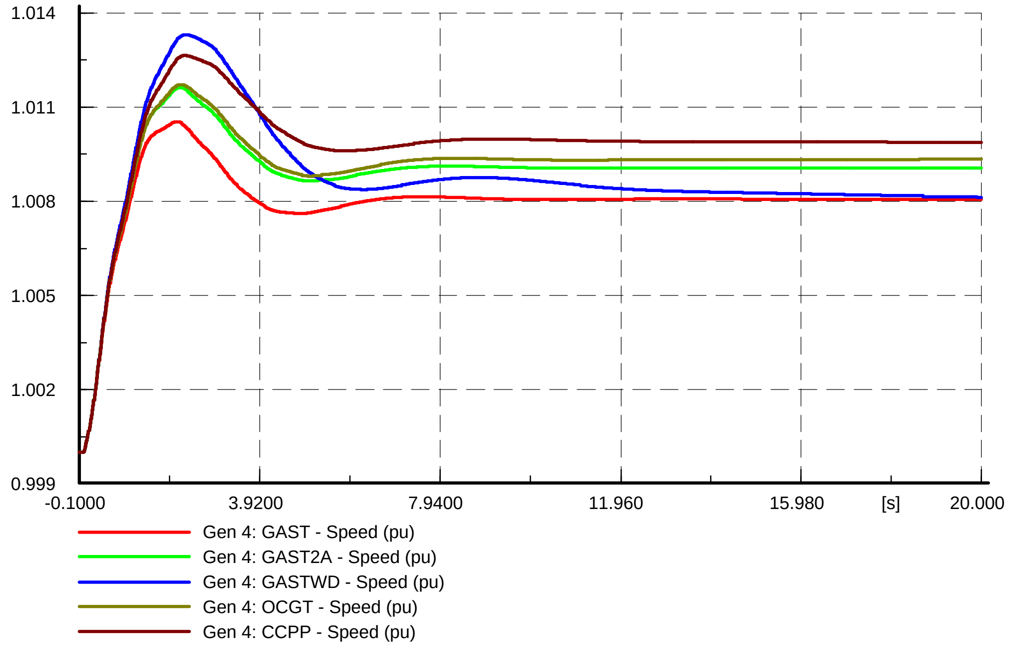
Date: 25/04/2011

Annex: /1









20316976	Development and Adaptation of Dynamic Models 100% Loadshed of Load D	Results 5.5.6.d External grid with low acceleration time constant	Date: 25/04/2011 Annex: /2

5.5.7 CRITICAL CLEARING TIME FOR GENERATOR 4

This study was performed to determine the critical clearing time for Generator 4 for a three-phase fault on the generator terminals. The critical clearing time is the maximum time allowed before a synchronous machine becomes unstable, i.e. pole slip or going out-of-step with the rest of the network. This is also known as loss-of-synchronism of the generator with the rest of the network.

Expected outcome:

It is expected that the machine remain stable (i.e. in synchronism with the rest of the system) for faults that are cleared by Zone 2 distance protection. Zone 2 distance protection time delay is set to 400 ms. The 400 ms is a typical time delay setting for Zone 2 distance protection schemes applied on the South African transmission network, i.e. on the overhead lines, while the Zone 1 distance protection is set to protect only the overhead line. The Zone 2 distance protection is set to detect faults on the equipment connected to the bus bar to which the overhead line is connected. In other words, if there is a transformer connected as well, the Zone 2 protection is set to detect faults in the transformer. The Zone 2 distance protection therefore overlaps the transformer unit protection schemes (i.e. transformer differential protection). The Zone 2 distance protection therefore acts as a backup protection to the transformer differential protection scheme. The expected value of 400 ms for the machine to stay in synchronism is to allow any Zone 2 distance protection to clear faults elsewhere on the network before the machine loses synchronism with the rest of the network.

High inertia system: (Result 5.5.7a to Result 5.5.7f)

During the fault (three-phase fault on Generator 4 terminals), the generator terminal voltage dropped to zero, as no impedance was modelled in the fault. The generator current increases and there is a change of the power factor during the fault period. During the fault condition, the speed of the generator increases as the source of the generator (the gas turbine) is producing active power while the generator is not delivering any active power to the network.

Result 5.5.7a and Result 5.5.7b show that the generator has not pole slipped for a fault with duration of 390 ms.

Result 5.5.7c and Result 5.5.7d show that the generator with the GASTWD model has pole slipped for a fault with duration of 400 ms.

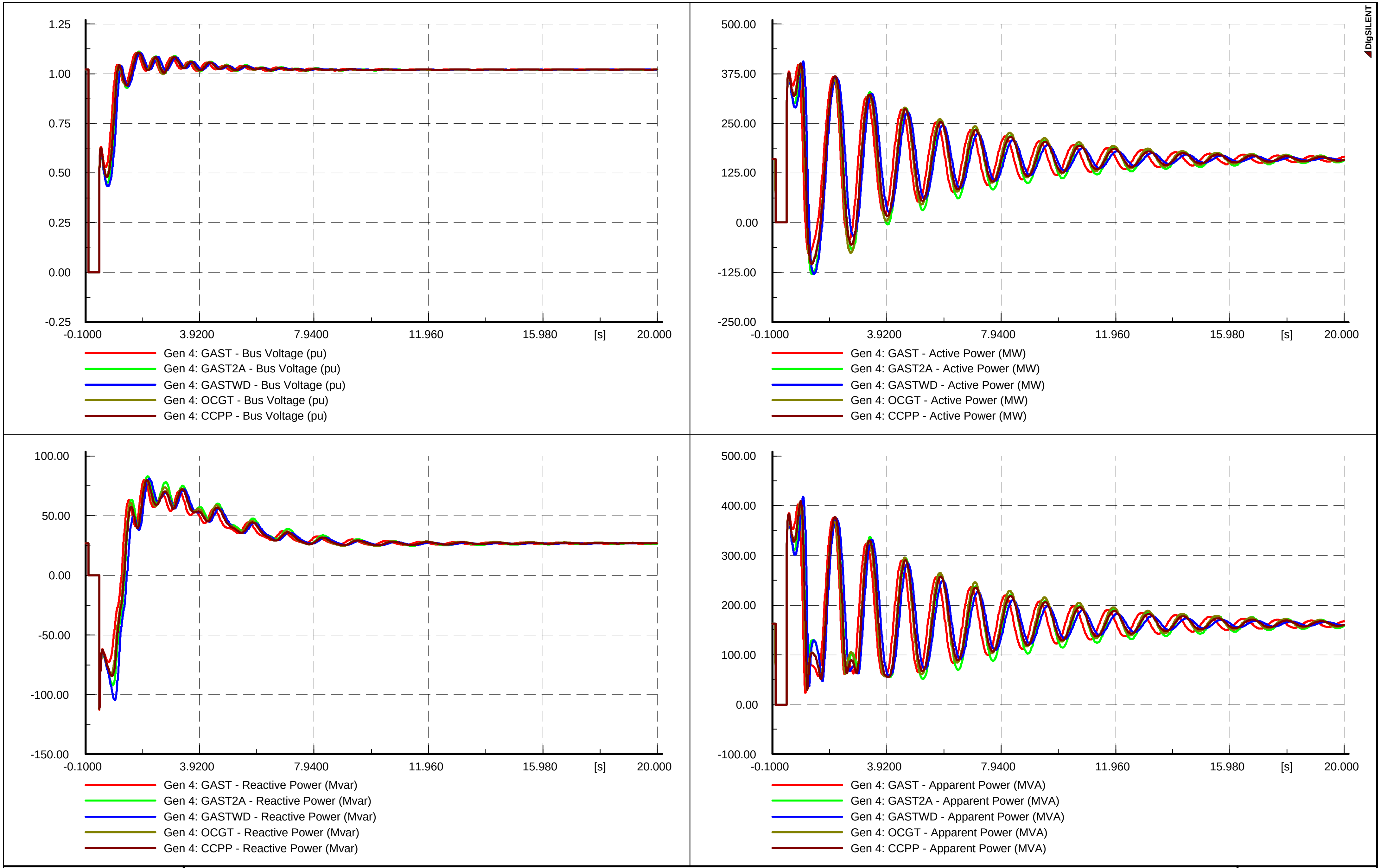
Result 5.5.7e and Result 5.5.7f show that the generator with all models except GAST has pole slipped for a fault with duration of 410 ms.

Additional simulations (graphical results not included) were done and it was found that the generator with the GAST model pole slipped for fault with duration of 430ms.

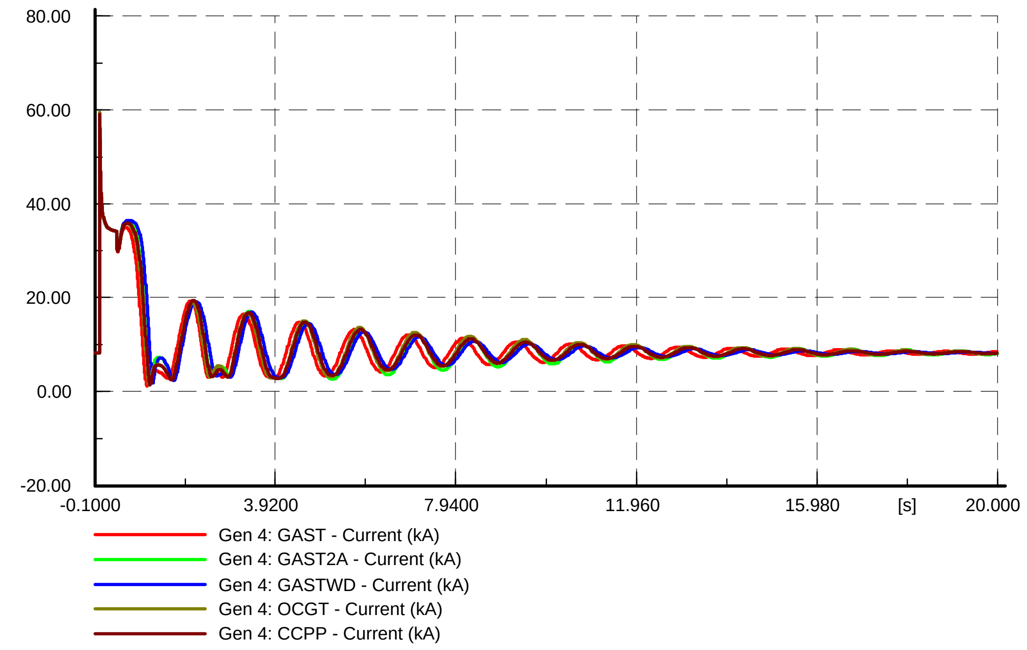
Conclusion:

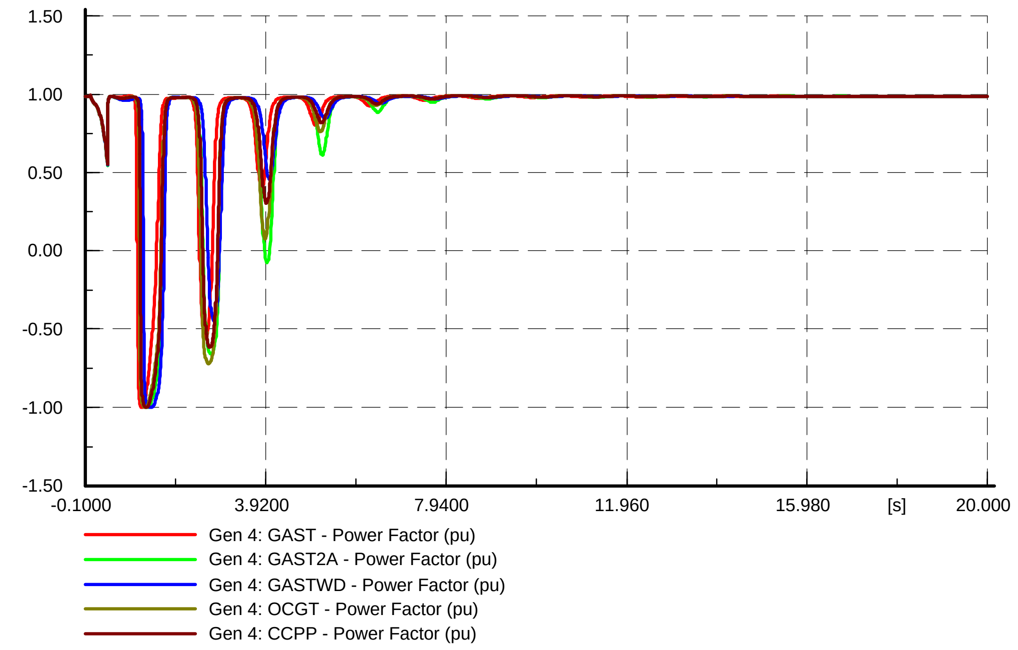
All the gas turbine models show that the critical clearing time of Gen 4 is 400 ms, except for GASTWD that showed the critical clearing time as 390 ms. The result is that the new and existing models (with the exception of GAST and GASTWD) provided similar results. This confirms the aim of this simulation to show that the new models do not replace the existing models, and the expectation that the machine should stay in synchronism for 400 ms, in order for the Zone 2 protection to operate.

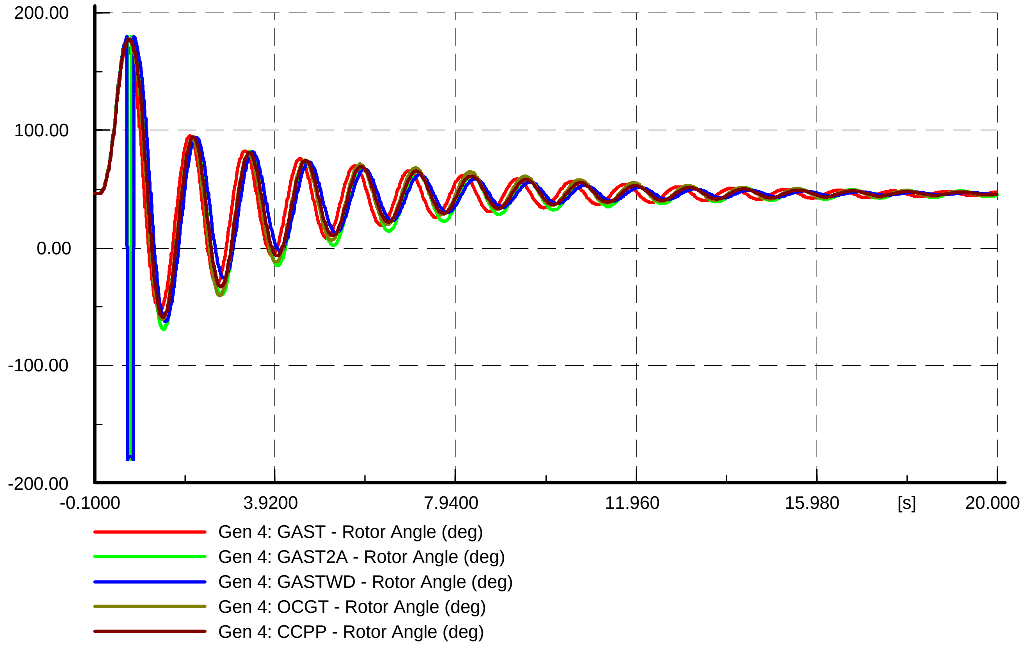
The difference between the models is because of the various controller types used in each of the models. GASTWD, OCGT and CCPP make use of PID controllers, while GAST and GAST2A uses lead-lag compensator. The lead-lag compensator’s response is different for the disturbance modelled in this section.

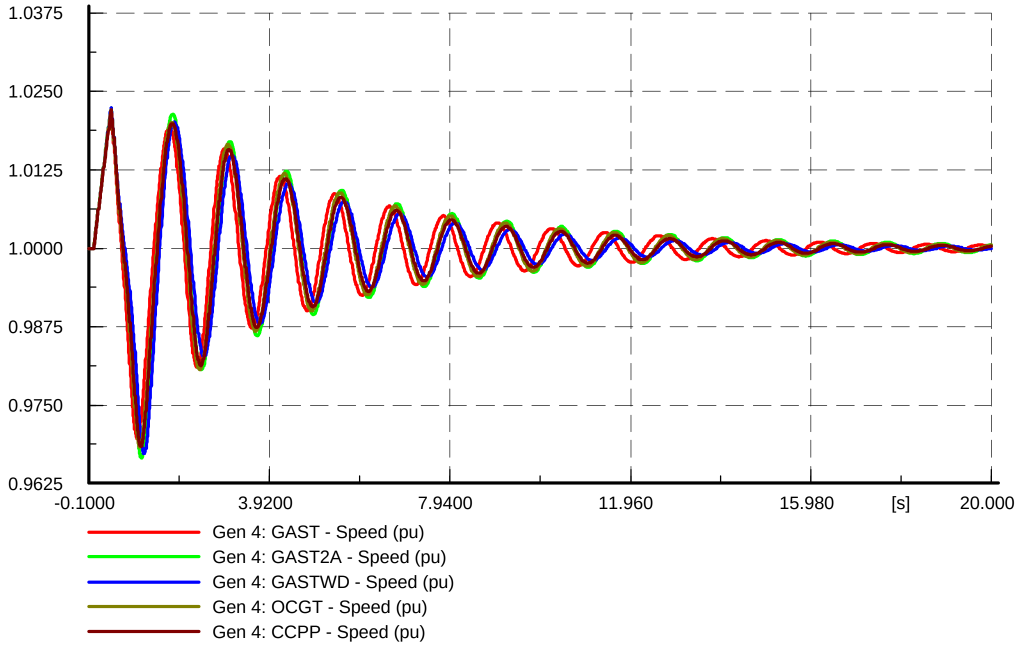


20316976	Development and Adaptation of Dynamic Models	Results 5.5.7.a	Date: 25/04/2011
	Critical clearing time for Generator 4 (3 Phase fault on terminals, 390 ms)	External grid with high acceleration time constant	Annex: /1









20316976

Development and Adaptation of Dynamic Models

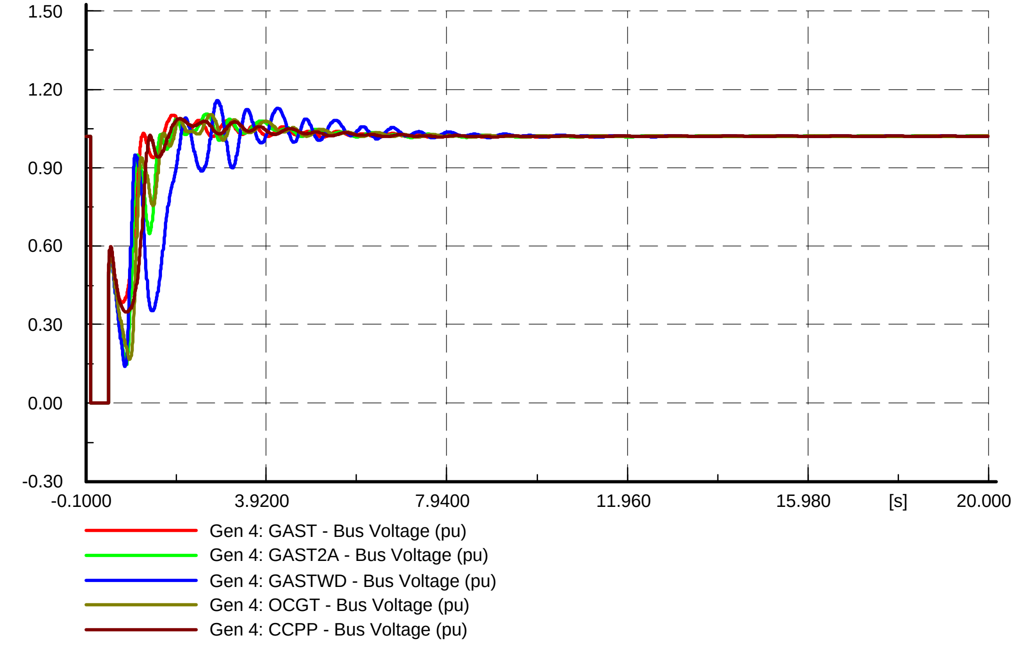
Critical clearing time for Generator 4 (3 Phase fault on terminals, 390 ms)

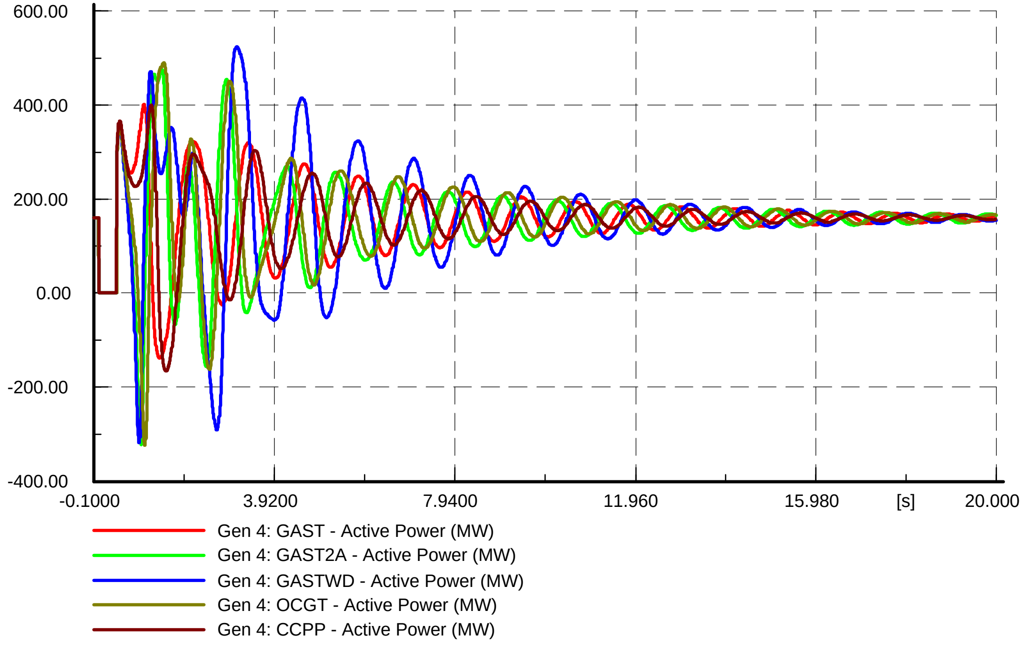
External grid with high acceleration time constant

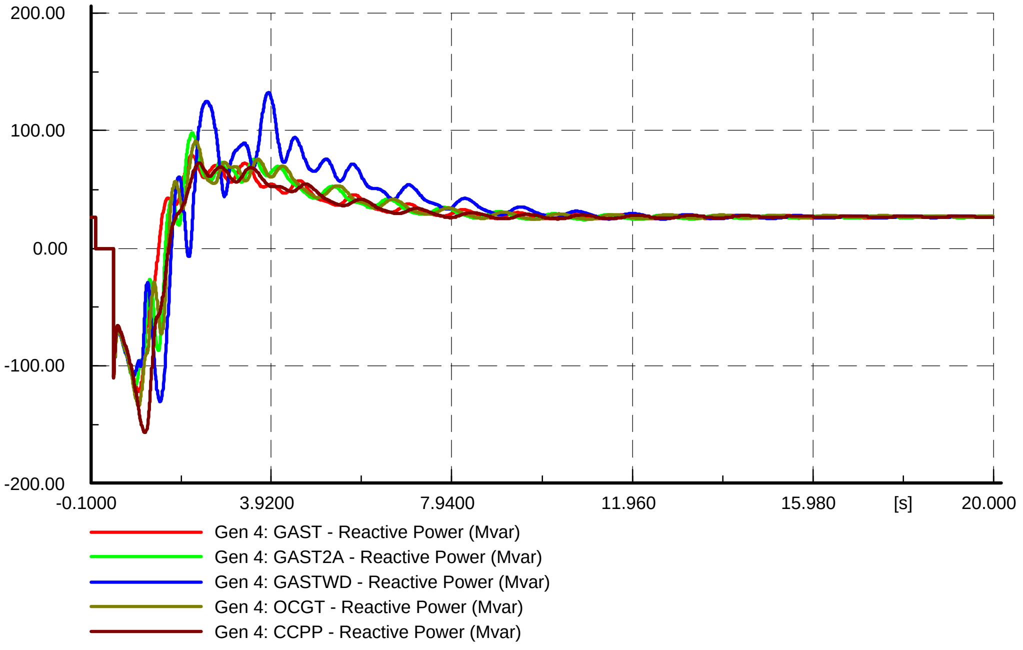
Results 5.5.7.b

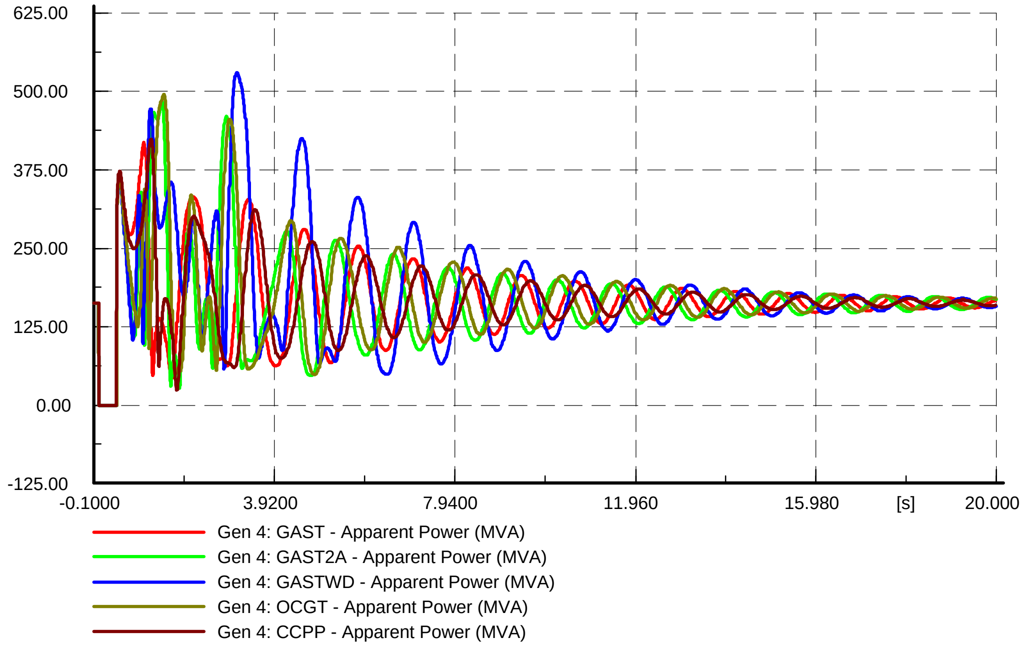
Date: 25/04/2011

Annex: /2









20316976

Development and Adaptation of Dynamic Models

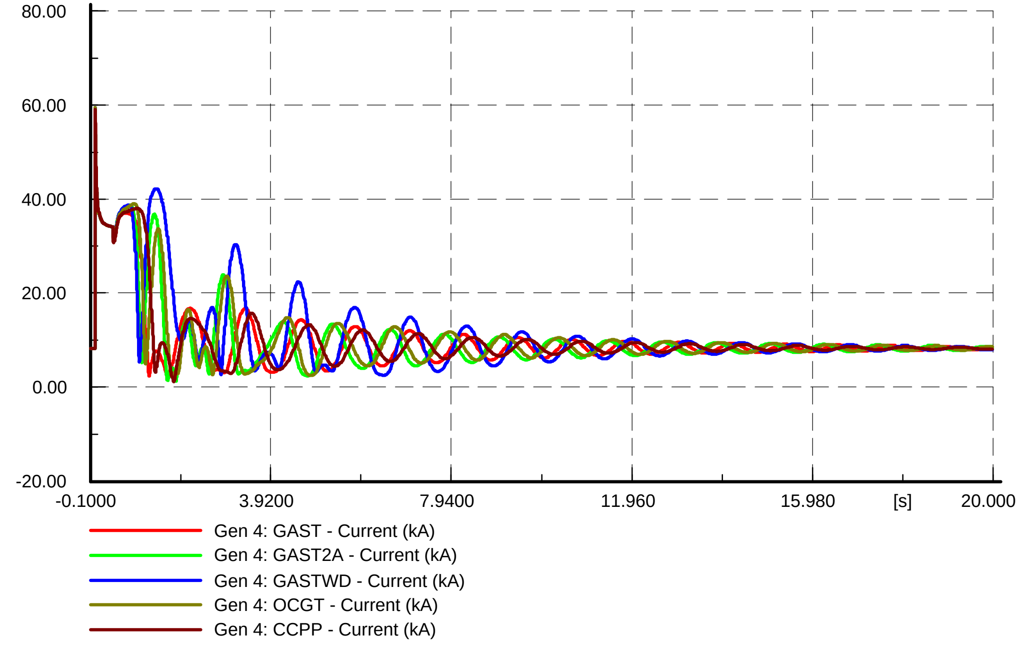
Results 5.5.7.c

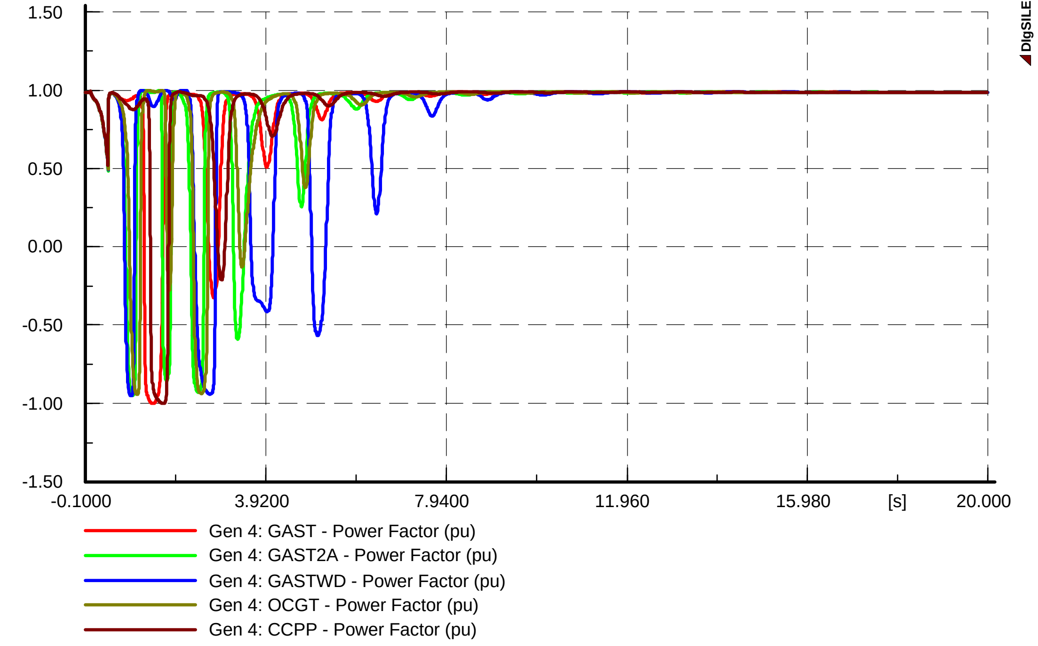
Date: 25/04/2011

Critical clearing time for Generator 4 (3 Phase fault on terminals, 400 ms)

External grid with high acceleration time constant

Annex: /1





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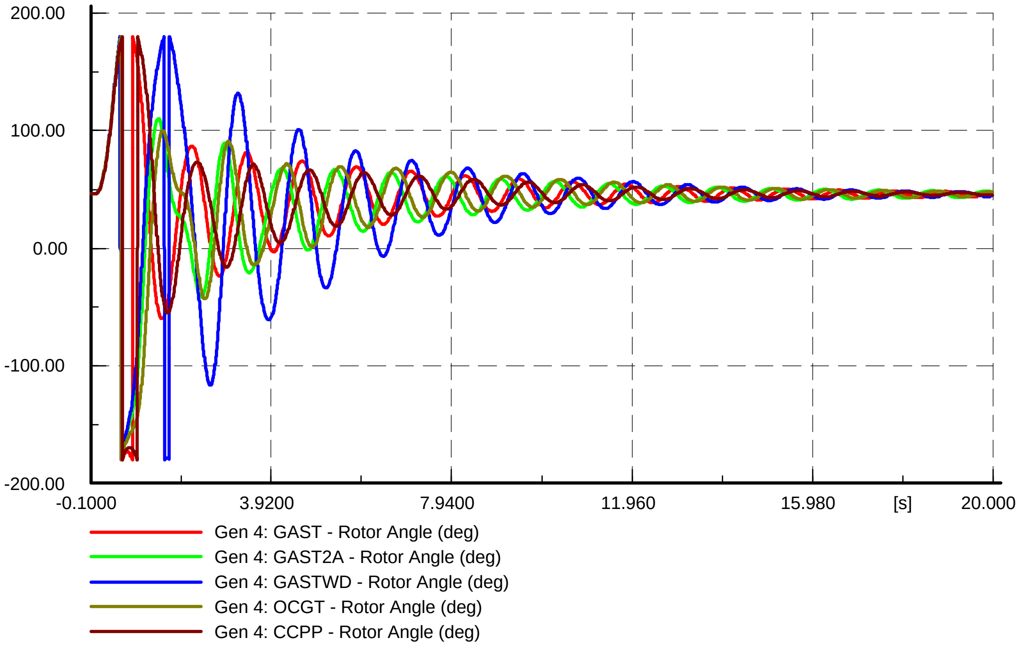
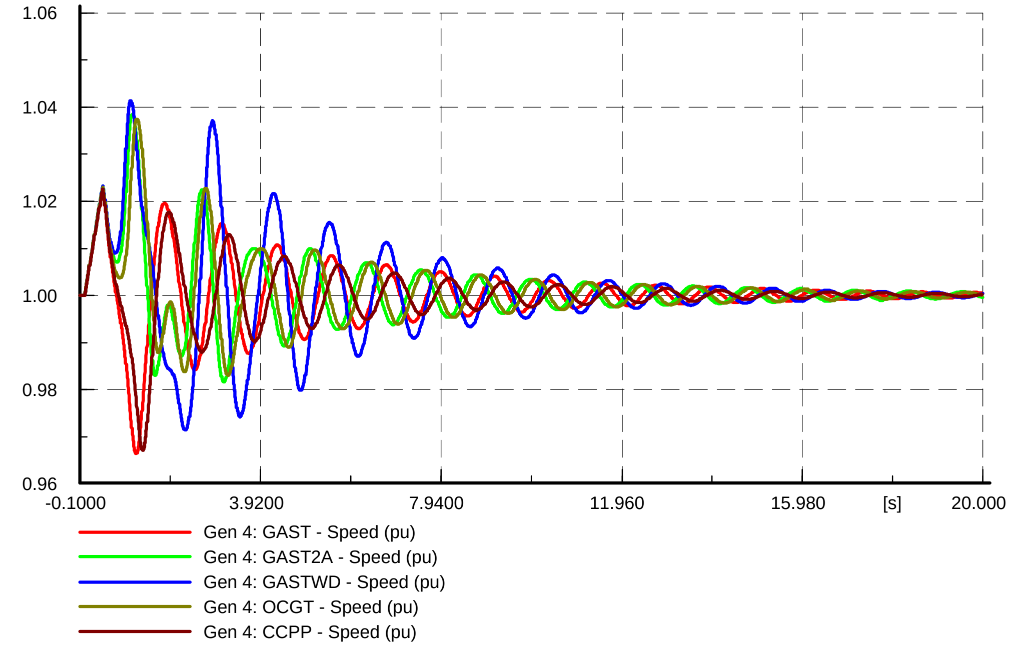




20316976

Development and Adaptation of Dynamic Models

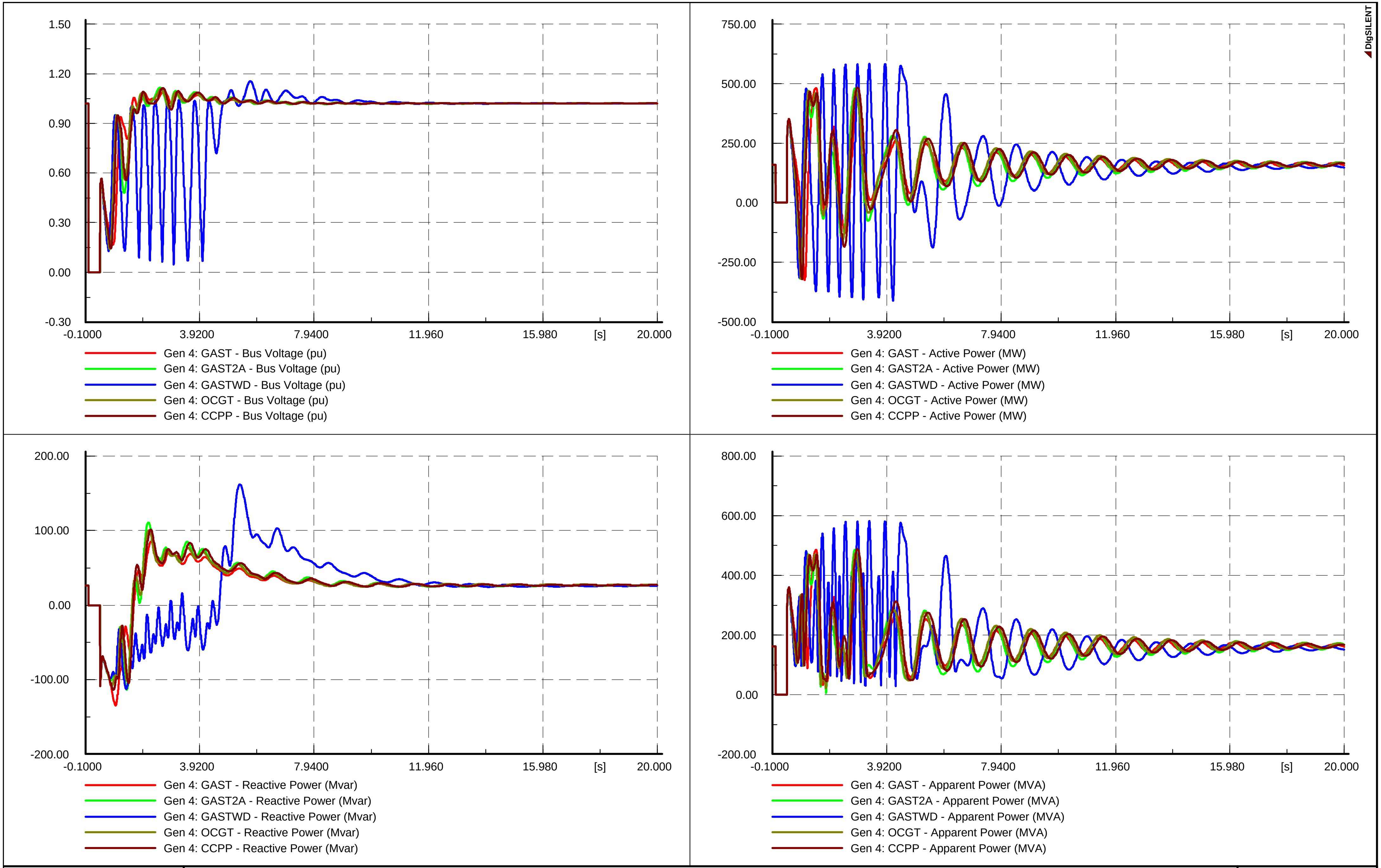
Critical clearing time for Generator 4 (3 Phase fault on terminals, 400 ms)

External grid with high acceleration time constant

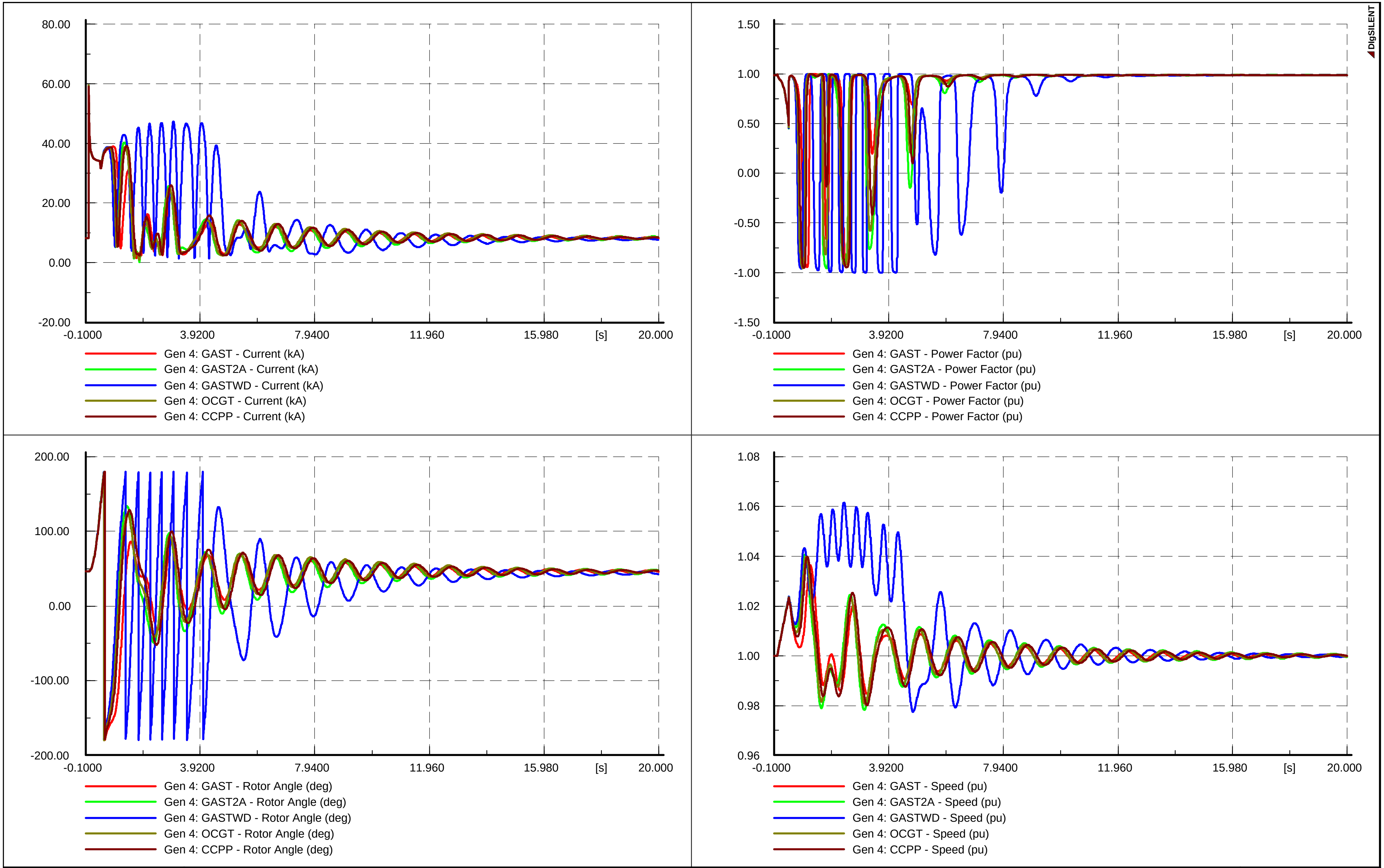
Results 5.5.7.d

Date: 25/04/2011

Annex: /2



20316976	Development and Adaptation of Dynamic Models	Results 5.5.7.e	Date: 25/04/2011
	Critical clearing time for Generator 4 (3 Phase fault on terminals, 410 ms)	External grid with high acceleration time constant	Annex: /1



20316976

Development and Adaptation of Dynamic Models

Results 5.5.7.f

Date: 25/04/2011

Critical clearing time for Generator 4 (3 Phase fault on terminals, 410 ms)

External grid with high acceleration time constant

Annex: /2

5.5.8 THREE-PHASE FAULT ON GAS TURBINE TERMINALS (ACCELERATION CONTROL)

For this disturbance, a three-phase fault on the gas turbine generator (Gen 4) terminals was modelled. Two faults with different durations were modelled. The first fault was modelled with duration of 200 ms and the second fault was modelled with duration of 300 ms.

Expected outcome:

The purpose of this simulation was to model the acceleration control loop of the OCGT model. If the turbine is experiencing excessive acceleration, this control loop should become the prominent of the three control loops. (Refer to Section 3.9.4 for a description of the three major control loops). The final purpose of this simulation is to show the additional feature of the new gas turbine model, as the existing models do not include this feature.

Fault with duration of 200 ms (Result 5.5.8.a)

The graphs show the three major control loops of the model (speed controller, acceleration control and temperature control). For this disturbance, the speed controller is the dominant controller. The graphs show that only the speed controller and the acceleration control is active. The temperature controller is not active, as the gas turbine temperature is within the set limits (i.e. not operated at full load). The low value gate output for this disturbance is only selecting the speed controller for its output, as this is the lowest value of the three control loop outputs.

During this disturbance, the machine speed will increase (a positive value for acceleration), as the mechanical power input from the turbine to the generator is larger than the active power delivered by the generator, and this causes acceleration of the machine. Therefore, both the speed controller and the acceleration control loops will be active. The speed controller is active to limit the speed of the machine, and the acceleration control is active to limit the acceleration of the machine. In this case, the speed controller is the dominant control loop (for a brief period only) and will therefore control the output of the turbine.

Fault with duration of 300 ms (Result 5.5.8.b)

The graphs show the three major control loops of the model (speed controller, acceleration control and temperature control). For this disturbance, the graphs show that the speed controller is initially the dominant control loop of the model (the low value gate uses the loop with the lowest value). Due to the longer duration of the disturbance, the machine is experiencing acceleration for a longer period. In this case, the acceleration control loop becomes the dominant control loop. The acceleration control loop is the dominant control loop for a short period (up to approximately 4 seconds after the disturbance), where after the speed controller loop takes over as the controlling loop again.

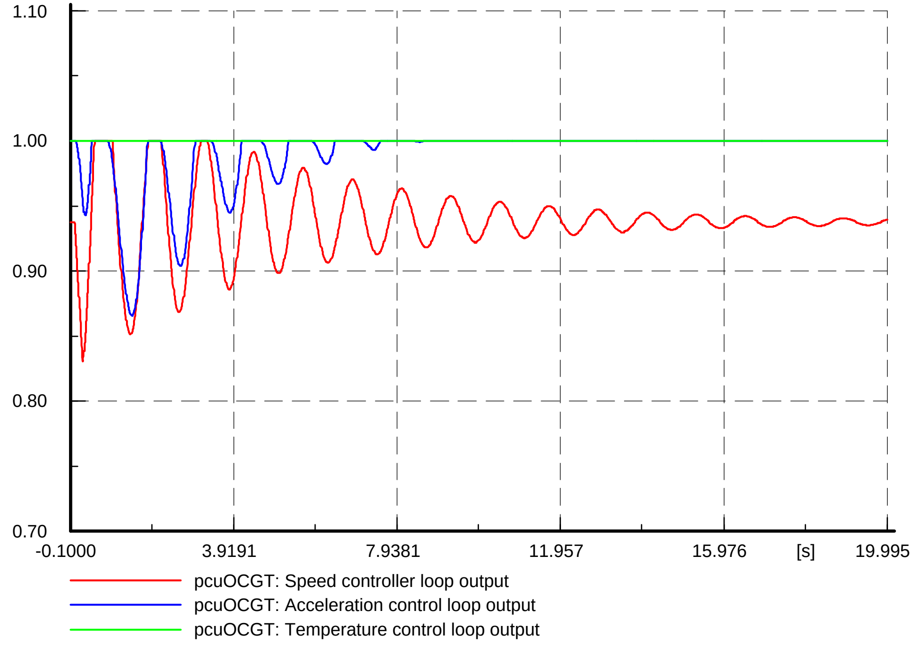
By comparing the results for the two disturbances, it can be seen that the longer disturbance, causes the machine to reach higher speeds. The longer disturbance also causes the machine to experience longer periods of acceleration. Due to this, the acceleration control loop is the dominant control loop for a period.

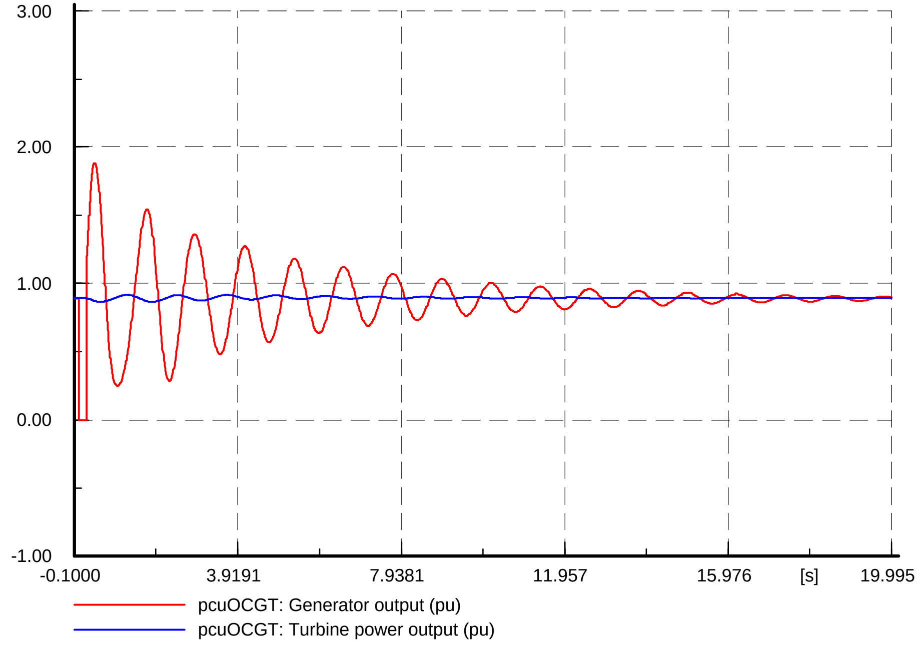
Conclusion:

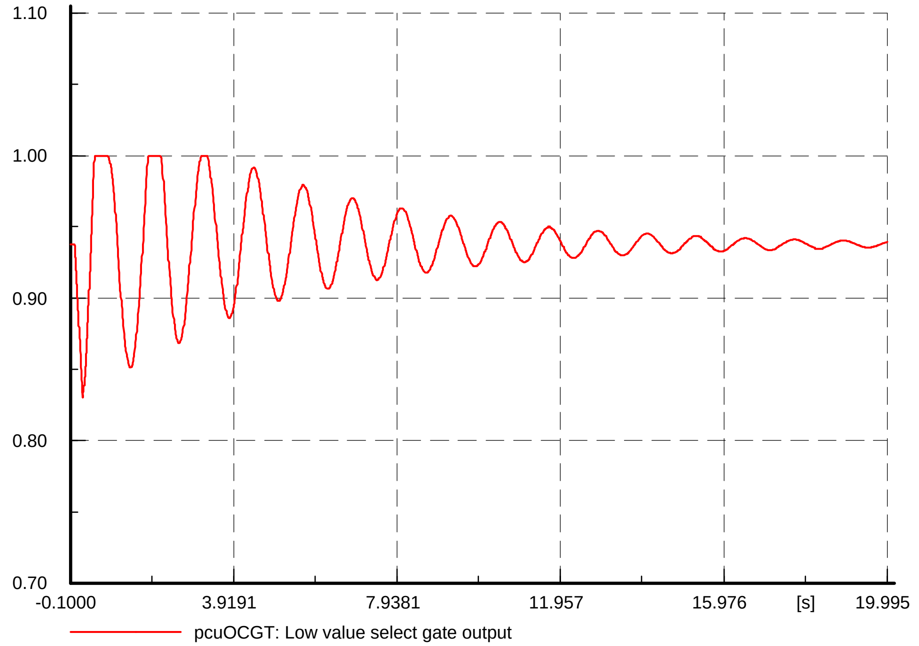
The acceleration control loop is the dominant control loop after disturbances that cause extended periods of acceleration of the machine.

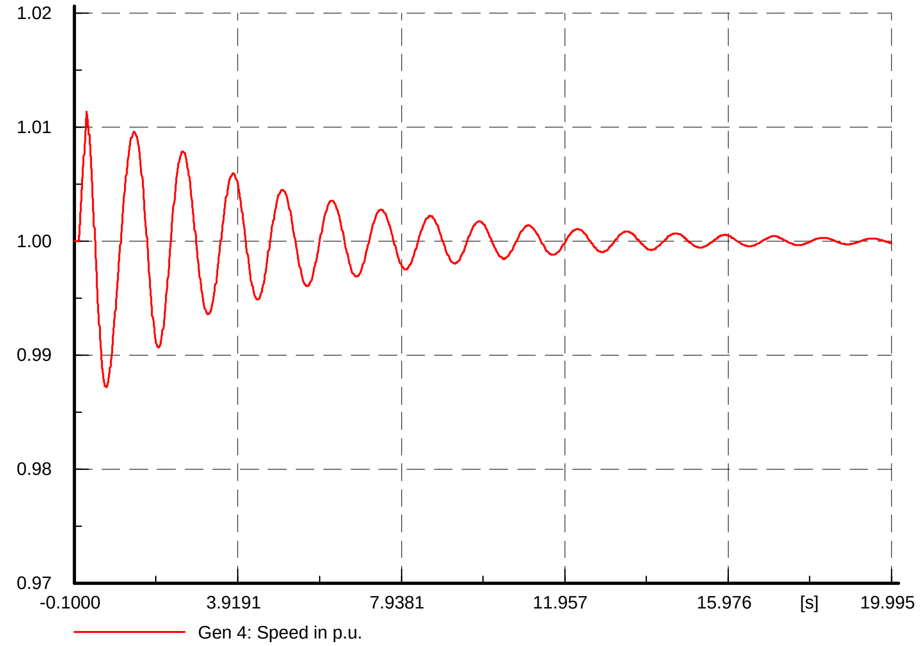
During the fault period, the gas turbine still delivered active power, although the generator did not delivered any active power. During this period the machine speeds up (i.e. acceleration) and therefore the both the speed control and acceleration control loops were active. For the shorter disturbance, the speed control loop was the dominant loop, while for the longer disturbance the dominant loop was the acceleration control loop (only for a short period). By comparing the two sets of graphs for the two disturbances, it can be seen that the turbine output power reached lower values during the period where the acceleration control loop was dominant, compared to the results for the short duration disturbance. This shows that the acceleration control loop worked and controlled the acceleration of the machine.

The simulation shows the additional feature of the new gas turbine model (OCGT), as the existing models do not include this feature.

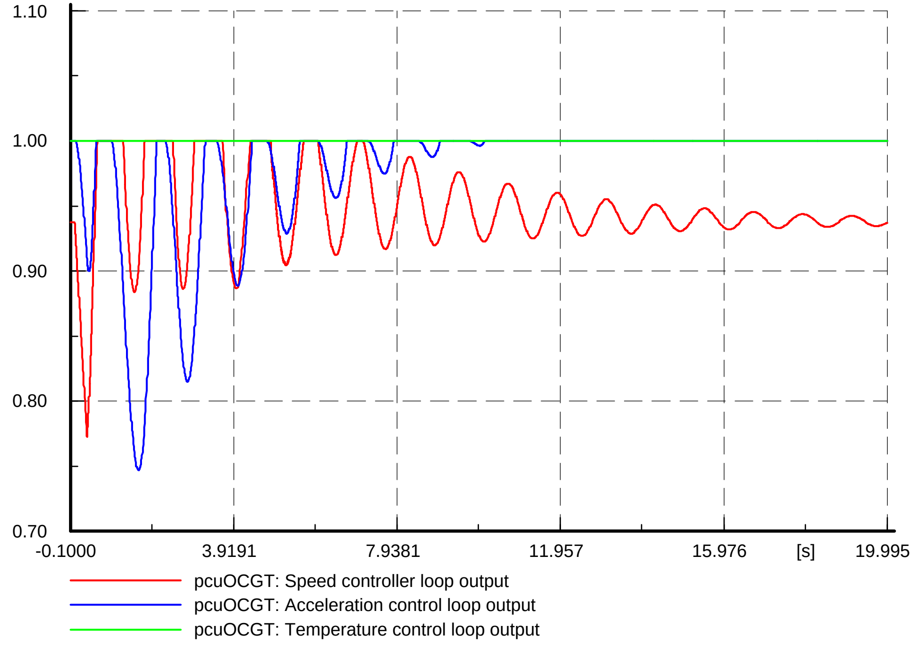


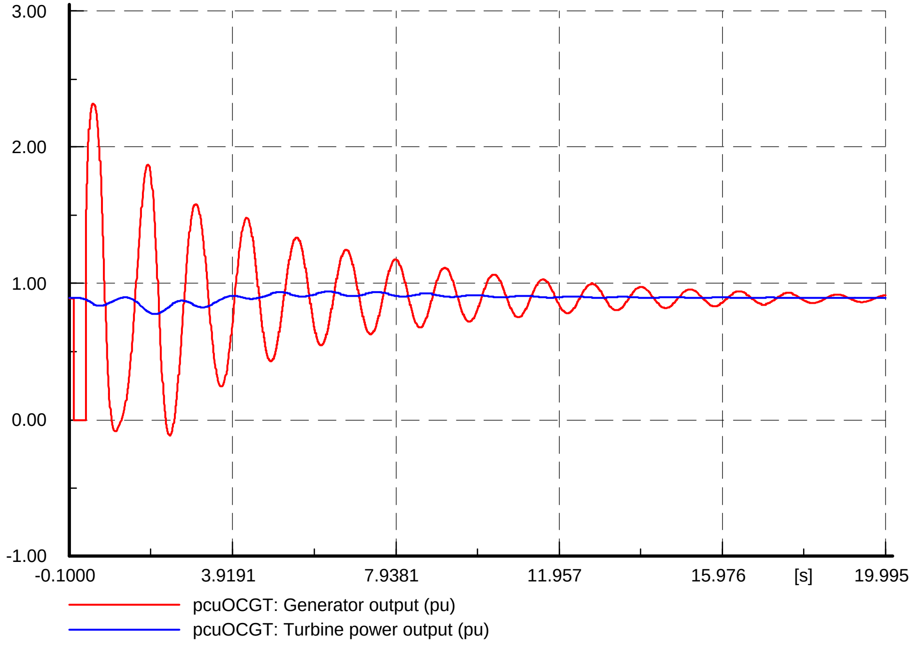


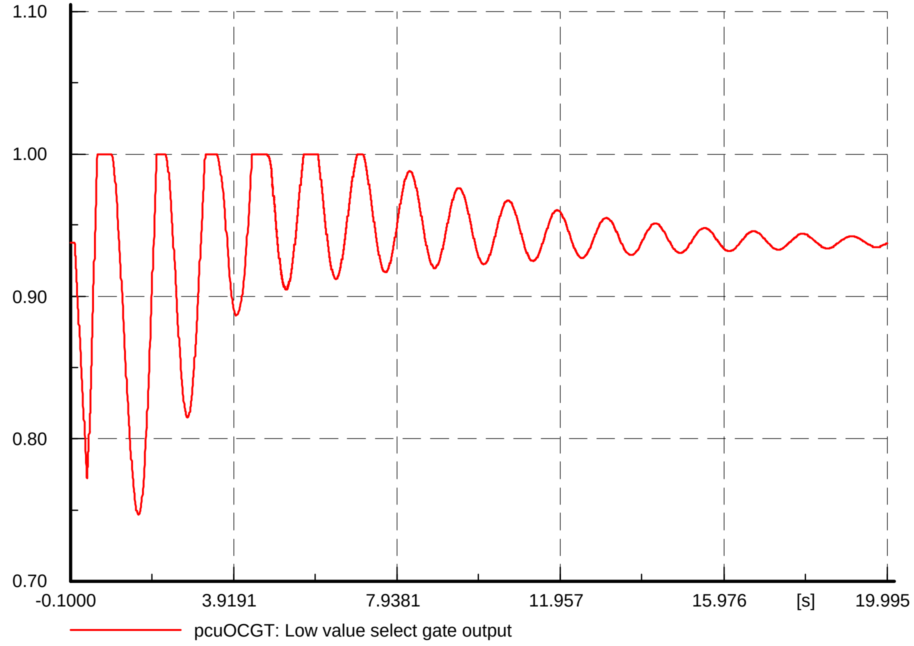


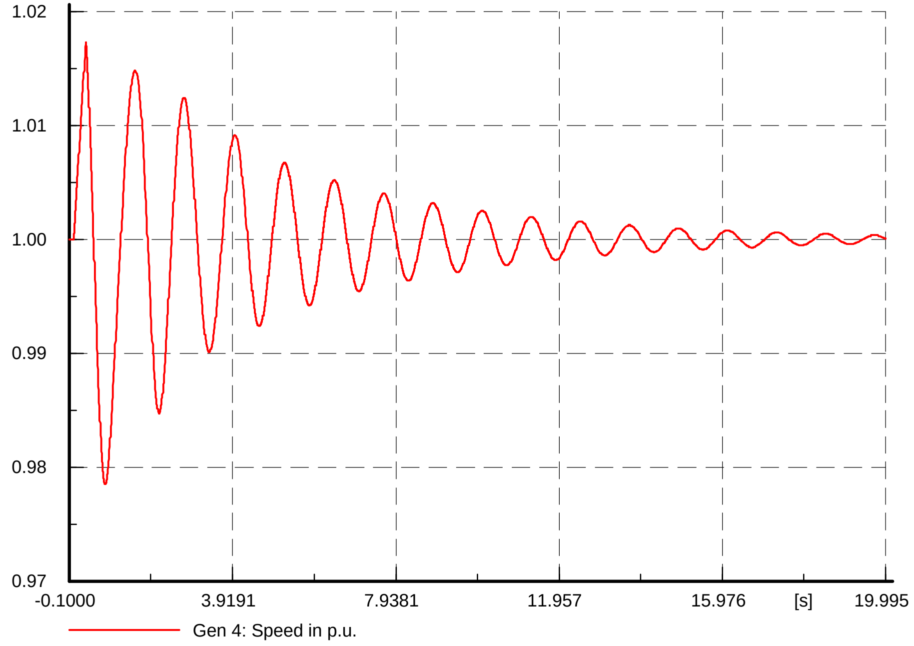


20316976	Development and Adaptation of Dynamic Models Major control loops	3 Phase fault on Generator 4 terminals (duration 200 ms)	Results 5.5.8.a	Date: 25/04/2011 Annex: /1









20316976

Development and Adaptation of Dynamic Models

Results 5.5.8.b

Date: 25/04/2011

Major control loops

3 Phase fault on Generator 4 terminals (duration 300 ms)

Annex: /1

5.5.9 SUPPLEMENTAL FIRING

Supplemental firing is used to increase the total power output of the combined cycle power plant. By switching off supplemental firing, the total power output of the plant is decreased. In order to model the increase in power output, the first simulation modelled a change in supplemental firing from 0.0 to 0.2 pu. The second simulation modelled a change from 0.2 to 0.0 pu in supplemental firing. The advantage of supplemental firing is that additional heat is added to the steam turbine without increasing the output from the gas turbine. In plants where various gases are available, but not suitable for use in the gas turbine, it can be used for supplemental firing.

Expected outcome:

Simulation 1: An increase in total power output.
Simulation 2: A decrease in total power output.

Results (Result 5.5.9.a)

The results show the response of the gas turbine and the steam turbine power output to a step input of 0.0 to 0.2 pu for the supplemental firing. Due to the supplemental firing, the total output of the combined cycle power plant (CCPP) plant model increased from 160 MW to approximately 168 MW. Although the steam turbine output increased, the gas turbine output decreased. This change in the gas turbine output is due to the disturbance. The gas turbine experienced a small increase in speed. For a short moment after the change in supplemental firing, the gas turbine and steam turbine still delivered the same output power. However, as the steam turbine output power increased the gas turbine output decreased. This caused the gas turbine to experience an increase in speed. The result is that the speed controller reduced the gas turbine output, to reduce the speed. Due the long time constant of the boiler (drum), the response of the plant takes several minutes (about 20 minutes) to settle at the new value.

Conclusion:

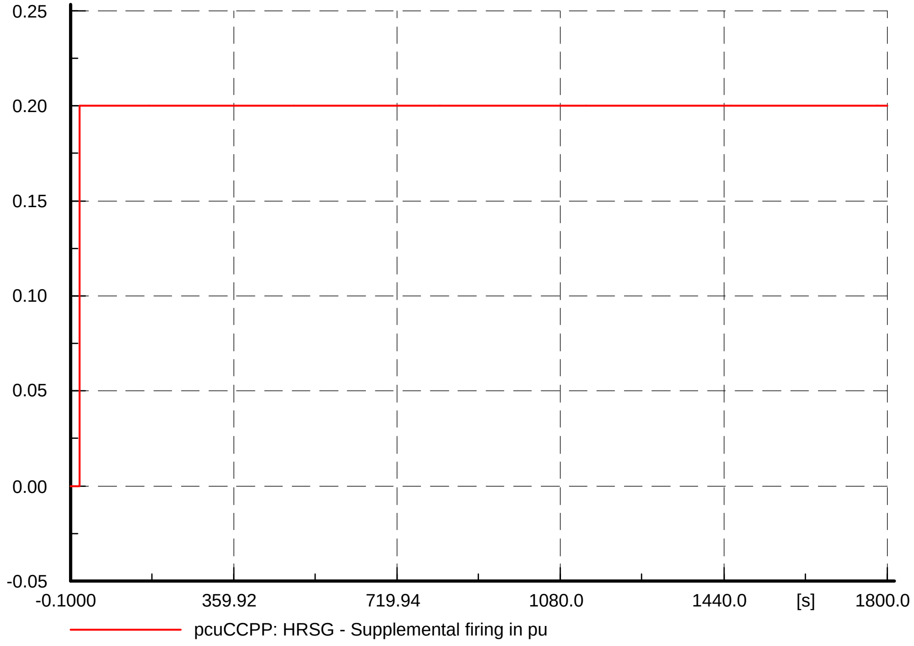
The supplemental firing, when applied, increase the total power output of the combined cycle power plant. The simulation shows the additional feature of the new combined cycle power plant model (CCPP).

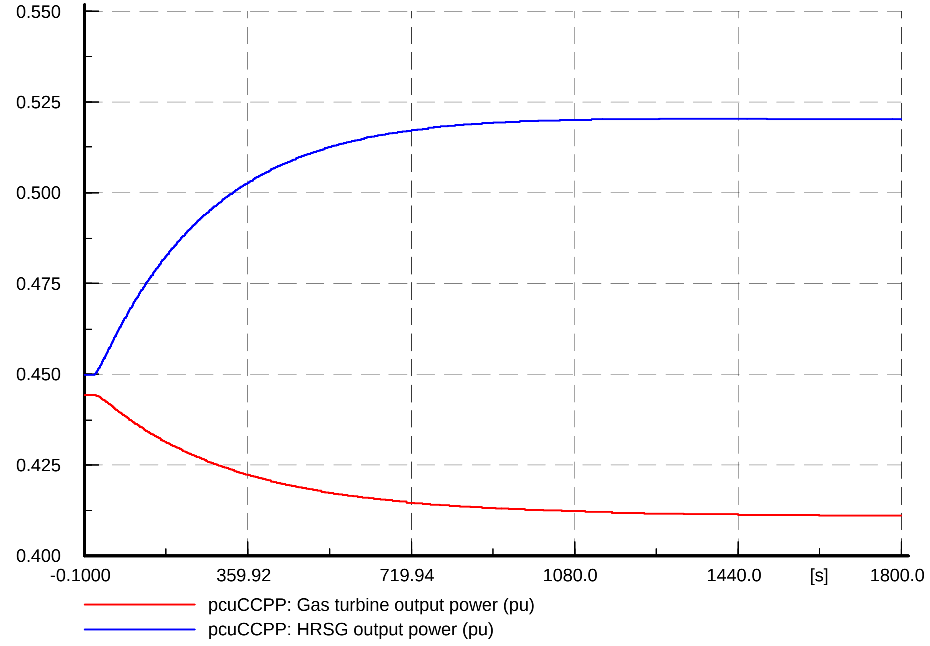
Results (Result 5.5.9.b)

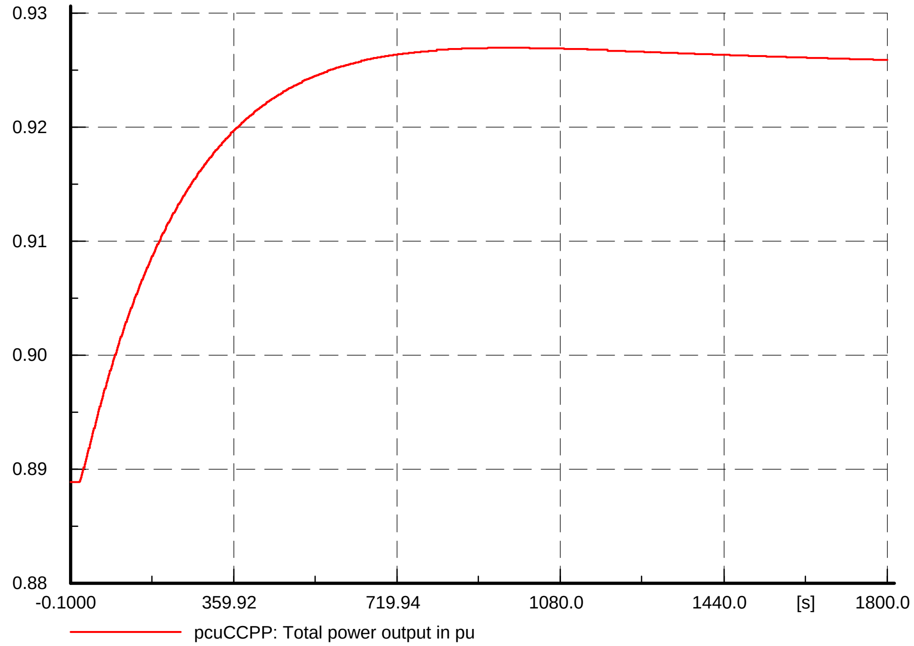
The result of switching off supplemental firing is that the total output power of the combined cycle power plant decreased from 160 MW to approximately 152 MW. However, the steam turbine output decreased while the gas turbine output increased. As with the change in supplemental firing from 0.0 to 0.2 pu, the opposite occurred for the change from 0.2 to 0.0 pu. The gas turbine experienced a small decrease in speed. As a result, the gas turbine speed controller increased the gas turbine output to increase it speed.

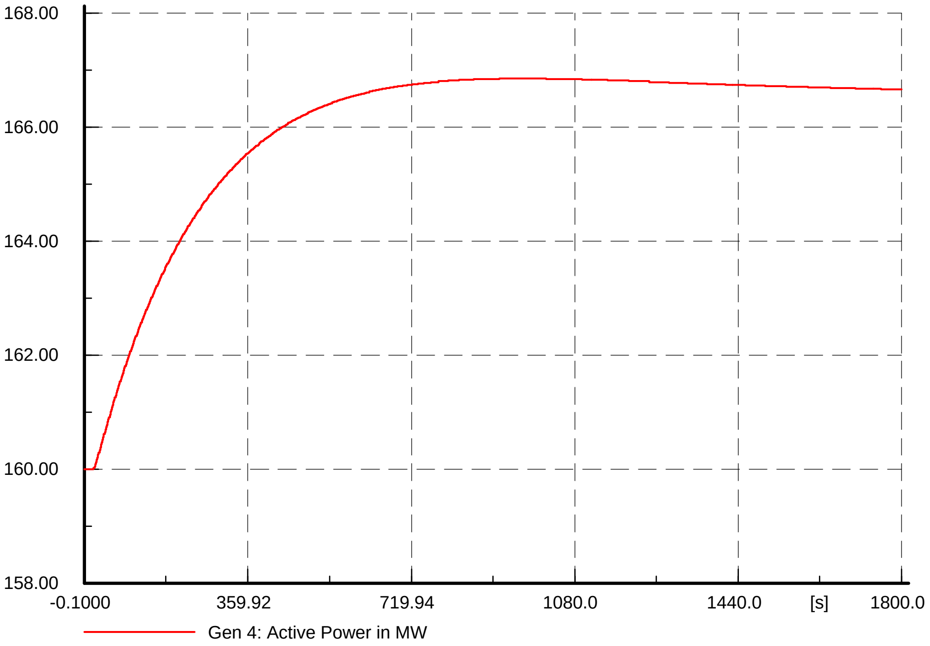
Conclusion:

The total power output of the combined cycle power plant decreases when the supplemental firing is switched off (after being switched on). The simulation shows the additional feature of the new combined cycle power plant model (CCPP).

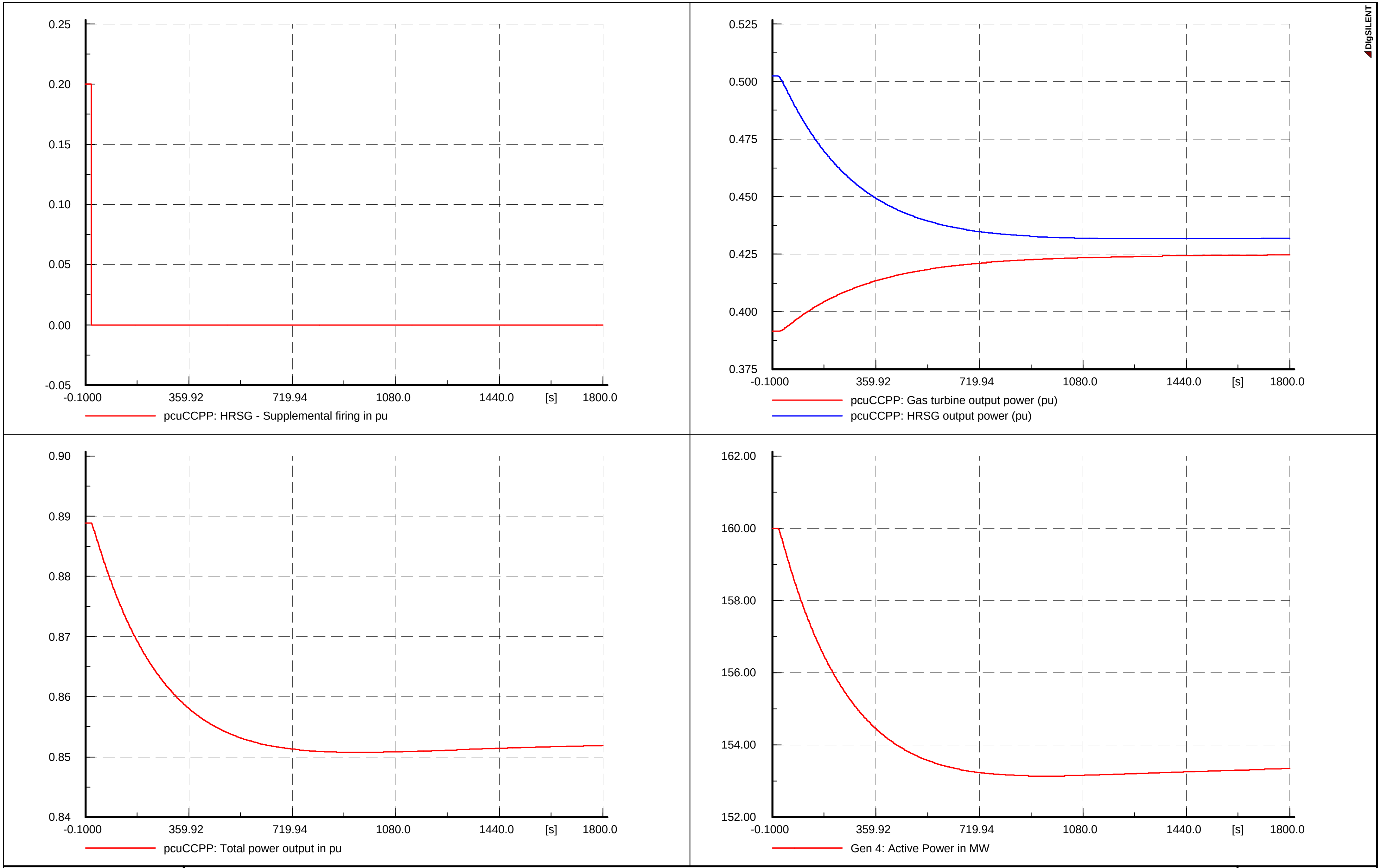








20316976	Development and Adaptation of Dynamic Models Change in Qs - Supplemental firing	Results 5.5.9.a Qs changed from 0 to 0.2	Date: 25/04/2011 Annex: /1



20316976	Development and Adaptation of Dynamic Models	Results 5.5.9.b	Date: 25/04/2011
	Change in Qs - Supplemental firing	Qs changed from 0.2 to 0	Annex: /1

5.5.10 BYPASS VALVE

The Bypass valve is used to extract steam from the heat recovery steam generator to be used elsewhere.

For this disturbance, a change in the control parameters of the combined cycle power plant model was modelled. For this simulation, the change from zero opening to 0.1 pu opening was modelled.

Expected outcome:

A decrease in the total power output of the combined cycle power plant, after several minutes as the time constant for the boiler drum is relative long (i.e. 300 s).

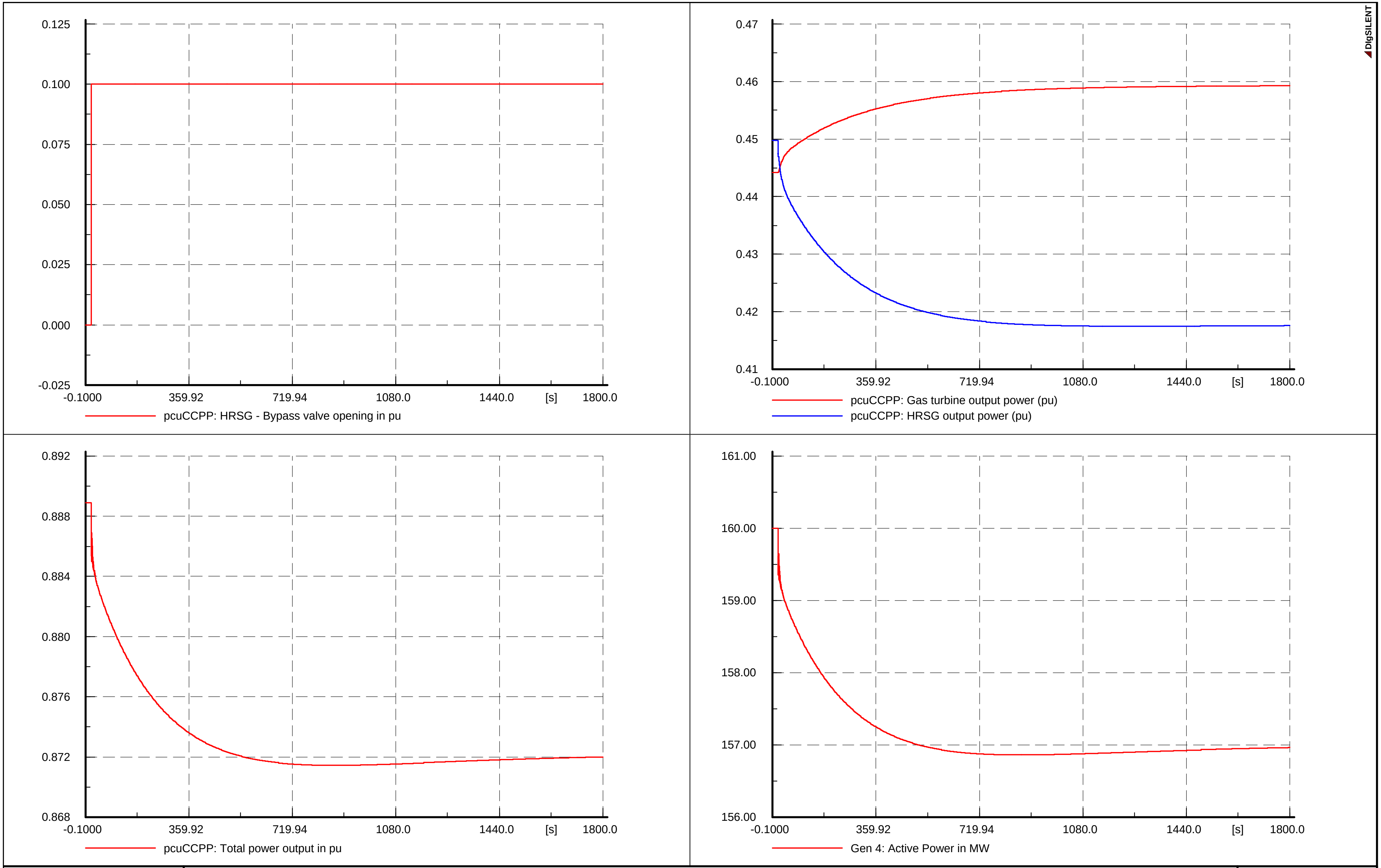
Results (Result 5.5.10)

The opening of the Bypass valve (from completely closed to 0.1 pu open) caused the total output power of the combined cycle power plant to decrease from 160 MW to approximately 156.5 MW. Although the steam turbine output decreased, the gas turbine power increased. Due to the change (disturbance), the gas turbine experienced a decrease in speed. The gas turbine speed controller increased the gas turbine output power in order to increase the turbine speed.

The total output power of the combined cycle power plant takes several minutes to settle at the new value due the long time constants of the boiler drum.

Conclusion:

This simulation shows that if the bypass valve is opened, the total power output of the combined cycle power plant model is reduced. The simulation shows the additional feature of the new combined cycle power plant model (CCPP).



20316976	Development and Adaptation of Dynamic Models Change in Bv - Bypass Valve	Results 5.5.10 Bv changed from 0 to 0.1	Date: 25/04/2011 Annex: /1

5.6 CONCLUSION

The aim of this dissertation was to adapt and test the gas turbine (OCGT) and combined cycle power plant (CCPP) models in the power system simulation software under different circumstances and to evaluate their performance against existing models. The new models were adapted as new gas turbine and combined cycle power plant equipment with new or alternative features are developed and need to be represented in power system simulation software. The models have to keep up with the development of new equipment in order to model the new equipment.

The new models (OCGT and CCPP) were tested against the existing models to show that the new models are correct, or that their responses are identical to the existing models, for the same configurations and settings. As far as it was possible, the models were set up identically to prove that all the models' responses are the same for the same disturbances in the same network with the same configuration. It is concluded that the OCGT and CCPP models represent the gas turbine and the combined cycle power plant adequately and compare satisfactorily with proven existing models

Additional features of the new models were investigated separately to show the differences between the models, as this is the main reason why new models are developed and adapted to be used in the power system simulation software.

It was demonstrated that the different gas turbine models' responses are almost identical for certain disturbances for the first few seconds, where after the models develop small variations between each other. These variations are small enough to ignore. This is true for power systems where the grid supplying the network under investigation has a significant high inertia (or acceleration time constant). Grids with high acceleration time constants tend to be more stable for disturbances in the network, and the network will respond slower to disturbances than networks with smaller acceleration constants.

It was shown that networks with smaller acceleration time constants responds faster to disturbances in the network, but will not maintain nominal speed (or frequency) for certain types of disturbances in the network. There are disturbances that cause the various models to respond substantially different from each other once the disturbance has passed and the network has settled at a new steady state operating point. The main reason for this is the non-zero droop settings of the generator governors. Droop settings will cause the speed of the generators to change with a change in the loading of the generator.

From the simulations, it was shown that the new gas turbine model (OCGT) does not replace the existing gas turbine models, but complements them and has an added feature. The new combined cycle power plant model (CCPP) contains features that are not found in existing steam turbine models. This model was developed to model combined cycle power plant. The CCPP model is a complete new model for combined cycle power plant, as currently no model exists for combined cycle power plant. The simulations show the new features of combined cycle plant.

5.7 SUMMARY

In this chapter, the various gas turbine models were evaluated against each other by means of dynamic stability simulations. The power system simulation software has the ability to generate revisions of a network that made it easy to compare the various gas turbine models in the same network with the same disturbances.

It was shown that the new gas turbine model does not replace the existing gas turbine models, but complement them and has an added feature, namely acceleration control. It was shown, that the new combined cycle power plant model is a complete new model for combined cycle power plant, as currently no model for combined cycle power plant exists.

From the simulation results, it is clear that the various gas turbine models will respond differently to the same disturbance due the differences between the models.

- Chapter 6 -

6 CONCLUSIONS AND RECOMMENDATIONS

6.1 INTRODUCTION

This dissertation’s main aim was to adapt a gas turbine and combined cycle power plant dynamic model, developed by a Cigre Task Force [1], to be used in the power system simulation software, called DigSilent PowerFactory [3]. Due to the advantages in overall efficiency and lower emissions when compared to conventional coal fired power plants, combined cycle power plants have gained popularity and have become a significant portion in power generation across the world. In South Africa, the main power source is still coal fired power stations, but in recent times, gas turbine power plants were built. Due to changes in the world to minimise carbon-dioxide footprints, there is demand for cleaner methods of power generation.

Approximately two-thirds of the generation capacity in a combined cycle power plant is produced by the gas turbines. The other third is generated by the steam turbine, and the mere fact that the steam is available, the overall efficiency of the power plant is improved and the emissions per developed active power unit (in kW or MW) are decreased.

Gas turbines and their controls are significantly different from the controls of a conventional steam turbine plant. In particular, the maximum output power of the gas turbine is very dependent on the deviation of its operating frequency from the rated frequency (or speed of the gas turbine), and the ambient conditions in which the gas turbine operates.

In an effort to provide the industry with a single document and simulation model that summarises the unique characteristics, controls and protection of combined cycle power plants, Cigre Task Force 25 was formed [1]. This Task Force developed a combined cycle power plant simulation model, as no models existed in any power system simulation software.

6.2 FINDINGS AND DEDUCTIONS

The simulation results show that the adaption of the models, (OCGT and CCPP) in the power system simulation software, was successful.

The first aim was to show that the new gas turbine model (OCGT) does not replace the existing gas turbine models (i.e. GAST, GAST2A and GASTWD), but complements them with similar characteristics and an added feature namely acceleration control. It was shown that the various gas turbine models are settable to achieve similar results for various disturbances in a network. The existing gas turbine models (GAST, GAST2A and GASTWD) are widely accepted and applied in power system dynamic analysis software [2], [3], [8].

The second aim was to show that the combined cycle power plant model is a new model, containing features that are not found in existing steam turbine models, namely supplemental firing and a bypass facility to extract steam for the steam turbine. The complete model is a new model developed by Cigre to model combined cycle power plants [1].

It was shown that the different gas turbine models' responses are almost identical for certain disturbances for the first few seconds, where after the models develop small variations between each other. These variations are small enough to ignore. This is true for power systems where the grid supplying the network under investigation has a significant high inertia (or acceleration time constant). Grids with high acceleration time constants tend to be more stable for disturbances in the network, as the network will respond slower to disturbances than networks with smaller acceleration time constants.

This is important to realise, as the question will arise, why alternative models are required if the existing models are adequate for modelling gas turbine equipment. The new models were tested against the existing models to show that the new models are correct, or that their responses are identical to the existing models. Additional features of the new models were investigated separately to show the differences between the models, as this is the main reason why new models are developed and adapted to be used in the simulation software.

The new models were also developed as new gas turbine and combined cycle power plant equipment with new or alternative features are developed. The models have to keep up with the development of new equipment in order to model the new equipment accurately.

6.3 RECOMMENDATIONS

The gas turbine model (OCGT) and the combined cycle power plant model (CCPP) can be used for power system dynamic analysis.

The OCGT model combines all the characteristics of a gas turbine in one model. The characteristics include PID speed governor control, droop or isochronous governor mode, temperature control, fuel flow control, the gas turbine dynamics and acceleration control. The existing models only include some of the characteristics of the OCGT model. Therefore, if more detail studies are required the OCGT model is the recommended model to use for power system studies.

The CCPP model contains all the characteristics of the OCGT model and includes the heat recovery steam generator (HRSG) equipment. No current model in power system simulation software exists that includes HRSG equipment. Therefore, any studies that include HRSG equipment (or if combined cycle power plant is studied), the CCPP model is required.

Actual manufacturer data must be obtained to model specific gas turbine and combined cycle power plants, when studying specific cases.

6.4 FIELDS FOR FURTHER STUDY

As the adaptation of the model for the various types of combined cycle power plants (e.g. Multi-shaft) was outside the scope of this dissertation, for further studies the CCPP model can be adapted to model these types of power plants.

Actual measurements of the response of the gas turbine and combined cycle power plant could be obtained, and modelled to confirm the accuracy of the models. The actual implemented settings for the controllers are required to set up the models correctly. This will help to represent the actual machine response accurately.

6.5 CONCLUSION

The purpose of this dissertation was the adaptation of the Cigre models for open cycle gas turbines (OCGT) and combined cycle power plants (CCPP). This was achieved successfully, and the models are available for dynamic studies of gas turbines and combined cycle power plants.

The open cycle gas turbine model (OCGT) contains all the features of the existing gas turbine models (GAST, GAST2A and GASTWD). The OCGT model does not replace the existing models, but complement them, with an added feature, namely acceleration control. The OCGT model is settable to achieve the same results as the existing model, but additional features can be modelled.

The combined cycle power plant model (CCPP) was developed to model combined cycle power plant, as currently no model exists for these types of plant. Additional features of the steam turbine part of the combined plant include supplemental firing and a bypass facility to extract steam from the steam turbine boiler.

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Appendixes

8 APPENDIX A

8.1 SIMULATION NETWORK DATA

8.1.1 GENERATOR DATA [5], [6] and [7]

The following data is typical data obtained from [5], [6] and [7]. To simplify matters the same data was assumed for the various generators with the exception of a few parameters The typical value range was obtained from [5] and [6] and are for Thermal Units [5] / Solid Rotor [6] generators. Thermal units are generally the round rotor type generator:

Generator Parameters	**External Grid**	**Generator**				**Typical value range [5]**	**Typical value [6]**
		Gen 1	**Gen 2**	**Gen 3**	**Gen 4**		
Nominal Apparent Power Sgn (MVA)	Swing Bus	250	200	150	200	-	-
Nominal Voltage (kV)	220	16.5	18.0	13.8	11.0	-	-
Rated Power Factor (pu)	-	0.85	0.90	0.85	0.90	-	-
Inertia Constant H (s), Based on Sgn	Infinite (1)	4.0	3.5	3.0	6.2	-	-
Synchronous Reactance Xd (pu)	-	2.00	1.90	1.65	1.90	1.0 – 2.3	1.90
Synchronous Reactance Xq (pu)	-	1.90	1.85	1.55	1.85	1.0 – 2.3	1.85
Transient Reactance X'd (pu)	-	0.28	0.25	0.20	0.25	0.15 – 0.4	0.25
Transient Reactance X'q (pu)	-	0.60	0.50	0.45	0.50	0.3 – 1.0	0.50
Sub-transient Reactance X"d (pu)	-	0.20	0.18	0.16	0.18	0.12 – 0.25	0.20
Sub-transient Reactance X"q (pu)	-	0.20	0.18	0.16	0.18	0.12 – 0.25	0.20
Leakage Reactance X_l (pu)	-	0.15	0.15	0.15	0.15	0.1 – 0.2	-
Leakage Resistance R_l (pu)	-	0.008	0.0072	0.0064	0.0072	-	-
Transient Time Constant T'd0 (s)	-	8.00	6.50	6.00	6.50	3.0 – 10.0	-
Transient Time Constant T'd (s)	-	-	-	-	-	-	0.55
Transient Time Constant T'q0 (s)	-	1.50	1.00	0.90	1.00	0.5 – 2.0	-
Sub-transient Time Constant T"d0 (s)	-	0.05	0.05	0.04	0.05	0.02 – 0.05	-
Sub-transient Time Constant T"d (s)	-	-	-	-	-	-	0.02
Sub-transient Time Constant T"q0 (s)	-	0.05	0.05	0.04	0.05	0.02 – 0.05	-
Main Flux Saturation Parameter 1.0	-	0.10	0.10	0.10	0.10	-	-
Main Flux Saturation Parameter 1.2	-	0.30	0.30	0.25	0.30	-	-
Zero Sequence Reactance X_0 (pu)	-	0.15	0.15	0.15	0.15	-	-
Zero Sequence Resistance R_0 (pu)	-	0.006	0.006	0.006	0.006	-	-
Negative Sequence Reactance X_2 (pu)	-	0.20	0.18	0.16	0.18	-	-
Negative Sequence Resistance R_2 (pu)	-	0.008	0.007	0.006	0.007	-	-
Minimum Reactive Power Limit (Mvar)	-	-80	-60	-45	-60	-	-
Maximum Reactive Power Limit (Mvar)	-	130	105	80	85	-	-
X/R Ratio	-	25	25	25	25	-	-
Generator Neutral Earthed	-	Solid	Solid	Solid	Solid	-	-
Short Circuit Current $I''k_{MAX}$ (kA)	10	-	-	-	-	-	-
X/R and X_0/R_0 Ratio	10	-	-	-	-	-	-
Z_2/Z_1 and X_0/X_1 Ratio	1.0	-	-	-	-	-	-
Voltage Controlled	Yes	Yes	Yes	Yes	Yes	-	-
Controlled Bus Voltage (pu)	1.025	1.02	1.02	1.015	1.02	-	-
Active Power Setting (MW)	Swing Bus	180	140	100	160	-	-

Note:

1. The inertia constant was changed to a relatively low value for the low inertia system studies. A value of 5 seconds was used.

8.1.2 TRANSFORMER DATA

Typical data was used for the transformers:

Transformer Data	Transformer Tx 1	Transformer Tx 2	Transformer Tx 3	Transformer Tx 4
Nominal Rating (MVA)	250	200	150	200
Nominal Voltage HV (kV)	220	220	220	220
Nominal Voltage LV (kV)	16.5	18.0	13.8	11.0
Impedance (%)	12	11	10	12
Vector group	YNd1	YNd1	YNd1	YNd1
HV Neutral Earth	Solid	Solid	Solid	Solid
Tap-changer Range (%)	± 5	± 5	± 5	± 5
Tap-changer Taps	9	9	9	9
Auto Tap-changer	Yes	Yes	Yes	Yes
HV Voltage Set-point (pu)	1.02	1.02	1.02	1.02
X/R Ratio	35	30	25	30

8.1.3 LINE DATA

The actual data for Zebra conductor was used for the line data. Digsilent PowerFactory calculates the line impedance based on the tower structure, the type of conductor (both phase and earth) used:

Line Data	Line 1	Line 2	Line 3	Line 4	Line 5	Line 6	Line 7
Nominal Rating (kV)	220						
Conductor	Zebra						
Conductor / Phase	Single						
Current Rating (A)	648						
Positive Sequence Resistance R_1 (Ω/km)	0.0683						
Positive Sequence Reactance X_1 (Ω/km)	0.4237						
Zero Sequence Resistance R_0 (Ω/km)	0.4347						
Zero Sequence Reactance X_0 (Ω/km)	0.9793						
Length (km)	30	30	25	40	35	40	100

8.1.4 LOAD DATA

Load Data	Load A	Load B	Load C	Load D	Load E	Load F
Active Power (MW)	125	120	100	100	50	90
Power Factor – Lagging (pu)	0.95					
Voltage Dependence on P	1.6					
Voltage Dependence on Q	1.8					

8.1.5 VOLTAGE CONTROLLERS / EXCITATION CONTROLLER

The values for the parameters were assumed as follows. The ranges were obtained from [2]:

Voltage Controller Data	**Generator Gen 1 to Gen 4**	**Range**
Type	IEEE Type 1	-
Measurement Delay, Tr (s)	0.02	0 ≤ Tr < 0.5
Controller Gain, Ka (pu)	250	10 ≤ Ka < 500
Controller Time Constant, Ta (s)	0.03	0 ≤ Ta < 1
Exciter Constant, Ke (pu)	1.0	-1 ≤ Ke ≤ 1
Exciter Time Constant, Te (s)	0.2	0.01 ≤ Te < 1
Stabilisation Path Gain, Kf (pu)	0.05	0 ≤ Kf < 0.3
Stabilisation Path Time Constant, Tf (s)	1.5	0.01 ≤ Tf
Saturation Factor 1, E1 (pu)	3.9	0 ≤ E1
Saturation Factor 2, Se1 (pu)	0.1	0 ≤ Se(E2) < 1
Saturation Factor 3, E2 (pu)	5.2	E1 ≤ E2
Saturation Factor 4, Se2 (pu)	0.5	Se(E1) ≤ Se(E2)
Controller Output Minimum, Vrmin (pu)	-10	-10 < Vrmin < 0
Controller Output Maximum, Vrmax (pu)	10	0.5 < Vrmax < 10

8.1.6 PRIME MOVERS / SPEED GOVERNING CONTROLLERS (GEN. 1 TO 3)

The values for the parameters were assumed as follows. The ranges were obtained from [2]:

Prime Mover Data	**Generator Gen 1 to Gen 3**	**Range**
Type	IEEE Type G1	-
Controller Gain, K (pu)	20	5 ≤ K ≤ 30
Governor Time Constant, T1 (s)	0.2	0 ≤ T1 < 5
Governor Derivative Time Constant, T2 (s)	1.0	0 ≤ T2 < 10
Servo Time Constant, T3 (s)	0.5	0.01 < T3 ≤ 1
High Pressure Turbine Time Constant, T4 (s)	0.6	0 < T4 ≤ 1
Intermediate Pressure Turbine Time Constant, T5 (s)	0.5	0 ≤ T5 < 10
Medium Pressure Turbine Time Constant, T6 (s)	0.8	0 ≤ T6 < 10
Low Pressure Turbine Time Constant, T7 (s)	1.0	0 ≤ T7 < 10
High Pressure Turbine Factor, K1 (pu)	0.3	-2 ≤ K1 ≤ 1
High Pressure Turbine Factor, K2 (pu)	0.0	K2 = 0
Intermediate Pressure Turbine Factor, K3 (pu)	0.2	0 ≤ K3 < 0.5
Intermediate Pressure Turbine Factor, K4 (pu)	0.15	0 ≤ K4 < 0.5
Medium Pressure Turbine Factor, K5 (pu)	0.1	0 ≤ K5 < 0.35
Medium Pressure Turbine Factor, K6 (pu)	0.2	0 ≤ K6 < 0.55
Low Pressure Turbine Factor, K7 (pu)	0.1	0 ≤ K7 < 0.3
Low Pressure Turbine Factor, K8 (pu)	0.05	0 ≤ K8 < 0.3
Valve Opening Time, Uo (pu/s)	0.3	0.01 ≤ Uo ≤ 0.3
Valve Closing Time, Uc (pu/s)	-0.3	-0.3 ≤ Uc < 0
Maximum Gate Limit, Pmax (pu)	2.0	0.5 ≤ Pmax ≤ 2
Minimum Gate Limit, Pmin (pu)	0.0	0 ≤ Pmin < 0.5

8.1.7 PRIME MOVERS / SPEED GOVERNING CONTROLLERS (GEN. 4)

The values for the parameters were assumed as follows, with parameters that represent the same equipment being set the same for the various models: Typical values and the ranges as provided in the PSSE documentation were used [2].

Prime Mover Data	**Generator Gen 4**	**Range**
Type	**GAST**	**-**
Speed Droop	0.05	0 < Pmax < 0.1
Controller Time Constant, T1 (s)	0.5	0.01 < T1 < 0.5
Actuator Time Constant, T2 (s)	0.15	0.01 < T2 < 0.5
Compressor Time Constant, T3 (s)	3	0.01 < T3 < 5
Ambient Temperature Load Limit, AT (pu)	1.0	0 < AT ≤ 1
Turbine Factor, Kt (pu)	1.3	0 < Kt < 5
Frictional Losses, Dturb (pu)	0.01	0 < Dturb < 0.5
Turbine Rated Power, Pturb (0 = pturb = pgen) (MW)	0.0	
Controller Minimum Output, Vrmin (pu)	0.0	0 ≤ Vrmin < 1.0
Controller Maximum Output, Vrmax (pu)	1.0	0.5 < Vrmin < 1.2

Prime Mover Data	**Generator Gen 4**	**Range**
Type	**GAST2A**	**-**
Fuel System Prop. Characteristic, a (pu)	1.0	0.5 < a < 50
Turbine 1st Factor, af1 (pu)	700	500 < af1 < 1000
Turbine Characteristic Constant, af2 (pu)	-0.3	-1 < af2 < 1
Fuel System Time Constant, b (s)	0.05	0.01 < b < 2
Turbine 2nd Factor, bf1 (pu)	550	300 < bf1 < 700
Turbine Characteristic Torque, bf2 (pu)	1.3	0.9 < bf2 < 1.5
Fuel System I/L Factor (0/1), c (pu)	1.0	c = 0 of c = 1
Turbine Characteristic Speed, cf2 (pu)	0.5	0 ≤ cf2 ≤ 1
Fuel System Delay, Ecr (s)	0.01	0 ≤ Ecr < 0.5
Temperature Controller Delay, Etd (s)	0.1	0 ≤ Etd < 0.5
Turbine Factor, K3 (pu)	0.45	0.5 < K3 < 1
Radiation Shield Prop. Factor, K4 (pu)	0.8	0.5 < K4 < 1
Radiation Shield Integration Factor, K5 (pu)	0.2	0.05 < K5 < 0.5
Compressor Factor, K6 (pu)	0.25	0.1 < K6 < 0.5
Fuel System Feed Back Factor, Kf (pu)	0.0	0 ≤ Kf ≤ 1
VCE Upper Limit, Max (pu)	1.5	0.5 < Max < 1.8
VCE Lower Limit, Min (pu)	-0.15	-0.2 < Min < 0.1
Fuel Control Delay Time, T (s)	0.01	0 ≤ T ≤ 0.05
Radiation Shield Time Constant, T3 (s)	15	10 < T3 < 25
Thermocouple Time Constant, T4 (s)	2.5	1 < T3 < 5
Temperature Controller Gain, T5 (pu)	3.3	1 < T5 < 5
Temperature Control, Tc (grd F)	1002	Tc > Tr
Turbine Time Delay, Tcd (s)	0.25	0 ≤ Tcd < 0.5
Fuel System Delay, Tf (s)	0.4	0.05 < Tf < 0.8
Turbine Rated Power, Trate (MW)	180	0.8 x MBase ≤ Trate ≤ 1.05 x MBase
Rated Exhaust Temperature, Trate (grd F)	1001	700 < Trate < 1050
Temperature Controller Time Constant, Tt (s)	400	100 < Tt < 600
Speed Controller Gain, W (pu)	25	0 < W < 30
Speed Controller Derivative Time Constant, X (s)	0.5	0 ≤ X

Prime Mover Data	**Generator Gen 4**	**Range**
Type	**GAST2A**	**-**
Speed Controller Time Constant, Y (s)	0.4	0.01 < Y < 0.5
Speed Controller Isochronous / Droop (0/1), Z (pu)	1	Z = 0 or Z = 1

Prime Mover Data	**Generator Gen 4**	**Range**
Type	**GASTWD**	**-**
Valve Positioner Prop. Characteristic, a (pu)	1.0	0.5 < a < 50
Turbine 1st Factor, af1 (pu)	700	500 < af1 < 1000
Turbine Characteristic, Constant, af2 (pu)	-0.3	-1 < af2 < 1
Valve Positioner Time Constant, b (s)	0.05	0.01 < b < 2
Turbine 2nd Factor, bf1 (pu)	550	300 < bf1 < 700
Turbine Characteristic, Torque, bf2 (pu)	1.3	0.9 < bf2 < 1.5
Valve Positioner I/L Factor (0/1), c (pu)	1.0	c = 0 of c = 1
Turbine Characteristic, Speed, cf2 (pu)	0.5	0 ≤ cf2 ≤ 1
Combustor Delay Time, Ecr (s)	0.01	0 ≤ Ecr < 0.5
Turbine Exhaust Delay, Etd (s)	0.1	0 ≤ Etd < 0.5
Turbine Factor, K3 (pu)	0.45	0.5 < K3 < 1
Radiation Shield Proportional Factor, K4 (pu)	0.8	0.5 < K4 < 1
Radiation Shield Integral Factor, K5 (pu)	0.2	0.05 < K5 < 0.5
Compressor Factor (Min. Flow), K6 (pu)	0.25	0.1 < K6 < 0.5
Speed Controller Derivative Factor, Kd (pu)	0.0	0 ≤ Kd ≤ 20
Power Controller Droop, Kdroop (pu)	0.05	0 ≤ Kdroop ≤ 0.1
Fuel System Feed Back Factor, Kf (pu)	0.0	0 ≤ Kf ≤ 1
Speed Controller Integral Factor, Ki (1/s)	5.0	0 < Ki ≤ 10
Speed Controller Proportional Factor, Kp (pu)	10.0	0 ≤ Kp ≤ 20
VCE Upper Limit, Max (pu)	1.5	0.5 < Max < 1.8
VCE Lower Limit, Min (pu)	-0.15	-0.2 < Min < 0.1
Fuel Control Delay Time, T (s)	0.01	0 ≤ T ≤ 0.05
Radiation Shield Time Constant, T3 (s)	15	10 < T3 < 25
Thermocouple Time Constant, T4 (s)	2.5	1 < T4 < 5
Temp. Contr. Derivative Time Constant, T5 (pu)	3.3	1 < T5 < 5
Temperature Control Set point, Tc (grd F)	1002	Tc > Tr
Turbine Dynamics Delay Time, Tcd (s)	0.25	0 ≤ Tcd < 0.5
Power Transducer Delay Time, Td (s)	0.5	0 < Td
Fuel System Delay, Tf (s)	0.4	0.05 < Tf < 0.8
Rated Exhaust Temperature, Tr (grd F)	1001	700 < Tr < 1050
Turbine Rated Power, Trate (MW)	180	0.8 x MBase ≤ Trate ≤ 1.05 x MBase
Temperature Controller Time Constant, Tt (s)	400	100 < Tt < 600

Prime Mover Data	**Generator Gen 4**
Type	**OCGT**
Acceleration limit set-point, aset (pu)	0.01
Intentional dead-band, dbd (pu)	0.0
Intentional upper error limit, up_err (pu)	0.02
Intentional lower error limit, low_err (pu)	-0.02
Speed governor derivative gain, Kdg (pu)	0.0
Acceleration control integral gain, Kia (pu)	10.0
Speed governor integral gain, Kig (pu)	5.0
Temperature control integral gain, Kit (pu)	1.0
Integral gain for outer loop MW control, Kmwi (pu)	0.0
Proportional gain for outer loop MW control, Kmwp (pu)	0.0
Acceleration control proportional gain, Kpa (pu)	0.0
Speed governor proportional gain, Kpg (pu)	10.0
Temperature control proportional gain, Kpt (pu)	3.3
Turbine gain, Kt (pu)	1.3
Maximum fuel flow command, max_fuel (pu)	1.0
Minimum fuel flow command, min_fuel (pu)	0.15
Maximum limit on outer loop MW control loop, rfmax (pu)	1.0
Minimum limit on outer loop MW control loop, rfmin (pu)	0.0
Electrical power feedback droop, Rp (pu)	0.05
Governor feedback droop, Rv (pu)	0.05
Acceleration control differentiator time constant, Ta (s)	0.1
Speed governor derivative time constant, Tdg (s)	0.5
Heat transfer lag time constant, Tdgt (s)	15.0
Ambient temperature, Temp (°C)	26.0
Heat transfer lead time constant, Tngt (s)	10.0
Electrical power feedback time constant, Tp (pu)	5.0
Turbine transfer function denominator time constant 1, Ttd1 (s)	0.25
Turbine transfer function denominator time constant 2, Ttd2 (s)	0.0
Thermocouple time constant, Tthcp (s)	2.5
Turbine transfer function numerator time constant 1, Ttn1 (s)	0.0
Turbine transfer function numerator time constant 2, Ttn2 (s)	0.0
Fuel system time constant, Tv (s)	0.4
Maximum valve opening, Vmax (pu)	1.0
Minimum valve opening, Vmin (pu)	0.0
Full-speed no-load fuel flow, Wfo (pu)	0.25

Prime Mover Data	**Generator Gen 4**
Type	**CCPP**
Acceleration limit set-point, aset (pu)	0.01
Intentional dead band, dbd (pu)	0.0
Intentional lower error limit, low_err (pu)	-0.02
Intentional upper error limit, up_err (pu)	0.02
Speed governor derivative gain, Kdg (pu)	0.0
Acceleration control integral gain, Kia (pu)	10.0
Speed governor integral gain, Kig (pu)	5.0
Temperature control integral gain, Kit (pu)	1.0
Integral gain for outer loop MW control, Kmwi (pu)	0.0
Proportional gain for outer loop MW control, Kmwp (pu)	0.0
Acceleration control proportional gain, Kpa (pu)	0.0
Speed governor proportional gain, Kpg (pu)	25
Temperature control proportional gain, Kpt (pu)	3.3
Turbine gain, Kt (pu)	1.3
Maximum fuel flow command, max_fuel (pu)	1.0
Minimum fuel flow command, min_fuel (pu)	0.15
Maximum limit on outer loop MW control loop, rfmax (pu)	1.0
Minimum limit on outer loop MW control loop, rfmin (pu)	0.0
Electrical power feedback droop, Rp (pu)	0.05
Governor feedback droop, Rv (pu)	0.05
Acceleration control differentiator time constant, Ta (s)	0.1
Speed governor derivative time constant, Tdg (s)	0.5
Heat transfer lag time constant, Tdgt (s)	15.0
Ambient temperature, Temp (°C)	26.0
Heat transfer lead time constant, Tngt (s)	10.0
Electrical power feedback time constant, Tp (pu)	5.0
Turbine transfer function denominator time constant 1, Ttd1 (s)	0.25
Turbine transfer function denominator time constant 2, Ttd2 (s)	0.0
Thermocouple time constant, Tthcp (s)	2.5
Turbine transfer function numerator time constant 1, Ttn1 (s)	0.0
Turbine transfer function numerator time constant 2, Ttn2 (s)	0.0
Fuel system time constant, Tvgt (s)	0.4
Maximum valve opening, Vmax (pu)	1.0
Minimum valve opening, Vmin (pu)	0.0
Full-speed no-load fuel flow, Wfo (pu)	0.25
Ratio of HRSG output to GT output, Kst (pu)	0.5
HRSG Supplemental firing, Qs (pu)	0.0
HRSG Minimum steam reference, Pref (pu)	0.5
HRSG Drum time constant, Tdrum, (s)	300
HRSG Pressure loss due to flow friction, Km (pu)	0.1
HRSG Bypass valve opening, Bv (pu)	0.0
HRSG Governor proportional gain, Kp (pu)	10.0
HRSG Governor integral gain, Ki, (pu)	2.0
HRSG Actuator time constant, Tvst (s)	0.5
HRSG Turbine lag time constant, Tdst (s)	10.0
HRSG Turbine lead time constant, Tnst (s)	3.0

9 APPENDIX B

9.1 BUILDING BLOCK DESCRIPTION

This section gives short descriptions of the basic building blocks that make up a dynamic model.

9.1.1 CONSTANT

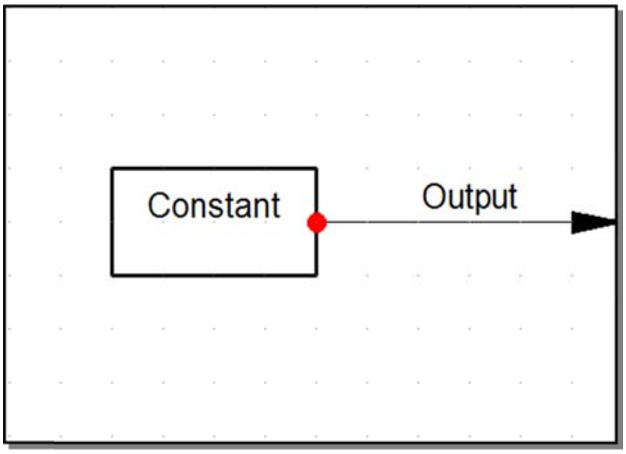

Figure 9.1: Constant block

The constant block is used to produce an output signal with a constant value. The value of the constant is settable by the user, or the actual value can be set inside the block.

9.1.2 SUMMATOR

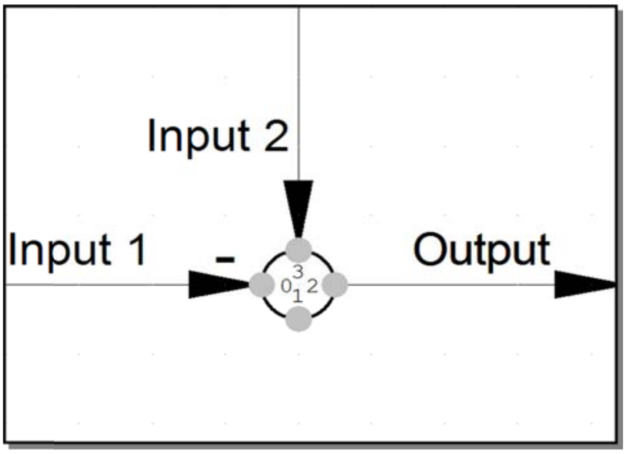

Figure 9.2: Summation block

The summation block is used to summate or subtract 2 or 3 inputs and produces the result on the output. The summation or subtraction is selected individually for each of the inputs. Any of the connection points can be used for the input and output signals.

9.1.3 MULTIPLIER

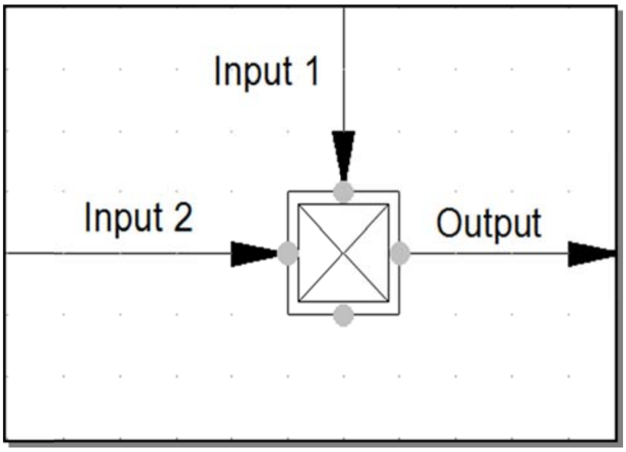

Figure 9.3: Multiplier block

The multiplier block is used to multiply 2 or 3 inputs and produces the result on the output. This block can also be used to quadrate the same value, when one input is connected to two input points of the block. Figure 3.4 shows the connection.

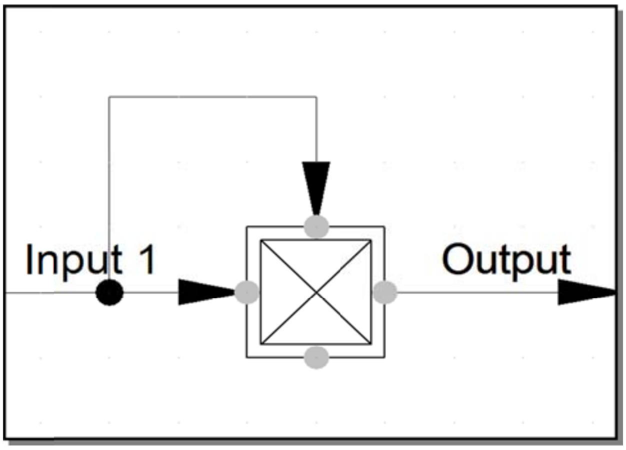

Figure 9.4: Quadrate block

This block can also be used to calculate the power of three of the input signal, when one input is connected to three input points of the block. Figure 3.5 shows the connection.

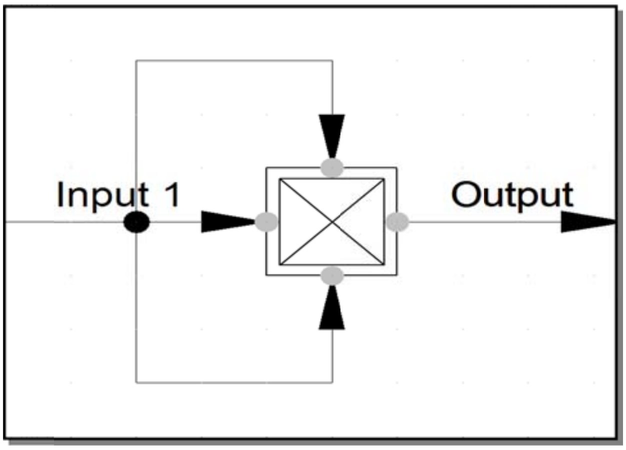

Figure 9.5: Power of three block

9.1.4 DIVIDER

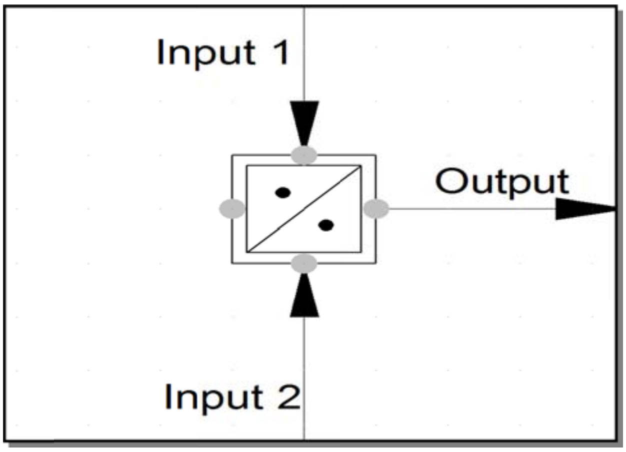

Figure 9.6: Divider block

The divider block is used to divide two inputs and produces the result on the output. Care must be taken as to the connection of the inputs, as the incorrect connection will produce the wrong result. The output is calculated as Input 1 / Input 2.

9.1.5 DEAD-BAND

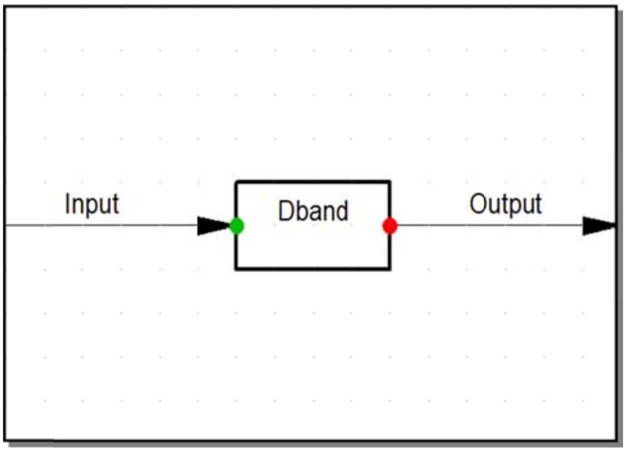

Figure 9.7: Dead-band block

The dead-band block is used to produce an output only if the input signal is greater than the dead-band value K, according to the following relationship:

- Output = if(abs(Input) > K, Input, 0)

  With abs: absolute

The output is equal to the input if the absolute value of the input is greater than the value K otherwise the output is zero.

9.1.6 INTEGRATORS

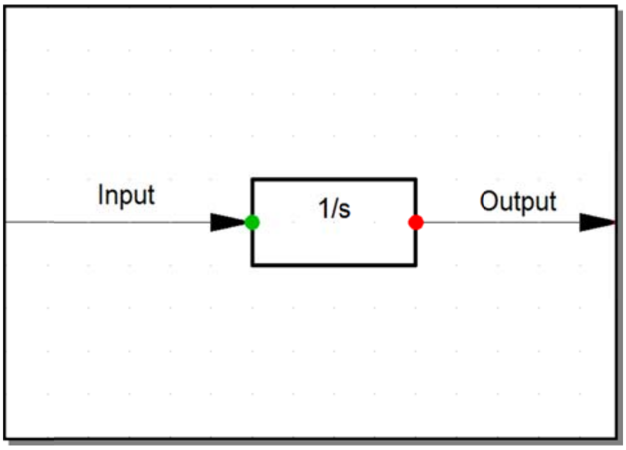

Figure 9.8: Integrator function block

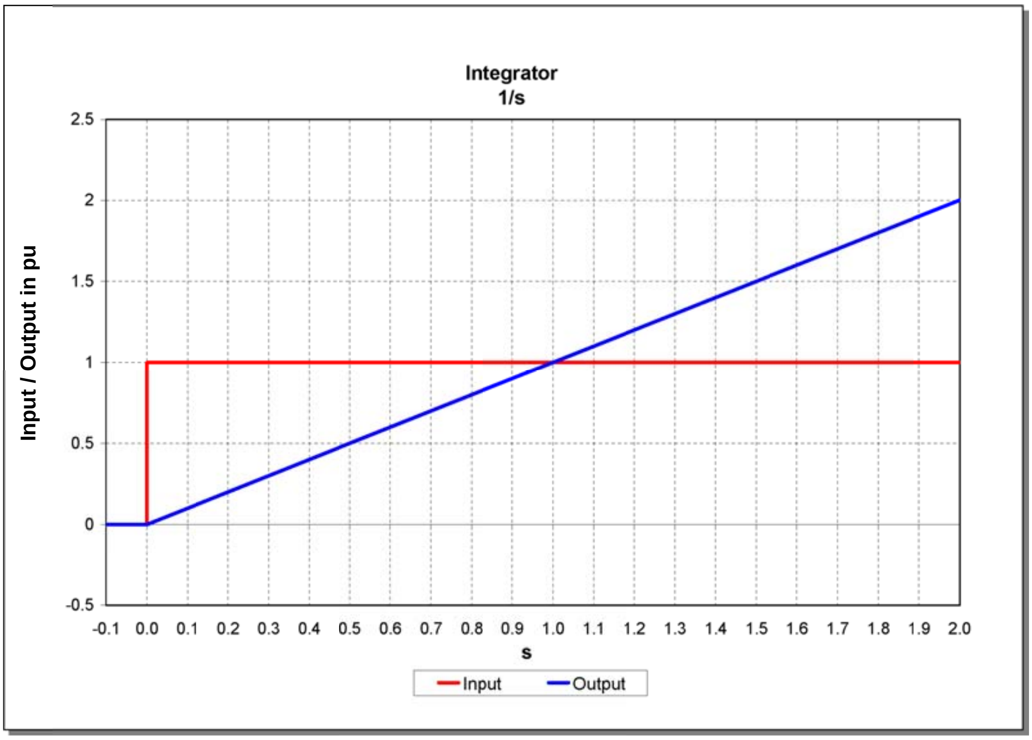

Figure 9.9: Integrator function block input / output relationship

The integrator function block integrates the input signal and produces the output. Integration is effectively the calculation of the area below a curve. From the graph it can be seen that at s = 1.0, the output (blue curve) equals 1. The area below the input (red curve) at s = 1 is calculated as 1 * 1 = 1. At s = 2 the area is 1 * 2 = 2 and that equals the output of 2.