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Dynamic Models for Turbine-Governors in Power System Studies

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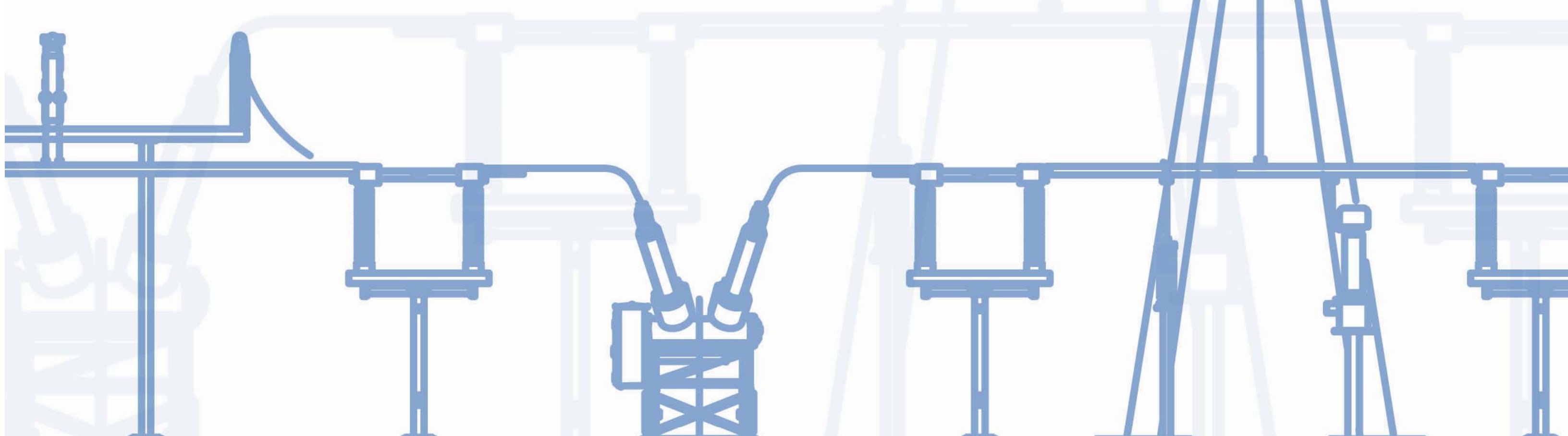
Power System Stability Subcommittee
Task Force on Turbine-Governor Modeling



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TASK FORCE ON TURBINE-GOVERNOR MODELING

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1. INTRODUCTION

1.1 The Scope of Work

The scope of this Task Force is to review and make recommendations related to the use of models for turbine-governors for power system simulations. Recent documents published by CIGRE[1], the Western Electricity Coordinating Council (WECC) and others have provided newly developed models for use in modeling thermal turbine-governors, modern combined cycle power plants and hydro turbines. There is, however, significant benefit in reviewing the recommendations in these documents and consolidating them with the usage of older models that still exist in commonly used simulation programs. The Task Force report is presented in three parts. Chapter 2 deals with steam turbines. Chapter 3 deals with gas turbines and combined cycle power plants. Chapter 4 deals with hydro turbines. Each document provides the following:

1. A summary of a hierarchy of models for the various turbine-governor systems.
2. A discussion of the various existing models in most commercial software tools and what may be considered as legacy models.

Additionally, in this first chapter a section is provided below that discusses the primary motive for modeling turbine-governors and the recommended usage of the models.

In many grid-codes around the world outside of North America, model performance and validation requirements are defined and must be complied with as these are deemed statutory (e.g. [1]). The Task Force has intentionally avoided the discussion of grid-codes since such discussion are outside the scope of this group. What is presented here is for guidance and educational purposes. This document is not a standard, nor does it claim to address or comment about any standards or grid-codes.

This Task Force was approved as a working body of the Power System Stability Subcommittee, of the Power system Dynamic Performance Committee of the IEEE Power & Energy Society in June, 2007.

1.2 The Purpose of Turbine-Governor Models in Power System Studies

Power system stability studies deal primarily with transient angle and voltage stability, small-signal stability and frequency control and stability. Reference [2] provides the definitions of each of these types of stability problems. In practice, it is not possible to completely decouple these various stability problems from one another. Particularly in smaller systems, a single event could give rise to multiple inter-related phenomena of frequency, voltage and angular stability problems.

[1] The International Council on Large Electric Systems (www.cigre.org).

With this understanding in mind, the modeling of turbine-governors is particularly important for studies related to transient angular stability, frequency stability and control, and to a lesser extent small-signal stability.

For small-signal stability issues it is possible for the turbine-governor to have a slight negative damping effect in the frequency range of electromechanical modes of rotor oscillation. However, the most widely accepted means of improving the damping of both local and inter-area modes of rotor oscillation on synchronous generators is the application of power system stabilizers (PSS) [3, 4]. If the PSS is properly tuned, it will more than compensate for any such negative damping associated with the turbine-governor and the more influential source of negative damping, high-gain automatic voltage regulators [3].

For transient rotor angle stability the turbine-governor model is of key importance. The important aspect of the turbine-governor dynamics is the initial response of the turbine-governor in the initial second or two following a grid disturbance. A clear example is the concept of fast-valving [4]. This control is characterized by the sudden action to close the intercept valves on a steam turbine following a nearby fault to reduce the mechanical power on the generator shaft and thus minimize the acceleration of the shaft and likelihood of rotor angle instability. Any such controls that will suddenly affect the mechanical output of the turbine following a nearby grid fault would be critical to model for transient stability studies. Another example is the implementation of acceleration controls on gas turbines, which in some cases may come into play during a nearby severe grid fault. Not all gas turbines have acceleration controls. Also, where droop is implemented through feedback of electrical power and the use of a proportional-integral (PI) or proportional-integral-derivative (PID) controller in the turbine-governor controls (typical on many modern gas turbines), this can have some affect on transient stability and it should be modeled accordingly.

The modeling of the turbine-governor is of greatest importance when studying frequency control and stability. Frequency response is an important aspect of power system performance. Figure 1-1a) shows a generic response of a typical power system to a large loss of generation. Here it is assumed that only a single event occurs, being the sudden and unplanned loss of major generation, due, for example, to a failure of mechanical equipment. As can be seen in Figure 1-1a), there are essentially three periods of response, (i) the initial period of response down to the nadir of the frequency deviation, which is governed by the initial rate of decline of frequency, which is a function of total system inertia and the primary-frequency control that is provided by turbine-generator governors, (ii) the initial recovery in frequency which is determined by both primary-frequency response (turbine-governor control and droop) and the load behavior [5], and (iii) the final return of the system frequency to nominal frequency as achieved by automatic generation control (AGC). A discussion of AGC is outside the scope of this Task Force. Figure 1-1b) shows a real example of frequency response for the loss of a large generating unit in WECC – the event is from 2008.

Clearly, when attempting to simulate such events for planning studies, the modeling and representation of turbine-governor response on conventional generating units is of great importance.

A detailed look at Figure 1-1b) reveals the following facts:

1. Initial rate of frequency decline = 0.25Hz/s
2. Minimum Frequency: 59.72 Hz reached 6 seconds after the event
3. Final settling frequency before AGC action: 59.85 Hz reached 20 seconds after the event.

This is a typical trace for a large loss of generation in WECC – the event in this case was the loss of 2.7 GW of generation. The WECC has over 300 GW of installed generation capacity. In comparison, for a system such as the Electric Reliability Council of Texas (ERCOT), which is roughly one-fifth the size of WECC, frequency dips (nadir) as low as 59.6 Hz have been seen and rates of change of approximately 0.2 Hz/s [6], for the loss of a large generating unit/plant in Texas. Going to an even smaller system such as Ireland, with a large penetration of wind generation, and one can see dips of up to 1 Hz (nadir)[2] and rates of decline of 0.5 Hz/s [7, 8]. New Zealand has reported rates of change of frequency as high as 1 Hz/s and a dip of 2.5 Hz (nadir)[3], back in 2003 [9]. With the increased penetration of wind generation these numbers may all be larger now, and in the case of New Zealand and Ireland they may increase significantly.

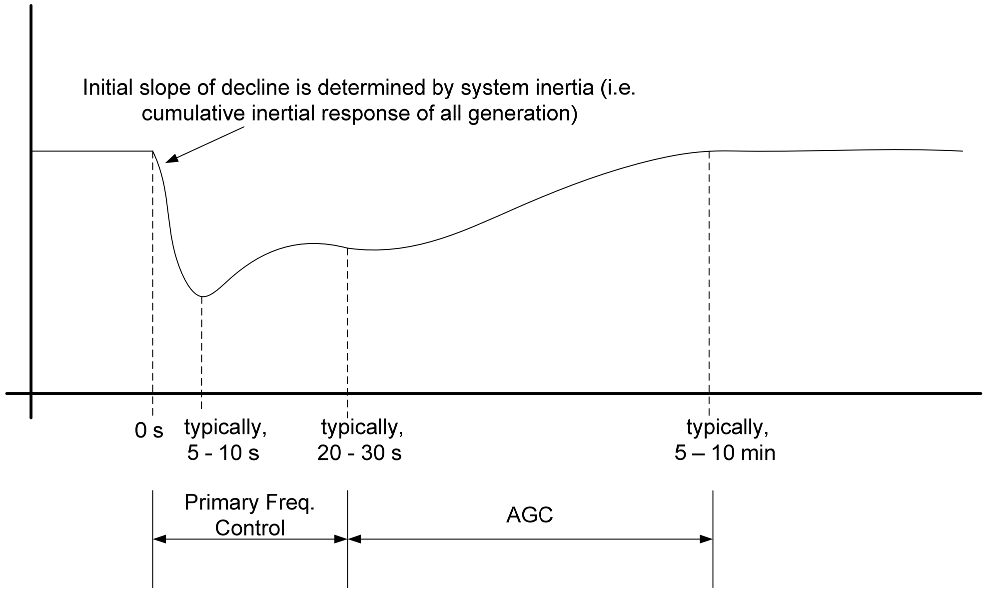
It should be noted that the examples discussed above are simply examples to illustrate some concepts. The actual frequency response of these systems varies from event to event since the response is dependent on (i) the amount of the MW loss and (ii) the amount of MWs of responsive generation that is on-line.

With this background it is easy to appreciate that it is difficult to provide an exact specification of model performance requirements for all possible systems and study conditions. Furthermore, it is impractical to expect a model to be appropriate for all possible systems and study conditions; therefore a hierarchy of models is needed.

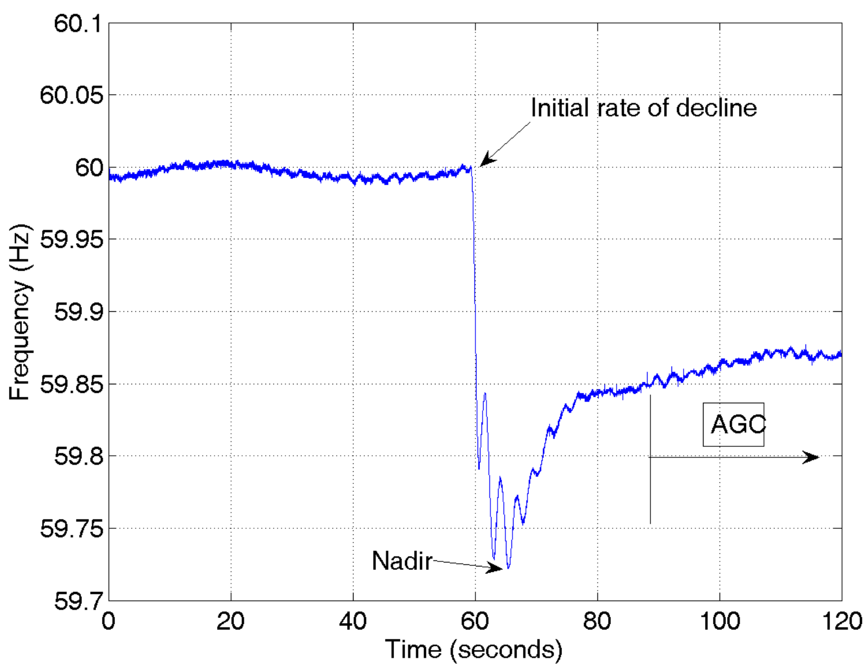
In general, however, it may be said that for typical planning studies the focus is on simulating events for which the system remains predominantly intact (i.e. no major islanding events). For large systems (e.g. North America) the typical deviation in frequency is at most around +/- 1% with an initial rate of change of at most 0.3 Hz/s. For smaller, island systems, the deviation may be as high as +/- 5% with an initial rate of change of at most 1 to 1.5 Hz/s.

Validation of modes for frequency response is not a simple task, and the type of model and model validation required is dependent on the time scale of interest.

[2] For Ireland the nominal system frequency is 50 Hz so a 1 Hz dip = 2 % drop in frequency.
[3] For New Zealand the nominal system frequency is 50 Hz so a 2.5 Hz dip = 5% drop in frequency.



a) Typical power system frequency response



b) Example of actual system frequency response on the WECC system.

Figure 1-1: Power system frequency response to a major loss of generation.

2. STEAM TURBINES

2.1 Steam Turbine Modeling

Large steam turbines are used in fossil fuel power plants. Fossil fuel plants typically burn coal to heat a boiler that produces high-temperature, high-pressure steam that is passed through the turbine to produce mechanical energy (see [10] for a detailed account of steam turbines design and functionality). Other fuels that may be used in fossil fuel power plants are crude oil or crude oil and natural gas (including, liquid petroleum gas). Nuclear power plants also use large steam turbines.

This chapter will provide a brief overview of steam turbine models. It will review some of the most commonly used legacy models (i.e. models that have been in use for several decades) as well as those models recently developed or augmented and typically recommended for planning studies. An overview will also be given of more detailed models that are used in specialized studies, such as when attempting to match plant response for longer-term dynamics. The last section provides brief modeling guidelines and recommendations for modeling steam turbines in power system studies.

2.2 Simple Steam Turbine Models (TGOV1, IEEESO and IEEEG1)

The simplest steam turbine model is the TGOV1 model shown in Figure 2-1. This model represents the turbine-governor droop (R), the main steam control valve motion and limits (T_1, V_{MAX}, V_{MIN}) and has a single lead-lag block (T_2/T_3) representing the time constants associated with the motion of the steam through the reheater and turbine stages. The ratio, T_2/T_3, equals the fraction of the turbine power that is developed by the high-pressure turbine stage and T_3 is the reheater time constant.

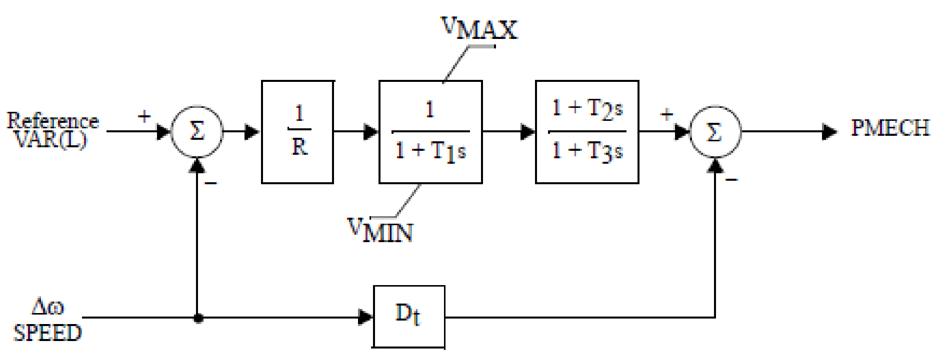


Figure 2-1: The TGOV1 steam turbine model (Courtesy of Siemens PTI).

In 1973 [11] the IEESGO model was introduced, which is also a legacy model, and only slightly more detailed than the TGOV1 model. It is shown below in Figure 2-2. In this case droop is effected by the gain K_1 (equivalent to 1/R in the TGOV1 model) and two turbine fractions are introduced (K_2 and K_3) to represent different stages in the steam turbine.

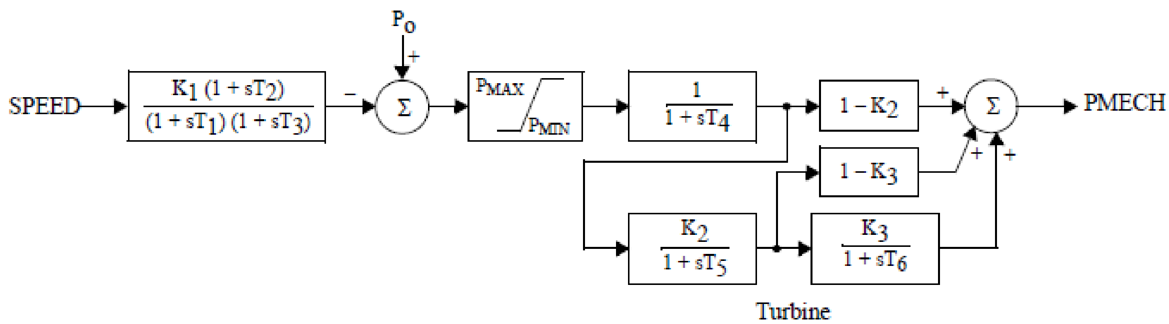


Figure 2-2: The IEESGO steam turbine model (Courtesy of Siemens PTI).

The IEEEG1 model, initially developed and described in [11], is the next level model of a steam turbine, and the one recommended for use. The IEEEG1 model includes the rate limits on the main control valve (Uo and Uc) as well as four steam-stages and the ability to model cross-compound units, as shown in Figure 2-3. More recently the LCBF1 model was developed in the WECC for use with this and other turbine models – it is presently implemented in several software programs such as GE PSLF® and Siemens PTI PSS®E. The LCFB1 model is a simple representation of an outer-loop MW controller (see [5] and [9] for a detailed discussion of this controller and its effects). The combination of these two models is shown in Figure 2-3. This model has been shown to be effective in capturing the behavior of large steam turbine generators that are operated on outer-loop MW control [12]. A simple illustrative example is given here from [12].

The LCFB1 model can be used with any turbine-governor model including the IEEEG1 and the TGOV5 models discussed below. It can also be used with the hydro turbine models discussed in chapter 4.

The list of parameters for the IEEEG1 and LCFB1 models are provided, and described, in Appendix A.

The IEEEG1 model, in combination with the LCFB1 model when needed, is applicable for large interconnected grid simulations when looking at relatively small frequency deviations, that is, in the range of +/- 0.5% change in frequency. All of the models described above assume constant steam pressure and temperature. Figure 2-4 shows an example (from [12]) of simulating the behavior of a large steam turbine using the IEEEG1 + LCBF1 model for grid stability studies. The LCBF1 model is needed only in cases where there is a active secondary outer-loop MW controller in the plant. This is not always the case.

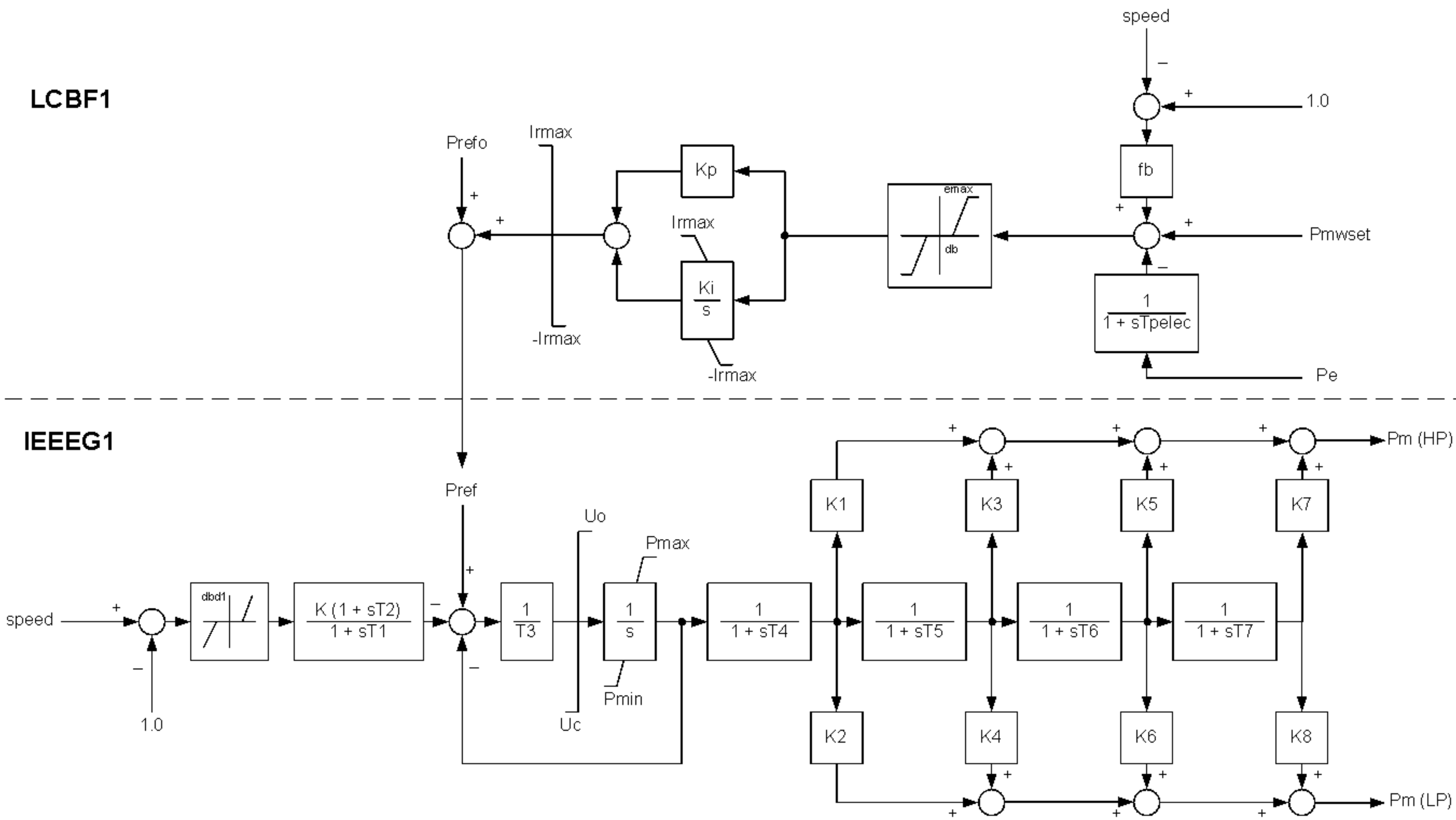


Figure 2-3: The IEEEG1 steam turbine model, together with the LCFB1 outer-loop MW-controller (top part of the model). <u>Note:</u> the IEEEG1 allows for the modeling of cross-compound units and has two sets of turbine fractions (K1, K3, K5 and K7) for the high-pressure (HP) turbine and (K2, K4, K6 and K8) for the low-pressure (LP) turbine. Also, the augmented version of IEEEG1 shown here allows for deadband in the speed-error.

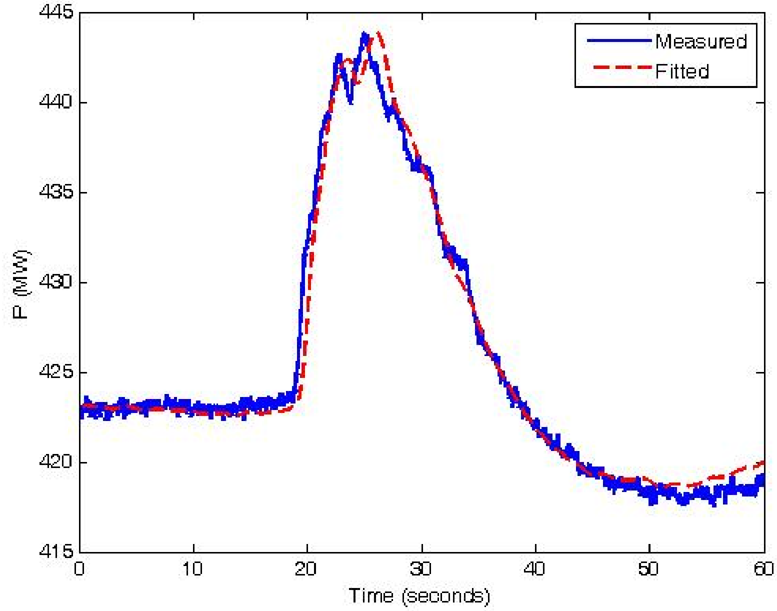


Figure 2-4: Steam turbine generator measured and simulated (fitted) response for a system wide frequency disturbance caused by the loss of a large block of generation elsewhere in the system [12]. The model used was that shown in Figure 2-3. The unit is a 496 MVA steam turbine generator.

2.3 Simple Steam Turbine Model with Basic Boiler Dynamics [13]

There are a number of key assumptions behind the IEEEG1 model presented above. These are, that:

1. steam pressure and temperature remain constant under all conditions,
2. the unit is in boiler-follow control mode – that is, the main steam control valve (MCV) is used primarily for regulating power and the boiler follows the turbine in producing additional steam as needed, and
3. there is essentially an unlimited source of steam from the boiler to be provided once the main steam control valve opens.

It is not difficult to realize that all these assumptions are quite simplistic and not truly indicative of the physics of a steam turbine. Assumption two can indeed be true for many steam turbines – that is, boiler-follow control. However, assumptions one and three are clearly extreme simplifications. Any significant sudden change in the MCV would constitute a sudden drop in steam throttle pressure. Thus, turbine output power would proportionally drop. Thus, additional fuel would need to be spent to increase steam production and steam pressure in the boiler. These pressure drop effects would actually be most significant under a boiler follow control strategy. Modern steam turbine controls often employ a coordinate control scheme where the movement of the MCV is controlled by a combined coordinated effort of regulating power (due to droop response) and main steam pressure. In this section we will illustrate some of these pressure transient effects through an actual recorded turbine response to a system frequency event. The modeling of the boiler dynamics discussed here is based on [14]. The model presented here is a significant simplification from that in [14].

Figure 2-5 shows the response of a large steam turbine unit to a system frequency event as recorded by the plants digital control system (DCS). The sampling rate is one sample per second. As shown in Figure 2-6, an attempt to fit the unit response with the model in Figure 2-3 does not yield a very good match due to the pressure fluctuations (see [13]).

Field testing on the unit confirmed a droop setting of 5% and that the re-heater time constant is of the order of 10 seconds. The reason for the large discrepancy between the fitted and measured response in Figure 2-6 is that the IEEEG1 model is not adequate for representing the full dynamics of this unit. This can be seen in Figure 2-5. As shown, when the frequency disturbance occurs the main steam control valve immediately begins to open based on droop control action. This leads to a sudden drop in main steam pressure. Thus, the coordinated control scheme begins to shut the valve to restore pressure and as pressure is restored the valve begins to open again. If this coordinated control is not represented, together with some representation of the boiler dynamics, it is clear from the turbine response that the inherent assumption of “constant steam pressure and temperature” in the IEEEG1 model is not valid for this case where we have coordinated control.

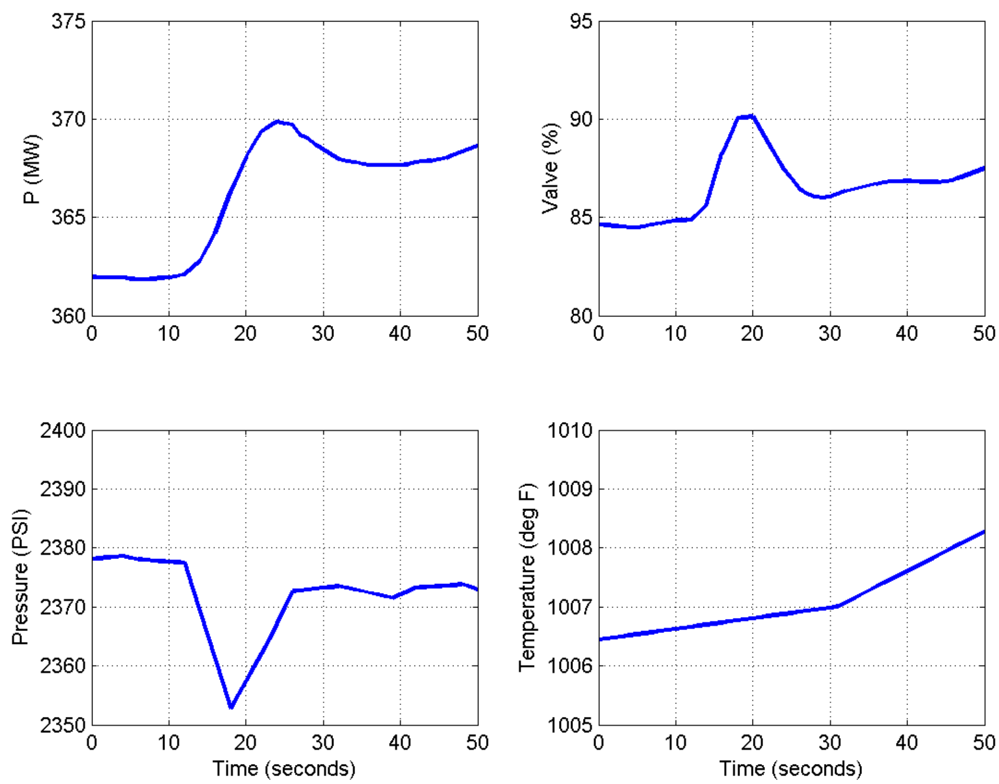


Figure 2-5: Steam turbine response to a grid frequency disturbance. A 446 MW steam turbine operated in coordinate boiler control.

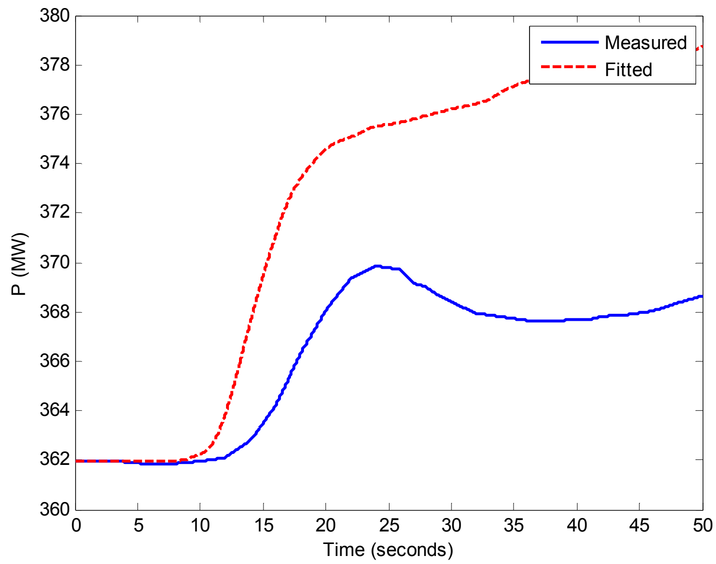


Figure 2-6: Fitting/simulating the MW response of the unit using the IEEEG1 model.

As such the IEEEG1 model was augmented with a simple representation of the boiler dynamics and a coordinated pressure control loop – this is shown in Figure 2-7. In addition, it is noted (as shown in Figure 2-8) that the actual output power to valve position characteristic of the main steam control valve is not quite linear. With these modifications the event was simulated and parameters for the simple boiler dynamics and control optimized. This yielded the fit shown in Figure 2-9. As can be seen the fit is much better since we now have a simple representation of the observed valve movement and pressure fluctuation. The model is still a vast simplification of the actual controls and thus the fit is still not a perfect match. That, however, is not the goal here. The goal was to simply illustrate the nature of the dynamics involved and to achieve a fit that is somewhat closer to the actual mechanical system behavior.

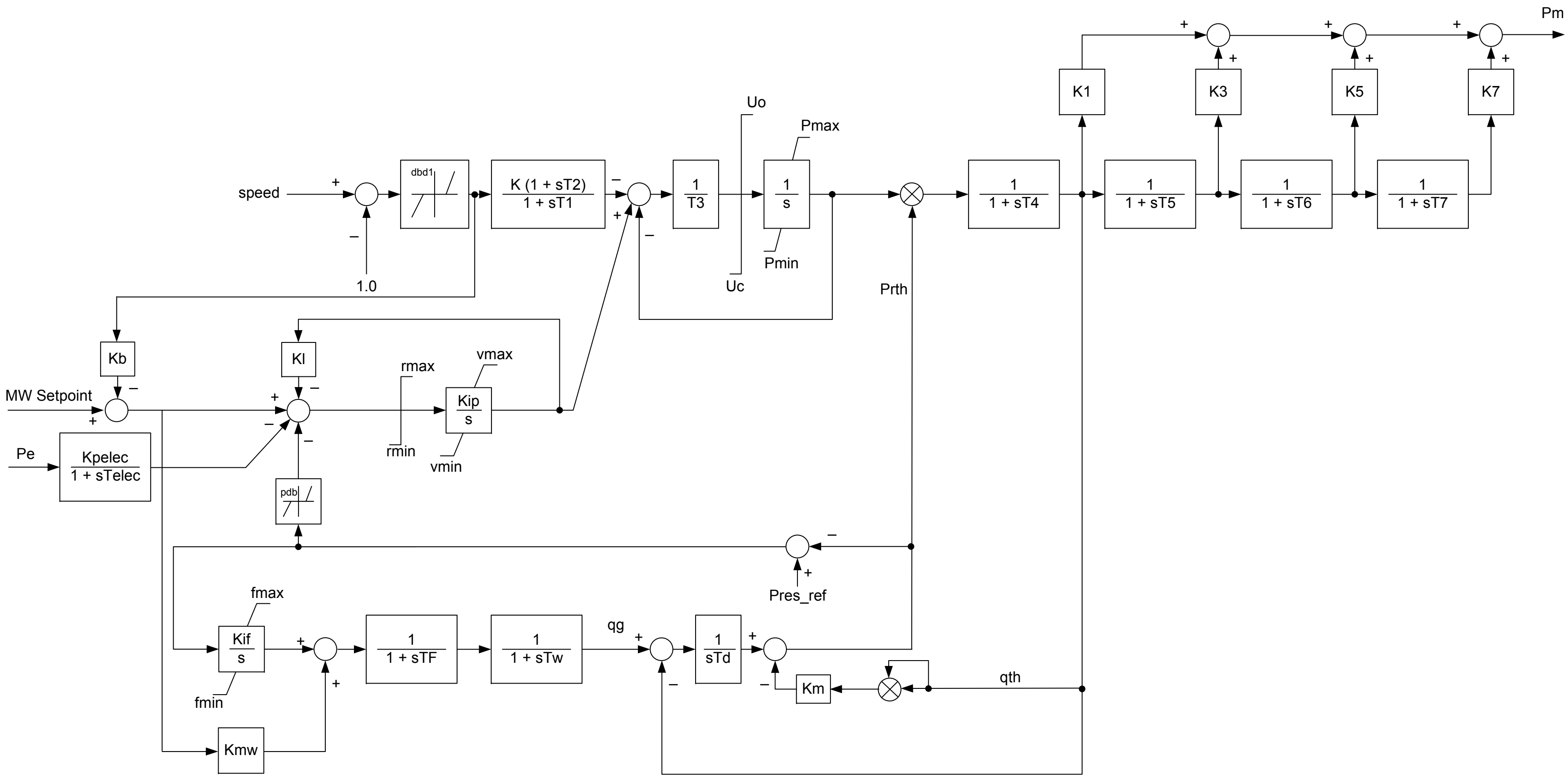


Figure 2-7: Augmented IEEEG1 model with simplified boiler dynamics and coordinate control.

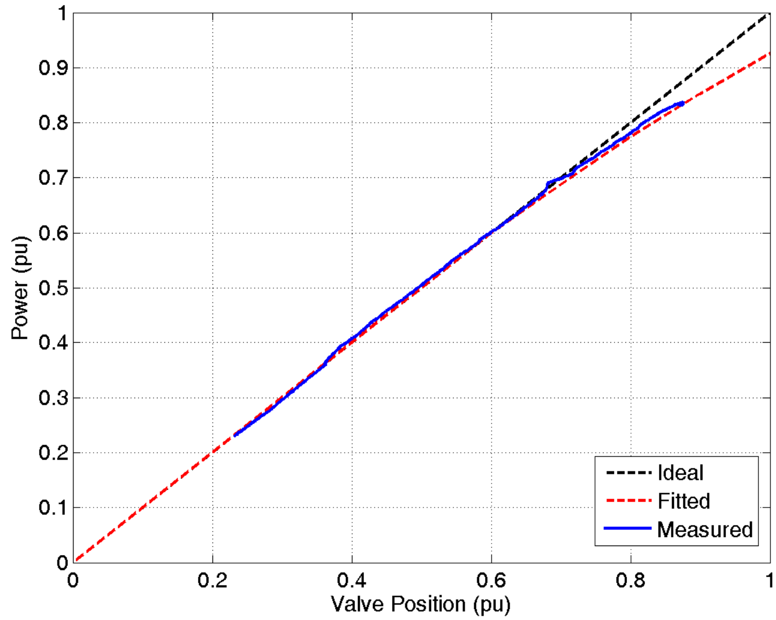


Figure 2-8: Main Control Valve (MCV) characteristic. Measured power versus MCV position is shown at relatively constant steam pressure, during quasi steady-state conditions while unloading the unit from near base-load.

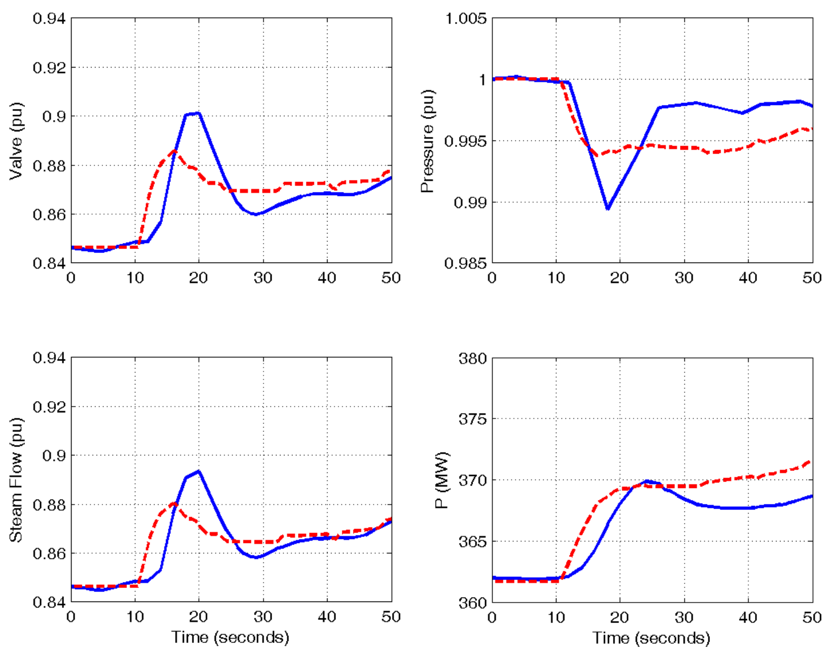


Figure 2-9: Simulated and measured response of the steam turbine using the model shown in Figure 2-7.

2.4 Detailed Steam Turbine Model – the TGOV5 model

In this section a more detailed steam turbine and boiler system model is presented going even deeper into the actual control behavior and turbine dynamics. The presentation is based around the TGOV5 model, presently available in some commercial software tools (e.g. Siemens PTI PSS®E). The TGOV5 model was originally developed based on reference [14]. More detail can be found on the modeling and behavior of boilers in [15].

2.4.1 The TGOV5 model

Figure 2-10 shows the block diagram of the TGOV5 model. It is evident that the turbine and droop control model are identical to the IEEEG1 model. The key additional features are:

1. the added boiler dynamics (parameters C_B, K_9, C_1) that determine the current steam throttle pressure (P_T), which when multiplied by the valve position yields the available current mechanical power at the steam turbine inlet,
2. the coordinated controls acting on current electrical power (P_{ELEC}), frequency error (Δf), and pressure error (P_E), to determine the power order (P_o), and
3. an emulation of the drum pressure controller (parameters K_I, T_I, T_R, T_{R1}, C_{MAX} and C_{MIN}) and fuel dynamics, that is, for example, the process of pulverizing coal[4] and combusting it to heat the boiler drum (parameters T_D, T_F, T_W, K_{11} and K_{10}).

Fitting parameters to the TGOV5 model is quite a difficult task and requires extensive testing of the plant. In addition, there are many subtle variations on the various control aspects. For this reason, models of this complexity are rarely used or appropriate for large scale power system simulations such as in the case of the North American power system.

[4] As noted previously, the fuel source in the case of a steam turbine may come from other sources as well, e.g. oil or gas.

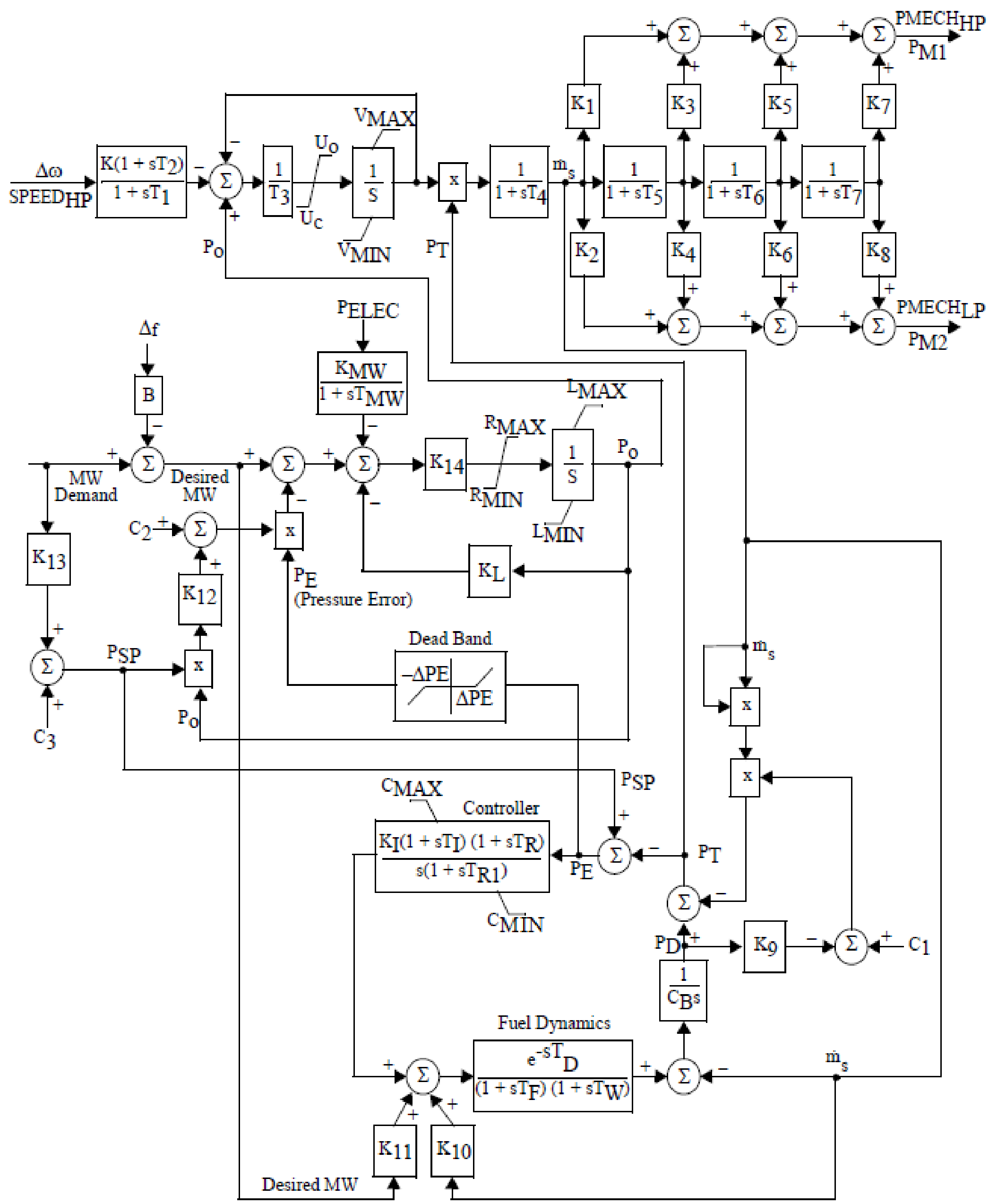


Figure 2-10: The TGOV5 model (courtesy of Siemens PTI).

2.4.2 An Example use of the TGOV5 Model

Here a brief outline is given on a case study of using the TGOV5 model in South Africa to represent a coal fired power station. The process revealed a subtle but important variation that was needed to accurately simulate the primary frequency control strategies and the boiler limit pressure controller.

When a unit is normally operated in boiler-follow or coordinated control, the boiler pressure is maintained predominantly using the main fuel control of the boiler. The limit-pressure controller is only activated when there is a large deviation in the boiler pressure from target. The governor valve takes over control of the boiler pressure to prevent the boiler pressure from falling further. The switch over logic results in an initial large drop of real power as the governor valve quickly closes to restore the boiler pressure (see Figure 2-11).

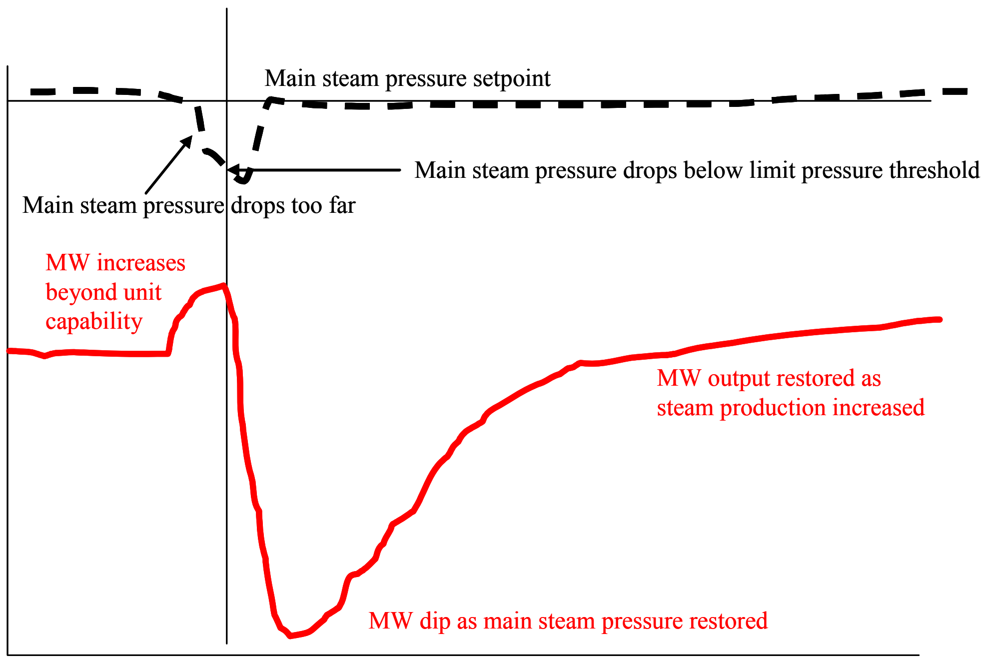


Figure 2-11: Description of limit pressure controller activation

A typical incident when the limit-pressure controller is engaged is when there is a significant frequency deviation. The initial governor action can result in a drop of the steam pressure and activation of the limit-pressure controller.

When this pressure controller is active the governor valve will try to maintain the boiler pressure for a few minutes. The typical control strategy is that the pressure controller is activated when the pressure deviation exceeds a predefined limit. This behavior should typically be modeled for under-frequency load shedding studies, when large frequency deviations studies are being performed.

Figure 2-10 shows the TGOV5 model. The deadband function in the pressure controller ($-\Delta Pe/\Delta Pe$ at the center of the figure) is implemented such that if the pressure error is within the deadband the output of the block is zero, and otherwise the output starts from zero and linearly increases – that is the output of the block is off-set by the deadband. In this case studied, this was found to be inappropriate for the particular controls in this plant. In that case the deadband has no off-set and jumps to the input value once outside the deadband. This was implemented with a step function the output of which is 1 if the pressure error is outside the deadband and zero if it is within – this is shown in Figure 2-12. This is essentially the issue of type 1 versus type 2 deadband implementation as discussed in [16] – the deadband implemented here is type 1 (see Appendix B).

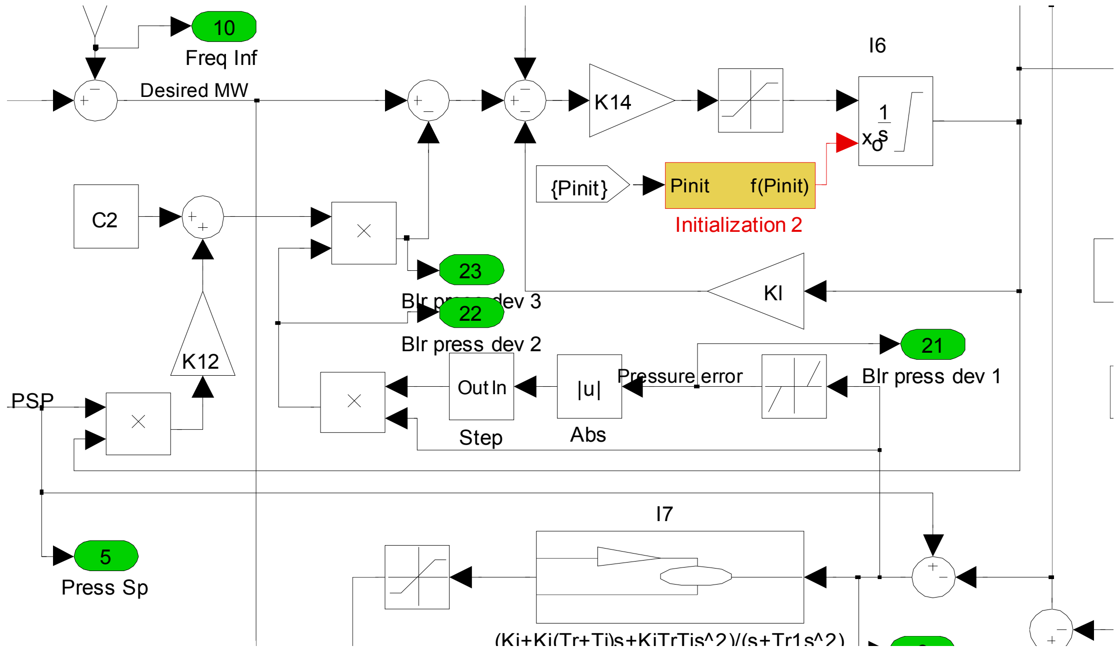


Figure 2-12: TGOV5 model zoomed in on limit pressure controller modification

The results of this modification to the model proved reasonably successful as shown in Figure 2-13 and 2-14, below. ESKOM – for whom this model was developed – will continue to monitor the performance of the model. It should be noted, however, that the results shown in the figures below are much longer in duration (600 seconds) than typical simulation studies. This illustrates the importance of understanding the actual plant controls when attempting to match the exact plant response for long term dynamics.

2.4.2.1 Primary Frequency Control Deadband and Limiter

For the normal operating conditions the primary frequency control is not required to be activated and a maximum deadband of 0.15 Hz is allowed in ESKOM (South Africa). The deadband also had to be applied to the TGOV5 governor model to improve accuracy of long term dynamic studies.

The deadband is mathematically modeled as:

If x > deadband then y = x – deadband

Else if x < - deadband then y = x + deadband

Else y = 0

This is a type 2 deadband (Appendix B), where the output is off-set by the deadband.

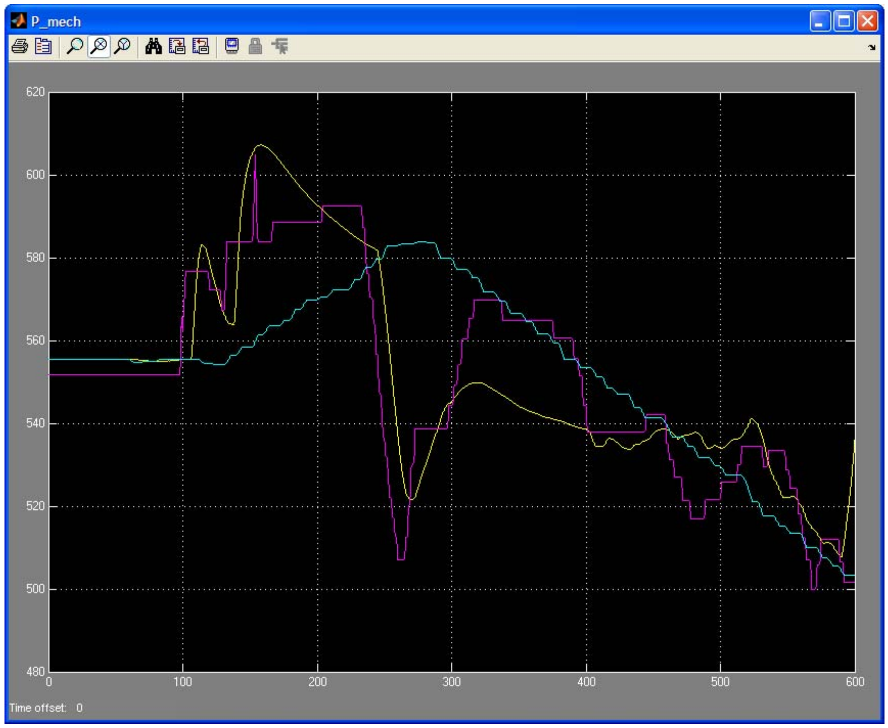


Figure 2-13: Unit 1 actual MW response (purple line), simulated MW response (yellow line) and Target MW setpoint (light blue line)

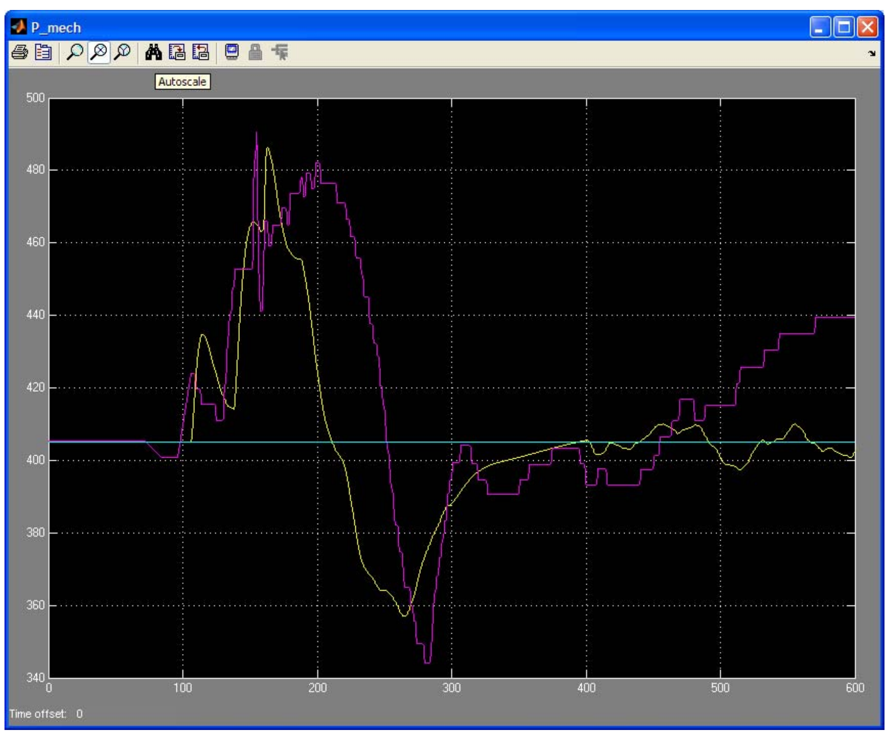


Figure 2-14: Unit 2 actual MW response (purple line), simulated MW response (yellow line) and Target MW setpoint (light blue line)

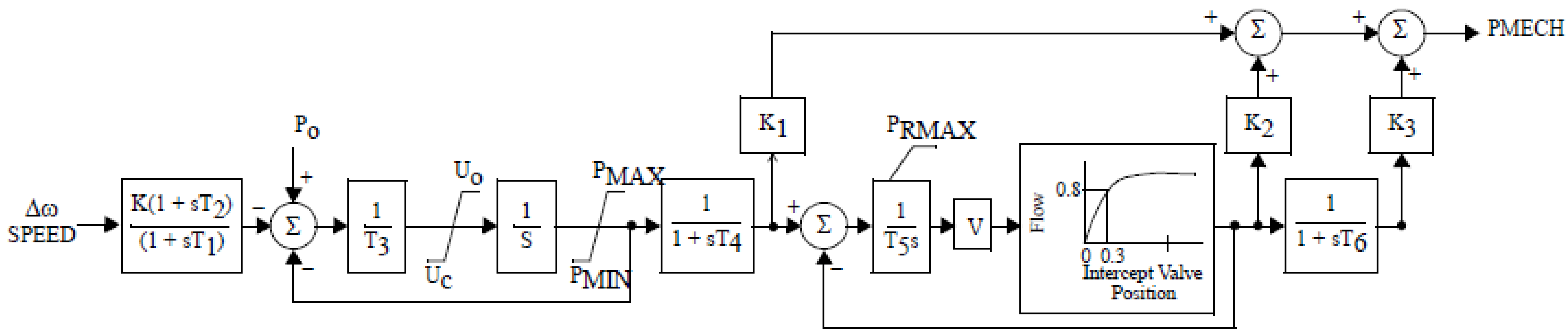
2.5 Simulating Fast-Valving

To represent fast-valving (see [4] for a detailed discussion of fast-valving) the TGOV3 model may be used. It is the IEEEG1 model augmented with the addition of a simple representation of the intercept valve actuator (T_5) and the intercept valve fast closing characteristics (T_I, T_A, T_B and T_C). This is shown in Figure 2-15. The non-linear valve position versus steam-flow characteristics is also represented. A more detailed representation of fast-valving, including a representation of the triggering logic, can be found in the TGOV4 model, which is a model available in the Siemens PTI PSS®E program, and possibly other software tools. A detailed description of that model is outside the scope of this document, but can be found in the documentation of those programs.

2.6 Other Aspects of Steam Turbine Modeling

Above we have presented a hierarchy of several steam turbine models, including their governors. The IEEEG1 model represents the turbine and steam valves with a simple set of time-constants and turbine-fractions, and a simple actuator model, respectively.

The model in section 2.3 introduces a simple representation of the boiler dynamics, and the TGOV5 model provides a more detailed representation of the boiler dynamics and controls. In the IEEEG1 model the turbine speed-governor action is represented by a simple proportional gain K and a lead/lag block (1 + sT2)/(1 + sT1). The proportional control acting on speed error is common and still used in many designs of steam turbines. However, in modern systems the speed-governor may incorporate an electrical power feedback. Figure 2-16 illustrates this type of speed-governor.



T_I [VAR(L+1)]: TIME to initiate fast valving.

T_A [CON(J+8)]: Intercept valve, v, fully closed T_A seconds after fast valving initiation.

T_B [CON(J+9)]: Intercept valve starts to reopen T_B seconds after fast valving initiation.

T_C [CON(J+10)]: Intercept valve again fully open T_C seconds after fast valving initiation.

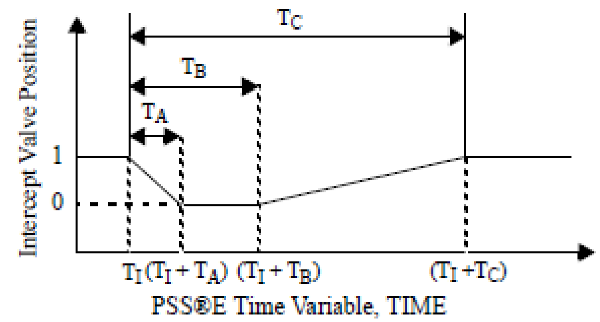


Figure 2-15: The TGOV3 model (Courtesy of Siemens PTI).

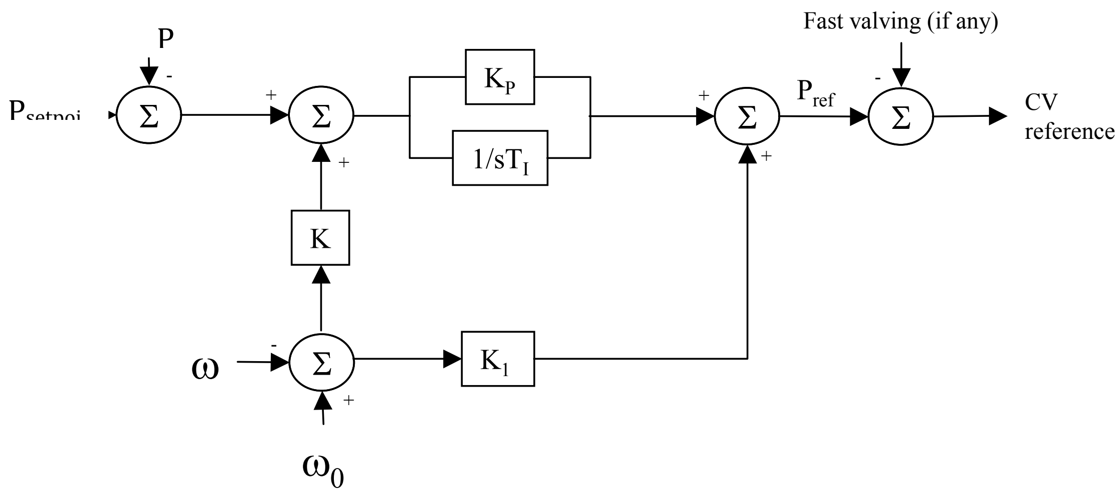


Figure 2-16: Generic scheme of a speed governor with one operating mode, using electrical power feedback.

Parameters:

P = measured power;

$P_{setpoint}$ = governor power setpoint

P_{ref} = Power reference

ω = measured speed;

ω_0 = reference speed;

1/K = speed droop;

K_P = proportional gain;

$1/T_I$ = integral gain;

K_1 = proportional gain of the speed loop.

In some turbines, two operating modes are possible:

- Automatic Mode: This is the most common in normal operation. This mode allows the power of the unit to be controlled to Psetpoint + KΔf + AGC requested by the transmission grid. That is, the control strategy shown in Figure 2-16. Namely, the PI controller acting on the power error (Psetpoint – P) maintains the unit power output – under steady-state frequency conditions – at Psetpoint. If frequency deviates, then the speed error feedback acts to off-set the power error (KΔf). Automatic Generation Control (AGC) can also affect this input, but a discussion of AGC is outside the scope of this paper.

- Direct Mode: In this strategy the valve is directly controlled. In some cases this control strategy is used during particular operating situations: lines energizing, islanding operation after blackouts, etc. If such conditions occur, switching to direct mode improves the generating unit stability margins.

The switch goes from the automatic mode to the direct mode automatically when two criteria are met:

- $(P_{setpoint} - P)/P_{setpoint} > P_{Threshold}$ (10 % in France)
- $\Delta f > f_{Threshold}$ (200 mHz in France)

After the switch from the automatic mode to the direct mode, the AGC (if any) also goes out of service. In addition the generating unit output power is not predictable anymore. It can have different values depending on the disturbance severity. The switch goes back to the automatic mode through a manual action of the plant operator. This is all depicted in Figure 2-17.

Finally, where fast-valving is applied for transient stability improvement, this may be modeled as directly controlling the simulated control valve, as depicted in Figure 2-16. Refer to [4] (Figure 17.3) for an example of a typical fast-valving valve action sequence. Also, in the above section the example of the TGOV3 model was given to show how fast-valving may be simulated.

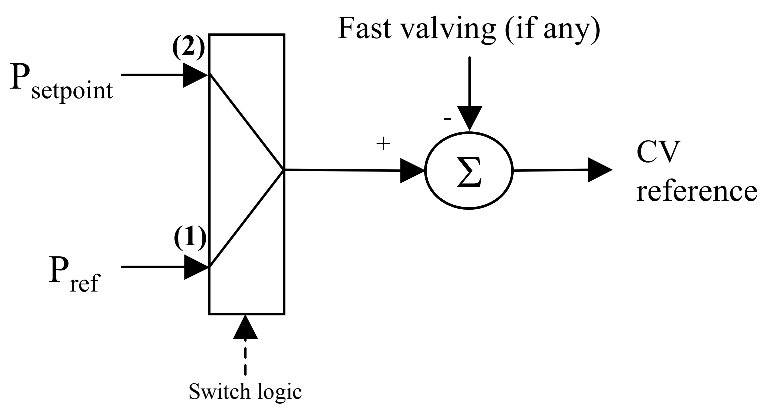


Figure 2-17: Generic scheme of a speed governor with two operating modes.

2.7 Modeling Guidelines and Summary

This chapter has presented several existing models for steam turbine generators. The IEEEG1 model is available in the common commercial power system simulation tools and one of the most commonly used for representing steam turbines. It is certainly the one recommended for use in typical planning studies, particularly for large interconnected systems. For studies related to system islanding or scenarios that result in large deviations in frequency, i.e. more than several percent change, the accuracy of the IEEEG1 model is limited.

The most important aspects of the steam turbine model are still as pointed out in the previous work [5], namely:

1. Check all turbine-governor models so that they are properly modeled based on the MW capability of the turbine and not generator MVA base. In some software platforms many of the turbine-governor models are on generator MVA by default and so Pmax and droop will need to be adjusted accordingly. If this is not done properly, then the unit will exhibit unrealistic response during simulated frequency events.
2. Represent the outer-loop MW controller (LCFB1 model) where such response is observed. The best way to capture this characteristic is by on-line disturbance monitoring (see example in section 2.2 or reference [12]).

Furthermore, for the IEEEG1 model the droop, valve time constant, turbine time constant and valve opening/closing rate may all be reasonably estimated by validation against appropriate test or on-line disturbance data. If an outer-loop MW controller is acting on the turbine-controls, the only effective way of validating its response (essentially integral gain) is from an on-line disturbance recording (see example in section 2.2, or reference [12]). The other parameters of the model, such as the turbine power fractions need to be obtained from the original equipment manufacturer data sheets. In addition, it should be noted that the estimated unit droop, based on measured response, when using the IEEEG1 model may vary depending on the operating condition. This is not due to an actual change in the droop constant, since that is an actual fixed gain in the controls, but a consequence of the non-linear nature of the steam turbine response (e.g. changing steam pressure, temperature, non-linear valve characteristics etc.).

The TGOV5 model is a significantly more detailed model and able to represent the boiler dynamics and coordinate control for studies where this may be deemed as necessary, e.g. studies on small islanded systems. However, the burden of model validation for TGOV5 is tremendous and it would certainly not be warranted for large systems such as the North American systems at this time. In [5] it was shown that good agreement between overall system frequency response between simulation and measurement has been achieved both for the WECC and ERCOT systems using simple models such as the IEEEG1 model.

As an example, in a current typical planning case for WECC, there are roughly 3000 synchronous generator models. For modeling steam turbines, there are 267 IEEEG1 models, many with associated LCBF1 models, 20 TGOV1 models and 3 CCBT1 models. TGOV1 is a very simple model – it models droop, one time constant for the valve and one lead/lag block to represent the turbine (see section 2.2). CCBT1 is a coordinated boiler turbine model available in GE PSLF® and is similar to the TGOV5 model in complexity and parameters. In addition, roughly 30% of the units do not have a turbine-governor model at all. A significant portion of these are steam turbines in multi-shaft combined cycle plants, pumped hydro, distributed generation, or blocked governors and thus deliberately not modeled. These considerations aside, many of these units may still be missing an adequate model of the turbine-governor.

Considering a typical Eastern Interconnection planning case, there are more than 7000 synchronous generators in the case. Roughly 50% (one in every two) of the generators do not have a turbine-governor model. This is a significant problem. Considering the steam turbine models, the rough statistics are as follows:

- IEEEG1 218
- IEESGO 425
- TGOV1 265

The above discussion is simply intended to illustrate that there is a significant gap, particularly for the Eastern Interconnection, between the present level of modeling and what could be achieved by using simple models such as IEEEG1. Namely, there are a vast number of units in the system wide model without an adequate turbine-governor model, or proper representation of the turbine rating, or the mode of turbine-governor operations (i.e. whether the turbine is under (i) droop-control, or (ii) has an outer-loop MW controller, or (iii) base-loaded and thus unable to respond).

While this document has identified a recommended turbine-governor for steam turbines (IEEEG1 + LCFB1), it is recognized that many governors are currently modeled using what may be considered as obsolete or legacy models (e.g. TGOV1, IEESGO, etc.). In many cases these models and associated data were provided to the owner by the equipment vendor when the unit was commissioned many years ago. It is also recognized some owners may not have the resources or expertise to convert such obsolete or legacy models to newer models. It is not the intention of this task force report to force conversion of such obsolete or legacy models to recommended models. Use of the best models available is the preferred course of action. For new plants or plants that have undergone an upgrade, a validated turbine-governor model should be requested from the vendor.

3. GAS TURBINES AND COMBINED CYCLE POWER PLANTS

3.1 Gas Turbine Modeling

Several documents published in the last decade give a comprehensive account of gas turbine modeling and dynamics behavior [9], [17], [18]. This section will provide a summary of some of the key concepts from these documents.

First the theory of the operation of gas turbines is briefly described. Then a summary will be given on some of the most commonly used legacy models (i.e. models that have been in use for several decades) as well as those models recently developed and typically recommended for planning studies. An overview will also be given of more detailed models, which is augmented with further details in the appendices. The last section of this chapter provides brief modeling guidelines and recommendations for modeling gas turbines in power system studies.

3.1.1 Brief Overview of Gas Turbine Theory

The gas turbine consists of an axial compressor, a combustion chamber, and a turbine (Figure 3-1). The air, to support the combustion process, is compressed through the axial compressor and then mixed with fuel in the combustion chamber, where the combustion process takes place. Ideally, the compression process between compressor inlet (1) and compressor outlet (2) is an isentropic process; i.e. the process is adiabatic and reversible. The combustion process between points (2) and (3) is ideally a constant pressure process. Isentropic expansion of the hot gases in the turbine, between points (3) and (4), produces real work on the turbine shaft. Finally, the working fluid (typically air) cools under constant pressure between points (4) and (1).

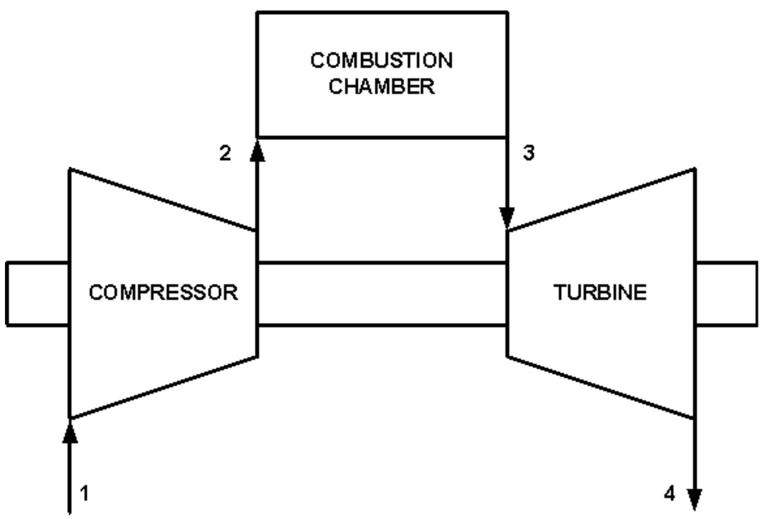


Figure 3-1: Gas Turbine

In reality the process is non-ideal. Reference [18] provides a detailed qualitative description of the thermodynamic process and thus the physical reason behind the dependency of maximum gas turbine power output as a function of ambient air temperature and turbine shaft speed. In summary, the temperature T_3 (at the inlet of the turbine) is the highest temperature in the cycle and may be assumed to be essentially the temperature of the hot gases entering the first stage of the turbine. In practice, this temperature will need to be kept below a certain limit in order to preserve the life of (reduce fatigue and stress on) the hot gas-path parts in the turbine. However, it is extremely difficult to measure this temperature in practice since (i) insertion of thermocouples with a fast response time into this region of the turbine can be difficult, and (ii) there is not one temperature but rather a spread of temperatures across the combustion chamber/cans. Therefore, it is common to measure the temperature at the exhaust of the turbine (T_4), and through controlling of this temperature T_3 is maintained below its limit.

In essence if we are operating at the maximum T_3 limit (essentially at base-load or rated turbine output) then if ambient air temperature rises, or the shaft speed decreases, both these will result in a lower air mass-flow through the compressor and turbine. Thus, to maintain the temperature limit the amount of fuel being mixed and combusted with the air must be reduced, thereby reducing the output power of the turbine. That is, the steady-state rated output of the gas turbine is dependent on both ambient inlet air temperature and the speed of the compressor [18]. It can be shown that this rather complex thermodynamic process can be simplified for the purpose of power system simulation analysis to a set of algebraic equations [17], [18] as shown in Figure 3-2.

In Figure 3-2 W_a is dimensionless airflow (in per unit), ω_c and $\Delta\omega_c$ are the per unit dimensionless speed and change in dimensionless speed (in pu), T_{a0} and T_a are ISO and current ambient air temperature (in degrees Kelvin), P_{a0} and P_a are ISO and current ambient air pressure (in psi) and θ_{IGV}, θ_0 and θ_{max} are the current, initial and maximum inlet guide vane opening angle, respectively. For a more detailed explanation of why the speed and airflow quantities are referred to as dimensionless see [17], [18] and [19].

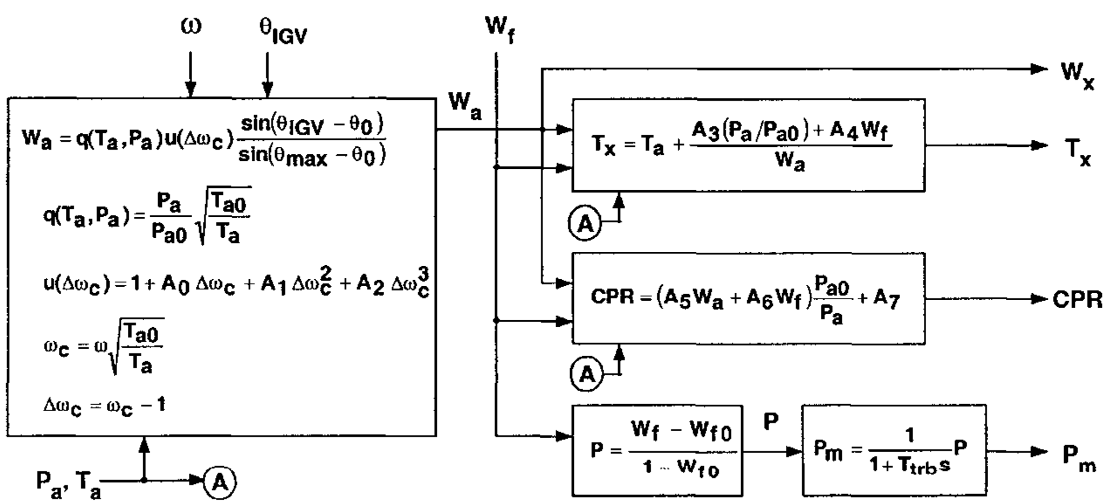


Figure 3-2: A simplified gas turbine thermodynamic model [17] (IEEE© 2001).

By combining the gas turbine controls with this simplified gas turbine model (as shown in Figure 3-3) we can arrive at a rather detailed model of a gas turbine. One example is the *gegt1* model discussed in Appendix D.

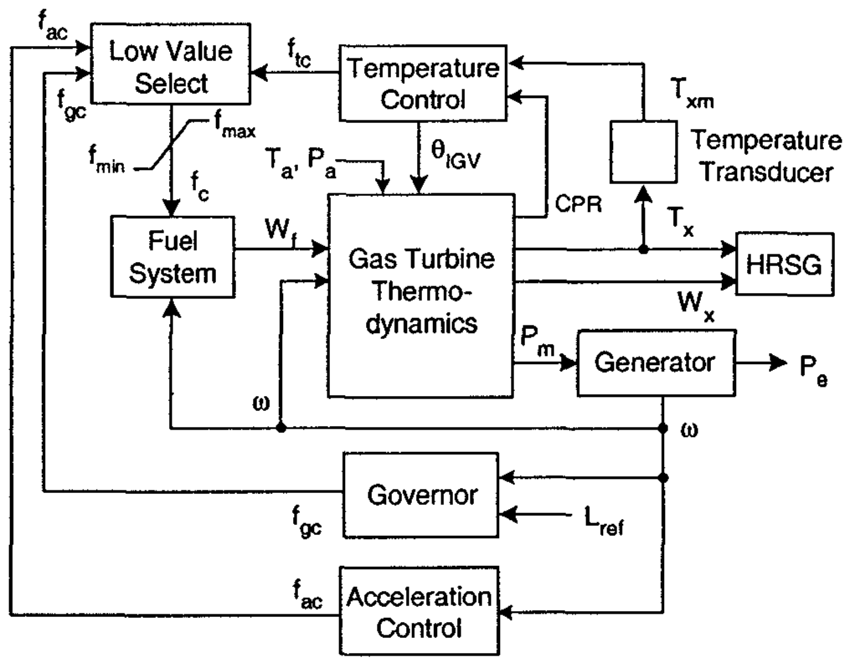


Figure 3-3: Gas turbine controls [17] (IEEE© 2001).

3.1.2 Modeling Gas Turbines

A hierarchy of models will be presented here for heavy-duty gas turbines.

3.1.2.1 GAST

The GAST model is still used in both the WECC and the Eastern Interconnection. There are roughly 50 units in the WECC database and over 400 in the Eastern Interconnection, with the GAST model.

Figure 3-4 shows the GAST model. A slightly modified version also exists. This is the most simplistic representation of a gas turbine. It assumes a simple droop control, constant load limit (rating of the turbine) and three time constants, one to represent the fuel valve response (T_1), one to represent the turbine response (T_2) and one to represent the load limit response (T_3). This model completely neglects all aspects of the physics of a heavy-duty gas turbine. This model is not recommended.

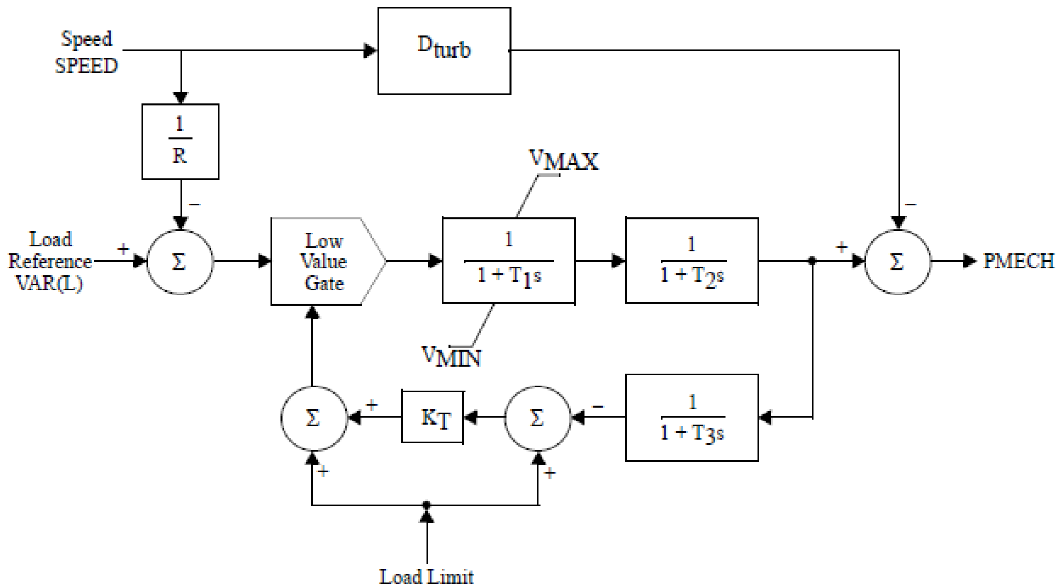


Figure 3-4: The GAST model. (Courtesy of Siemens PTI)

3.1.2.2 GAST2A

The paper by William Rowen in 1983 [20] presented one of the first models for gas turbines. This model is no longer used in WECC, but over one hundred gas turbines in the Eastern Interconnection are still modeled using this model. The model block diagram and parameters may be found in [20] or the user's manual of some of the commercial power system software. For the sake of brevity we provide only the following general comments related to this model. It makes the simplifying assumption of neglecting inlet guide vane controls (IGV), since for simple-cycle gas turbines the IGVs are typically wide open. Furthermore, it assumes a proportional speed governor control and a constant exhaust temperature limit – this latter assumption is for simplicity, the actual exhaust temperature limit (as explained in section 3.1.1) is not constant.

The GAST2A model was developed in the 1980's to represent GE Mark II and Mark IV controls. Since that time, the turbine controls have evolved to include a governor with a proportional-integral controller and droop developed with electrical power as the feedback signal. Also, significant changes have been made to the gas turbine design to improve performance with higher efficiency and lower emissions. These changes include going from rich burn combustion to a lean burn to achieve higher performance. Newer controls for the gas turbines include guide vane dynamics regulating airflow to maintain the lean burn. The complexity of the newer controls results in more complex models. This is discussed below.

The GAST2A model is not recommended for use for modern gas turbine installations.

3.1.2.3 GGOV1

The GGOV1 model was developed as a general purpose turbine-governor model to be used for dynamic simulation studies. Figure 3-5 shows the GGOV1 model.

In terms of the gas turbine, much of the same simplifying assumptions as in the case of GAST and GAST2A are made, namely, neglecting the IGV controls, assuming a temperature limit/load limit that is constant[5] and not explicitly representing any ambient or other effects. The steady-state developed mechanical power from the model is given by

$P_{mech} = K_{turb}*(Wf - W_{fnl})$

The parameter W_{fnl} is the fuel flow for full speed, no load conditions, and this allows a representation of fuel consumed at no-load for running the axial compressor [9].

The main improvement over GAST and GAST2A is the flexibility of the model to provide for various governor control options and feedback signals, namely:

1. PID control on speed error signal consisting of speed, speed reference and droop signal.
2. PI control on speed error signal consisting of speed, speed reference and droop signal.
3. P control on speed error signal consisting of speed and speed reference.

The droop signal can be obtained from a number of feedback signals, i.e. electrical power, governor output or valve stroke. For the third control design in which only proportional control is used, the reciprocal of the value for the proportional gain defines the droop. There is no need for another signal to define droop.

The output of the governor blocks is the signal "fsrn", which goes through a Low Value Select block. The other two signals are "fsra" and "fsrt", which represent the acceleration controller (if any) and the temperature limit controller. Controls for those two signals can be disabled by setting the values for aset and Ldref to large values.

If "fsrn" is selected, it becomes the input signal for the remaining blocks that represent the dynamics of the fuel stroke. The valve stroke model includes a time constant, T_{act} and rate limiters. The fuel stroke demand has a set of limits with Vmax and Vmin. Normally Vmax is set to 1.0 pu on the turbine MW base. Vmin is set to a value for the minimum fuel flow for a gas turbine.

The remaining blocks are used to represent the turbine model. The turbine gain K_{turb} was described above. The time constant Teng is a transport delay typically set to zero for gas turbines[6]. The constant flag is set to 1 if the turbine has a liquid fuel source that is shaft driven to represent the dependence of fuel flow on shaft speed.

The block containing the lead-lag function with time constants Tb and Tc can be used to represent the lags for the gas turbine to changes in fuel flow. Typically, Tc = 0, and Tb = 0.1 seconds.

---

[5] A speed damping factor can be modeled to influence the temperature limit as a rather gross approximation of the speed dependence of the turbine rating. This is, however, not very accurate.

[6] Teng can be set to 15/N+60/nN, when using this mode to represent a reciprocating diesel engine. In this case N is the rpm of the engine and n is the number of cylinders firing per revolution. Note that for diesel engine applications the value for "flag" should be set to 1.

The signal “fsra” is the output of the acceleration control. The speed signal is processed through a block consisting of a derivative with a lag filter. The output of this block provides the steady rate of change in the speed, and it is compared to the value assigned to *aset*. For GE gas turbines this parameter aset = 0.01 pu/sec.

The signal “fsrt” provide a limiter which is similar to the temperature limiter of the 1983 Rowen model. The difference is that instead of calculating temperature from fuel flow the model uses fuel flow as the input and sets the limit based on a value of maximum load in per unit, as specified by the parameter, Ldref. The user is required to know the values of the maximum gas turbine MW output for different ambient conditions and to assign those values to Ldref. The lead-lag and lag blocks for the limiter are the same as those in the Rowen model.

Note that the selected value of “fsr” out of the low value select block is used as input for the three controllers. In the event that the low value select block selects either “fsrn” or “fsrt”, then the controller whose output is selected will function as a PI controller based on the feedback loops[7]. If “fsra” is selected, then the control for the acceleration limiter is an integral controller. The controllers whose output is not selected use the signal “fsr” for tracking to avoid wind-up.

Typical values for the GGOV1 model parameters are provided in Appendix C.

The principal elements of gas turbine controllers are represented in GGOV1. That is, the speed/power governor, which normally has control at turbine loads between about 70 percent and 100 percent of maximum, the acceleration control (some GTs do not have this), and a temperature limit controller that enforces the maximum output limit. The temperature limit controller is one of several that can override the simple speed/power governor function; it is singled out for consideration because it is the one most frequently of importance in grid studies. The explicit modeling of the individual controllers of a complete gas turbine control system is not appropriate for grid studies, both because of the great difficulties of managing the data required and because controllers concerned with the internal engine variables have only relatively small influence on the behavior seen by the grid. Accordingly the GGOV1 model provides a description of the behavior to be expected of a properly built and adjusted gas turbine engine, but not a detailed description of any particular engine.

GGOV1 is appropriate for studies where the plant remains connected to a large grid; that is for all conventional grid interconnection studies in for example continental North America. The model can also be used for load rejection studies to simulate the system response to loss of loads. For large load pick up studies (e.g. black start) the model may not provide accurate results since guide vane dynamics, which are not included in the GGOV1 model, may become a factor.

[7] Note in the case of “fsrn” there is also the option of a derivative control (K_{dgov}) to allow for modeling of a PID control for the governor.

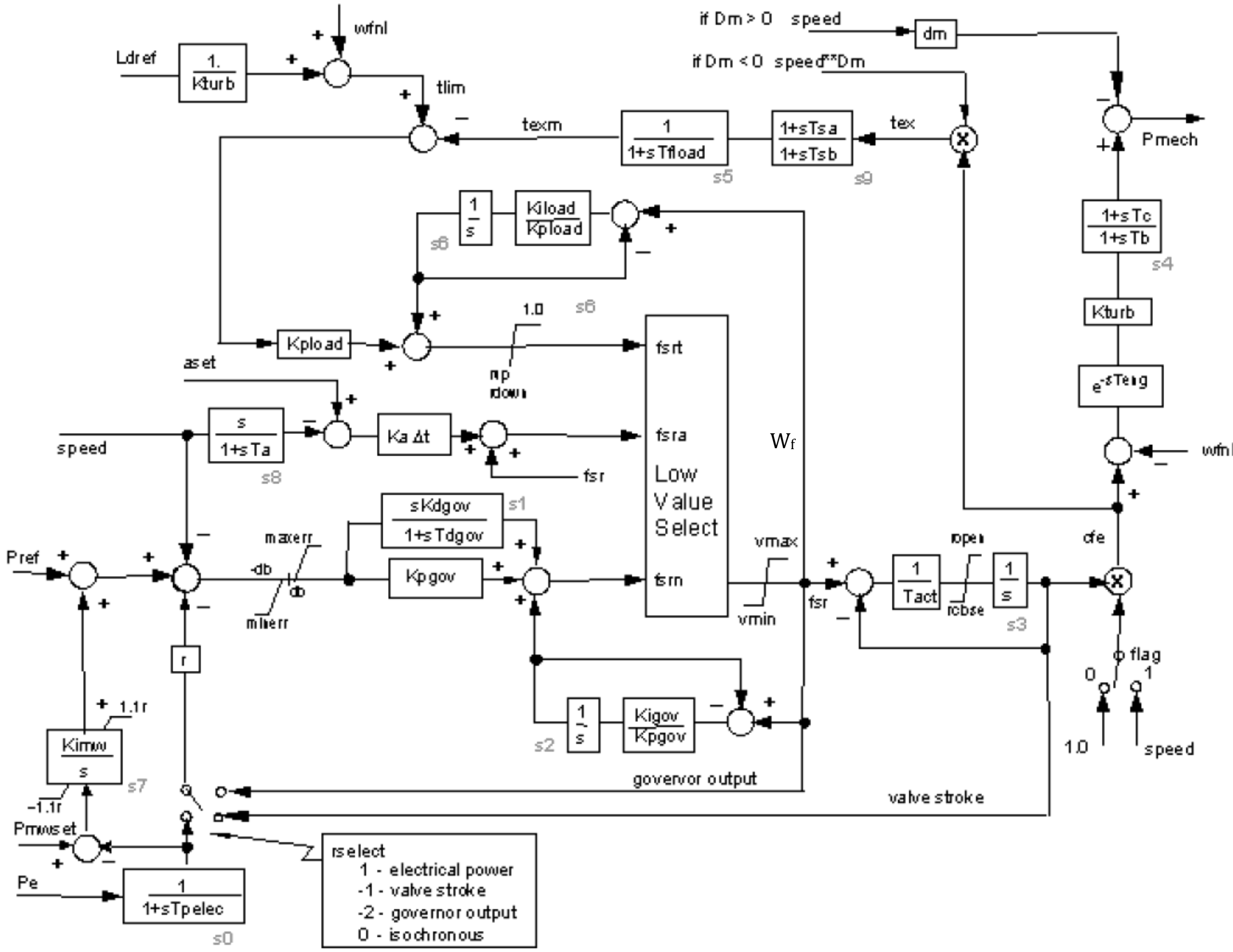


Figure 3-5: GGOV1 gas turbine model. (Courtesy of GE). Note that for this model the shown implementation of the PI controllers through feedback of "fsr" inherently incorporates the tracking logic among the three controllers feeding into the low value select gate.

As an example, Figure 3-6 shows the fitted response of a large heavy-duty gas turbine using the GGOV1 model. The response shown was for a system frequency event resulting in an initial drop down to 59.7 Hz, with the frequency eventually settling down at 59.85 Hz. This response was recorded on a unit in the WECC system.

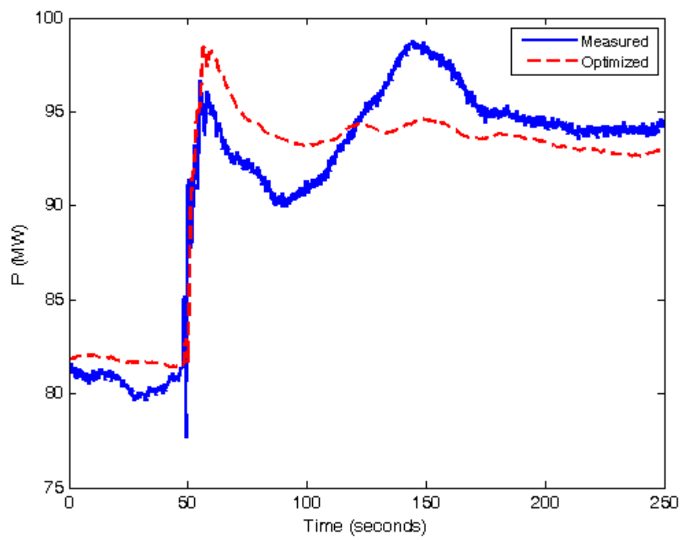


Figure 3-6: Measured response of a heavy-duty gas turbine (in a combined cycle power plant) as compared to the simulated response of the GGOV1 model [21].

3.1.2.4 CIGRE model [9]

As mentioned in the discussion of the GGOV1 model, there is no explicit attempt to capture the dependence of the maximum turbine output power on ambient conditions and shaft speed. In the case of ambient conditions, primarily ambient temperature effects, the user must accordingly adjust the Ldref parameter. In terms of the dependence of the maximum turbine output on shaft speed (system frequency), in GGOV1 the simplifying assumption, which is not very accurate, may be made of introducing a damping term in the speed (Dm).

At the same time that GGOV1 was developed, the CIGRE gas turbine model was developed [9]. This model is shown in Figure 3-7. A detailed description, with typical parameters is given in [9]. In essence, perusal of Figures 3-5 and 3-7 reveals that GGOV1 and the CIGRE model are very similar. The major difference is the inclusion of a second order transfer function to represent gas turbine dynamics, if necessary, and a polynomial fit of the maximum power output limit as a function of ambient air-temperature and shaft speed with a simple user-defined look-up table (y=F(x)). This user-defined look-up table requires data from the turbine vendor. An example curve is given in Appendix C of reference [9].

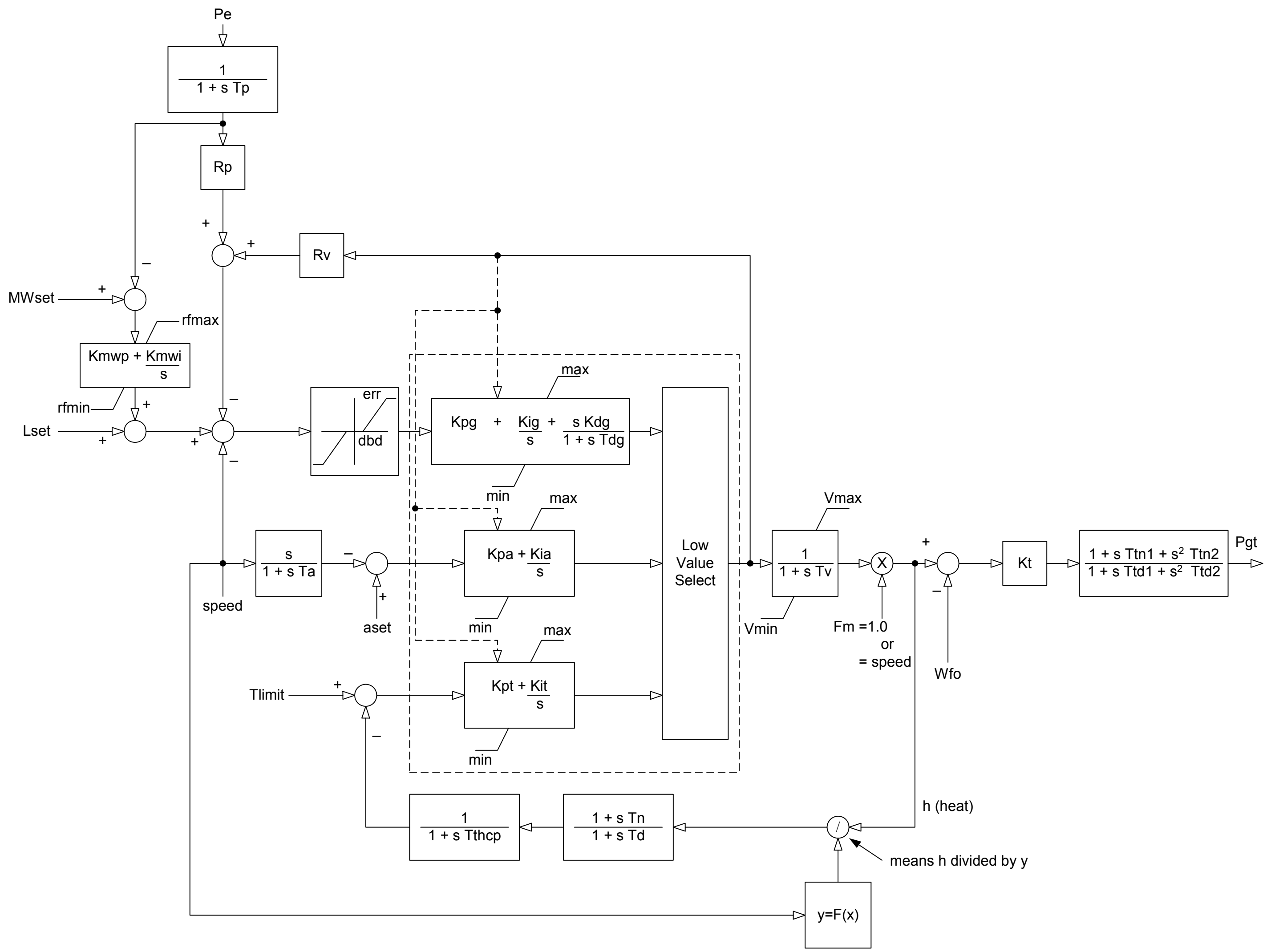


Figure 3-7: CIGRE gas turbine model [9].

3.1.2.5 Simplified but Explicit Modeling of the Ambient and Speed Dependence [16]

Starting with the detailed gas turbine model in Figure 3-2, we can simplify the model significantly by making a few assumptions, while maintaining a quite simple explicit representation of maximum turbine output dependence on ambient temperature and system frequency (shaft speed). One such example is that given in [16]. First assume that air pressure does not change. Since airflow and power changes are proportional to air pressure, and the typical diurnal (or even seasonal) air pressure variations at a given site are rarely that significant (no more than a few percent) it is therefore reasonable to neglect the pressure effects on the turbine power output. This eliminates the P_a/P_{a0} term in the algebraic expressions in Figure 3-2. Second, we are interested primarily in the maximum turbine output. Thus, the inlet guide vanes (IGV) may be assumed to be wide open. This is because for a simple cycle gas turbine, typically, the IGVs are wide-open once the unit is loaded and for combined cycle operation the IGVs are wide-open (i.e. at their maximum angle) once the unit reaches its maximum load [9]. Thus, we will ignore the IGV dynamics and this further simplifies the equations. So we are left with accounting for power variations due to ambient air temperature and turbine speed.

In [17], the manufacturer's axial compressor airflow characteristic data was used to determine the parameters of $\bar{u}(\Delta\omega_c)$, the power-speed correction factor. Such data is difficult to obtain. However, the manufacturer will typical provide the turbine maximum power output as a function of turbine speed and compressor inlet air temperature as a set of curves provided with the unit data. The power output of the turbine is directly proportional to fuel flow [17]. Furthermore, the maximum allowable fuel flow for a given steady-state condition is determined by the turbine temperature limit, which in turn is a function of the fuel air mixture and thus airflow [17], [18]. Therefore, it is fair to assume that the maximum power output of the turbine will follow the same general trend (cubic function shown in equation Figure 3-2) as the airflow equation. Thus, making the simplifying assumptions quoted above (neglecting pressure changes and IGV dynamics – i.e. IGVs wide-open) a little simple algebra leads us to the equations below [16], which describe the maximum power output of the turbine as a function of speed and ambient air temperature variations.

$$
\begin{aligned}
&m(T) = a_1 - a_2(T - T_o) \\
&u(\Delta\omega_c) = 1 + a_3\Delta\omega_c + a_4\Delta\omega_c^2 + a_5\Delta\omega_c^3 \\
&\Delta\omega_c = \omega\sqrt{\frac{273.15 + T_o}{273.15 + T}} - 1 \\
&\Delta\omega_c\big|_{\omega=1} = \sqrt{\frac{273.15 + T_o}{273.15 + T}} - 1 \\
&u_o = 1 + a_3\,\Delta\omega_c\big|_{\omega=1} - a_4\left(\Delta\omega_c\big|_{\omega=1}\right)^2 + a_5\left(\Delta\omega_c\big|_{\omega=1}\right)^3 \\
&P_{\max} = \frac{u(\Delta\omega_c)}{u_o}m(T) \\
&F(T,\omega) = \frac{P_{\max} + KtWfo}{1 + KtWfo}\frac{1}{\omega}
\end{aligned}
\qquad (2)
$$

Thus the model shown in Figure 3-8 is established. Here $u(\Delta\omega_c)$ is the power-speed correction factor, u_o is the power-speed correction factor at nominal mechanical speed, m(T) is the power-temperature correction factor, *Pmax* is the maximum turbine power for the given condition, Kt is the turbine gain (see Figure 3-8), W_{fo} is the full-speed no-load fuel flow (see Figure 3-8), T, which is a user entered value, is the current ambient inlet air temperature (in degrees C) and T_o is the temperature under which the turbine capability curve used to fit $u(\Delta\omega_c)$ is defined – typically ISO[8], 15^{o}C. The coefficients a_1 to a_5 are determined by using standard polynomial curve fitting techniques for fitting the functions m(T) and $u(\Delta\omega_c)$ to the manufactured supplied power versus temperature and power versus speed curves, respectively. In [16] this approach was found to provide a curve fit that predicted maximum power to an accuracy of within 1% of the manufacturer supplied numbers for speed variations of up to +/- 2% and temperatures ranging between 5^{o}C to 40^{o}C.

What is shown in Figure 3-8 as the function y=F(x) is actually implementing the function shown above in equation (2). The temperature is assumed to be a user entered parameter (T), while the input to the function is the speed variable.

Important Note: All the models and techniques presented here are intended primarily for the purpose of power system simulations. It is most certainly not claimed, nor should it be assumed, that these models represent the actual turbine controls or that they can be used to assess turbine performance for design, economic or energy/efficiency calculations. In some cases, determination of primary and secondary response capability of a power plant as part of grid-code compliance may also require vendor specific functionality or models not available in the models described above.

[8] Note: ISO conditions are defined as ambient conditions for which the ambient temperature is 15^{o}C, the relative humidity is 60% humidity and sea-level ambient air pressure.

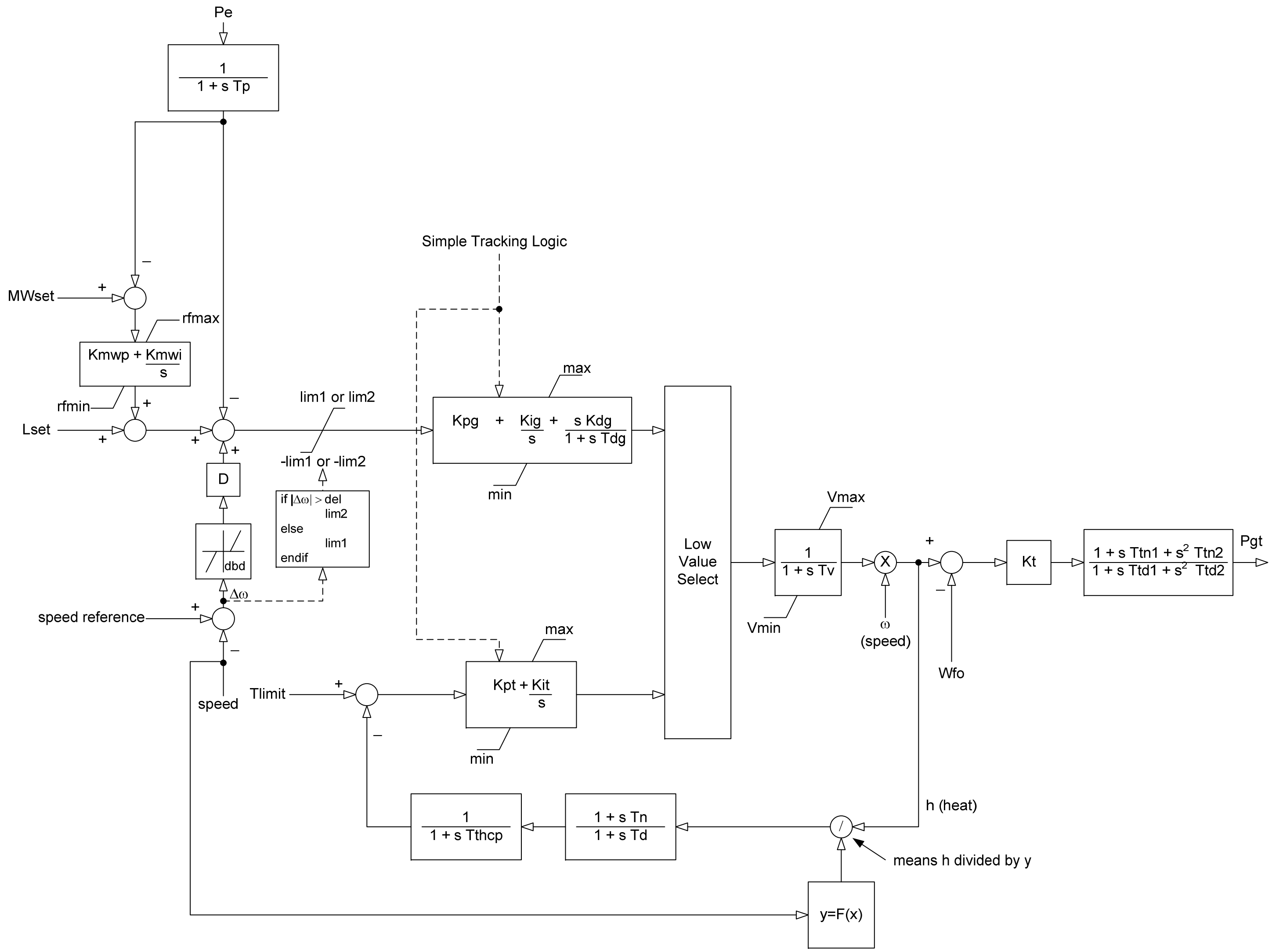


Figure 3-8: Simplified gas turbine model (GT1) from [16].

3.1.2.6 Vendor Specific Models

Vendor specific models are also available. For example, going back to the simplified thermodynamic model of the gas turbine in Figure 3-2, by combining it with the turbine controls the so-called *gegt1* model is developed, which is available in the GE PSLF® program. This model, originally developed in [17], has been updated to the latest control strategy for the current GE fleet. Experience has shown that this model is able to achieve quite a good match between field measured and simulated response in the gas turbine power, exhaust temperature, compressor pressure ratio and inlet-guide vane response. This model is, however, quite complex and not necessarily appropriate for large interconnected power system simulation studies. Appendix D provides a description of the model.

Another example of a more detailed vendor specific model is provided in Appendix E for some of Alstom's fleet of turbines.

These models are still quite simplified compared to the actual turbine controls.

As discussed in detail in section 3.1.1, the fuel and airflow combined basically affect the power of the gas turbine. For some heavy-duty gas turbine designs (e.g. Alstom's GT24 and GT26), the fuel is directed into two combustors, namely the EV (environmental) and SEV (sequential environmental) burners. The capacity of the EV burners are 60% that of the SEV burners. In the load and temperature controllers these fuel flows are regulated separately by their own PI controllers. As such, for such sequential dual-stage combustor design turbines the simple generic models such as GGOV1 or the CIGRE model will not be able to adequately represent the gas turbine dynamic behavior, particularly in cases where the grid-code requires that the model be used to assess the performance of a power plant and to grade it for primary and secondary reserves payments. Where models are to be used for such detailed performance requirements, vendor specific models may be required and the turbine manufacturer should be consulted.

3.2 Combined Cycle Power Plants

A good reference on combined cycle power plants (CCPP) and their models is the CIGRE document [9]. In this document we will not discuss the operational characteristics of CCPP nor give a detailed overview; this can be found in [9] and other such documents.

This section will provide a brief overview of combined cycle power plants and the recently developed models for CCPP. The last section of this chapter provides brief modeling guidelines and recommendations for modeling combined cycle power plants in power system studies.

3.2.1 Brief Overview of CCPP

A combined cycle power plant (CCPP) consists of at least one gas turbine (GT), a steam turbine (ST), a heat-recovery steam generator (HRSG), and an electric generator. A variety of combinations exist, employing multiple gas turbines, HRSGs, and generators in several possible configurations. If the gas and steam turbines are on separate mechanical shafts, then the plant is referred to as a multi-shaft CCPP. If the gas turbine, steam turbine and electric generator are all connected in tandem on a single mechanical shaft then the plant is referred to as a single-shaft CCPP.

For multi-shaft CCPP the gas turbine model is the same as that discussed in the previous section. The output of the gas turbine model is then connected to the HRSG and steam turbine model. In its simplest form, the HRSG and steam turbine model is as shown in Figure 3-9, from [9].

For a single-shaft CCPP the models in Figure 3-7 and 3-9 may be combined as discussed in [9] to form a single-shaft CCPP model.

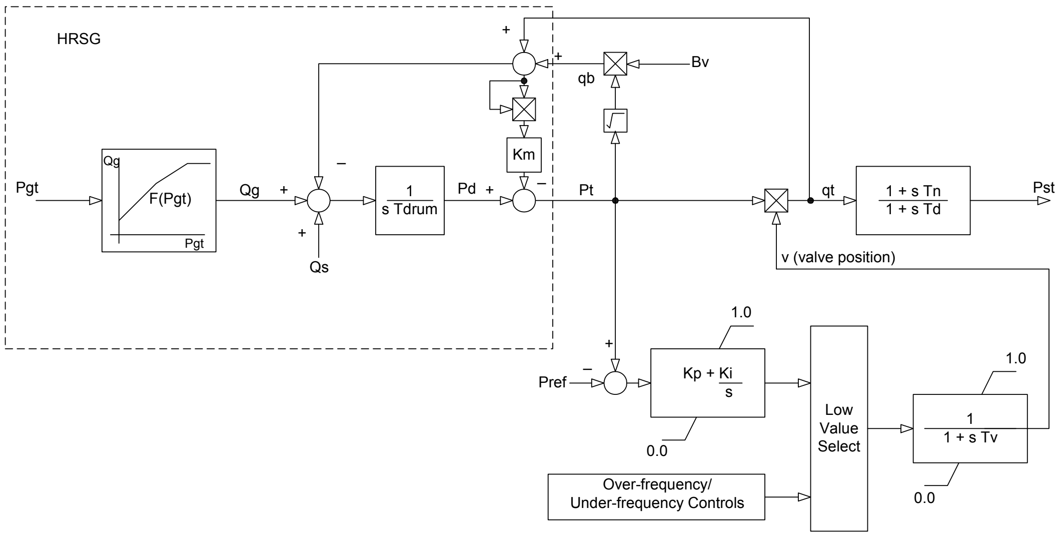


Figure 3-9: Generic HRSG/ST model. [9].

The parameters and a description of the model in Figure 3-9 can be found in detail in [9].

Appendix F provides an example detailed account of CCPP modeling for a given vendor with a list of parameters.

The next section gives an example description of how these CCPP models may be implemented and used in power system simulations tools.

3.2.2 Implementation of the CIGRE HRSG and ST model in ERCOT

Combined cycle power plants became popular in the 1990s, and became a significant part of the generation mix in many regions [9, 22]. Initially, there were no adequate dynamic models for these plants combining both the GT, HRSG, and ST portions into an integrated model. The publication of the CIGRÉ models in 2003 [9] provided a "unified" model in block diagram form, particularly for the representation of the HRSG and the steam turbine controls. ERCOT contracted Siemens PTI to incorporate these models in Siemens PIT PSS®E simulation program. These models have thus been available as user-written models since 2007. However, as of the publication date of this document, these models were not yet part of the program's library of standard simulation models.

Here we present a brief discussion of this software implementation of the models, since the process highlights some key important aspects of CCPP modeling.

The implementation of the CIGRE models resulted in the creation of 3 dynamic user-written models:

- UCBGT – a generic combustion turbine model
- UHRSG – a generic heat recovery steam generator and steam turbine model
- UCCPSS – a generic single-shaft combined cycle turbine model

In addition, an auxiliary program call PARCC was developed to adjust the dispatch of individual units in the CCPP plant in the power flow case to known operating points. PARCC does not change the net output of the CCPP plant.

The power output of the steam turbine (ST) in a CCPP is a function of the available heat in the HRSG, which is related to the dispatch (MW output) of the GTs in the CCPP. The relationship between the power output of the GTs and the maximum power output of the ST in a CCPP must be accounted for in a dynamic simulation, but is often misrepresented during power flow development. In addition, maintaining the proper relationship between the dispatch of the GTs and the power output of the ST of a given CCPP can be tedious while exploring different system or plant dispatch scenarios. Therefore, an auxiliary program called PARCC was developed to facilitate maintaining the proper relationship between ST and GTs. This relationship is provided, for different operating levels and even different seasons, by the user in a spreadsheet form (e.g. as shown in Table 3-1 for a multi-shaft CCPP). The PARCC program calculates the total power output of the CCPP, as given in a power flow case, and readjusts the dispatches of the individual units in the CCPP based on the characteristics of the CCPP as described in the associated spreadsheet, while maintaining the total power output of the CCPP.

The PARCC also allows for gross or net generation representation in the power flow case; multiple spreadsheets can be developed to account for seasonal variations in output; and enhancements such as duct firing and inlet-air cooling can be included. Ideally, the data to populate the spreadsheets should come from the facility owner and/or manufacturer and carefully reviewed for consistency. The output of individual machines is often monitored for operational purposes, and examining historical output data can often yield sufficient information to populate the spreadsheet.

The UCBGT model is an implementation of the CIGRE gas turbine model described above. For applicable assumptions and attributes see [9]. This model can be used as a governor model for a simple-cycle GT, or for one or more GTs that are part of a CCPP. Each GT in a multi-shaft CCPP must be modeled using UCBGT, or another appropriate GT model. Figure 3-10 shows a plot comparing the response of the UCBGT (CIGRE model) and GGOV1 models.

Table 3-1: Example of input data for GT/ST MW relationship.

	GT 1	GT 2	ST	
Summer	Gross MW	Gross MW	Gross MW	AUX Load
Dispatch 1	40	0	0	4
Dispatch 2	0	40	0	4
Dispatch 3	58	0	11	5
Dispatch 4	0	58	11	5
Dispatch 5	74	0	25	6
Dispatch 6	0	74	25	6
Dispatch 7	51	51	34	6
Dispatch 8	80	80	60	8

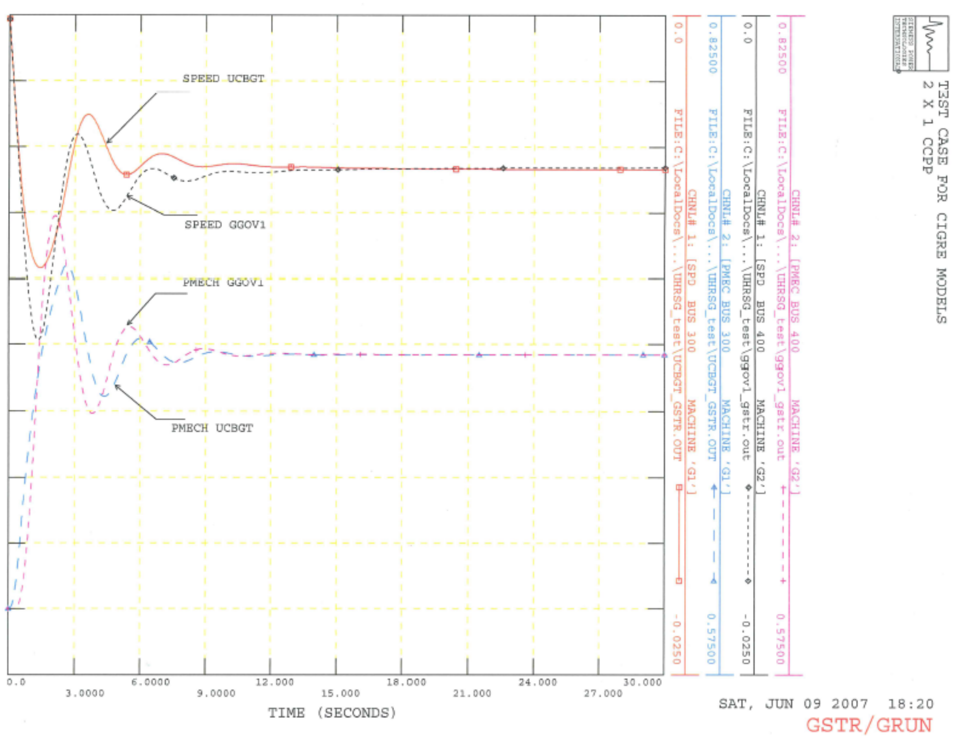


Figure 3-10: comparing the response of GT models UCBGT and GGOV1 [23].

The UHRSG model is an implementation of the CIGRE generic heat recovery steam generator and steam turbine model shown in Figure 3-9. The model allows up to 6 GTs to be connected to a common HRSG. The total power output of the gas turbines is related to heat through a look-up table, and the steam production is proportional to the total heat from the exhaust of the combustion turbines plus the heat provided by any supplemental firing, if used. Figure 3-11 shows the model as implemented. Comparing this to Figure 3-9 we see the addition of the weighting coefficients and summing of the GT outputs to develop the total heat going into the HRSG model.

The look-up table, relating the total power output of the GTs in the CCPP to the available heat (and thus steam flow) for the ST, should be consistent with the generation dispatch in the power flow case that provides the initial condition for the dynamic simulation. In other words, the look-up table in the UHRSG model should also be consistent with the CCPP characteristic, as expressed in the spreadsheets used as input to the auxiliary program PARCC. Otherwise the dynamic simulation will not be properly initialized (will not be in steady state) and the power output of the steam turbine would drift away to the set point determined by the data in the look-up table.

The auxiliary program PARCC is used to convert the CCPP dispatch scenarios and create the look-up table for the UHRSG model. The PARCC program is also used to enforce such dispatch scenarios to the power flow case, thus allowing the proper initialization of the dynamic simulation.

An additional set-point (Q_{imb}) has also been implemented, corresponding to the heat imbalance between the available heat calculated (based on the look-up table) from the power output of the combustion turbines and the heat (and steam flow) required to maintain the power output of the steam turbine, as given in the power flow case. This heat imbalance is calculated during the initialization of the UHRSG model and is added to the heat produced by supplemental firing. This ensures a correct initialization of the dynamic simulation, even when the dispatch of the generation units in the CCPP as expressed in the power flow case is inconsistent with the data in the look-up table. This heat imbalance could be positive or negative, and it is necessary to modify the power flow case and/or the look-up table in the UHRSG model, as described above, to bring this initialization imbalance back to zero.

While the GTs associated with this model can be any of several from the software library, care should be exercised to ensure the turbine rating base is consistent throughout, including the look-up table. The steam model is usually operated in sliding pressure mode, but can be in pressure control mode at light loading. In addition to the heat from the GTs, the response of the steam turbine to system transients will be influenced by the HRSG drum time constant. While typical values can be used, actual values from test or examination of historical data would be preferred.

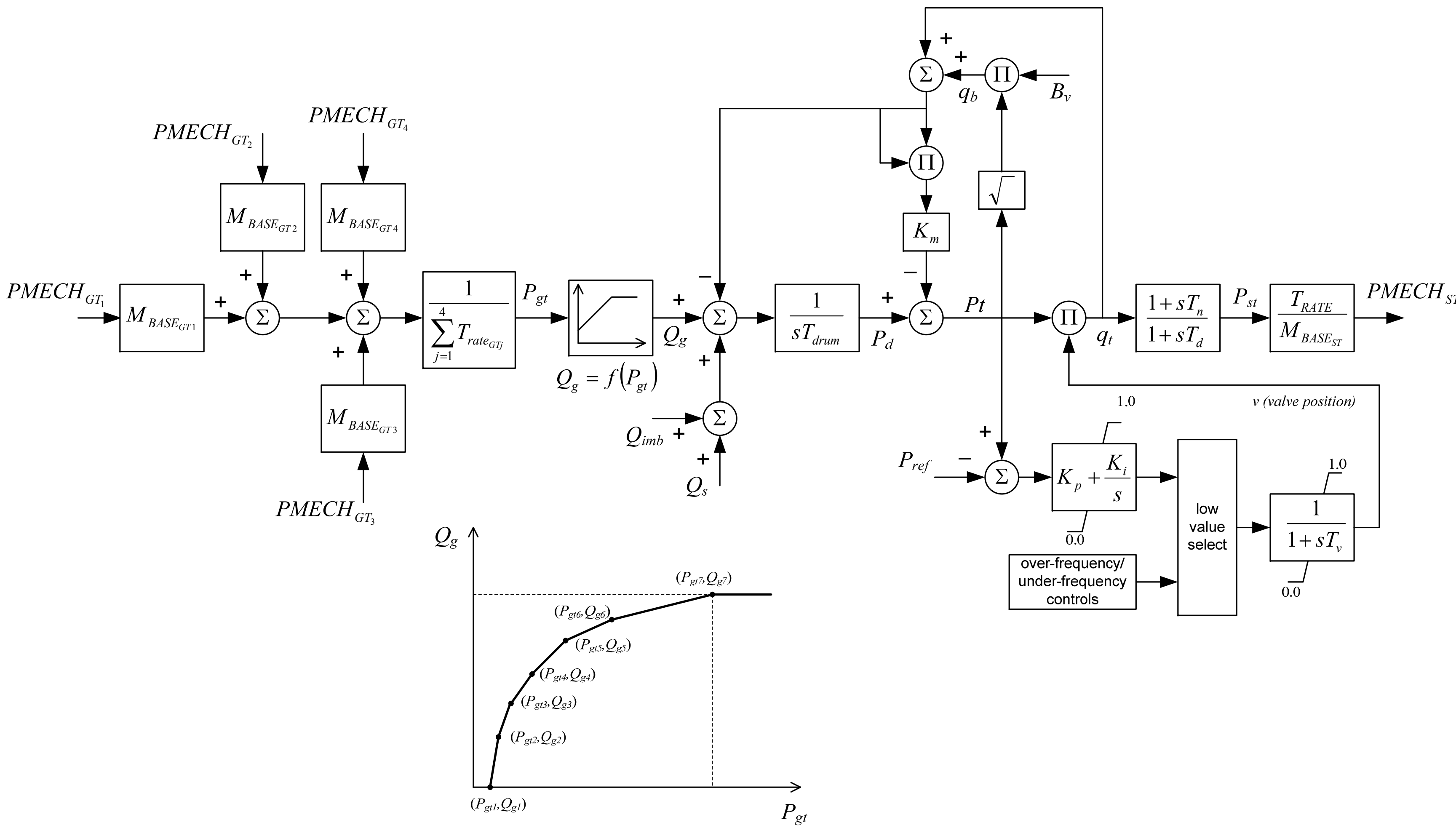


Figure 3-11: Siemens PTI PSS®E implementation of the CIGRE HRSG and ST model [23].

The UCBGT and UHRSG models are combined to create the UCCPSS generic single-shaft combined cycle turbine model. The model automatically calculates the power distribution between GT and ST sections via the dispatch spreadsheet and look-up table. However, it should be noted that determining the proportion of total output by the GT and ST sections would require detailed heat balance analysis. Absent such an analysis, the typical look-up table in [9] may be used.

ERCOT was motivated, in part, to sponsor implementation of the CIGRE model because of two frequency events where actual system response was significantly worse (larger frequency decline) than simulations suggested. Lack of an adequate CCPP model was one of several factors identified [22, 24]. Figure 3-12 illustrates an improvement in simulated system response using the CCPP models compared to the "traditional" modeling approach used in ERCOT. In this case the system frequency drops lower using the CCPP models, as actual measured system response would suggest.

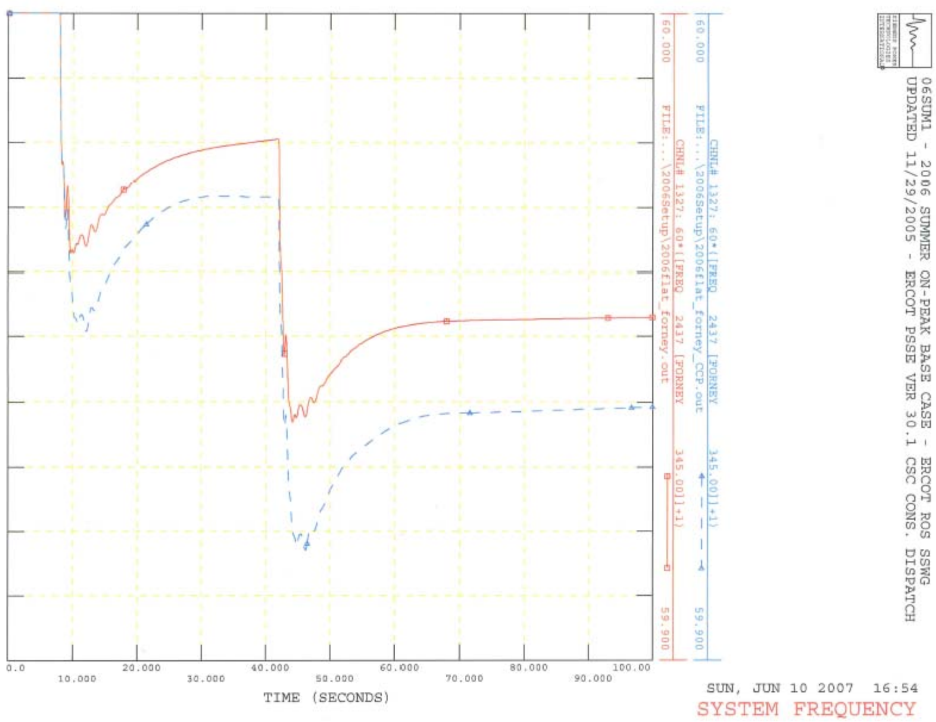


Figure 3-12: Illustrates an improvement in simulated system response using the CCPP models [23]

3.3 Modeling Guidelines and Summary

This chapter has presented several existing models for gas turbines and for components of combined cycle power plants. The GGOV1 model is available in the common commercial power system simulation tools and is the most commonly used gas turbine model in the WECC. It is one of the recommended models for use in typical planning studies, particularly for large interconnected systems. The CIGRE model discussed in this document is also a recommended model for system planning studies. As discussed in the introduction of this report (section 1.1), some regions have specific grid-code requirements, in such regions clearly vendor or grid-code recommended models should be used if available. For studies related to system islanding or scenarios that result in large deviations in frequency, i.e. more than several percent change, the accuracy of the GGOV1 and CIGRE model are limited and vendor specific models may be needed. In general, a simple model is better than no model.

The most important aspects of the gas turbine model are still as pointed out in previous work [5, 9, 17, 18, 25], namely:

1. Check all turbine-governor models so that they are properly modeled based on the MW capability of the turbine and not generator MVA base, in particular the maximum power and droop.
2. Represent the outer-loop MW controller, where such response is observed. The best way to capture this is by on-line disturbance monitoring. This controller is part of the GGOV1 and CIGRE models.
3. Be cognizant of the ambient temperature dependence of the gas turbine maximum output [18], and appropriately adjust the load limit of the model for seasonal studies.
4. Be cognizant of the dependence of the gas turbine maximum output as a function of large frequency variations [9, 17, 18]. This may be relevant in some studies. For most studies where the system frequency deviation is not more than +/- 1%, the change may be quite small and can be neglected – see [9, 17] for examples of output variation as a function of temperature and frequency.

The GGOV1 model, which is the model most extensively used these days for modeling gas turbines, has a significant number of parameters. Appendix C presents an example set of typical parameters. Many of these parameters can be fitted from either disturbance data recordings or field tests, and this is highly recommended. Other parameters of the model, such as the turbine rating, control mode (e.g. electrical power feedback or not) can be identified from the vendor documentation or a perusal of the controls. Some of the parameters, such as the maximum/minimum controller error and acceleration control settings, cannot be easily fitted based on tests or disturbance data. These should be set to vendor recommended values, or typical values shown here if better information is not available.

In a current typical planning case for WECC, there are roughly 3000 synchronous generator models. The GGOV1 model is used for modeling gas turbines almost exclusively; there are 992 instances of the GGOV1 model[9]. In addition, roughly 30% of the units do not have a turbine-governor model at all. A significant portion of

[9] It should be noted that in previous years, in WECC, some steam turbines were also modeled using GGOV1. This is not a recommended practice as it results in a confusion of the various turbine types in the planning database.

these are steam turbines in multi-shaft combined cycle plants, pumped hydro, distributed generation, or blocked governors and thus deliberately not modeled. These considerations aside, many of these units may still be missing an adequate model of the turbine-governor. Considering a typical Eastern Interconnection planning case, there are more than 7000 synchronous generators in the case. Roughly 50% (one in every two) of generators do not have a turbine-governor model. This is a significant problem. Considering the gas turbine models, the rough statistics are as follows:

- GAST 412
- GAST2A 120
- GGOV1 133

The above discussion is simply intended to illustrate that there is a significant gap, particularly for the Eastern Interconnection, between the present level of modeling and what could be achieved by using simple models such as GGOV1 or the CIGRE model. Namely, the vast number of units without an adequate turbine-governor model, or proper representation of the turbine rating, or the mode of turbine-governor operations (i.e. whether the turbine is under (i) droop-control, or (ii) has an outer-loop MW controller, or (iii) base-loaded and thus unable to respond).

While this document has identified recommended turbine-governor models for gas turbines (GGOV1 or CIGRE model) and combined cycle power plants (CIGRE model), it is recognized that many governors are currently modeled using what may be considered as obsolete or legacy models (e.g. GAST, GAST2A, etc.). In many cases these models and associated data were provided to the owner by the equipment vendor when the unit was commission many years ago. It is also recognized some owners may not have the resources or expertise to convert such obsolete or legacy models to newer models. It is not the intention of this task force report to force conversion of such obsolete or legacy models to recommended models. Use of the best models available is the preferred course of action. For new plants or plants that have undergone an upgrade, a validated turbine-governor model should be requested from the vendor.

4. Hydro Turbines

4.1 Modeling Hydro Turbines

When contemplating models of the turbine-governors of a hydro plant, it is best to consider the governor and turbine models separately (Figure 4-1):

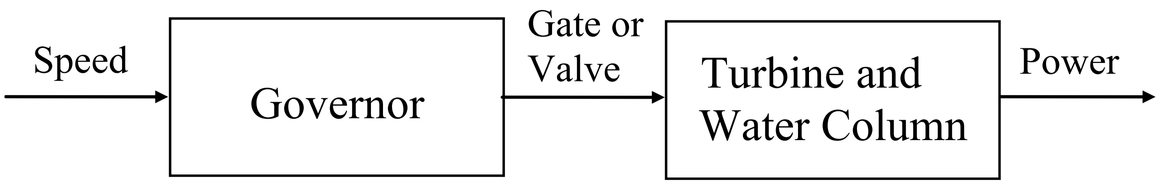


Figure 4-1: General Governor-Turbine Block Diagram

In the long period of time that has passed since an IEEE task force has provided general recommendations for turbine-governor models for power system studies [11], the requirements for representing hydro plants has changed significantly. Power system studies in that era were focused on transient stability studies primarily limited to the first or second swing of a generator after a fault or switching operation. The response of a hydro prime mover control system has a minimal impact during the time frame of these studies; consequently the computer model representation was as simplistic as possible to reduce the demand on very limited computer resources. The simplified model of a hydro governor consisted of a transfer function of two poles and a single zero for the governor and a single pole and single zero for the water column (Figure 4-2):

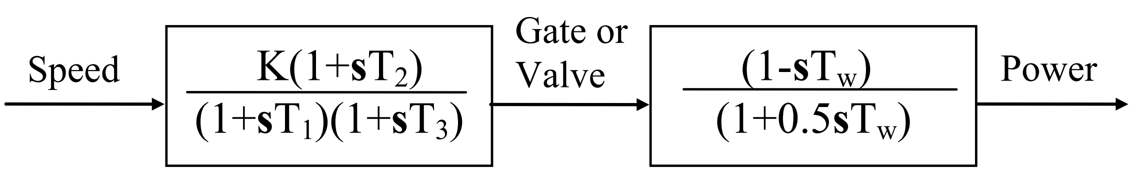


Figure 4-2: Simple hydro turbine model.

This simple representation can only provide accuracy within a very limited bandwidth, and is therefore inadequate for general use. For the most part, use of this model has disappeared in large system databases, but vestiges of these models for older plants are still occasionally found. This model, sometimes referred to as the IEEE Type 2 model is obsolete, and inclusion of this model structure in commercial programs should be discontinued.

In 1992, an IEEE working group provided updated recommendations for hydro plant models [26] which are accurate for a wide variety of power system studies. This paper is an excellent reference on the subject, particularly in its coverage of nonlinear behavior of a water column. The depth and width of coverage cannot be duplicated herein, and it is recommended that it be reviewed, particularly if detailed models of facilities with surge tanks or multiple penstocks with manifolds are desired. A key conclusion made in the paper is that computer resources are no longer a reason to sacrifice model accuracy. However, for many hydro facilities, it is difficult to assess how much detail is warranted for a hydraulic system model. For most power system studies involving large system representation, very detailed models may not be necessary for most plants.

4.2 Hydro Governors

Computer model representation of the governor, i.e., the control system that moves gates, valves, or blade positions is straightforward and appropriate models for several types of controller designs have been in use for quite some time. Figures 4-3 to 4-6 show typical control designs employed in hydro plants.

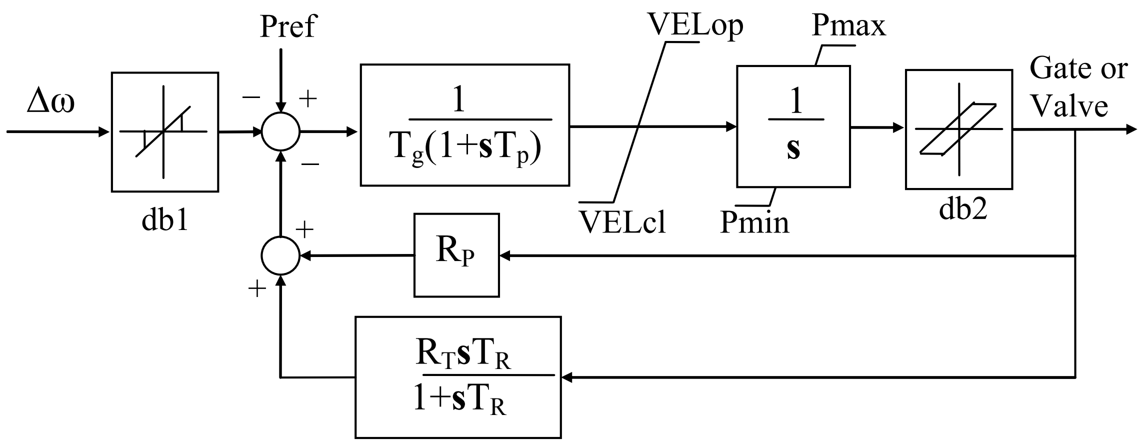


Figure 4-3: Mechanical hydraulic governor.

Figure 4-3 represents a traditional mechanical-hydraulic governor, consisting of a hydraulic pilot valve, main servo, and dashpot temporary droop. Figures 4-4 to 4-6 represent more modern electronic or digital implementations where response is tuned using PID, double derivative, or lead-lag compensation design. A common variation in the PID design is to exclude the droop feedback from the derivative term. In practice, the derivative term is often not used.

Since the primary use of these models in power system studies is to represent the unit response to small changes in system frequency, these governor models will be adequate for most present equipment, as the control design for frequency regulation is very straightforward and mature, with the simple PID structure the most commonly found. However, governors have other control objectives, such as load set-point response, which influences overall governor designs and results in unique variations in control structures among manufacturers and models. For example, Figure 4-7 depicts a variation in the PID design of Figure 4-4.

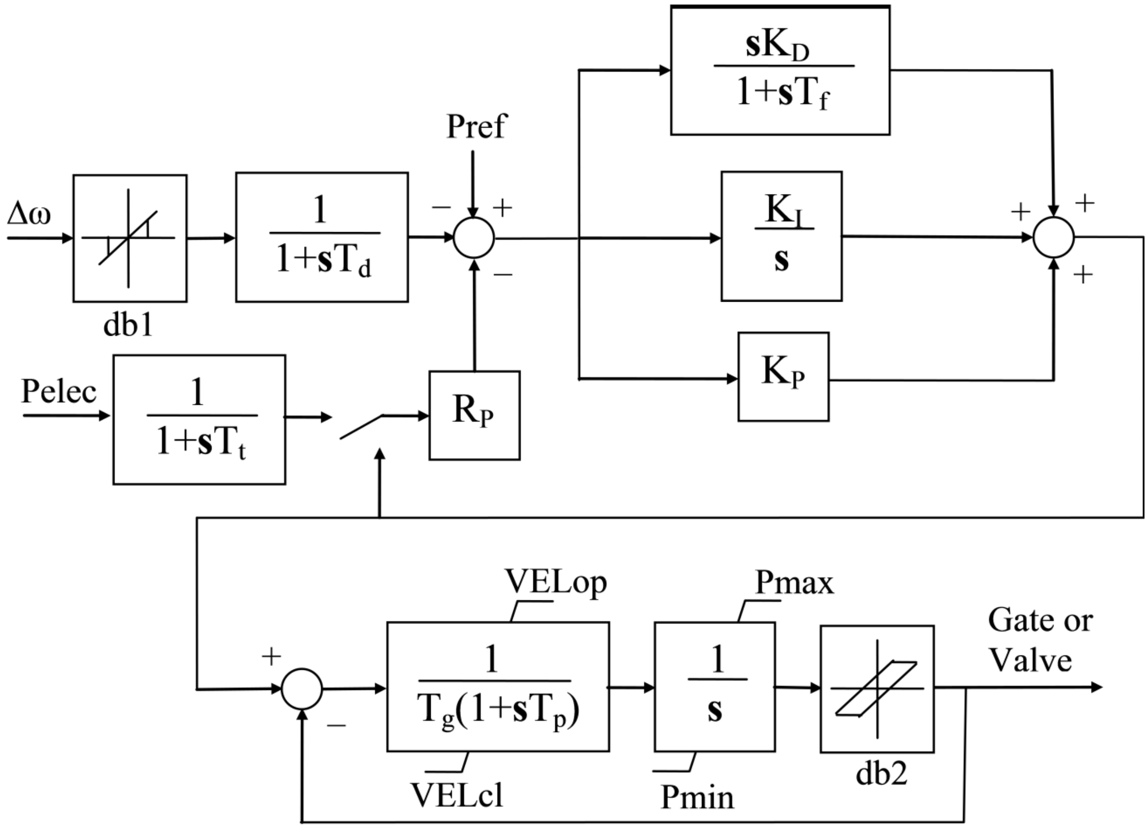


Figure 4-4: PID governor.

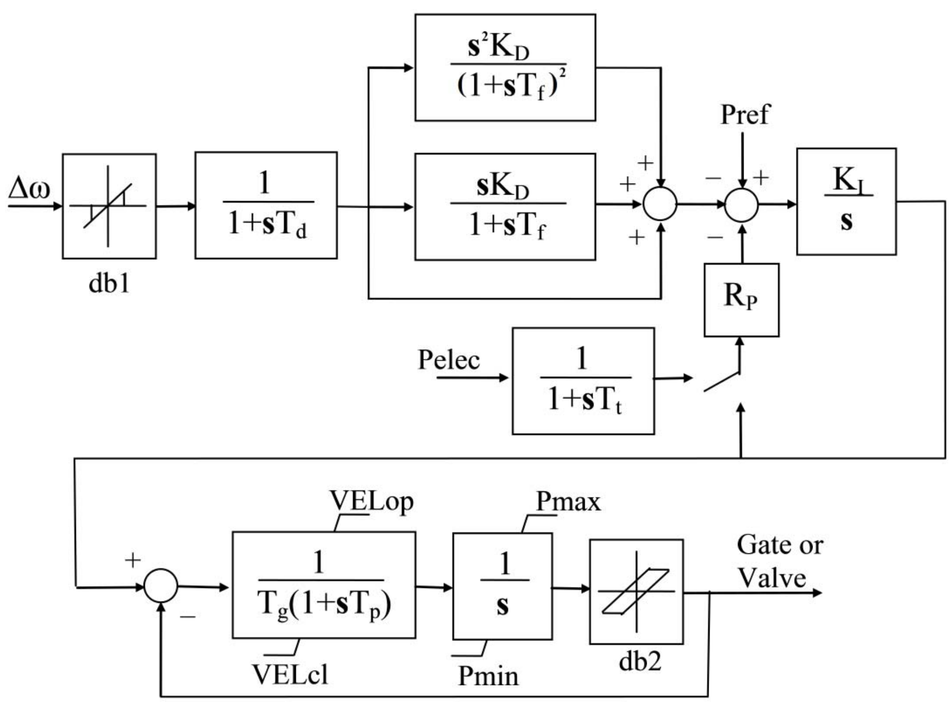


Figure 4-5: Double-derivative governor.

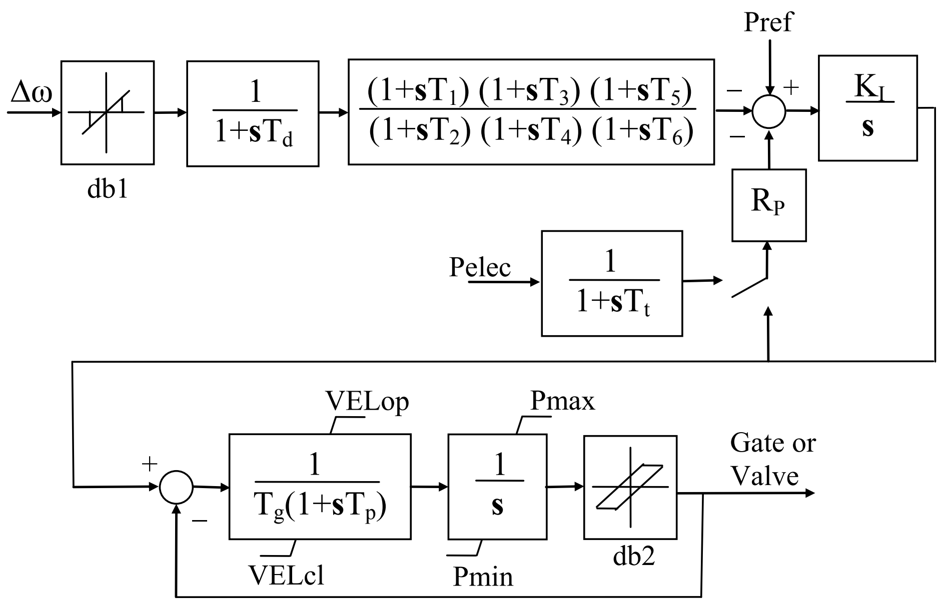


Figure 4-6: Lead-lag governor.

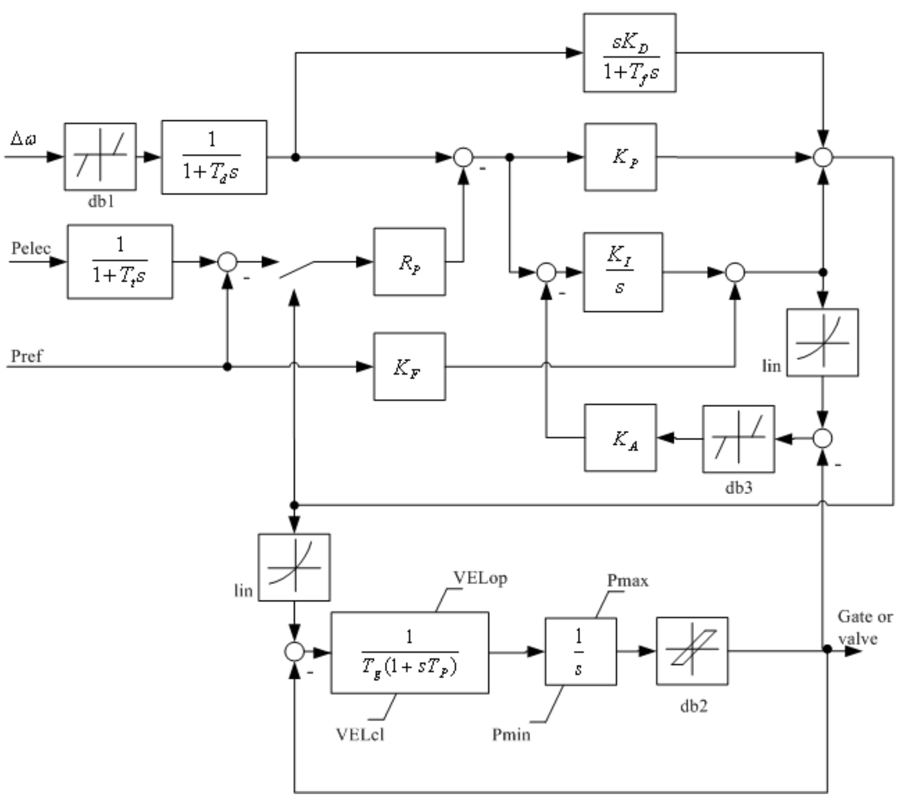


Figure 4-7: PID governor with set-point control, anti-windup, and nonlinear compensation. For this figure all signals entering a summing junction are positive unless otherwise noted as being subtracted (negative).

The modifications to the basic design are primarily to improve upon the practical drawbacks to integral compensation, particularly when changing the load via P_{ref}, and to compensate for some nonlinearity, discussed below. In most power system studies, manipulation of P_{ref} is not necessary. If it desired, it would be most likely be derived from slower, outer control loops (e.g., unit load controllers, AGC, etc.) Unless a large transient response is being delivered as a load reference change, the effects of such detail will not impact system studies. If modeling such response is necessary, manufacturer specific models such as this may be required. This need appears rare at this time. In this particular example, the compensation for system nonlinearity may not be necessary in the model if the nonlinearity is omitted from an otherwise linear model. If such detail is included, then detailed representation of the nonlinearity must be present in the model to obtain a correct response.

4.3 Hydro Water Column

The second stage of the model relates gate or valve position to mechanical power delivered to the generator shaft. The simplified model used in computer programs has traditionally been the idealized, linear turbine representation (Figure 4-8), which depends on only one parameter, the water starting time constant, T_W:

$$\text{Gate or Valve} \longrightarrow \boxed{\frac{(1-\mathbf{s}T_w)}{(1+0.5\mathbf{s}T_w)}} \longrightarrow \text{Power}$$

Figure 4-8: Linear Turbine-Water Column Model

At rated head and flow, T_W is calculated as

$$T_W = \left(\frac{L}{Ag}\right)\left(\frac{q_{base}}{h_{base}}\right)$$

Where

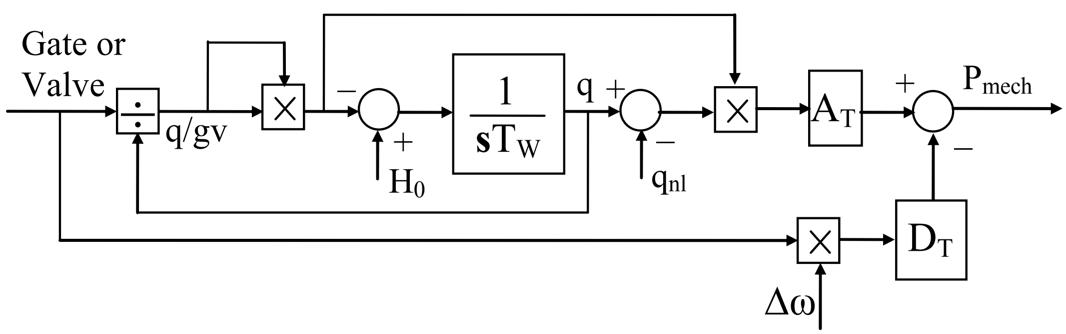
A	=	Penstock area, m^2
L	=	Penstock length, m
g	=	Acceleration due to gravity, m/sec^2
q	=	Flow of water through turbine, m^3/sec
h	=	Operating head at turbine admission, m

This model is a simplification of the more generalized version (Figure 4-9),

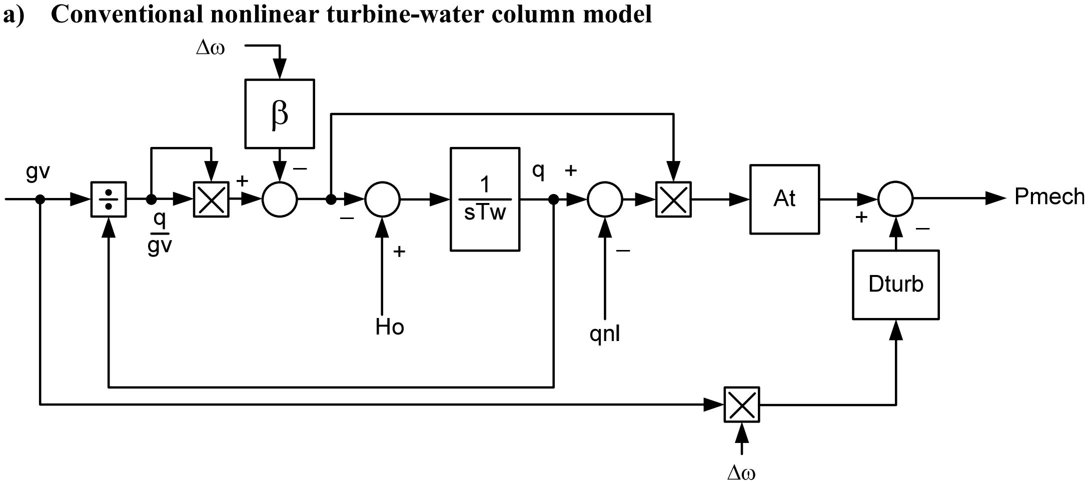
$$\text{Gate or Valve} \longrightarrow \boxed{\frac{a_{23}[1+(a_{11}-\dfrac{a_{13}a_{21}}{a_{23}})\,\mathbf{s}T_W]}{1+a_{11}\mathbf{s}T_W}} \longrightarrow \text{Power}$$

Figure 4-9: General Form of Idealized, Linear Model

where the constants a_{11}, a_{21}, a_{13}, a_{23} are partial derivatives of flow and torque based on head and gate position [27]. The simplified model results by setting a_{11}, a_{21}, a_{13}, a_{23} to 0.5, 1.5, 1, 1, respectively which assumes lossless operation at 100 percent speed and flow. The obvious disadvantage of this model is that the parameters require recalculation for simulating conditions for different flow conditions. Figure 4-10 a) is a more flexible, nonlinear model of the turbine and water column, which accounts for the effects of varying flow on the effective water starting time. This model can also incorporate the change in gains due to off nominal head variations (H_0), the offset for speed-no-load flow (q_{nl}), and the effective reduction of gate stroke (A_T). It also includes a turbine self-regulation gain term (D_T) which depends on both speed variation ($\Delta\omega$) and flow. Note that all per unit values in the governor/turbine models are normally based on the rated power output of the turbine and not on the generator model MVA base. Otherwise the gain A_T is used to scale the turbine/governor model to the generator base. Figure 4-10 b) shows a slightly modified version of this model where a water pressure coefficient (β) is added to the model [28]; when beta is negative (Francis turbine) the turbine is stable and when beta is positive (Kaplan turbine) the turbine is unstable without a controller.



a) Conventional nonlinear turbine-water column model



b) Modified nonlinear turbine-water column model

Figure 4-10: Nonlinear Turbine-Water Column Model (Inelastic)

Figure 4-11 shows the frequency response of the nonlinear model for full load, 75% load, 50% load and 25% load, for a penstock with a water starting time of 1.0 second at full load. Compared to the linear model, varying load with the nonlinear model results in large variations in response spanning more than a decade in frequency. If T_W is not recalculated for changes in power operating point, the linear model can lead to vastly different responses if a generating unit is modeled at less than full load. Since this is the norm in large system databases, the linear model should be replaced with a nonlinear, flow dependent model.

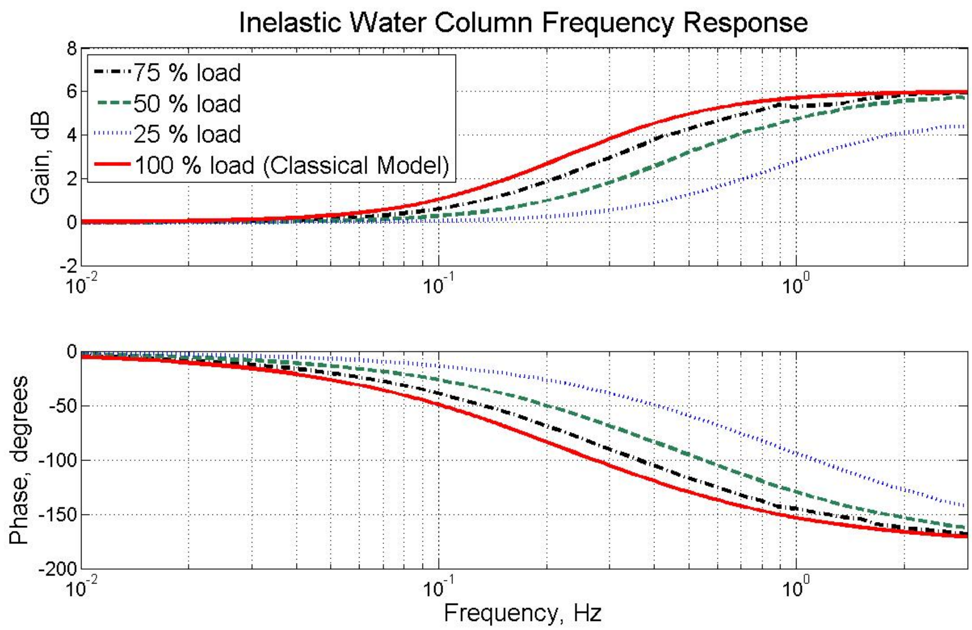


Figure 4-11: Effect of Flow on Inelastic Model

A more accurate model (Figure 4-12) for the water column is a travelling wave model which accounts for the water hammer effects due to elasticity of the water and the penstock. Additional constants used in this model are the penstock surge impedance (Z_0) and the wave travel time (T_e).

$$Z_0 = \left(\frac{1}{\sqrt{g \propto}}\right)\left(\frac{q_{base}}{h_{base}}\right)$$

$$T_e = \left(\frac{L}{a}\right)$$

where

$$a = \sqrt{\frac{g}{\propto}}$$

$$\propto = \rho g\left(\frac{1}{K} + \frac{D}{fE}\right)$$

and

ρ = Density of water

K	=	Bulk modulus of water
D	=	Internal diameter of penstock
f	=	Wall thickness of penstock
E	=	Young's modulus of pipe wall material

As compared to the inelastic model, the frequency response shows the true frequency response characteristic where the phase lag is not limited to 180 degrees, but continues to increase with frequency (Figure 4-13).

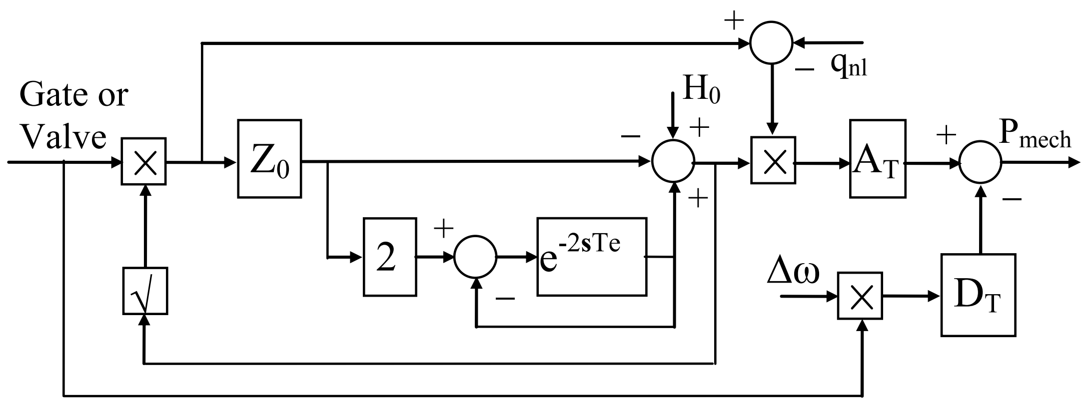


Figure 4-12: Travelling Wave Model of Water Column

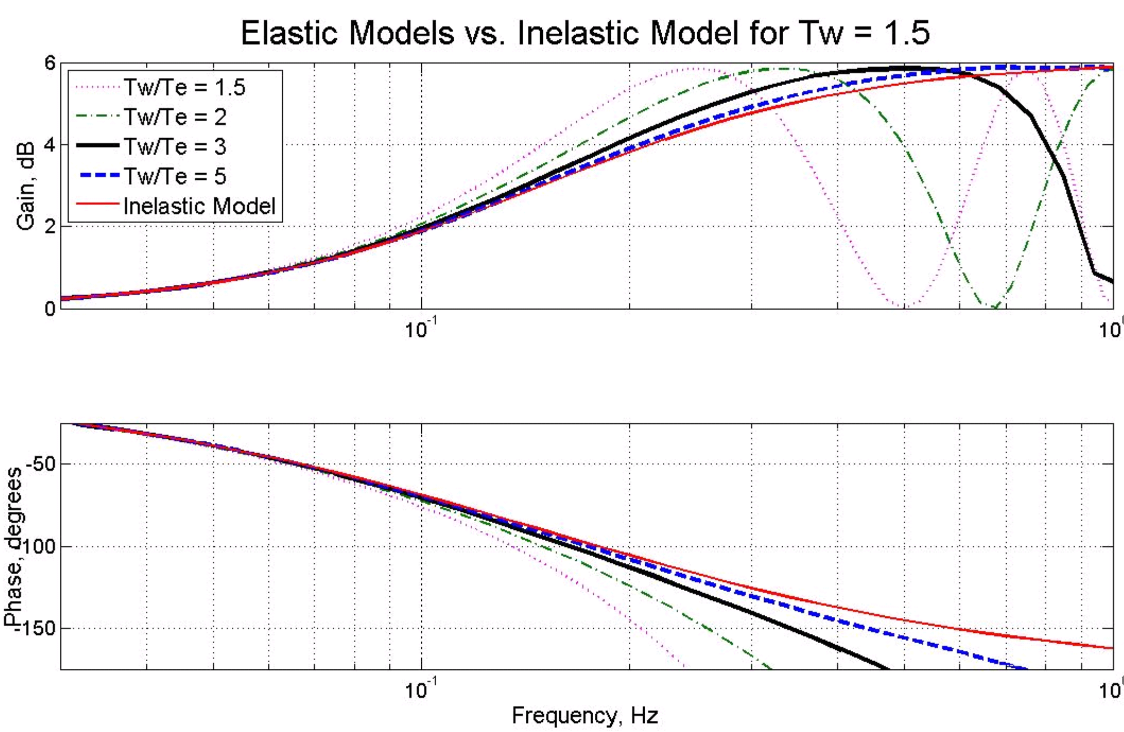


Figure 4-13: Elastic Water Column Model

Although the traveling wave model is the most accurate dynamic representation of a water column and penstock, it may or may not impact the results of studies for a particular facility. The need for a traveling wave model depends on the design of the hydro plant and the tuning of the governor. If a governor is set for a relatively fast response, e.g., derivatives are used, then differences between the elastic and inelastic penstock models may make a difference in some studies, especially those involving inter-area system oscillations. A lower ratio between the water time constant (T_W) and the wave travel time (T_e) will also increase the impact of the elastic model. Therefore, the need for the elastic penstock model must be evaluated on a case by case basis.

4.4 System Frequency Regulation Studies

In recent years, the desire to study frequency regulation in large systems has increased. In order to properly account for the short and long term response of a plant, some refinements to turbine-governor modeling were identified [25, 29]. An important effect of a hydro unit that impacts this response is the nonlinear gain relationship between power output and flow due to turbine efficiency. This variable gain can be characterized by measuring output power as a function of gate position, which results in a curve such as that shown below in Figure 4-14.

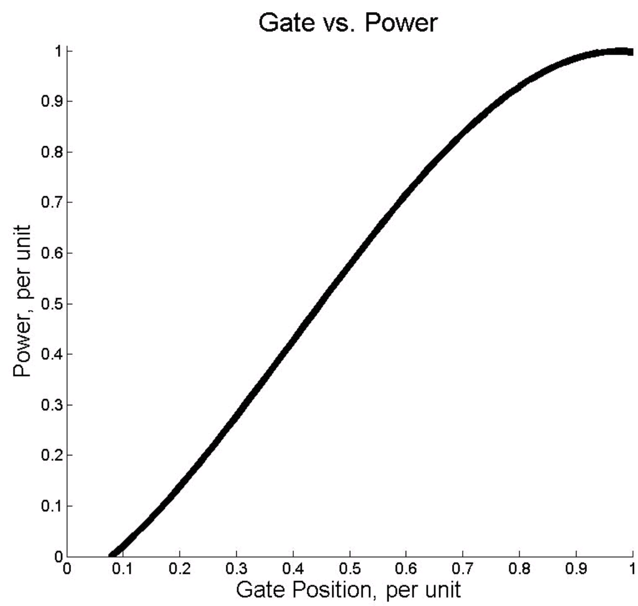


Figure 4-14: Gate Position vs. Power Output

The slope of the tangent at the operating gate position determines the effective gain modification and can be quite pronounced in some cases, varying the total effective gain of the turbine-governor from 50 to 200 percent. It is especially important to capture this effect in systems where the droop signal is not derived from the electrical power output, which is commonly the case. This effect can significantly impact the results of frequency response reserve (FRR) studies, and should be incorporated into the nonlinear turbine models as in Figures 4-15 and 4-16.

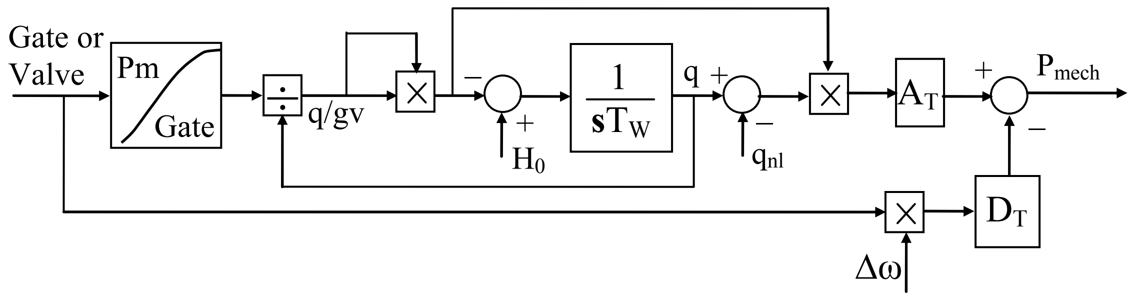


Figure 4-15: Nonlinear Turbine-Water Column Model (Inelastic)

The addition of the Gate vs. Power gain block in the model will account for the effects due to no-load flow and the fixed turbine gain, and therefore q_{nl} and A_T would not be used.

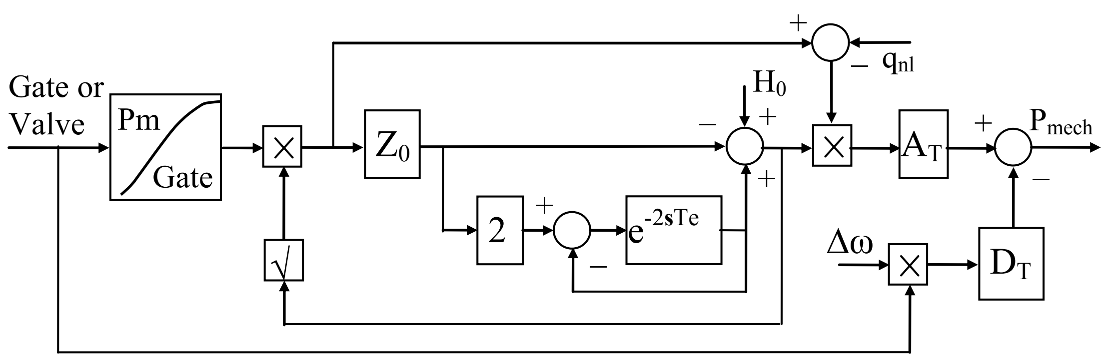


Figure 4-16: Travelling Wave Turbine-Water Column Model (Elastic)

For variable blade pitch Kaplan turbines, a further refinement [30] in the nonlinear gain capability of the model is necessary to accurately model large responses in turbine output, which requires inclusion of a time constant to account for slow adjustment of the blade angle, as depicted in Figure 4-17. In this design, the gate control signal is also sent to the blade pitch controller, whose effect on net turbine flow is delayed. The nonlinear gain effects of the wicket gates, blade control, and turbine efficiency are computed separately and then combined in this model.

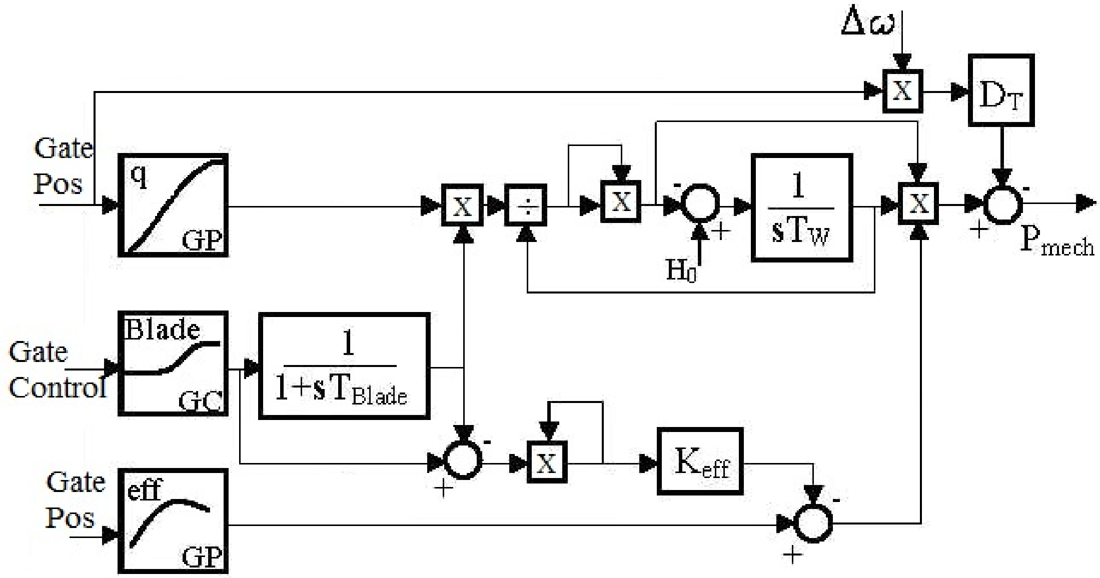


Figure 4-17: Variable pitch Kaplan turbine

4.5 Modeling Guidelines and Summary

We will refrain from repeating some of the key recommendation related in general to turbine-governor modeling (e.g. modeling of the outer-loop MW controller, ensuring the turbine rating is correctly modeled etc.), which have already been explained at length in the previous two chapter summaries, and equally apply to hydro turbines.

Linear models of a hydro penstock and water column should not be used for general power system studies. For some facilities, a travelling wave model may be necessary to correctly model dynamic behavior. Use of simpler models that do not accurately capture dynamic performance in the 0.1 Hz to 1 Hz frequency range can lead to errors in stability studies where inter-area oscillations are a concern. Inclusion of nonlinear effects in hydro models is necessary for system frequency regulation studies.

The most commonly used model in the present North American system databases are HYGOV (in WECC) and the HYGOV, WEHGOV and WPIDHY[10] (in the Eastern Interconnection) models. The WPIDHY model is a very rudimentary representation of a hydro turbine, a combination of a PI controller, and a simple turbine model similar to that show in Figure 4-2. This model is very simple and not recommended where better information is available. The WEHGOV model is essentially the combination of Figure 4-4 (PID controller) and Figure 4-15 (non-linear turbine dynamic model); this model is suitable for power system planning studies where the controls are known to be of the PID

[10] HYGOV is the name of a model available in GE PSLF® and Siemens PTI PSS®E. WEHGOV and WPIDHY are names of models available in Simens PTI PSS®E. These models may also be available in many other commercial power system simulation programs.

type. The WEHGOV model also incorporates a piecewise linear curve of flow versus power instead of assuming a linear relationship as implemented by a fixed turbine gain (A_T). The HYGOV model is essentially a combination of the Figure 4-3 (mechanical hydraulic controls) and Figure 4-15, this model is suitable for power system planning studies where the controls are known to be of the mechanical hydraulic type.

5. References

[1] Australian Energy Market Operator (AEMO), "Generating System Model Guidelines", Document Number 118-0009, 29 February 2008.

[2] IEEE/CIGRE Joint Task Force on Stability Terms and Definitions, Definition and Classification of Power System Stability, IEEE Transactions on Power Systems, 19(2), 1387–1401, August 2004.

[3] F.P. de Mello and C. Concordia, "Concepts of synchronous machine stability as affected by excitation control," IEEE Trans. PAS, vol. 88, pp.316–329, 1969.

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

[5] IEEE Task Force on Large Interconnected Power Systems Response to Generation Governing, *Interconnected Power System Response to Generation Governing: Present Practice and Outstanding Concerns*, IEEE Special Publication 07TP180, May 2007.

http://www.pes-store.org/continuing-education/interconnected-power-system-response-to-generation-governing-present-practice/p-13433.htm

[6] S. Sharma, S-H. Huang and N. D. R. Sarma, "System Inertial Frequency Response Estimation and Impact of Renewable Resources in ERCOT Interconnection", Proceedings of the IEEE PES GM 2011, Detroit, MI.

[7] A. Rogers, "DS3 Grid Code Workstream: Rate of Change of Frequency (RoCoF)& Voltage Control", Operations, EirGrid, DS3 Industry Forum, 14thMarch, 2012.

http://www.eirgrid.com/media/DS3%20Industry%20Forum%20-%20Grid%20Code%2014-03-12.pdf

[8] Transmission System Performance Report, Eirgrid 2010.

http://www.eirgrid.com/media/Transmission%20System%20Performance%20Report%202010.pdf

[9] CIGRE Technical Brochure 238, Modeling of Gas Turbines and Steam Turbines in Combined-Cycle Power Plants, December 2003. (www.e-cigre.org)

[10] J. K. Salisbury, Steam Turbines and Their Cycles, John Wiley & Sons, 1950.

[11] IEEE WG on Prime Mover, "Dynamic Models for Steam and Hydro Turbines in Power System Studies", IEEE Trans. on PAS, Vol PAS-92, Nov-Dec 1973, pp. 1904-1915.

[12] P. Pourbeik, C. Pink and R. Bisbee, "Power Plant Model Validation for Achieving Reliability Standard Requirements Based on Recorded On-Line Disturbance Data", Proceedings of the IEEE PSCE, March 2011.

[13] Power Plant Model Validation Using On-Line Disturbance Monitoring: Technical Update on Latest Results. EPRI, Palo Alto, CA: 2009. Product ID # 1017801. http://my.epri.com/portal/server.pt?Abstract_id=000000000001017801

[14] F. P. de Mello, "Boiler Models for System Dynamic Performance Studies", IEEE Trans. PWRS, pages 66-74, February, 1991.

[15] F. P. de Mello, Boiler Dynamics and Controls Course Notes. http://www.isa.org/~powid/boiler_dynamics_controls_course/DownloadpageforBoilerDynamicsandControlsCourseNotes.htm

[16] P. Pourbeik and F. Modau, "Model Development and Field Testing of a Heavy-Duty Gas-Turbine Generator", IEEE Trans. PWRS, May, 2008.

[17] K. Kunitomi, A. Kurita, H. Okamoto, Y. Tada, S. Ihara, P. Pourbeik, W. W. Price, A. B. Leirbukt and J. J. Sanchez-Gasca, "Modeling Frequency Dependency of Gas Turbine Output", Proceedings of the IEEE PES Winter Meeting, Jan 2001.

[18] P. Pourbeik, "The Dependence of Gas Turbine Power Output on System Frequency and Ambient Conditions", paper 38-101, presented at CIGRE Session 2002, August 2002, Paris, France.

[19] H. Cohen, G. F. C. Rogers, and H. I. H. Saravanamutto, Gas Turbine Theory, 4th Edition, Addison Wesley Longman Ltd, 1996.

[20] W. I. Rowen, "Simplified Mathematical Representations of Heavy Duty Gas Turbines", ASME Paper 83-GT-63 and ASME Journal of Engineering for Power, October 1983, pages 865-869.

[21] Automated Model Validation for Power Plants Using On-Line Disturbance Monitoring. EPRI, Palo Alto, CA: 2009. Product ID # 1016000.

http://my.epri.com/portal/server.pt?Abstract_id=000000000001016000

[22] R. Boyer, "Primary Governing and Frequency Control in ERCOT", IEEE PES General Meeting Conference Proceedings, Tampa, Florida, June 2007.

[23] L. Lima, Y-Chi Lin and J. Feltes, "Improved Modeling of Combined Cycle Power Plants in Steady State and Dynamic Simulation, Prepared for ERCOT", Siemens PTI Report R61-07, June 2007.

[24] ERCOT Dynamics Working Group, "August 19, 2004 Forney Plant Trip Event Simulation", www.ercot.com

[25] L. Pereira, J. Undrill, D. Kosterev, D. Davies, and S. Patterson, "A New Thermal Governor Modeling Approach in the WECC", IEEE Trans. PWRS, May 2003, pp 819-829.

[26] IEEE Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies, "Hydraulic Turbine and Turbine Control Models

for System Dynamic Studies," in IEEE Trans. Power Systems, Vol. 7, No. 1, pp. 167-179, Feb. 1992.

[27] D. G. Ramey, J. W. Skooglund, "Detailed Hydrogovernor Representation in System Stability Studies," IEEE Trans. Vol. PAS-89, No. 1, pp. 106-112, Jan. 1970.

[28] B.Khodabakhchian, G.T.Vuong and S.Bastien. "On the comparison between a detailed turbine-generator EMTP simulation and corresponding field test results." http://www.ipst.org/techpapers/1995/95_IPST_136-21.PDF

[29] S. Patterson, "Importance of Hydro Generation Response Resulting from the New Thermal Modeling – and Required Hydro Modeling Improvements", Proceedings of the IEEE PES General Meeting, Denver July, 2004.

[30] D. Kosterev, "Hydro Turbine-Governor Model Validation in the Pacific Northwest," IEEE Trans Power Systems, Vol. 19, No. 2, pp. 1144-1149, May 2004.

6. Further Reading on Turbine-Governor Modeling

[1] J. W. Barnett, “Speed Governing Design Considerations for Multivalve Condensing Steam Turbine Generators,” AIEE Transactions, volume 67, part II, 1948, pp 1567-61

[2] H. B. Ruud, S. B. Farnham, ”A New Automatic Load Control for Turbine Generators,” AIEE Transactions, volume 68, part II, 1949, pp 1337-42

[3] J. E. McCormack, C. N. Metcalf, “Load Frequency Control on the Northeast Interconnection,” Electric Light and Power (Chicago, Ill.), volume 27, February 1949, pp 70-73

[4] E. E. George, “Speed and Load Control of Interconnected Systems,” Electrical World, volume 123, March 1945, pp 84-85

[5] S. B. Crary, J. B. McClure, “Supplementary Control of Prime-Mover Speed Governors,” AIEE Transactions, volume 61, 1942, pp 209-14

[6] C. Concordia, H. S. Scott, C. N. Weygandt, “Control of Tie-line Power Swings,” AIEE Transactions, volume 61, June 1942, pp 306-14

[7] R. J. Caughey, J. B. McClure, ”Prime Mover Speed Governors for Interconnected Systems,” AIEE Transactions, volume 60, April 1941, pp 147-51

[8] C. Concordia, S. B. Crary, E. E. Parker, “Effect of Prime-Mover Speed Governor Characteristics on Power-System Frequency Variations and Tieline Power Swings,” AIEE Transactions, volume 60, 1941, pp 559-67

[9] P. G. Ipsen, J. R. Norton, “Prime Mover Speed Governors and the Interconnected System,” AIEE Transactions, volume 72, June 1953, pp 353-60

[10] Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies, “Dynamic Models for Fossil Fueled Steam Units in Power System Studies,” IEEE Transactions on Power Systems, volume 6, No. 2, May 1991, pp 753-61

[11] IEEE Committee Report, “Dynamic Models for Steam and Hydro Turbines in Power System Studies,” Transactions in Power Apparatus & Systems, volume 92, No. 6, Nov./Dec. 1973, pp 1904-15

[12] L. H. Johnson, T. D. Younkins, “Steam Turbine Overspeed Control and Behavior During System Disturbances,” Transactions in Power Apparatus & Systems, volume 100, No. 5, May 1981, pp 2504-11

[13] F. P. de Mello, D. N. Ewart, “MW Response of Fossil-Fueled Steam Units,” Transactions in Power Apparatus & Systems, volume 92, No. 2, March/April 1973, pp 455-63

[14] F. M. Hughes, “Improvement of Turbogenerator Transient Performance by Control Means,” Proceedings IEE, volume 120, No. 2, Feb. 1973, pp 233-40

[15] F. G. Dent, “Microgovernor – A Replacement of Existing Large Steam Turbine Governing Controls,” IEE Conference on Refurbishment of Power Station Electrical Plant, No. 295, Nov. 1988, pp 128-42

[16] P. A. L. Ham, N. J. Green, “Development and Experience in Digital Turbine Control,” IEEE Transactions on Energy Conversion, volume 3, No. 3, Sept. 1988, pp 568-74

[17] C. Concordia, F. P. de Mello, L. Kirchmayer, R. Schultz, “Effect of Prime- Mover Response and Governing Characteristics on System Dynamic Performance,” American Power Conference, 1966, volume 28, pp1074-85

[18] J. L. Woodward, “Hydraulic-Turbine Transfer Function for Use in Governing Studies,” Proceedings IEE, volume 130, March 1968, pp 424-26

[19] J. M. Undrill, J. L. Woodward, “Nonlinear hydro Governing Model and Improved Calculation for Determining Temporary Droop,” Transactions in Power Apparatus & Systems, volume 86, No. 4, April 1967, pp 443-53

[20] P. W. Agnew, “The Governing of Francis Turbines,” Water Power, pp. 119-27, April 1974

[21] R. Oldenburger, J. Donelson, “Dynamic Response of a Hydroelectric Plant,” Transactions AIEE, volume 81, Pt. III, 1962, pp 403-18

[22] P. L. Dandeno, P. Kundur, J. P. Bayne, “Hydraulic Unit Dynamic Performance Under Normal and Islanding Conditions – Analysis and Validation,” Transactions in Power Apparatus & Systems, volume 97, No. 6, Nov./Dec. 1978, pp 2134-43

[23] Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies, “Hydraulic Turbine and Turbine Control Models for Dynamic Studies,” Transactions on Power Systems, volume 7, No. 1, Feb. 1992, pp 167-179

[24] S. Hagihara, H. Yokota, K. Goda, K. Isaobe, “Stability of a Hydraulic Turbine Generating Unit Controlled by PID Governor,” Transactions in Power Apparatus & Systems, volume 98, No. 6, Nov./Dec. 1979, pp 2294-98

[25] F. R. Schleif, A. B. Wilbor, “The Coordination of Hydraulic Turbine Governors for Power System Operation,” Transactions in Power Apparatus & Systems, volume 85, July 1966, pp 750-58

[26] G. R. Berube, L. M. Hajagos, “Modeling Based on Field Tests of Turbine/Governor Systems,” PES Winter Power Meeting, volume 1, Feb. 1999, pp 567-73

[27] C. K. Sanathanan, “A Frequency Domain Method for Tuning Hydro Governors,” IEEE Transactions on Energy Conversion, volume 3, No. 1, March 1988, pp 14-17

[28] L. N. Hannett, J. W. Feltes, B. Fardanesh, “Field Tests to Validate Hydro Turbine-Governor Model Structure and Parameters,” IEEE Transactions on Power Systems, volume 9, No. 4, Nov. 1994, pp 1744-51

[29] L. N. Hannett, J. W. Feltes, B. Fardanesh, C. Crean “Modeling and Control Tuning of a Hydro Station with Units Sharing a Common Penstock,” IEEE Transactions on Power Systems, volume 14, No. 4, Nov. 1999, pp 1407-

[30] C. D. Vournas, G. Papaioannou, “Modelling and Stability of a Hydro Plant with Two Surge Tanks,” IEEE Transactions on Energy Conversion, volume 10, No. 2, June 1995, pp 368-75

[31] B. Strah, O. Kuljaca, Z. Vukic, “Speed and Active Power Control of Hydro Turbine Unit,” IEEE Transactions on Energy Conversion, volume 20, No. 2, June 2005, pp 424-34

[32] D. Kosterev, “Hydro Turbine-Governor Model Validation in Pacific Northwest,” IEEE Transactions on Power Systems, volume 19, No. 2, May 2004, pp 1144-49

[33] L. Pereira, J. Undrill, D. Kosterev, D. Davies, S. Patterson, “A New Thermal Governor Modeling Approach in the WECC,” IEEE Transactions on Power Systems, volume 18, No. 2, May 2003, pp 819-29

[34] Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies, “Dynamic Models for Combined Cycle Plants in Power System Studies,” IEEE Transactions on Power Systems, volume 9, No. 3, Aug. 1994, pp 1698-1708

[35] H. E. Wickert, N. S. Dhaliwal, “Analysis of P.I.D. Governors in Multimachine Systems,” IEEE Transactions on Power Systems, volume 19, No. 3, May 2004, pp 1144-49

[36] L. N. Bize, J. D. Hurley, “Frequency Control Considerations for Modern Steam and Combustion Turbines,” IEEE PES Winter Meeting, volume 1, Jan. 1999, pp 548-53

[37] L. N. Hannett, A. H. Khan, “Combustion Turbine Dynamic Model Validation from Tests,” IEEE Transactions on Power Systems, volume 8, No. 1, Feb. 1993, pp 152-58

[38] L. N. Hannett, G. Jee, B. Fardanesh, “A Governor/Turbine Model for a Twin-Shaft Combustion Turbine,” IEEE Transactions on Power Systems, volume 10, No. 1, Feb. 1995, pp 133-40

[39] N. Nakimoto, K. Baba, “Performance of Gas Turbine-Based Plants During Frequency Drops,” IEEE Transactions on Power Systems, volume 18, No. 2, May 2003, pp 931-37

[40] K. Chan, A. E. Ariffin, Y. C. Chew, C. Lin, H. Ye, “Validated Combined Cycle Power Plant Model for System and Station Performance Studies, ”Power System Technology, 2004 PowerCon, volume 2, Nov. 2004, pp 1991-97

[41] K. Kunitomi, A. Kurita, Y. Tada, S. Ihara, W. W. Price, L. M. Richardson, G. Smith, "Modeling Combined-Cycle Power Plant for Simulation of frequency Excursions," IEEE Transactions on Power Systems, volume 18, No. 2, May 2003, pp 724-29

[42] J. Undrill and A. Garmendia, "Modelling of Combined Cycle Plants in Grid Simulation Studies," IEEE PES Winter Meeting, volume 2, Jan. 2001, pp 657-63

APPENDIX A: Parameters for the IEEEG1 + LCBF1 Model

IEEEG1:

Parameter	Description
K	Governor gain (1/droop) [pu]
T1	Lag time constant [s]
T2	Lead time constant [s]
T3	Valve position time constant [s]
Uo	Maximum valve opening rate [pu/s]
Uc	Maximum valve closing rate [pu/s]
Pmax	Maximum valve opening , on MW capability [pu]
Pmin	Minimum valve opening, on MW capability [pu]
T4	Time constant for steam inlet [s]
K1	HP fraction
K2	LP fraction
T5	Time constant for second boiler pass [s]
K3	HP fraction
K4	LP fraction
T6	Time constant for third boiler pass [s]
K5	HP fraction
K6	LP fraction
T7	Time constant for fourth boiler pass [s]
K7	HP fraction
K8	LP fraction
db1	deadband

For typical values for the parameters see reference [11].

LCBF1:

Parameter	Typical Value	Description
db	0	Controller dedband [pu]
emax	0.1	Maximum error [pu]
fb	0	Freqeuncy bias gain [pu/pu]
Kp	0	Proportional gain [pu/pu]
Ki	0.005 to 0.2 (typical range)	Integral gain [pu/pu/s]
Tpelec	2 to 5	Power transducer time constant [s]
lrmax	0.025 to 0.05 (typical range)	Maximum output [pu]
Pmwset	initialized by model	Power Setpoint [pu]

Note: The values above are typical for illustrative purposes. Actual values may differ on a case by case basis. Also, for the LCBF1 model, the model output must be on the same units as the turbine-governor Pref input.

APPENDIX B: Deadband

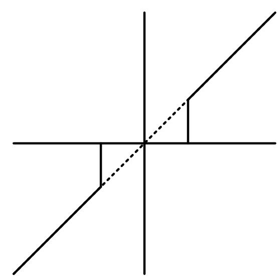
A deadband may be implemented one of two ways, with significantly different results in terms of the response of the control system to the implementation – see [16] for an example explanation for an actual control system on a turbine-governor.

The two types are:

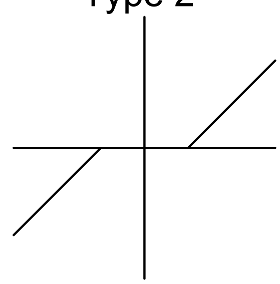
Type 1 – no offset in the output, that is once outside the deadband the input is passed straight through

Type 2 – the output is offset by the deadband, that is once outside the deadband the output starts from zero and then proportionally increases with the input.

Type 1



Type 2



Warning: the type 1 deadband shown above leads to a sudden jump in MW response on the transition point of going out of the deadband. This can be undesirable.

APPENDIX C: Typical Parameters for the GGOV1

Typical Values for the GGOV1 model are provided below. Validation of the values is recommended for in-service equipment. The value for the parameter MWCAP is the rating of the turbine and the base for all the other parameters. It is not equal to the MVA of the generator and is typically smaller. The generator is typically oversized (i.e. MWCAP < MVA) because of the possible range of maximum gas turbine output for seasons ranging from 120% of nameplate rating on an extreme cold day to 80% of nameplate rating on an extreme hot day.

The value for *Ldref* will vary for changes of ambient temperature. A value of 1.0 is valid for 59ºF, 14.70 psia, and for natural gas containing 96% CH_4.

Parameter	Typical Values
MWCAP	User Supply
r	0.04
rselect	1
T_{pelec} (sec)	1
Max_{ERR} (pu)	0.05
Min_{ERR} (pu)	-0.05
K_{PGOV}	10
K_{IGOV}	2
K_{DGOV}	0
T_{DGOV} (sec)	1
V_{MAX}	1
V_{MIN}	0.15
T_{ACT} (sec)	0.5
K_{TURB}	1.5
W_{FNL}	0.2
T_B (sec)	0.1
T_C (sec)	0
flag	1
T_{ENG} (sec)	0
T_{FLOAD} (sec)	3
K_{PLOAD}	2
K_{ILOAD}	0.67
L_{DREF}	1
D_M	0
R_{OPEN} (pu/sec)	0.1
R_{CLOSE} (pu/sec)	-0.1
K_{IMW}	0
P_{MWSET} (pu)	N/A
A_{SET} (pu)	0.01
K_A (pu)	10
T_A (sec)	0.1
db (pu)	0
T_{SA} (sec)	4
T_{SB} (sec)	5
R_{UP} (pu)	99
R_{DOWN} (pu)	-99

APPENDIX D: Vendor Specific Model for GE Heavy-Duty Gas Turbines

The model *gegt1* was written specifically to represent GE heavy duty gas turbines with Mark V, Mark VI and Mark VIe controls. This more detailed model represents not only the features present in models like GGOV1*,* but also the guide vane dynamics and its effect on the temperature limiter function. Block diagrams are presented in Figures D-1 and D-2. The model is available in the GE PSLF® power system simulation tool.

As shown in Figure D-1, the possible feedback signals for droop are electrical power and the fuel stroke reference (fsr). To represent the Mark V or Mark VI controls normally the parameter RVALVE is set to 0 and the value for droop is assigned to RPELEC.

The controls that are shown in Figure D-1 are quite close to those for the *ggov1*, model without the extra control loops for the droop signal. The inputs for the Low Value Select are FSRN, FSRT and FSRA. FSRN and FSRA are derived in the same manner as in the model *ggov1*, when the same blocks are used. The third signal, FSRT is derived differently, and the blocks to derive the error signal, ETEMP, for the temperature limiter are shown in Figure D-2.

The input signals for the blocks in Figure D-2 are the exhaust temperature TX, the compressor discharge pressure ratio CPR and the governor feed forward signal GVFF, which is derived as shown in Figure D-1. The exhaust temperature and compressor discharge ratio are calculated from the turbine model presented in [15], which was actually the first implementation of the *gegt1* model for the Mark V controls (see Figure 3-2, section 3.1.1.). The current implementation of the *gegt1* model in the GE PSLF® program uses the following equations for this simplified aero-dynamic model:

$$W_{AIR} = [I_{GV0} + K_{AF}\sin(I_{GV})] * [1 - a_0 \Delta N^{a_1} + a_2 \Delta N] * [\frac{\theta_{ISO}}{\theta_{AMB}}] \quad (1)$$

$$T_{EX} = T_{AMB} + \frac{a_3 * \frac{P_{ISO}}{P_{AMB}} + a_4 W_F}{W_{AIR}} \quad (2)$$

$$C_{PR} = a_7 + (a_6 W_F + a_5 W_{AIR}) * \frac{P_{ISO}}{P_{AMB}} \quad (3)$$

Where, ϑ_{ISO} is the absolute temperature at ISO conditions, ϑ_{AMB} is the absolute ambient temperature, P_{ISO} is the air pressure at ISO conditions, P_{AMB} is the ambient air pressure, W_{AIR} is the airflow in per unit with the base value at ISO conditions and maximum guide vane position, and ΔN is the rotor speed deviation in per unit.

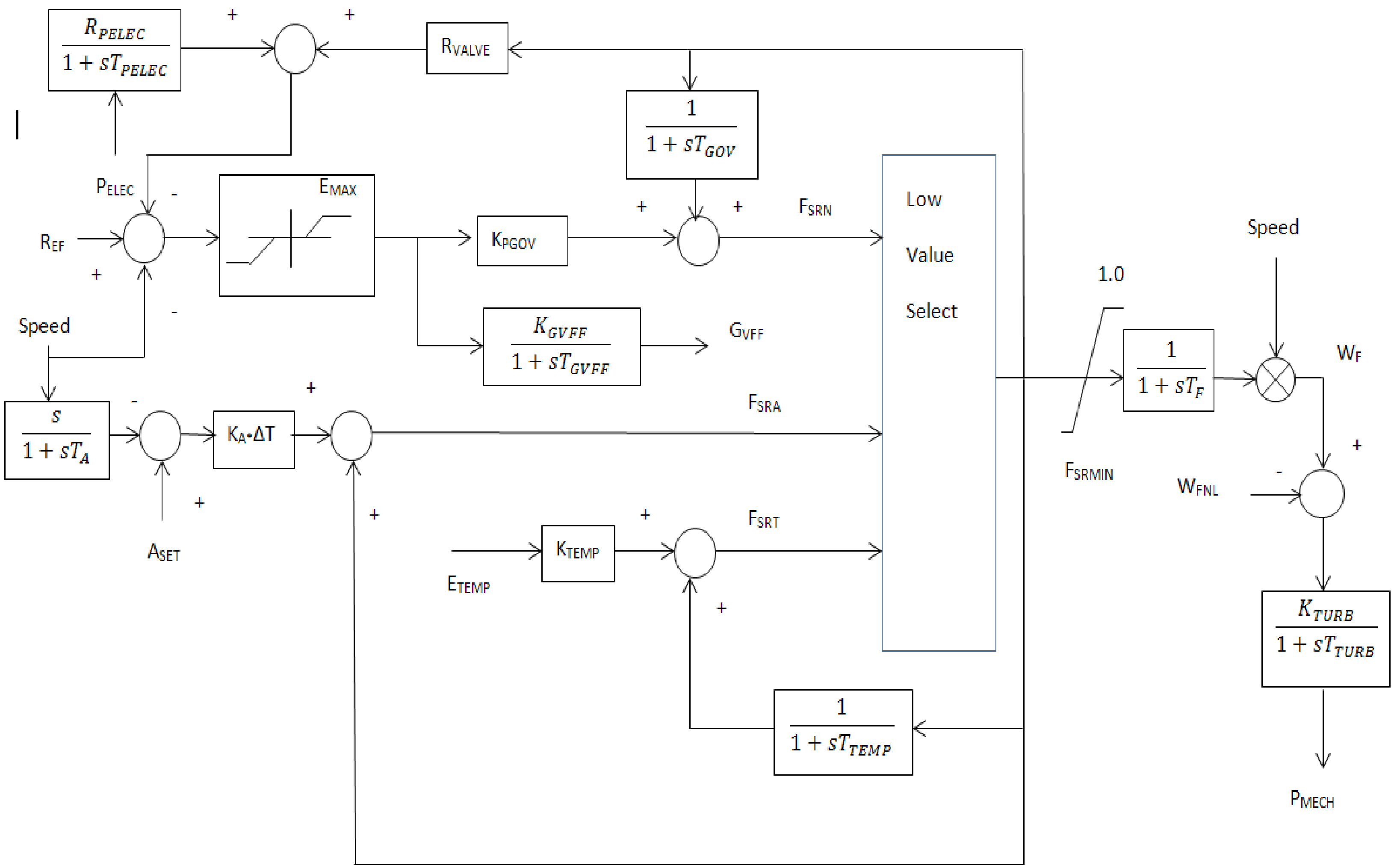


Figure D-1: Block Diagram of *gegt1* for the Three Control Functions.

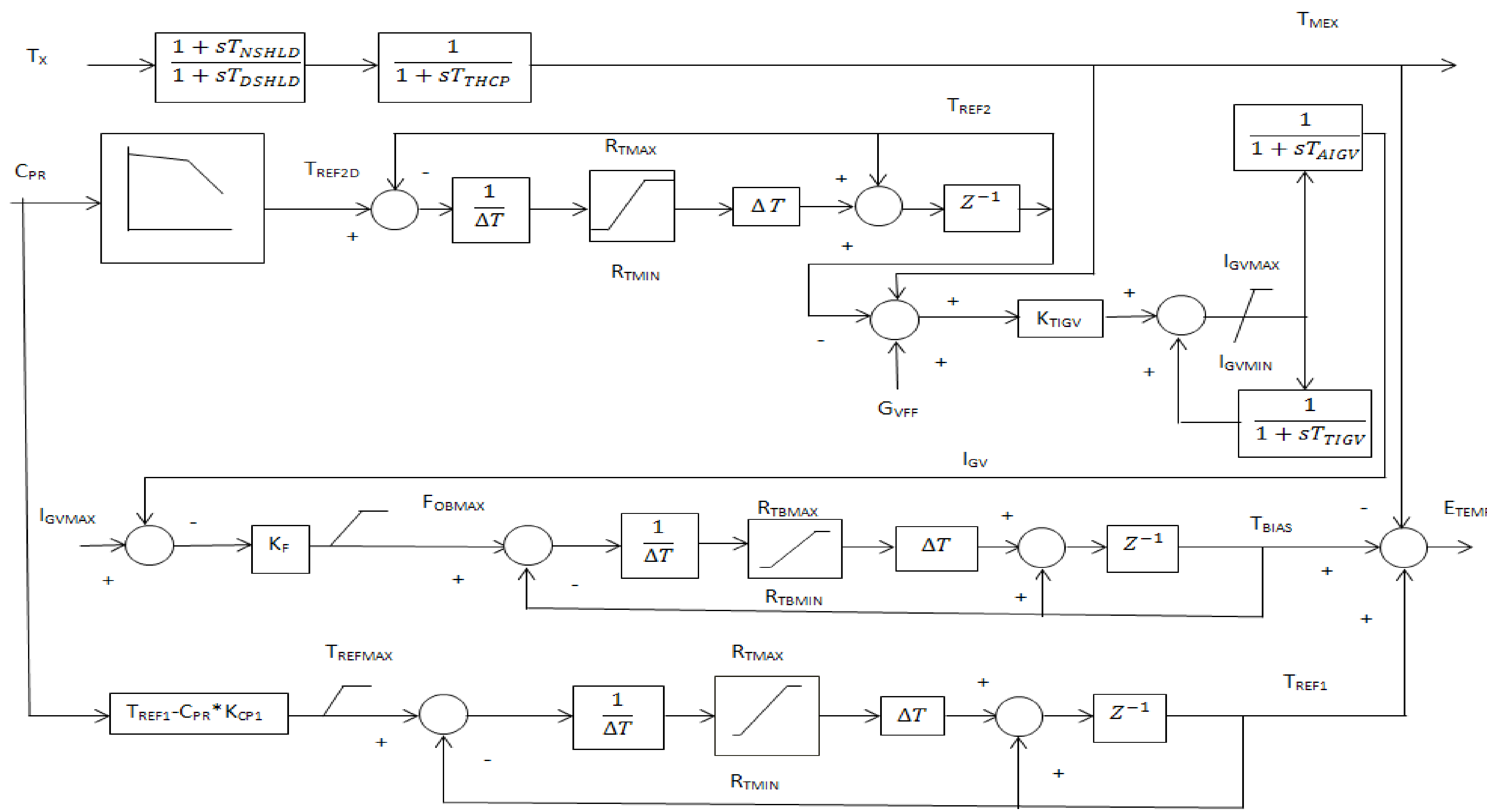


Figure D-2: Block Diagram of *gegt1* for the Guide Vane Dynamic Function.

One of the benefits of using the simple aero-dynamic model for the turbine is that it accounts for variation of air temperature and air pressure so that with the model parameters determined for a given gas turbine, variations for the seasons can be obtained by changing the values for the ambient temperature and pressure input to the model (see [15] and [18] for a theoretical explanation).

There are no typical values for the coefficients that appear in the equations of Figure 3-2 (section 3.1.1). The values for the parameters are identified by collecting values during steady state operation, and "fitting" the model response with the recorded measurements [15]. Listed in the table below are the values for the *gegt1* model parameters for 5 common GE gas turbines. These values were obtained from testing individual units. Two of the units are from Frame 7EA gas turbines and two are from Frame 7FA gas turbines. The fifth unit is a Frame 6B gas turbine. Note that for the values of the parameters a1 to a7 there is variation between the 2 Frame 7EA gas turbines while these parameters have the same values for the two Frame 7FA gas turbines. Variations between designs for the gas turbines may account for the differences between the two Frame 7EA gas turbines. Ideally one would like to say for one Frame model that the parameters would have the same values allowing one to specify a set of typical values for the model parameters. Due to known, previous observed differences, it is strongly recommended to collect data for the individual unit and identify the values for the parameters.

Parameter	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5
MWCAP	84.71	84.71	41.69	151.9	151.9
Model Series	MS7001EA	MS7001EA	PG6581(B)	MS7231FA	MS7001FA
Droop (Hz/MW)	0.0273	0.0459	0.05204	0.0418	0.06526
tpelec	2.5	1	2.4	2.5	1
kpgov	10	9	10	6	8
tgov	10	3.5	4	2.7	1.8
kisoc	0	0	0	0	0
ttemp	3.3	3.3	3.3	3.3	3.3
ktemp	0.004	0.004	0.004	0.004	0.004
aset	0.01	0.01	0.01	0.01	0.01
ka	10	10	10	10	10
ta	0.1	0.1	0.1	0.1	0.1
wfnl	0.184	0.15	0.2	0.169	0.17
fsrmin	0.14	0.05	0.1	0.14	0.14
tf	0.4	0.4	0.4	0.4	0.55
kturb	1.66	1.612	1.868	1.626	1.654
tturb	0.1	0.1	0.1	0.1	0.1
tthcp	3	3	3	3	3
tnshld	12	12	12	12	12
tdshld	15	15	15	15	15
igvmax	84	84	86.1	84	80
igvmin	41.8	42	42.1	41.8	43.5
ktigv	5	5	0.3	5	5
ttigv	1	1	4	1	1
taigv	3	3	3	3	3
kgvff	0	0	0	0	0
tgcff	0.1	0.1	0.1	0.1	0.1
kfb	100	100	100	100	100
fobmax	16.7	16.7	16.7	16.7	16.7
rtbmax	999	999	999	999	999
rtbmin	-1.11	-1.11	-1.11	-1.11	-1.11
rtmax	2	2	2	2	2
rtmin	-1	-1	-1	-1	-1
tref1	723.1	740	759.3	759.3	754
kcpr1	14.796	20	20.2	10.96	10.96
tref2	647.5	739.4	614	648.3	648.3
kcpr2	9.1057	23.47	0	0	0
tref3	0	0	775	723.6	723.6
kcpr3	0	0	23.72	8.855	8.855
tref	649	649	650	660	660
dbd	0	0	0	0	0
emax	0.03	0.03	0.03	0.03	0.03
rrmax	999	999	999	999	999
fsrol	1	1	1	1	1
flag	1	1	1	1	1
tamb	-8.88	-8.88	13.8	11.7	21.667
tiso	15	15	15	15	15
pamb	0.938	0.938	0.923	0.8135	0.8655
piso	1	1	1	1	1
a0	25	25	25	25	25
a1	2	2	2	2	2
a2	1.75	1.75	1.75	1.75	1.75
a3	128.13	90.678	76.32	81.37	81.37
a4	546.53	627.97	664.8	663.6	663.6
a5	5.956	7.919	9.002	8.504	8.504
a6	8.0323	2.0115	3.3626	5.521	5.521
a7	-0.8491	-1.1055	-1.986	0.1887	0.1887
igv0	-0.26117	-0.26117	-0.26117	-0.5645	-0.5645
kaf	1.2776	1.2776	1.2776	1.568	1.568

APPENDIX E: Vendor Specific Model for the ALSTOM GT26B Heavy-Duty Gas Turbine

E.1 Nomenclature

EV	Environmental Burner
FCM	Frequency control module
FFWD	Feed forward
FR	Frequency response
GT	Gas Turbine
PI	Proportional integral regulator
SEV	Sequential Environmental Burner
TIT	Turbine inlet temperature
VGV	Variable Guide Vane

E.2 Base Unit

The per unit power used in the turbine-governor model is based on the GT reference power PREF at ISO conditions, i.e. at an ambient temperature of 15°C. For the ALSTOM GT26B 2.3 its rating is 283 MW. Nominal frequency is 50Hz. The ALSTOM GT26B gas turbine is a product for the 50 Hz market and hence the parameters of the model provided here are related to this nominal frequency.

E.3 Model Application

The GT dynamic model described in this appendix can be used to investigate the dynamic behavior of the gas turbine and its impact on the grid during frequency disturbance events.

The model provides an accurate representation of its dynamic behavior with the governor operating in normal droop mode (3% to 4%) when connected to a grid. For investigating the frequency response behavior of the GT, the model can be used within a frequency range of 94% and 104% of the nominal frequency and a wide ambient temperature range (from -15°C to 50°C) with the appropriate model parameters.

For island or partial load rejection operation in an isolated grid, supplementary control features are required that are not in the scope of this appendix.

E.4 Alstom GT26B simplified dynamic model (Figure E-1)

The ALSTOM's GT26 gas turbine employs a sequential two combustion stage design (EV and SEV burners) instead of one combustion stage design used by other manufacturers. The former has the advantage that it offers higher efficiency as well as lower emission levels. The control structure, fuel/power management and the dynamics of the GT26 are therefore different from single stage combustion gas turbines.

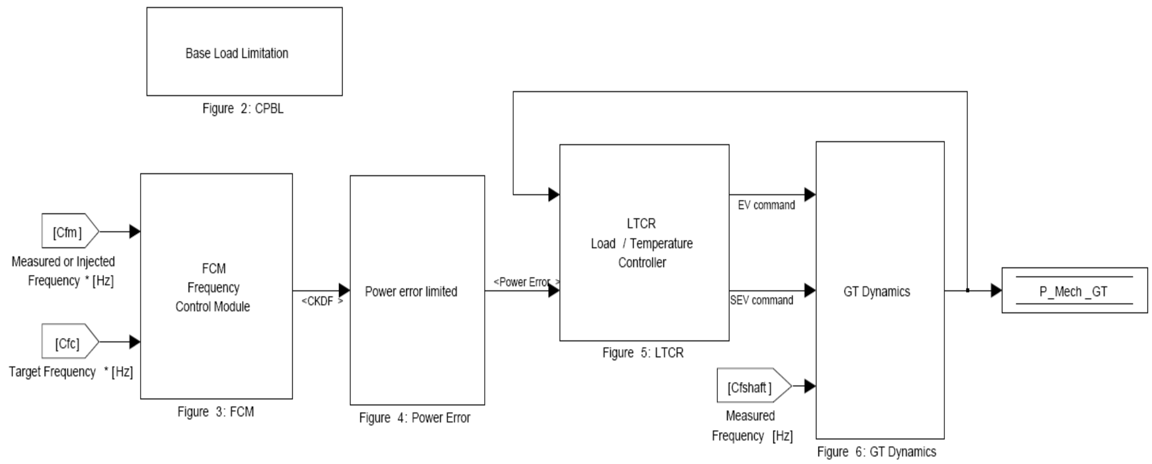


Figure E-1: Overview of the ALSTOM GT26B Simplified Dynamic Model.

The modeling of the ALSTOM's GT26 gas turbine-governor follows to a large extent the CIGRE modeling recommendations [9]. The model is based on a modular structure consisting of subsystems. Each subsystem provides a simplified model of a particular function of the gas turbine-governor system.

An overview of the GT26B simplified dynamic turbine-governor model, consisting of five subsystems, is shown in Figure E-1. These subsystems are described in the following sections of this appendix.

The basic interface to the generator model consists of two inputs and one output namely:

- Generator frequency or speed [Cfm or Cshaft].
- Generator electrical power [P_Elec_GT].
- Generator mechanical power [P_Mech_GT].

Base load limitation subsystem (Figure E-2)

A gas turbine's power output is dependent on ambient temperature as well as the turbine shaft speed. This has to be accounted for in the modeling of the gas turbine. The Base Load Limitation subsystem models this characteristic of the gas turbine by introducing a correction factor CPBL that is a function of the actual rotor speed and ambient temperature inputs.

For the ALSTOM's GT26 gas turbine model the factor CPBL is determined by different mathematical functions (lookup table or depending on whether extended frequency response support (iExtFR) is enabled or disabled. Extended frequency response support allows the gas turbine to provide a guaranteed minimum output level at frequencies down to 94% nominal even at high temperatures as stipulated in certain grid codes.

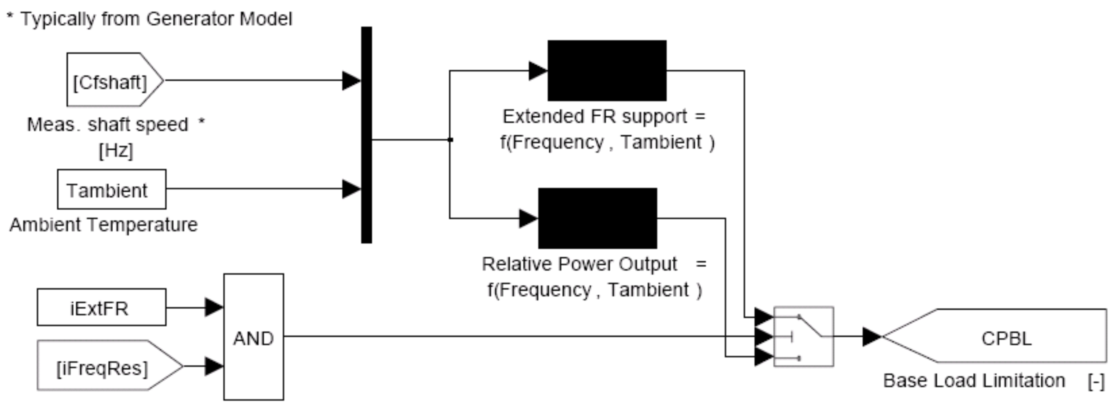


Figure E-2: Base Load Limitation Subsystem.

Frequency control subsystem (Figure E-3)

The Frequency Control subsystem (Figure E-3) in its base configuration consists of the control structure to enable the GT to provide frequency response (FR) capability.

Optional or additional frequency control features that may be required but not addressed in the appendix, such as island operation or partial load rejection capabilities, are also part of this subsystem.

Frequency response is the automatic variation of the commanded GT power as a function of the grid frequency. A change in power is related to the frequency error through a predefined power gain. The FR control, as shown in Figure E-4, is superimposed onto load control.

The ALSTOM's GT uses a dynamic dead band in the speed governor. This dead band generates a commanded load signal for frequency response operation and replaces the functionality of static dead bands.

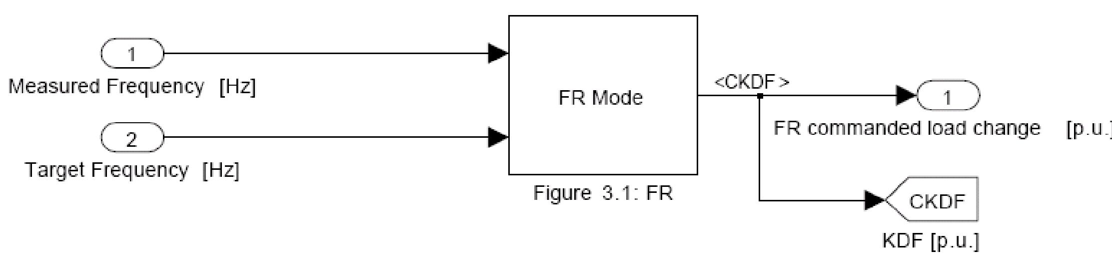


Figure E-3: Frequency Control Subsystem

The following operation modes can be selected based on the GT's operation concept:

- Frequency Response Mode OFF: The GT power output is controlled to match the load setpoint specified by the operator. The GT does not react to any frequency deviations in the grid. The parameter iFreqRes enables or disables the frequency response mode.

- Frequency Response Mode ON, Sensitivity ON: Sensitive mode incorporates a dynamic dead band around the frequency trend signal. As long as the measured frequency signal follows the trend signal within the dynamic dead band, the trend will be used to determine the GT power response. Otherwise, the measured frequency signal will be used. The parameter iDBsens switches frequency sensitive mode on or frequency limited sensitive mode on.
- Frequency Response Mode ON, Sensitivity OFF (Figure E-5): Frequency limited sensitive mode is generally chosen, if the GT is dispatched for base load operation and is not required to take part in frequency regulation, except for high frequency emergency conditions. Insensitive mode incorporates a static dead band around the frequency setpoint. The trend signal and dynamic dead band are in operation during insensitive mode, but do not have any influence on the power response as long as the frequency error remains within the predefined static dead band.

If accuracy is not important, this dynamic dead band could probably be replaced with a static dead band similar to the CIGRE generic model [9] but it does not reflect the true behavior of the ALSTOM GTs.

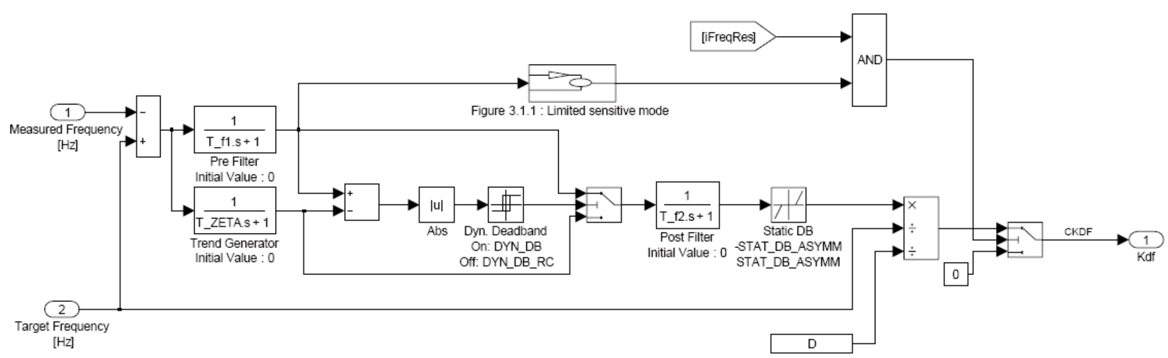


Figure E-4: Frequency Response Function

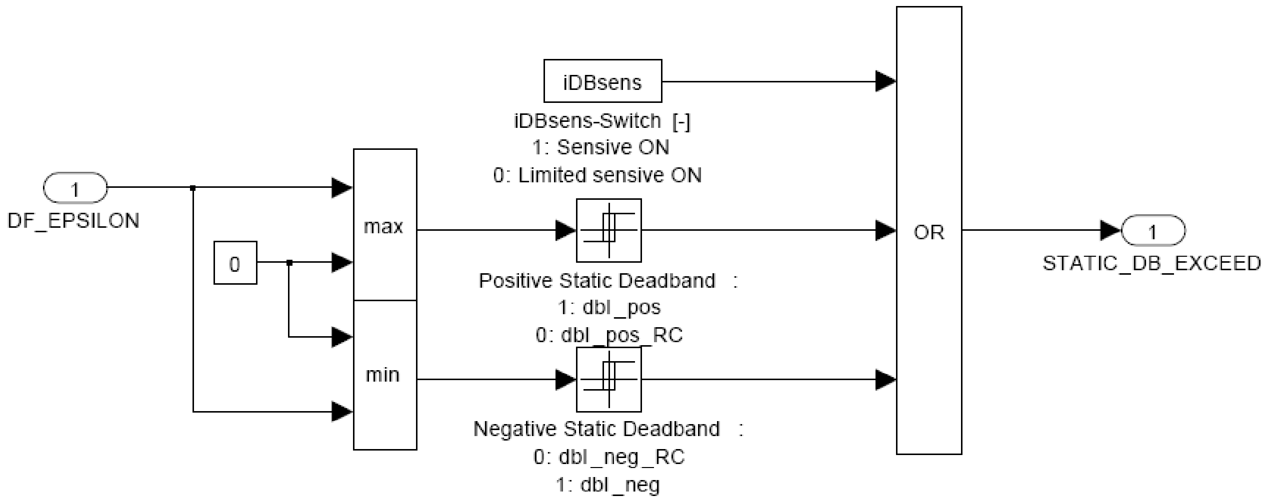


Figure E-5: Limited Sensitive Mode

Power error subsystem (Figure E-6)

The total commanded power is the sum of the power set point and the commanded power change due to frequency response as calculated in the frequency response controller above.

The change in commanded power is limited to a maximum predefined rate by a power rate limiter (LD_RT_LIM) in ALSTOM's GT control system. This rate limiter is not present in the CIGRE generic model hence implying the possibility to change the power at a very fast rate. This in reality is not true and will result in a flame instability situation. The power error function determines the difference between the current GT power and the required commanded GT power that will drive the GT load and temperature controllers.

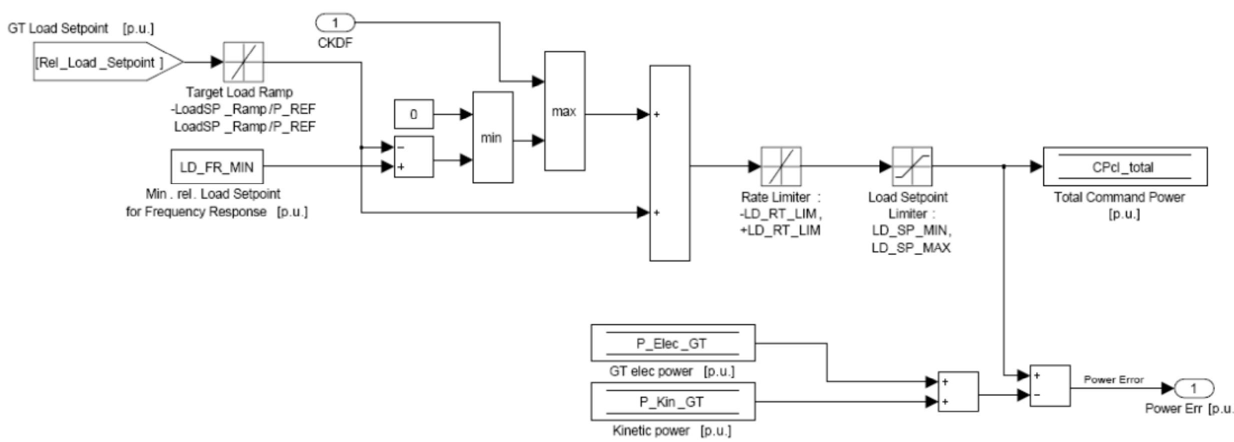


Figure E-6: Power Error Subsystem

Load and temperature control subsystem (Figure E-7)

The ALSTOM's GT26 Load/Temperature controller (LTCR) consists of two parts: Power Management and Power/Temperature Controller. The Power Management distributes the power error to the Power/Temperature Controllers according to the GT operating concept, and the Power/Temperature Controllers regulate the actuators to meet the power demand and keep the hot gas temperatures within safe limits.

Load control is used to vary fuel flow and VGV position until the commanded power is achieved.

Temperature control is used to ensure that the GT temperatures do not exceed the set point limits.

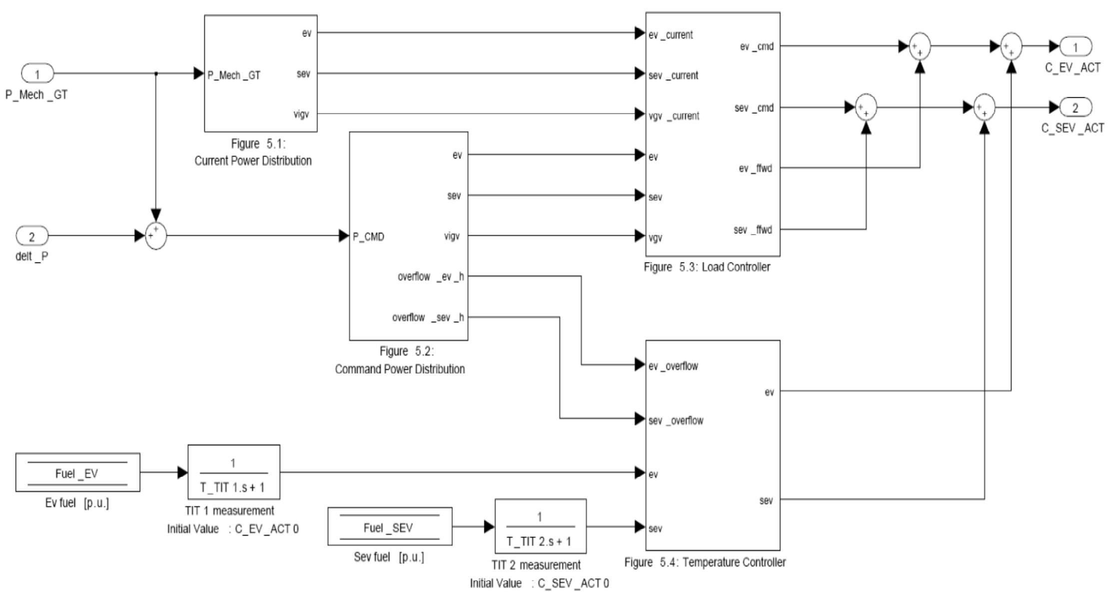


Figure E-7: Load and Temperature Control Subsystem

Current Power and Command Power Distribution (Figures E-8 & E-9)

Current Power and Command Power Distributions control functions divides the power between the predefined capacities of the gas turbine, starting from the smallest capacity and advancing towards the largest. The Current Power Distribution module is generating current EV, SEV, and VGV powers while Command Power Distribution in addition generates EV and SEV command power overflow. The influence of frequency and ambient temperature on EV, SEV and VGV powers is taken into account using the CPBL factor.

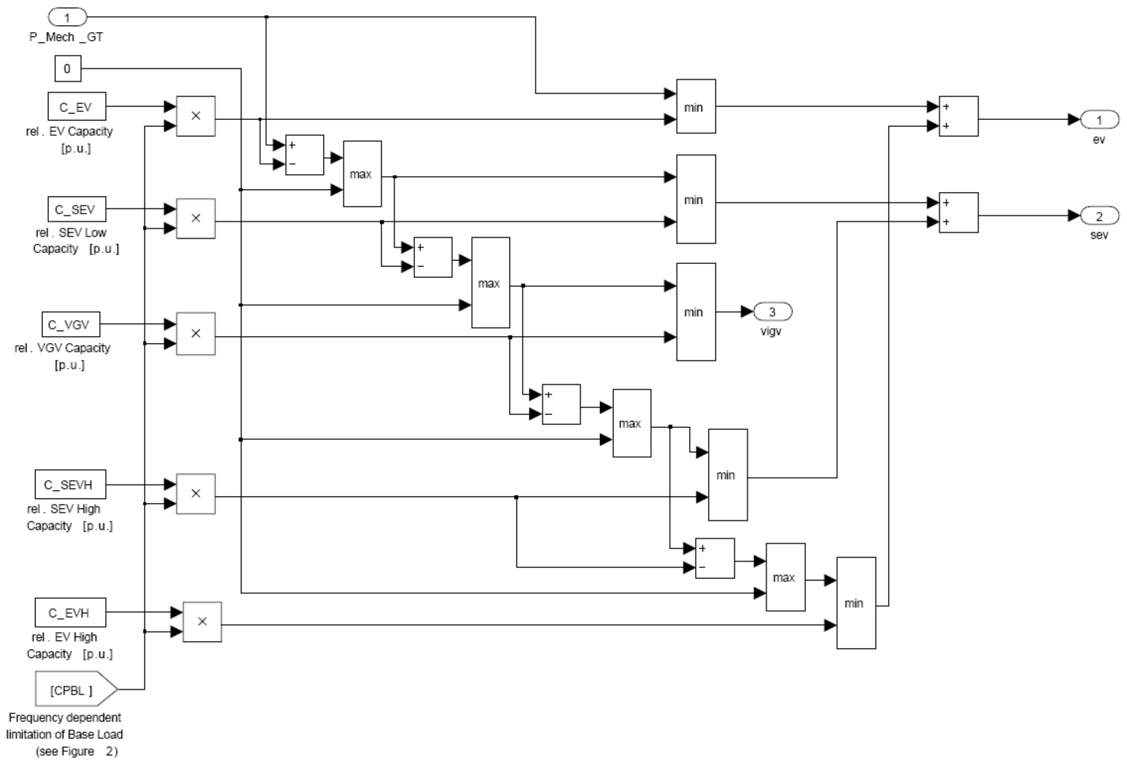


Figure E-8: Current Power Distribution Controller

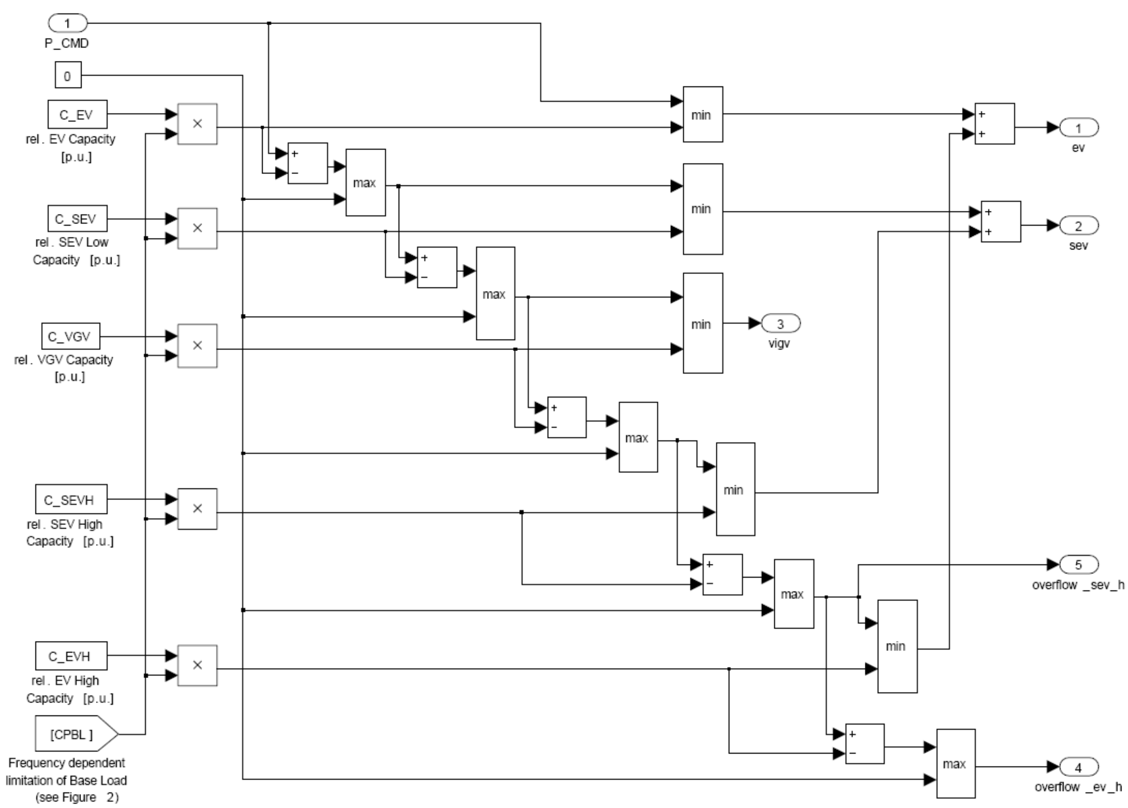


Figure E-9: Command Power Distribution Controller

Load Controller (Figure E-10)

Load Controller is used to vary fuel flows and VGV position until the commanded power is achieved. The Load Controller consists of the combined EV power controller, combined SEV power controller and combined VGV power controller. A combined power controller integrates advantages of both the feedback controller and feed forward controller. The EV and SEV fuel mass flow commands are sums of the outputs of the combined power controllers and the FFWD controllers from the VGV command.

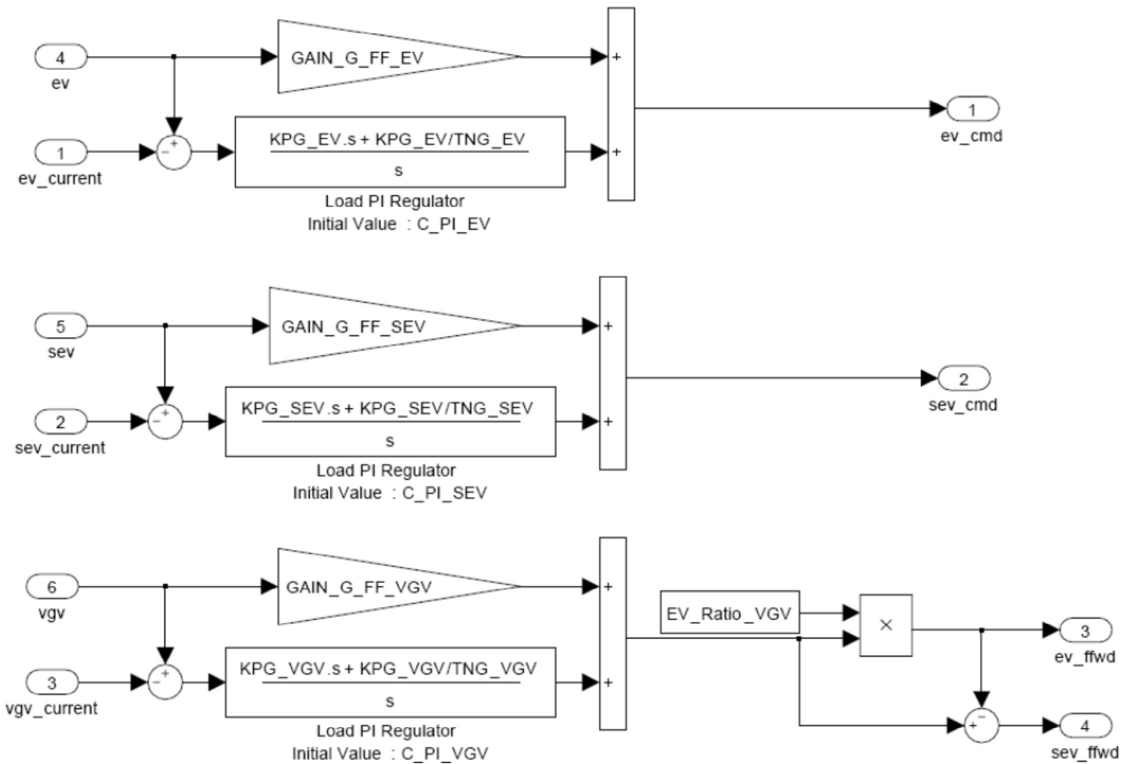


Figure E-10: Load Controller

Temperature Controller (Figure E-11)

Temperature controller is used to ensure that the GT temperatures do not exceed the set point limits. When the turbine is at its maximum power output, the temperature controller will take over command of the fuel mass flow as GT is approaching the temperature limits. Any fluctuation in ambient air conditions (mainly ambient temperature) and/or shaft speed will result in a change in the airflow through compressor. Therefore, the power output will change due to a change in fuel demand as determined by the temperature limiter.

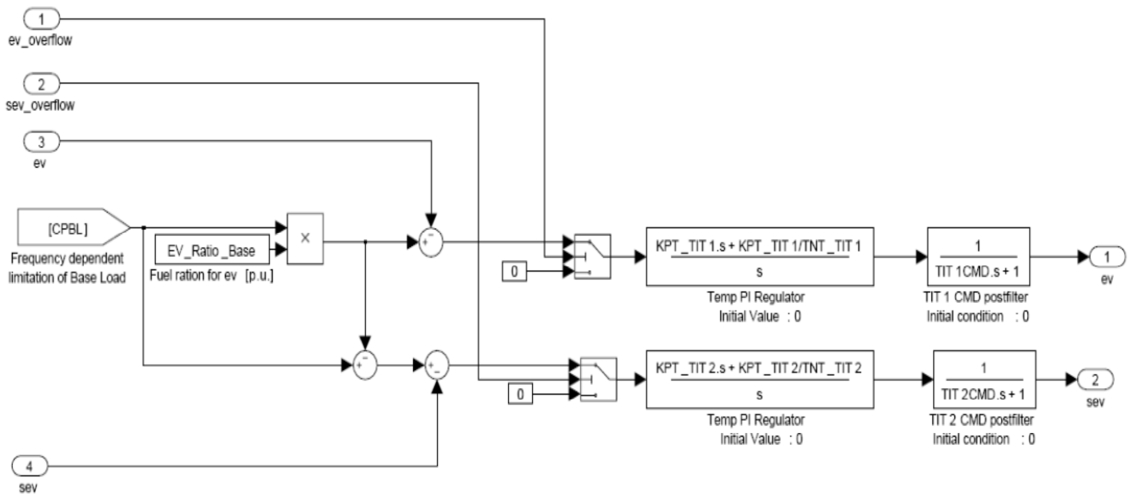


Figure E-11: Temperature Controller

GT dynamics subsystem (Figure E-12)

The dynamics of the Gas Turbine is represented by three transfer functions, i.e., representing the EV fuel system, the SEV fuel system and the gas turbine itself. These lag functions also include a series connected transport delay to simulate a dead time related to these systems.

The kinetic power of the GT P_Kin_GT is also calculated in this subsystem. P_Kin_GT is used basically together with the generator electrical power P_Elect_GT to determine the GT mechanical power for feedback control purposes.

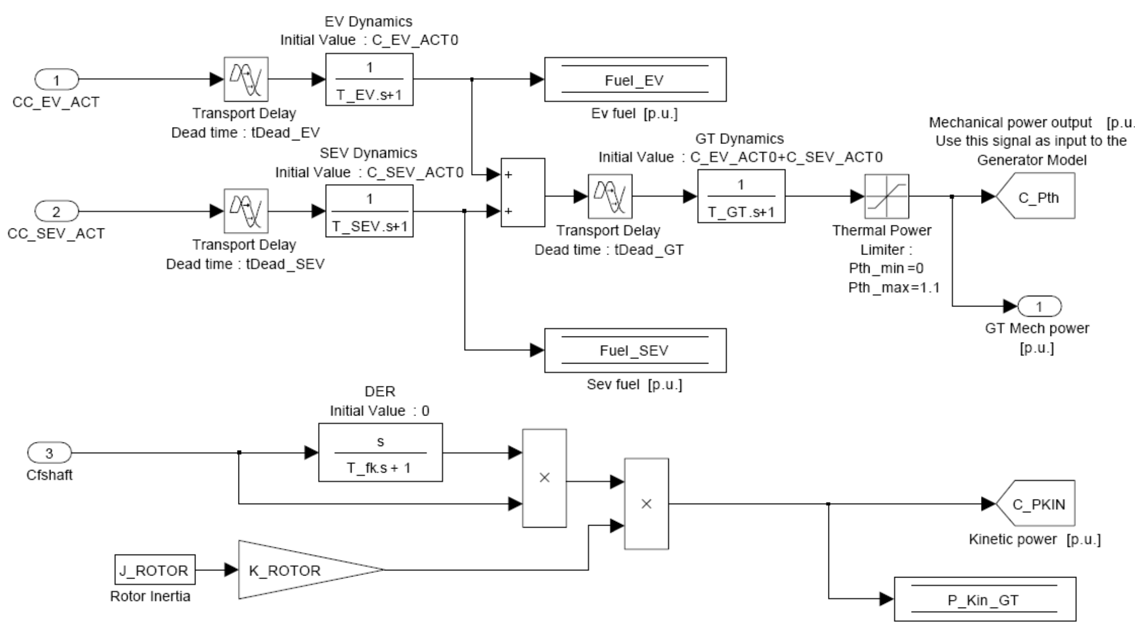


Figure E-12: GT Dynamics Subsystem

Model parameters

Parameter	Value	Unit
PREF	283	[MW]
Cfm	Variable	[Hz]
Cfc	50	[Hz]
CShaft	Variable	[Hz]
Tambient	15.0	[°C]
Relative Power Output	f (Freq., Tamb.)[1]	p.u.
Extended FR Support	f (Freq., Tamb.)[2]	p.u.
IExtFR	0 or 1	-
iFreqRes	0 or 1	-
T_f1	0.5	[s]
T_ZETA	8	[s]
DYN_DB	0.015	[Hz]
DYN_DB_RC	0.001	[Hz]
T_f2	0.5	[s]
STAT_DB_ASYMM	0	[Hz]
D	0.0315	-
iDBSens	0 or 1	-
db	0.015	[Hz]
dbl_pos	5	[Hz]
dbl_pos_RC	5-db	[Hz]
dbl_neg	-0.4	[Hz]
dbl_neg_RC	-0.4+db	[Hz]
Rel_Load_Setpoint	Variable	p.u.
LoadSP_Ramp	15/P_REF	p.u./s
LD_FR_MIN	0.496	p.u.
LD_RT_LIM	10 / P_REF	p.u.
LD_SP_MAX	1.2	p.u.
LD_SP_MIN	30/P_REF	p.u.

Parameter	Value	Unit
T_TIT1	0.9	[s]
T_TIT2	0.9	[s]
C_EV	0.0536	p.u.
C_SEV	0.1679	p.u.
C_VGV	0.75	p.u.
C_SEVH	0.0285	p.u.
C_EVH	0.0285	p.u.
GAIN_G_FF_EV	0.6	-
GAIN_G_FF_SEV	0.775	-
GAIN_G_FF_VGV	0.85	-
KPG_EV	0.00167*P_REF	p.u.
KPG_SEV	0.00167*P_REF	p.u.
KPG_ VGV	0.001176*P_REF	p.u.
TNG_EV	3	[s]
TNG_SEV	3	[s]
TNG_VGV	3	[s]
EV_Ratio_VGV	0.5	-
EV_Ratio_Base	0.49	-
KPT_TIT1	0.78	p.u.
KPT_TIT2	0.86	p.u.
TNT_TIT1	3.5	[s]
TNT_TIT2	3.5	[s]
TIT1CMD	1	[s]
TIT2CMD	1	[s]
tDead_EV	0.5	[s]
T_EV	1.5	[s]
tDead_SEV	0.5	[s]
T_SEV	1.5	[s]
tDead_GT	0.5	[s]
T_GT	2	[s]
Pth_min	0	p.u.

Parameter	Value	Unit
Pth_max	1.1	p.u.
T_fk	0.5	[s]
J_ROTOR	53567	[kgm^2]
K_ROTOR	$(2*pi)^2*1e-6/P\_REF$	p.u.

[1] At 15°C,

Relative Power Output = M0_15 + M1_15*Freq + M2_15*$Freq^2$ + M3_15*$Freq^3$

where, M0_15 = -6.65130e1; M1_15 = 3.77491e0; M2_15 = -6.99148e-2;
M3_15 = 4.28436e-4;

[2] At 15°C and based on the UK NGET Grid Code[11] requirements,

Extended FR Support = interp1(C633_F,C633_L, Freq, 'linear', 'extrap');

where, C633_F = [47,49.5,50.5,51.5] Hz & C633_L = [0.95,1,1,0.8] p.u;

[11] National Grid Electricity Transmission; "The Grid Code – Issue 4", 2010.

APPENDIX F: ALSTOM Combined Cycle Power Plant Models

F.1 ALSTOM Combined cycle power plant turbine-governor HRSG and steam turbine dynamic models

A conventional combined cycle power plant (CCPP) in its simplest form consists of one gas turbine (GT) with its hot exhaust gas being used to feed a heat recovery steam generator (HRSG) that subsequently produces steam to drive a steam turbine (ST). Such a configuration can produce an energy efficiency of up to 60%.

There are two basic schemes of CCPP, namely in multi-shaft (MS) and single-shaft (SS) configuration. Multi-shaft CCPP consists of GT and ST on separate power trains driving its own generators. In a single-shaft CCPP the GT and the ST are coupled on the same power train that is driving a single generator.

This appendix provides information on how ALSTOM models the MS and SS CCPP. For the MS CCPP a 2 on 1 arrangement, namely 2 GT with 2 HRSG and 1 ST, example for the model is used.

The ALSTOM GT26 model described in Appendix E on GT modeling is used in the modeling of the CCPP. References to GT parameters correspond to those described in Appendix E on ALSTOM GT modeling.

CCPP	Combined Cycle Power Plant
CLC	Closed Loop Control
CV	Control Valve
EV	Environmental Burner
FCM	Frequency control module
FFWD	Feed forward
FR	Frequency response
FS	Frequency support
GT	Gas Turbine
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IP	Intermediate Pressure
LP	Low Pressure
MS	Multi-Shaft

PI	Proportional integral regulator
SEV	Sequential Environmental Burner
SF	Supplementary Firing
SOC	Superordinate Control
SS	Single-Shaft
ST	Steam Turbine
TIT	Turbine inlet temperature
VGV	Variable Guide Vane
WSC	Water Steam Cycle

For the CCPP models described in this chapter, the per unit power base used for the GT and ST is based on their rated power at a given ambient temperature (normally at design point). Nominal frequency is 50Hz.

F.2 Model Application

The CCPP dynamic model described in this appendix can be used to investigate the dynamic behavior of the power plant and its impact on the grid during frequency disturbance events.

The model is valid under the following conditions:

- The model provides an accurate representation of its dynamic behavior with the governor operating in normal droop mode (e.g. 5%) when connected to a grid.
- Fuel gas operation.
- For investigating the frequency response behavior of the CCPP, the model can be used within a frequency range of 94% and 104% of the nominal frequency
- Ambient temperature range (from -9°C to 32°C).
- The load range is valid for a 50 to 100% GT load.
- For MS CCPP it is assumed that the load distribution between the two gas turbines is balanced.
- Supplementary firing is only possible for plant loading greater than 90%. Only detailed in MS model.

For island or partial load rejection operation in an isolated grid, supplementary control features are required that are not in the scope of this appendix.

F.3 Alstom KA26 CCPP simplified dynamic models

The ALSTOM's KA26 CCPP is available in two main configurations, namely:

- Multi-shaft configuration consisting of 1 or more gas turbines that provide exhaust heat to generate steam to drive 1 or more steam turbines each connected to its own generator.
- Single-shaft configuration consisting of 1 gas turbine that provides exhaust heat to generate steam to drive 1 steam turbine connected to a common generator.

The modeling of the ALSTOM's KA26 CCPP follows to a large extent the CIGRE modeling recommendations [9]. The model is based on a modular structure consisting of subsystems. Each subsystem provides a simplified model of a particular function of the CCPP plant consisting of:

- The GT26 gas turbine.
- The heat recovery steam generation (HRSG) system.
- The steam turbine.
- Related control systems

The CCPP models for the multi-shaft and single-shaft configurations are presented in following sections.

F.4 ALSTOM KA26 multi-shaft CCPP simplified dynamic model

The GT model for the GT26 is the same as that described in the previous section on gas turbine models. An overview of the HRSG and ST subsystems for the MS CCPP is shown in Figure F-1.

Heat recovery steam generation subsystem (Figure F-2)

The HRSG subsystem consists of HRSG steam generation and HRSG Control functions. The HRSG steam generation includes also the supplementary firing system. The HRSG dynamics is represented in a simplified manner by a lag function with a time constant TI being its thermal inertia. Each GT is connected to its own HRSG with its individual supplementary firing system. Each supplementary firing system dynamics is also represented by two lag functions in series with a time constant TL_SF being the thermal inertia due to changes of supplementary firing load. Normally supplementary firing is not available during frequency support operation of the ST or when the GT power is below 90%.

The HRSG steam production is derived from the GT relative power. A linear temperature correction of the ST base load value is applied so as to allow for the impact of air-cooled condenser at elevated ambient temperature.

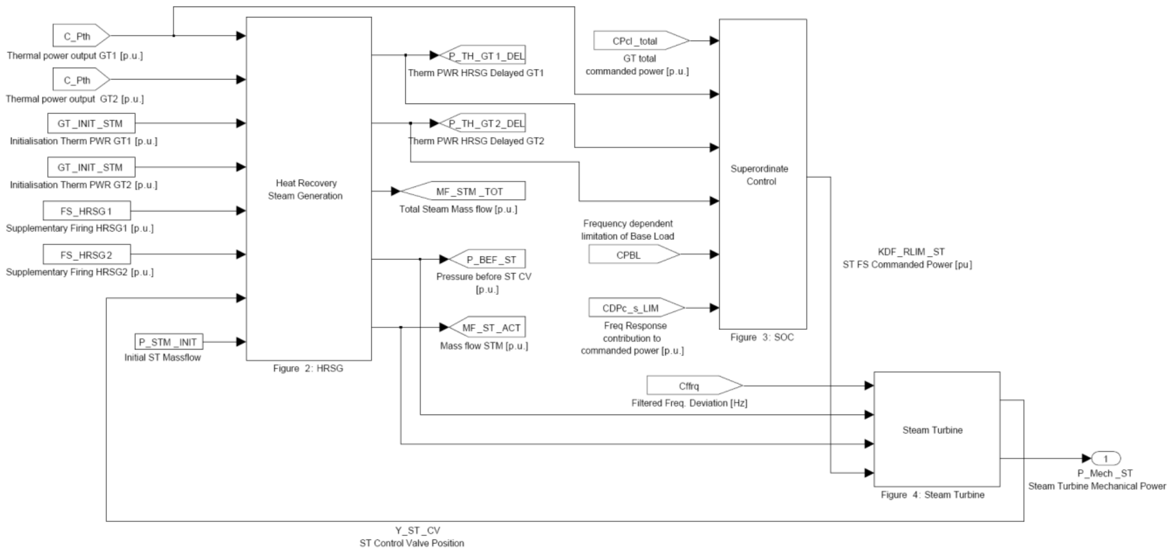


Figure F-1: Overview of the ALSTOM KA26 CCPP MS HRSG and ST Simplified Dynamic Model

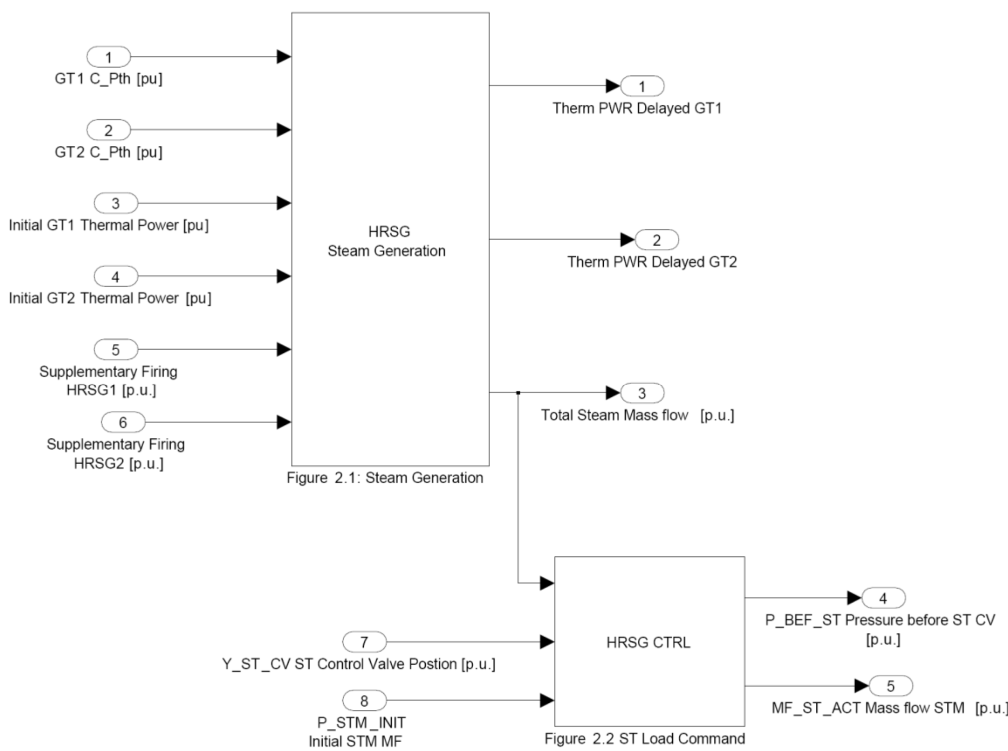


Figure F-2: Overview of the HRSG subsystem.

The HRSG steam generation and the HRSG control functions are implemented as shown in Figures F-3 and F-4, respectively.

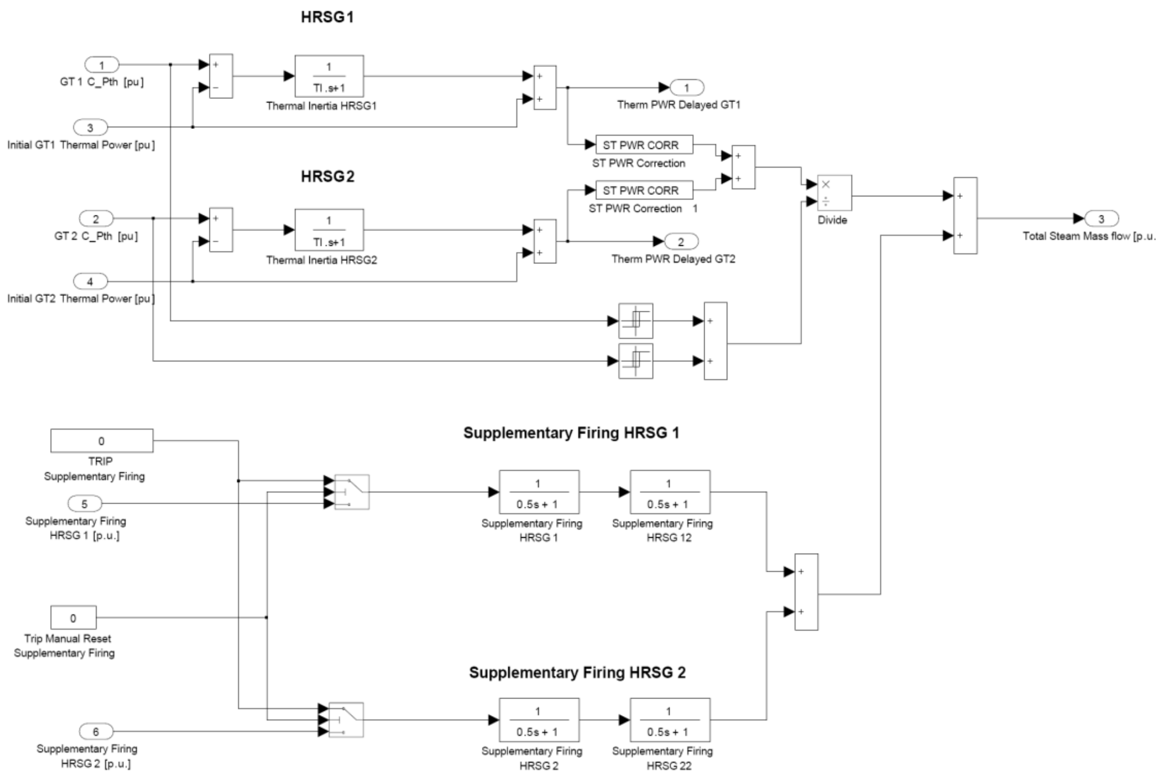


Figure F-3: HRSG Steam Generation Function.

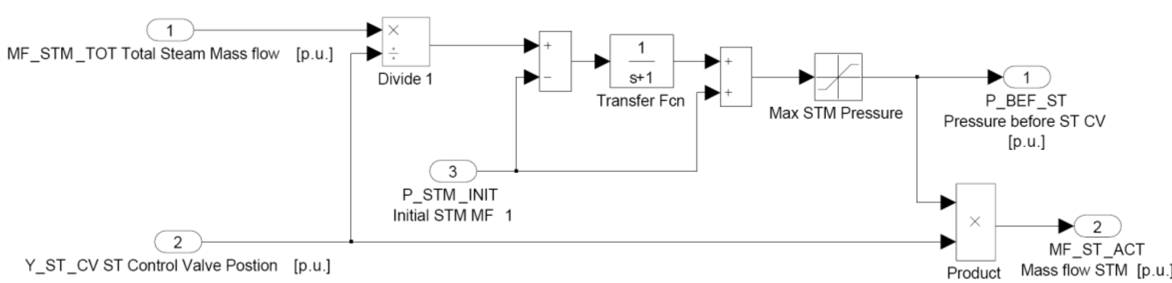


Figure F-4: HRSG Control Function.

Superordinate control subsystem (Figure F-5)

The SOC subsystem consists of control functions that provide additional features to the standard functionality of the steam turbine. These features may include primary frequency support, partial load rejection, islanding capabilities. In this appendix only the primary frequency support capability is presented. The implementation of this subsystem is shown in Figures F-5, F-6, F-7 and F-8.

The ST primary frequency support control function is an optional feature of ALSTOM's CCPP. In conventional combined cycle plant, the steam turbine is operated in sliding pressure mode and the output of the ST only follows the output of the GT and does not contribute to the primary frequency response of the plant.

The ST primary frequency support control function however allows ALSTOM's CCPP to actively participate in providing primary frequency response for a limited time. The temporary ST power contribution is superimposed on the command for power variation. The ST actual power variation is constrained by capacity limitations calculated in the ST FS Commanded Power Calculation function shown in Figure F-7.

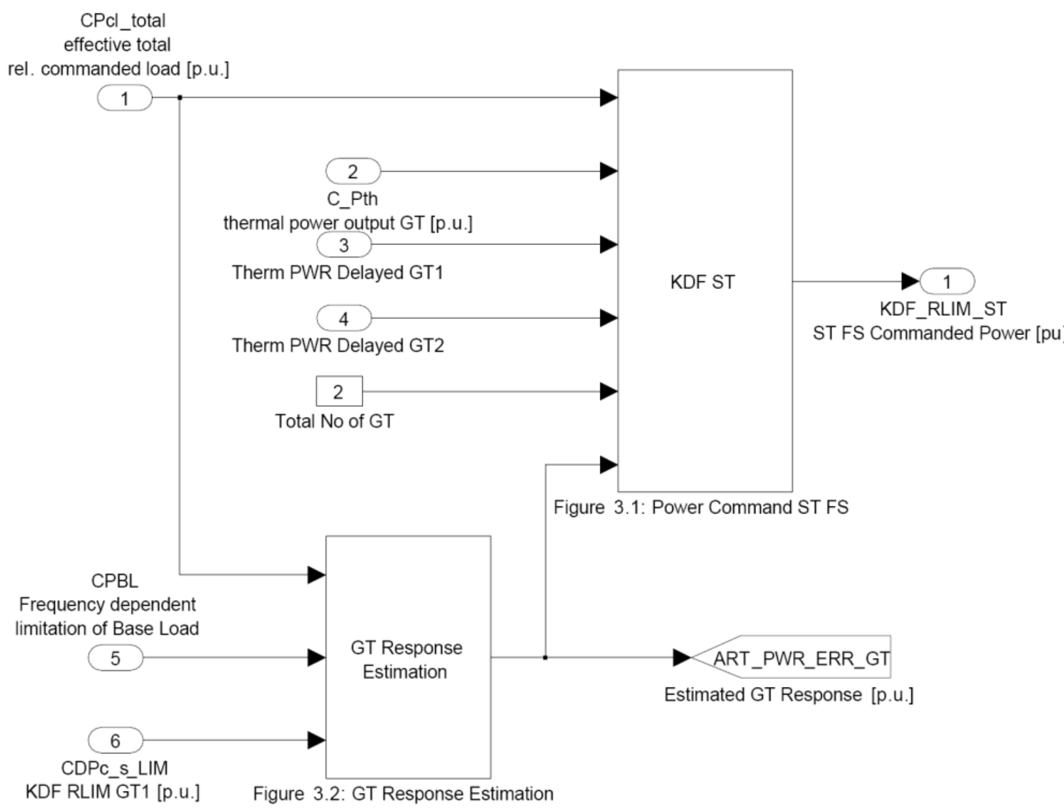


Figure F-5: Overview of SOC Subsystem with Primary Frequency Support Function.

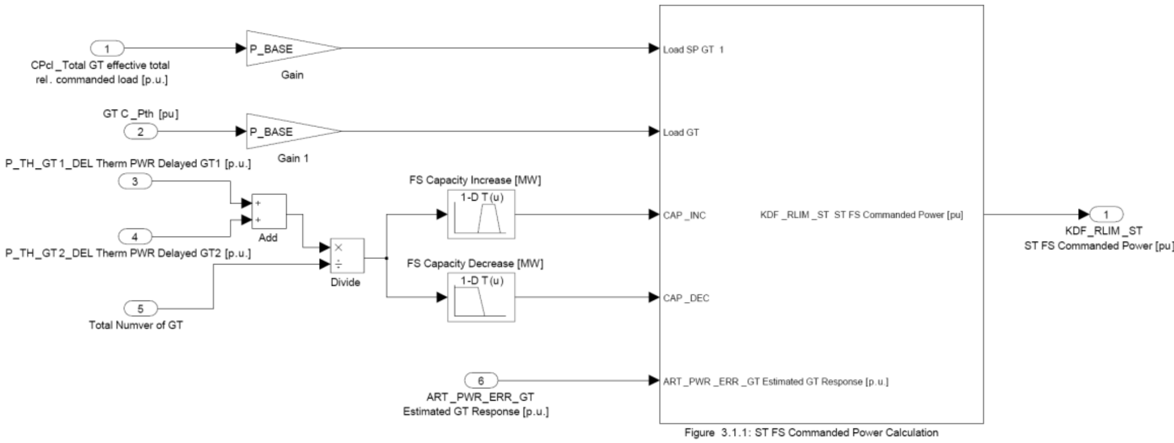


Figure F-6: Power Command ST Frequency Support Function.

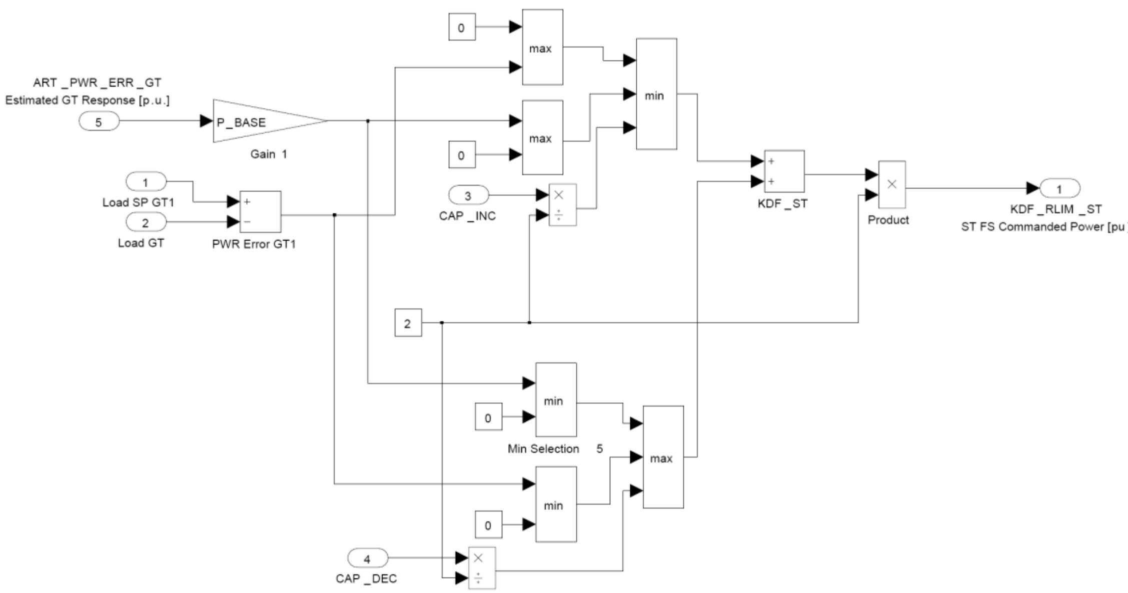


Figure F-7: ST FS Commanded Power Calculation Function.

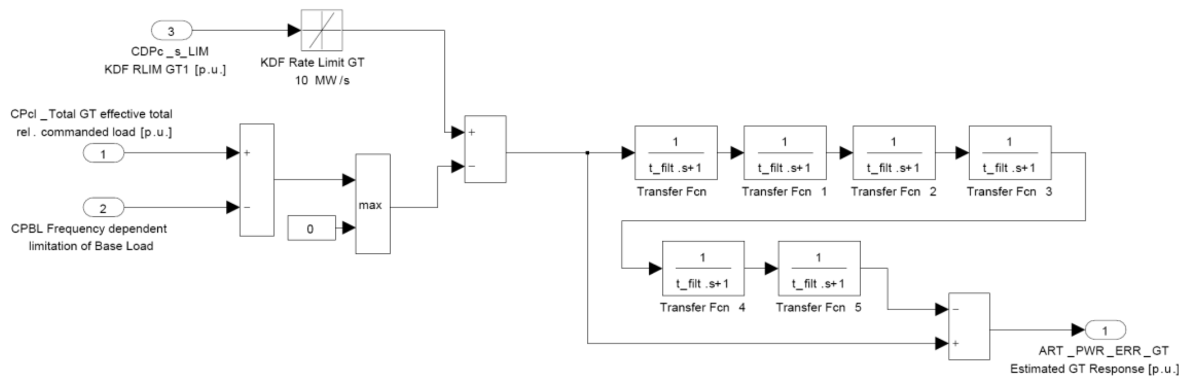


Figure F-8: GT Response Estimation Function.

Steam turbine subsystem (Figure F-9)

The steam turbine subsystem consists of the steam turbine control and the steam turbine dynamics functions. The overview of the implementation of this subsystem is shown in Figure F-9. For the interface between the HRSG and ST only live steam admission has been accounted for, .i.e, no reheat steam or low pressure steam. The output from this subsystem is the ST mechanical power that is used to drive the ST generator.

The steam turbine control valve command is generated in the ST control function as shown in Figure F-10. Project specific characteristics of the control valve and actuator are integrated to account for non-linear effects. The control valve operation is employed in general for positive frequency excursions exceeding a static dead band. The high-pressure servo motor model that is controlling the control valve is shown in Figure F-11.

Figure F-12 shows the implementation of the model describing the steam turbine dynamics. The dynamics of primary frequency support for the ST is also considered.

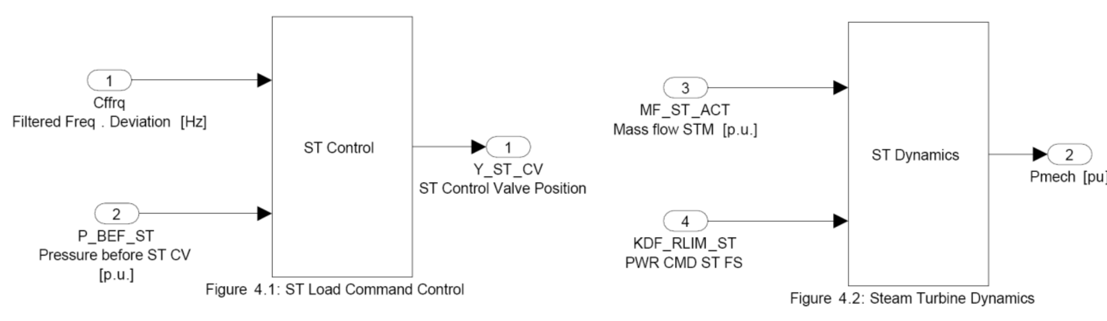


Figure F-9: Overview of Steam Turbine Subsystem.

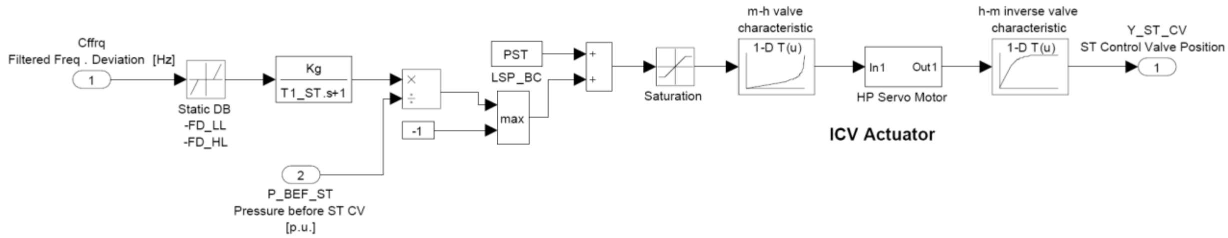


Figure F-10: Steam Turbine Load/Valve Control.

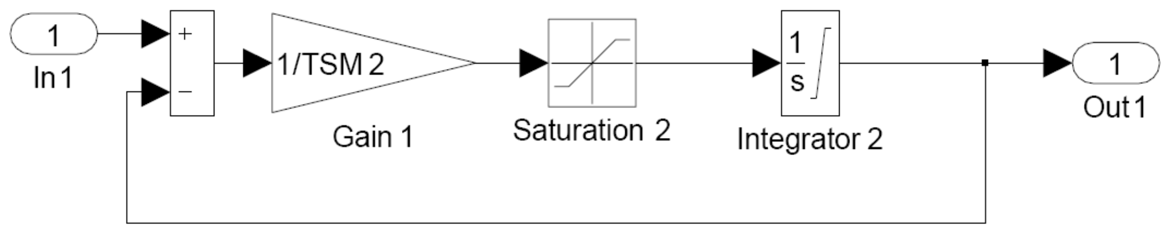


Figure F-11: HP Servo Motor Model.

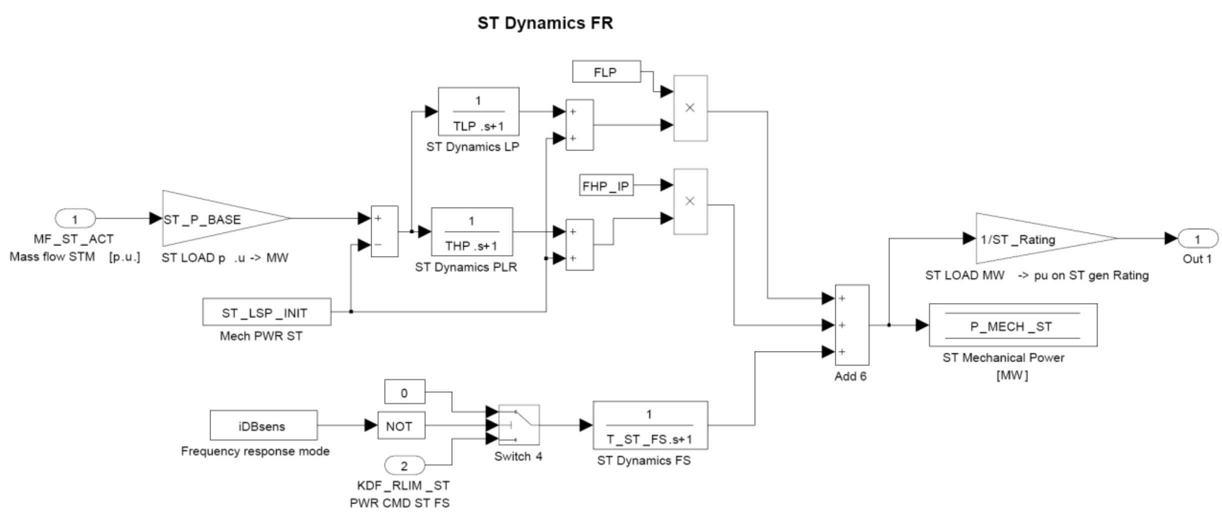


Figure F-12: Steam Turbine Dynamics.

ALSTOM multi-shaft CCPP HRSG and ST model parameters

Parameter	Value	Unit
GT_INIT_STM	Variable	p.u.
Cffrq	Variable	[Hz]
FS_HRSG1	Variable	p.u.
FS_HRSG2	Variable	p.u.
P_STM_INIT	Variable	p.u.
TI	145	s
ST PWR CORR	Look-up table	p.u.
P_STM_LIM (Max STM Pressure)	1.5	p.u.
P_BASE	Variable	MW
t_filt	0.5	s
FD_LL	-3	[Hz]
FD_HL	0.05	[Hz]
Kg	20	-
T1_ST	0.1	s
PST	1.0	p.u.
TSM2	0.1	s
CCL	-5	p.u./s
COP	0.1	p.u./s
ST_P_BASE	Variable	MW
ST_LSP_INIT	Variable	MW
TLP	0.2	s
THP	0.1	s
FLP	0.435	p.u.
FHP_IP	0.565	p.u.
T_ST_FS	0.5	s
iDBSens	0 or 1	-

Valve characteristics given by:
m_ICV=[0 0.029 0.671 0.723 0.767 0.785 0.802 0.817 0.831 0.855 0.866 0.876 0.884 0.893 0.907 0.919 0.93 0.943 0.957 0.967 0.98 1]

h_ICV =[0 0.01 0.21 0.23 0.25 0.26 0.27 0.28 0.29 0.31 0.32 0.33 0.34 0.35 0.37 0.39 0.41 0.44 0.48 0.52 0.6 1]
P_BASE for the GT @ 15°C is for this example 270.3 MW
ST_P_BASE @ 15°C is for this example 313.7 MW
GT_INIT_STM is the initial GT load setpoint in p.u.
Initial Load setpoint of ST ST_LSP_INIT is given by:
ST_LSP_INIT=interp1([0 0.2 1 2],[0 0.436 0.966 1],GT_INIT_STM)*ST_P_BASE
Initial ST massflow P_STM_INIT is given by:
P_STM_INIT=interp1([0 0.2 1 2],[0 0.436 0.966],GT_INIT_STM)+(FS_HRSG1+FS_HRSG2);

If supplementary firing is considered, the initial load setpoint of the ST will be corrected as given by:
ST_LSP_INIT=ST_LSP_INIT+(FS_HRSG1+FS_HRSG2)*ST_P_BASE
Frequency support capacities of the ST is given by:
For capacity increase
X1_GT_REL=[0 0.5 0.6 0.88 1 1.1]
Y_FS_INC=[0 0 27.2 27.2 0 0]
For capacity decrease
X2_GT_REL=[0 0.5 0.65 1 1.1]
Y_FS_DEC=[0 0 -27.2 -27.2 -27.2]
The Steam Flow Correction Table (ST PWR CORR) is given by:
[0 0.2 1 2]
[0 0.436 0.966 1]

F.5 ALSTOM KA26 single-shaft CCPP simplified dynamic model

An overview of the HRSG and ST systems for the SS CCPP is shown in Figure F-5. Similar to the model of the MS CCPP configuration the model consists of the following subsystems:

- Heat Recovery Steam Generation (HRSG).
- Superordinate Control (SOC).
- Steam Turbine (ST)

These subsystems are described in the following sections of this appendix.

The basic interface to the GT26 gas turbine and the generator model consist of:

- GT thermal power [C_PTH].
- GT frequency response contribution to GT commanded power [CDPc_s].
- GT target frequency in Hz [Cfc].
- GT measured frequency in Hz [Cfm].
- GT mechanical power [P_Mech_GT].
- ST mechanical power [P_Mech_ST].

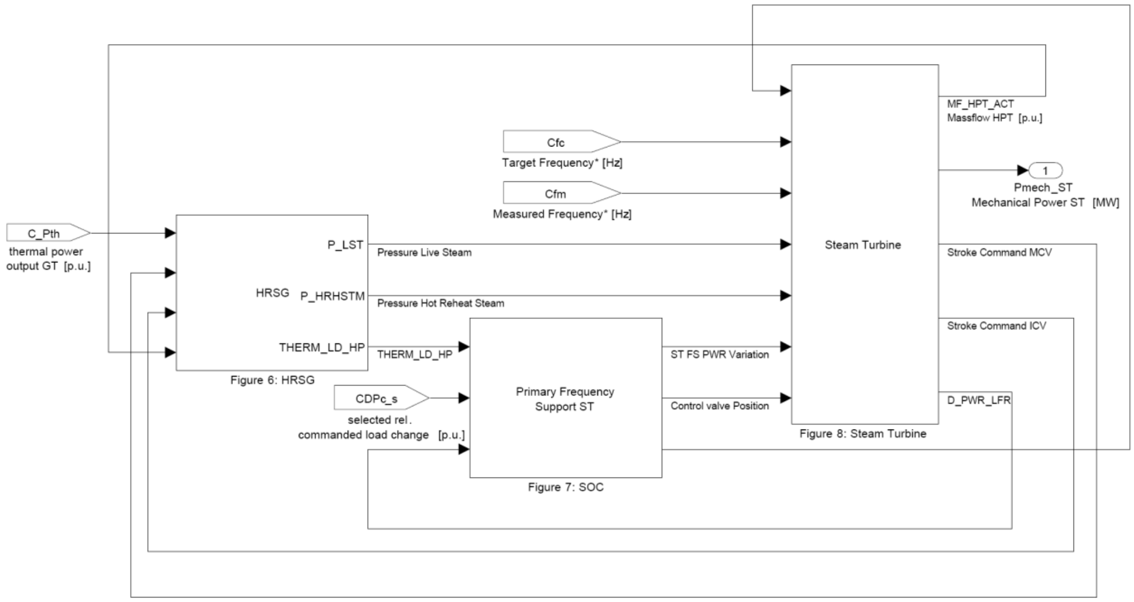


Figure F-13: Overview of the ALSTOM KA26 CCPP SS HRSG and ST Simplified Dynamic Model

Heat recovery steam generation subsystem (Figure F-13)

The implementation of the HRSG subsystem consists of the HP and IP sections as shown in Figure F-14. As in the MS configuration only the frequency support feature of the steam turbine is considered in this model.

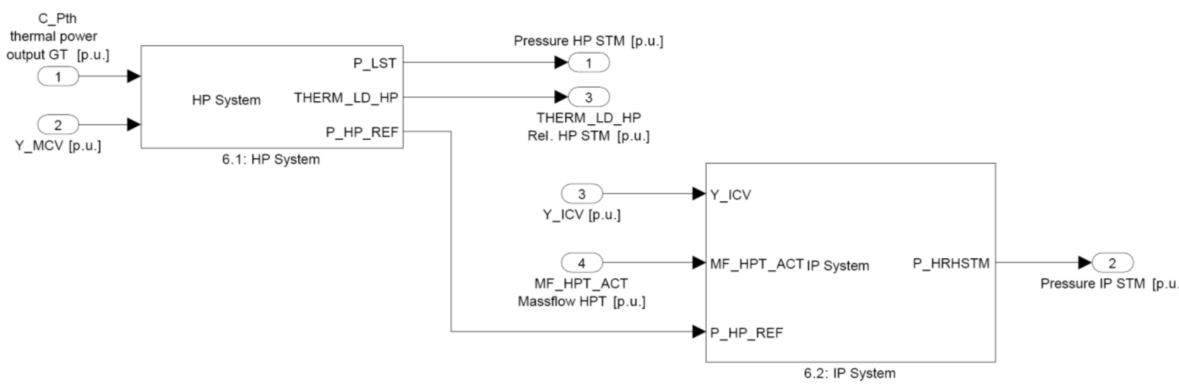


Figure F-14: Overview of HRSG Subsystem.

The dynamic behavior of the HP and IP Systems is defined by the following HRSG characteristics:

- Thermal inertia of the HP system.
- Storage capacity of the HP system.
- Storage capacity of the IP system.

The HP and IP systems are shown in Figures F-15 and F-17, respectively.

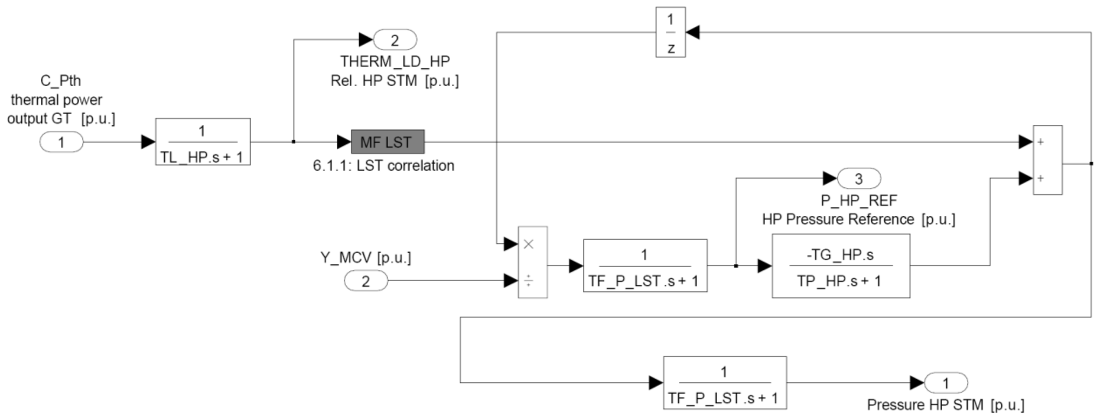


Figure F-15: HP System.

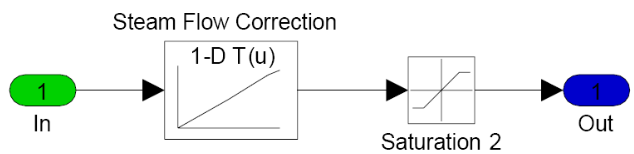


Figure F-16: Live Steam (LST) Correlation Function.

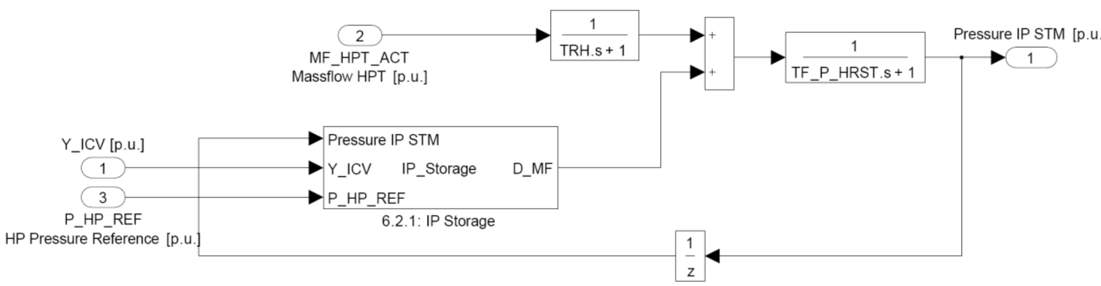


Figure F-17: IP System.

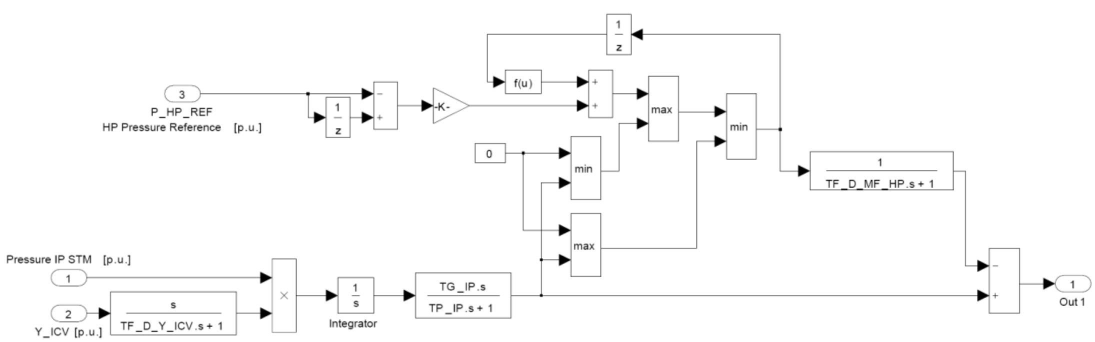


Figure F-18: IP Storage.

Superordinate control subsystem (Figure F-19)

The SOC subsystem consists of control functions that provide additional features to the standard functionality of the steam turbine. These features may include primary frequency support, partial load rejection, islanding capabilities. In this appendix only the primary frequency support capability is presented. The implementation of this subsystem is shown in Figures F-19.

The steam primary frequency support control function is an optional feature of ALSTOM's CCPP. In conventional combined cycle plant, the steam turbine is operated in sliding pressure mode and the output of the ST only follows the output of the GT and does not contribute to the primary frequency response of the plant.

The steam primary frequency support control function however allows ALSTOM's CCPP to actively participate in providing primary frequency response for a limited time. The temporary ST power contribution is superimposed on the command for power variation. The ST actual power variation is constrained by capacity limitations.

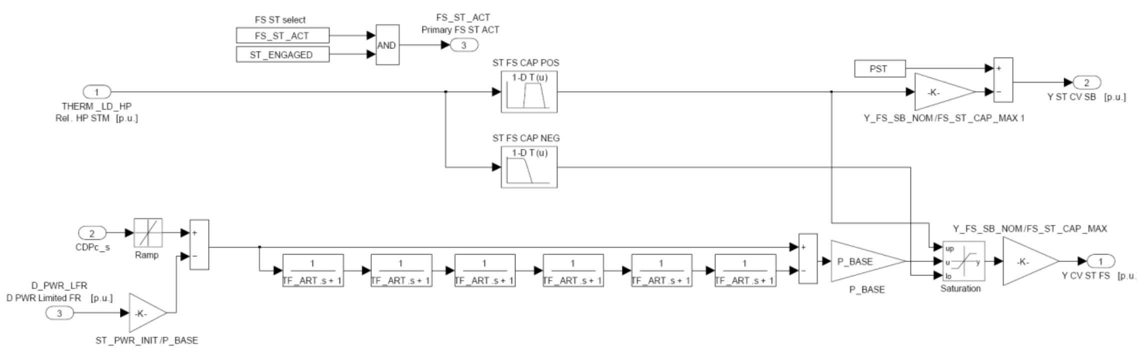


Figure F-19: SOC Subsystem with Primary Frequency Support Function.

Steam turbine subsystem (Figure F-20)

The steam turbine subsystem consists of the steam turbine control and the steam turbine dynamics functions. The overview of the implementation of this subsystem is shown in Figure F-20. For the interface between the HRSG and ST, the live steam admission, reheat steam and low pressure steam is taken into account. The output from this subsystem is the ST mechanical power that is added to the GT mechanical power to yield the total mechanical power to drive the single-shaft CCPP generator.

The steam turbine control valve command is generated in the ST control function as shown in Figure F-21. Figure F-22 shows the limited frequency support block. Project specific characteristics of the control valve and actuator are integrated to account for non-linear effects. The control valve operation is employed in general for positive frequency excursions exceeding a static dead band. This is however deactivated in case primary frequency support of the ST is selected. The HP and IP servo motor models that are controlling the respective control valves are shown in Figures F-23 and F-24.

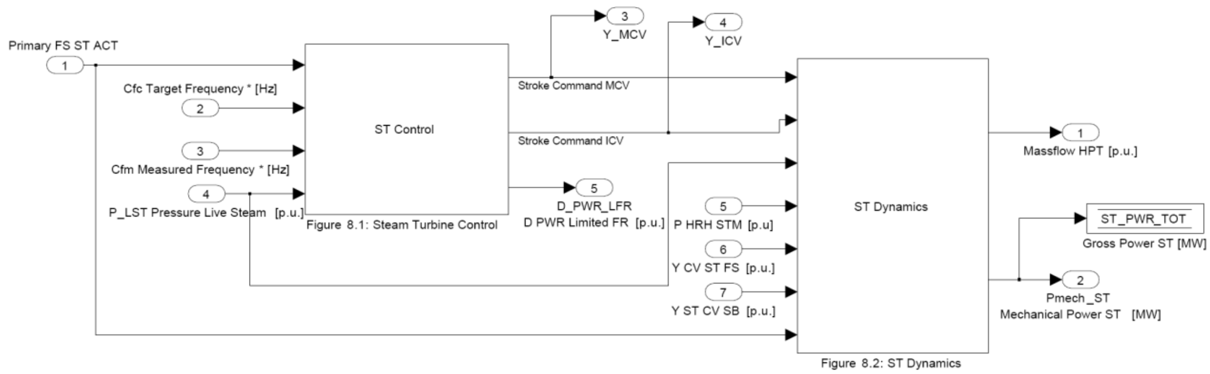


Figure F-20: Overview of Steam Turbine Subsystem.

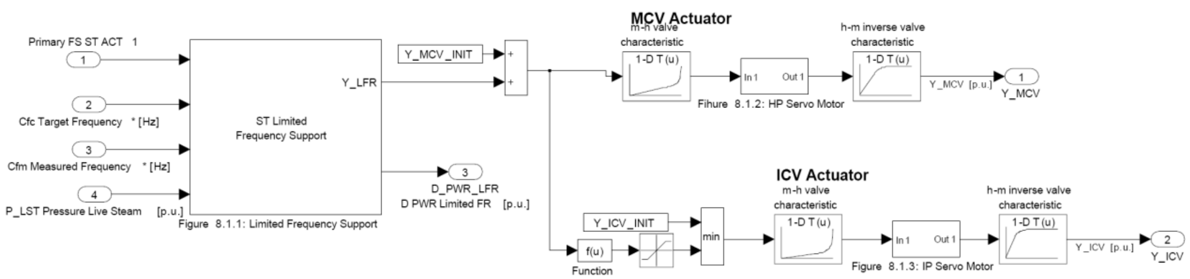


Figure F-21: Steam Turbine Control.

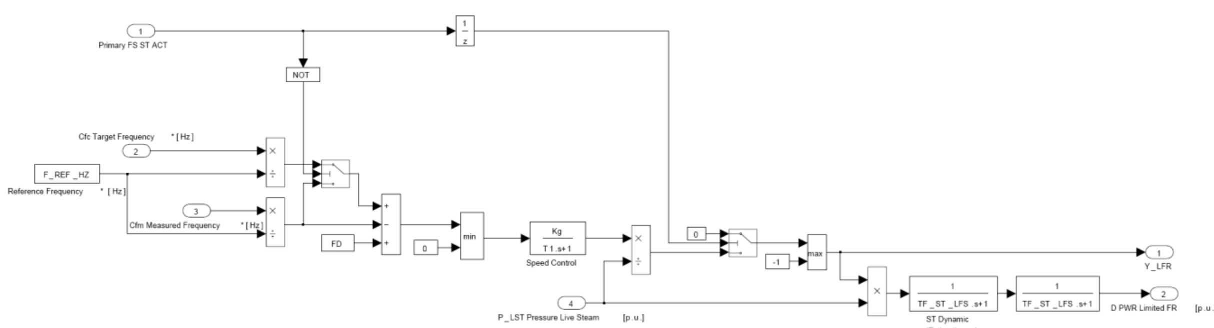


Figure F-22: Limited Frequency Support.

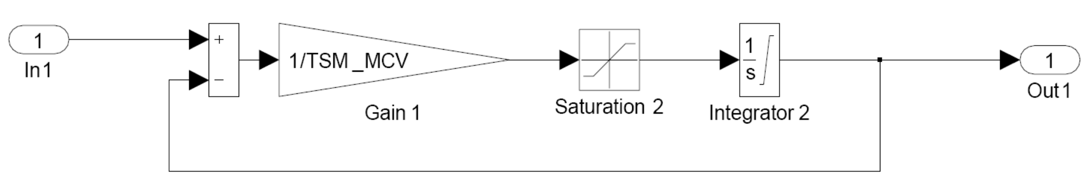


Figure F-23: HP Servo Motor Model.

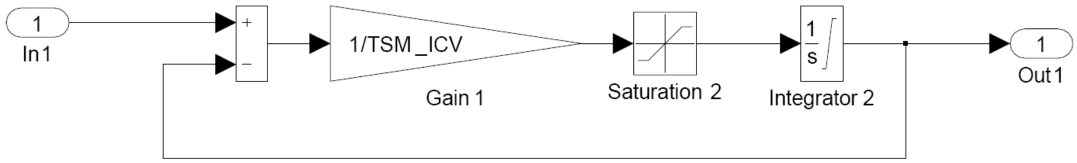


Figure F-24: IP Servo Motor Model.

Figure F-25 shows the implementation of the model describing the steam turbine dynamics

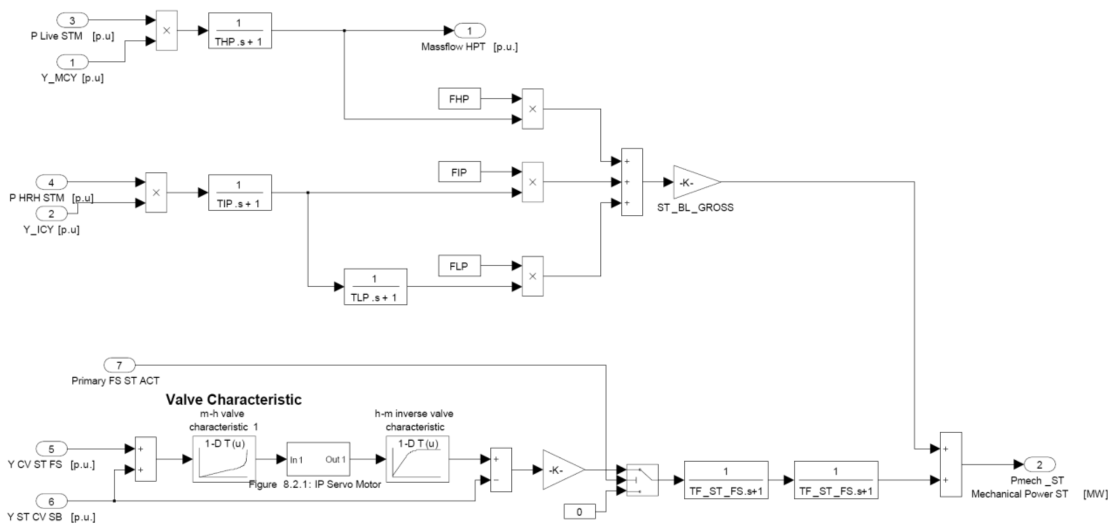


Figure F-25: Steam Turbine Dynamics.

ALSTOM single-shaft CCPP HRSG and ST model parameters

Parameter	Value	Unit
TL_HP	120	s
TF_P_LST	2	s
TG_HP	150	-
TP_HP	10	s
TRH	1.0	s
TF_P_HRST	2	s
TF_D_Y_ICV	0.1	s
TG_IP	130	-
TP_IP	150	s
TF_D_MF_HP	1.0	s
TF_ART	0.4	s
P_BASE	Variable	MW
Y_FS_SB_NOM	0.1	p.u.
Y_MCV_INIT	1.0	p.u.
Y_ICV_INIT	1.0	p.u.
FD	0.008	p.u.
Kg	20	-
T1	0.1	s
TF_FS_LFS	1.0	s
F_REF_HZ	50.0	Hz
TSM_MCV	0.1	s
CCL_MCV	-5.0	p.u./s
COP_MCV	0.1	p.u./s
TSM_ICV	0.1	s
CCL_ICV	-5.0	p.u./s
COP_ICV	0.1	p.u./s
THP	0.1	s
TIP	0.1	s
TLP	0.2	s
FHP	0.205	p.u.

Parameter	Value	Unit
FIP	0.315	p.u.
FLP	0.480	p.u.
TSM	0.1	s
CCL	-5.0	p.u./s
COP	0.1	p.u./s

Valve characteristics given by:
m_MCV=[0 0.026 0.668 0.724 0.75 0.793 0.828 0.857 0.882 0.892 0.902 0.919 0.933 0.945 0.95 0.959 0.967 0.973 0.98 0.987 0.993 1]
h_MCV=[0 0.01 0.26 0.28 0.29 0.31 0.33 0.35 0.37 0.38 0.39 0.41 0.43 0.45 0.46 0.48 0.5 0.52 0.55 0.59 0.66 1]
m_ICV=[0 0.041 0.654 0.722 0.75 0.774 0.795 0.814 0.846 0.859 0.872 0.882 0.892 0.901 0.908 0.922 0.933 0.943 0.954 0.968 0.98 1]
h_ICV=[0 0.01 0.15 0.17 0.18 0.19 0.2 0.21 0.23 0.24 0.25 0.26 0.27 0.28 0.29 0.31 0.33 0.35 0.38 0.43 0.5 1]
P_BASE for the GT @ 15°C is for this example 270.3 MW
GT_INIT_STM is the initial GT load setpoint in p.u.

APPENDIX G: Solar Turbine's Perspective on the GT Models

Solar Turbines maintains a variety turbine dynamic performance models with various levels of complexity and accuracy.

- Detailed first principles models are used internally for product development, transient performance prediction and validation of simpler models. These models contain company confidential information.
- Medium complexity models in formats developed by Solar for use both internally and by Solar's customers provide reasonable estimation of transient behavior to a wide range of scenarios.
- Models using formats requested by Solar's customers or using industry standard formats have been developed primarily for use by Solar's customers. These models provide reasonable estimates of transient behavior for a limited range of scenarios. Accuracy may drop off significantly as the magnitude of the scenario disturbance increases or conditions become extreme.

In recent years, requests for power system modeling data have increased dramatically. Many of these requests are for stability studies required by utilities for grid connection of Solar packages. A significant number of requests are for analysis of operation of Solar packages in small power generation islands. Sudden load acceptance and rejection capability and load shedding system design are often of concern.

Solar Turbine's customers use a variety of power system modeling programs. Their expertise in power system modeling and analysis varies from expert to novice. Some understand all the principles involved and can translate any model into one usable in their modeling program. At the other extreme are some who can do no more than fill in the parameters required by their modeling software and follow the user manual to run the model with little to no understanding of how the model works or what the parameters represent.

The support Solar needs to provide for its customer's modeling efforts has become significant, particularly for turbine governor modeling. IEEE AVR/Exciter models and generator model are well accepted and require relatively little support. Manufacturer specific gas turbine governor models are widely used and industry standard gas turbine governor models appear to have little application.

Implementation of Solar Turbines governor models in commercially available power system stability analysis programs has had mixed success with more problems than Solar would like to solve. Solar sees high value to itself, to its customers, to power system modeling program providers, and to utilities, in the development of standard turbine governor models with the same level of acceptance as the industry standard AVR/Exciter models.

Solar's review of current models:

GAST

This model is very simple.

It provides only for droop control mode.

It provides a reasonable estimate of mechanical power response to small changes in speed or load reference

The addition of an isochronous control mode and / or a KW control mode would make this more useful but more complicated.

GAST2A

This model does not appear to provide for a good description of a Solar Turbines governor controls.

CIGRE Gas Turbine Model

This model is fairly simple. It provides a reasonable estimate of mechanical power response to small changes in speed or load reference.

GGOV1

Solar has defined parameters for the GGOV1 model to model Solar Turbines. Results are very similar to the CIGRE model.

Problem areas modeling of integral term windup prevention:

When the output of proportional plus integral control (PI control) or proportional plus integral plus derivative control (PID control) are overridden by limiters or alternative control modes, the integral term of the control has the tendency to keep integrating (wind up) and cause a wide discrepancy between the PID output and the active control signal. This leads to poor response when the PID control needs to become active again.

Good PID controls have a wind up prevention feature to provide smooth transition when limiters or other controls modes become inactive. Solar has experienced problems in the modeling of integrator windup prevention. An IEEE standard approach may be beneficial.

Alternative Control Modes:

The most basic simple models assume turbine steady state output torque droops with frequency (droop control). More sophisticated models include turbine temperature limiting control, isochronous control, KW control, import export control, load rate limiters, and/or acceleration control. An IEEE recommendation for a standard set of control modes to be modeled may be beneficial.

Large Transients and Lean Combustion Control:

Large upset conditions push turbines into increasingly complex non-linear responses. In particular lean combustion controls can initially slow down turbine response and have significant impact on the overall response to large transients.

Lean combustion controls are evolving quickly and therefore difficult to establish standard models for. Possible ways to deal with this are

- Provide simple universal models for minor power or frequency transients and more complex manufacture specific models for large transients.
- In an otherwise simple model, provide power limiting and rate limiting functions that vary with time and /or load that allow modeling the behavior of changing responses without modeling the actual physics involved which could result in a more universal less manufacturer specific model for a wider range of upset conditions.