PARTICIPATION
OF
COMBINED CYCLE POWER PLANTS
TO
POWER SYSTEM FREQUENCY CONTROL:
MODELING AND APPLICATION

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ABSTRACT

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This thesis proposes a method and develops a model for the participation of a combined cycle power plant to power system frequency control.

Through the period of integration to the UCTE system, *(Union for Coordination of Transmission of Electricity in Europe)* frequency behavior of Turkey's grid and studies related to its improvement had been a great concern, so is the reason that main subject of my thesis became as "Power System Frequency Control".

Apart from system-wide global control action (secondary control); load control loops at power plants, reserve power and its provision even at the minimum capacity generation stage, (primary control) are the fundamental concerns of this subject.

The adjustment of proper amount of reserve at the power plants, and correct system response to any kind of disturbance, in the overall, are measured by the quality of the frequency behaviour of the system.

A simulator that will simulate a dynamic gas turbine and its control system model, together with a combined cycle power plant load controller is the outcome of this thesis.

Keywords – combined cycle power plant, gas turbine, governor, reserve power, plant outer control loop.

ÖZ

KOMBİNE ÇEVRİM ENERJİ SANTRALLARININ
ŞEBEKE FREKANSI KONTROLUNE KATILIMI:
MODELLENMESI VE UYGULAMASI

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Bu tez, kombine çevrim elektrik santralının şebeke frekansı kontrolune doğru katılımının nasıl olması gerektiği üzerine, bir metot önerisi ve model uygulaması içermektedir.

Mezuniyetimden bu yana, UCTE sistemine entegrasyon çalışmalarının sürdürüldüğü bu dönem boyunca, Türkiye şebekesinin frekansının davranışı ve iyileştirilmesi yönünde yapılan çalışmalar her zaman gündemin üst sırasında yer aldı. Bu sebeple de lisansüstü eğitimimin başlangıcında, tez çalışmamın ana konusu "Enterkonnekte Sistemlerde Frekans Kontrolu" olarak belirlendi.

Santrallardaki güç kontrol döngülerinin analizinin ve rezerv güç miktarının sürekliliğinin; bu gücün en küçük üretim biriminde bile, uygun şekilde, gerektiğinde sisteme temin edilebilir olmasının, bu konunun en temel kavramları olduğu söylenebilir.

Sistemin geneli için, santrallarda rezerv güç miktarının yeterince ayarlanıp ayarlanmadığı ve herhangi bir frekans sapmasına verilen kollektif cevabın doğruluğu, şebeke frekansının davranış karakteristiği incelenerek değerlendirilir.

Bu tez çalışmasının sonucunda, dinamik bir gaz türbini ve kontrol sistemi modeli ile birlikte, doğalgaz kombine çevrim santrallarında kullanılabilecek yük denetleyicisi modelini içeren bir simulasyon elde edilmiştir.

Anahtar Kelimeler — kombine çevrim santralı, gaz türbini, yük kontrolu, rezerv güç, santral dış kontrol döngüleri.

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CHAPTER 1

INTRODUCTION

In the last five to ten years, there has been a huge trend in building gas turbine based power plants, especially combined cycle ones throughout the world. One of the main reasons for building those plants was the high overall efficiency (nearly %57) together with their comparably short construction time and ease of operability. The other concern was unfavorable responses of existing steam power based thermal power plants (i.e. coal fired) resulting in unexpected grid behaviour after major disturbances.

As the spinning reserve requirements, rather than base load requirements of the systems increased, this has led to the construction of many simple cycle gas turbine power plants in the US. The requirement of fast governor response with appropriate spinning reserve was also a need in Europe as a consequence of which many combined cycle power plants with powerful gas turbines were built.

Together with the above reasoning, and through the agreements that were signed with Russia in terms of natural gas sales, Turkey was obliged to use vast amount of natural gas and also get rid of an electrical energy crisis, so that construction of combined cycle power plants was a remedy.

Under the light of the explanations in the preceding paragraphs, as larger blocks of gas turbine based power plants were added to the generation system, importance of understanding their grid related operational characteristics (most importantly their primary control capabilities) and having their accurate dynamic models for the prediction of their effect to the power systems became apparent. This calls for appropriate modeling of the gas turbines as well as the power plant load controllers, which act on gas turbine unit controllers.

1.1 Issues Related to Power System Frequency Control and Justification of the Thesis Study

Primary aim in interconnected power system operation is to continuously maintain the balance between generated and demanded power. However, due to the dynamics of the turbines, generation of electricity by mechanical and chemical means require certain time. On the other hand, "already generated" electrical energy for large power systems is practically impossible. So instantaneous tracking of demand by the generation is impossible to attain, but can only be compensated by the kinetic energy of rotating masses, namely the generator rotors. A continuous deficiency of generation with respect to demand means continuous decrease in the kinetic energy of rotating masses, which in turn results in decreasing system frequency. Complementary to this fact is of course the increase in frequency when generation exceeds the demand.

In case of a generation demand mismatch, which is likely to occur anytime, the first step is to maintain the balance. Since the recovery of the generation-demand balance is associated with some definite time constants in the order of "seconds", regarding the instant responses of generation units; a deviation in system frequency is inevitable. But once equilibrium is reached, rate of frequency change is zero. In other words, frequency is at a new steady state. (Please refer to Figure 1.1-1).

The second step is to bring the system frequency back to its initial, commonly agreed steady state operating point (nominal frequency which is 50 Hz in Turkey), or in other words, back to its initial "state". (Please refer to frequency trace of Figure 1.1-2) This can only be achieved by disturbing the previously established balance for a definite time on purpose, in the opposite direction of the primary cause, until reestablishing the balance when the system frequency is back at its initial steady state level. Those two control philosophies are named as *Primary and Secondary Controls* respectively. When Secondary Control is realized by a central system, it is called *Load Frequency Control (LFC).*

If there exists a generation-load unbalance against generation, the initial dip and quasi-steady state deviation of the frequency [1] are directly related to the magnitude of the power mismatch and primary response characteristics of the generating units. There is absolutely no global control but only distributed local control (each power plant) associated with these two frequency characteristics. (**$f_{dyn\ max}$** and **f** as indicated in Figure 1.1-1) Since magnitude of power mismatch, being an uncontrollable disturbance, is an outside effect, amount of ready-to-provide reserve power and its provision characteristics are vitally important to minimize the deviations in frequency and maintain reliable operation of the grid.

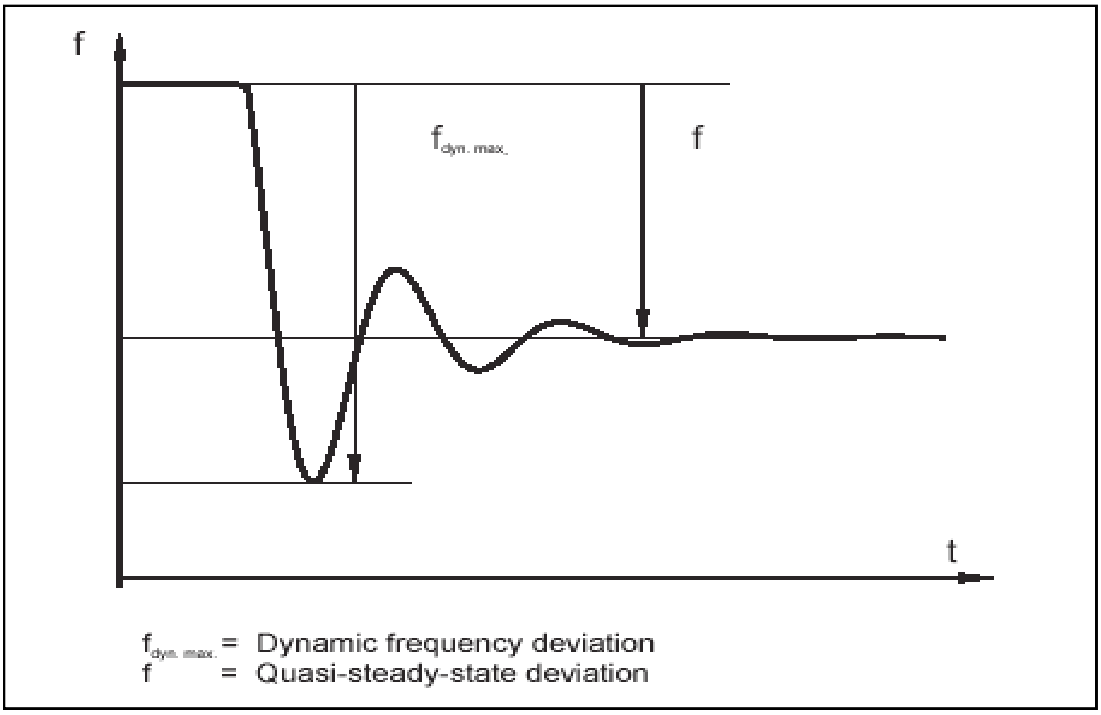

Figure 1.1-1: A theoretical view of frequency dynamics during a generation load unbalance *(UCTE Operation Handbook)*.

Above figure can be thought to be a velocity profile as a result of non-zero equivalent force, just like Newtonian mechanics. If there is a power/<u>force</u> mismatch, there is a "change in frequency"/<u>acceleration or speed change</u>. Once power/force equilibrium is maintained there is no "change in frequency"/ change in speed, however a new state different than the initial will be reached.

Generating units of the power plants are the only manipulating elements to control the frequency of the grid. Their primary action is the natural response "of them", and their secondary action is the global, forced control response "on them" via an outside controller (AGC software). That's why; interaction between primary control loops of generating units, namely the governors, and load control loops of power plants that they belong to, is really important and require careful study. Once also the different characteristics of different types of generating units (i.e. gas turbines, steam turbines and hydro units) are included subject might become more complicated.

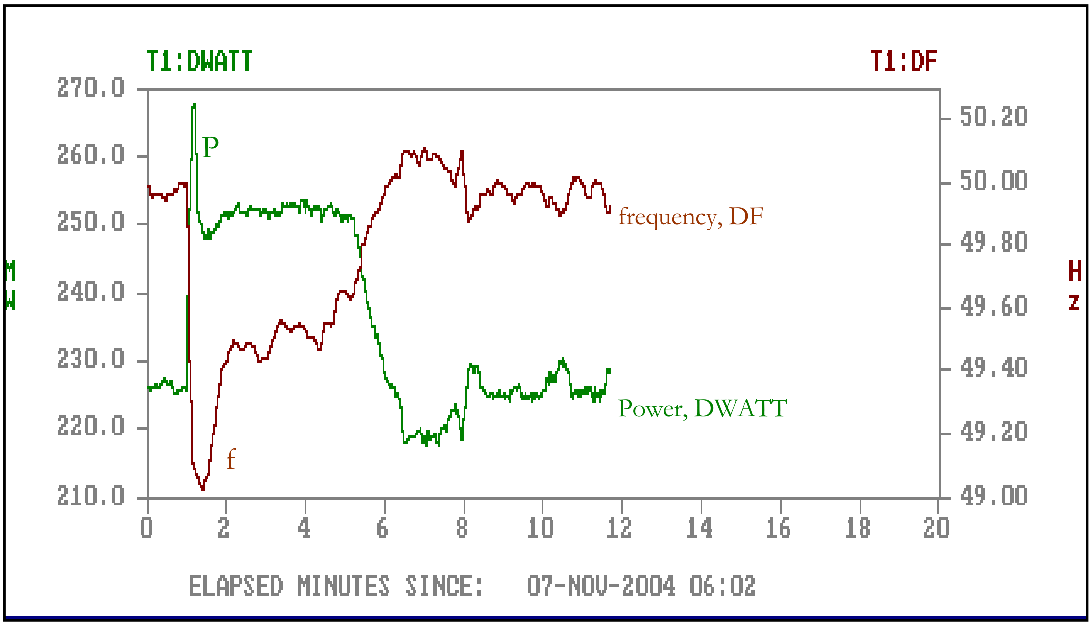

Figure 1.1-2: A real time plot of a generating unit's governor response (gas turbine) to a severe grid frequency disturbance. (DWATT being generating unit output, DF being grid frequency)

Those above mentioned characteristics, if not modeled correctly, result in reduced overall system reserve level and as a consequence, totally unexpected frequency response of the grid takes place compared to the ideal simulation cases.

1.2 Problem Statement

Generating units in a power plant, which are nominated to participate in system frequency control must be operated under "governor control" with "certain, predetermined amount of reserve power". Otherwise, units will either not respond to frequency deviations or; those respond, will have to swing (load/unload) in large magnitudes to an extent, frequency occurrences of which might worn out the machines. So "appropriate control loop implementation" is a must for all power plants.

1.3 Previous Work

Previous studies of Rowen and Undrill from General Electric, establish the basis for the modeling of gas turbines, and combined cycle power plants together with their fundamental control system characteristics.

Rowen's studies on gas turbine modeling [5], [6] and combined cycle power plants' operating characteristics, [7] are starting points of many scientific studies [11] as well as my thesis.

Undrill's studies on modeling, [3] and power plant operational issues [2], [4] clearly underline the mostly overlooked fact of power plants' operating mode effects on power system frequency control. Reference [10] is the outcome of a detailed work offering a new approach in modeling to be used in system-wide simulation studies. Accuracy of system-wide simulation studies can be increased with this new modeling approach that include real world facts prior of which are the power plant operating modes.

1.4 Contribution to the Problem Solution

This thesis study can be thought to be a "*bridge*" between academic environment and technical people working either as power plant crew or transmission system operators. It will serve for the healthy operation of Turkey's grid, such that the frequency characteristics of the system will be within internationally accepted limits.

The multivariable, complex, and dynamic nature of a gas turbine was implemented and analyzed in a simulation program which also includes a combined cycle power plant load controller that will enable units respond to a load control reference signal without sacrificing their primary response capabilities.

This thesis is expected to help technical people responsible from the desirable operation of the grid to understand the main problems and technical solution methods to those problems.

C H A P T E R 2

PROPOSED SYSTEM

Throughout this chapter, details of the critical load control loops on a power plant will be given and a successful control system architecture, in relation to system frequency control will be discussed. Application of this architecture was realized on MATLAB Simulink, which is also detailed in Chapter 5.

Implementation of AGC *(Automatic Generation Control)* in Turkish Power System, in the form of “set point assignments”, raised the question of “how the generating power plants would use these set point values to provide their responses”. In addition, analysis of the system frequency (Figure E-1 of Appendix E) indicates the poor control of the generation-load balance even when the AGC is in use. As a result of the poor control, frequency of the interconnected system exhibit inadvertent swings which are outside the acceptable limits defined by the operational quality standards.[1]

One of the reasons for this might be the misuse of the setpoint values sent to each power plant by the AGC software. If the update interval of the setpoint value is comparably long to be used as an instantaneous load reference, then, it should be used as a steady-state reference signal appropriately. If the power plant response, as a whole, is not controlled with respect to a desired frequency-power output characteristic, with active plant load controllers, primary behaviour of the units will be overridden by the error developed due to this time lag. There must be a common understanding between plants and interconnected system operators on how to interpret this set point and introduce it to the plant control loops. Poor understanding and lack of appropriate control systems design talent might result in plants exhibiting undesired responses.

In addition to plant load control loops, unit control loops must be understood properly. Although plant load control loops are generally designed by power plant engineering people, unit control loops are trademark designs of manufacturers and their perfect analysis is a must for the correct design and successful operation of plant load control loops. Please refer to Chapter 3 for unit control loop details.

A case study, analysis and successful control loop for a combined cycle power plant will be given in the rest of the chapter.

2.1 Power Plant Load Control Loops

Primary elements of combined cycle power plants are gas turbines, that burn natural gas with compressed air, resulting in high temperature, high pressure exhaust gases expanding through the turbine and heat recovery steam generator. Energy that results from gases expanding through the gas turbine is converted to the electricity via the

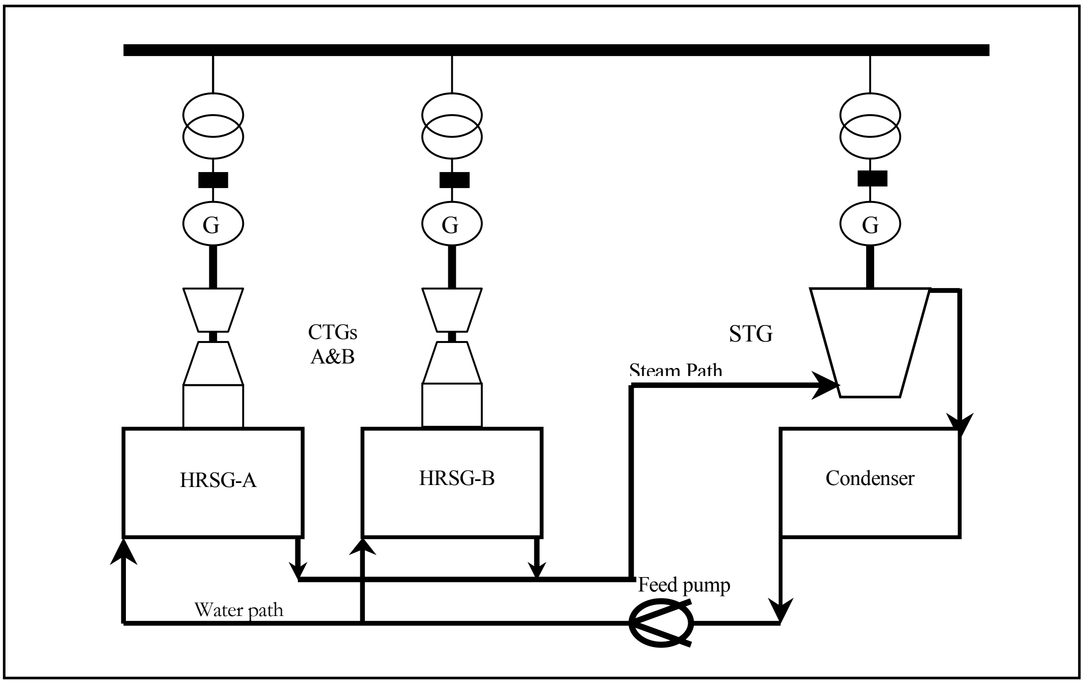

Figure 2.1-1: A symbolic layout of a “2 on 1”combined cycle power block

generator coupled to the turbine. Having left the turbine, exhaust gases are passed through a heat recovery steam generator (HRSG), which is basically a heat exchanger, so that pressurized steam is produced and sent to the steam turbine for power generation. As a result, efficient power production is achieved by combining two different cycles, hence the name *combined cycle power plant.*

Combined cycle power plants are constructed generally as “2 on 1 configuration” that is, steam extracted in the HRSGs of two gas turbines are fed in to one steam turbine, all of which are on separate shafts. Generally, such a configuration is called as a *block* or a *power block.* A combined cycle power plant can be built up using one or more of these power blocks of nearly identical characteristics.

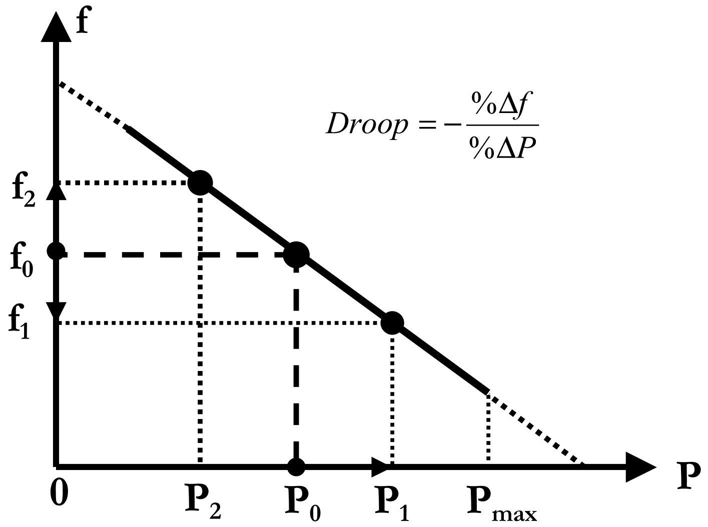

Figure 2.1.1-1: Speed-Droop Curve

Gas turbine’s response to frequency deviations via its internal control loops and its interaction with plant control loops are to be analyzed in detail. Due to the comparably long time constants associated with the heat recovery steam generator, steam turbine’s response, operating with fully open control valves, will be noticeable after a delay of a few minutes following the gas turbine’s response.

2.1.1 Gas Turbine Control Loop

Principal controller that is responsible from the gas turbine’s response to frequency deviations *(primary control loop)* is the governor. Governor’s principle action is to increase/decrease the instantaneous output regarding the instantaneous frequency deviation, according to the set “droop” value. The action of governor can be summarized with the very well known droop curve and equation.

Once the droop value is set to the required value (generally 4-8%), governor controls the corresponding output change for the instantaneous frequency deviation to be in accordance with the droop relation or in other words in accordance with the equation of the above speed-droop line.(Figure 2.1.1-1)

Fundamental control system block diagram of a gas turbine and related control signals are shown on Figure 2.1.1-2. When steady state conditions are taken into account, say at nominal grid frequency, steady state load on the machine is determined via the load reference.

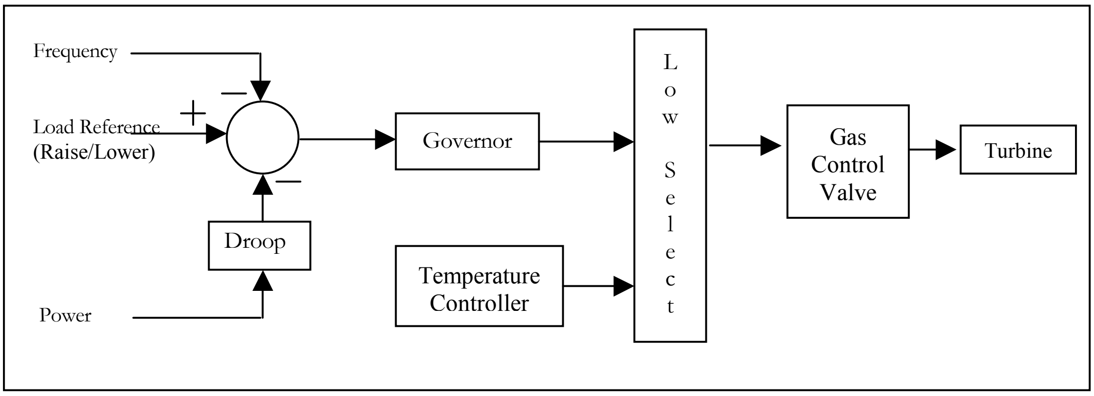

Figure 2.1.1-2: Fundamental gas turbine control system block diagram

During transient conditions of system frequency, even if the load reference is kept unchanged, error introduced to the governor due to the frequency deviation will result in a change in the output.

Just to underline, in order for a required increase in load to take place, machine must not be at its maximum operating limit determined by the temperature controller.

If gas turbine and its control system are perceived as a black-box system, load reference is the only interference point with an outside system, say, an interacting controller. This interaction takes place in the form of Raise/Lower pulses.

A detailed modeling for the gas turbine and its control system is carried out in the following chapter. Moreover, details and variation of the control variables can be observed in the MATLAB Simulink model that accompanies this thesis study.

2.1.2 Combined Cycle Power Plant Load Control Loop

Plant load control loop is responsible for making the plant output behave in accordance with a reference. Since combined cycle power plants are comprised of power blocks as mentioned before, load set point is generally assigned per block. Due to the fact that steam turbine part is just a follower (whatever amount of steam produced, is directly sent to the steam turbine), main manipulating elements are the gas turbines.

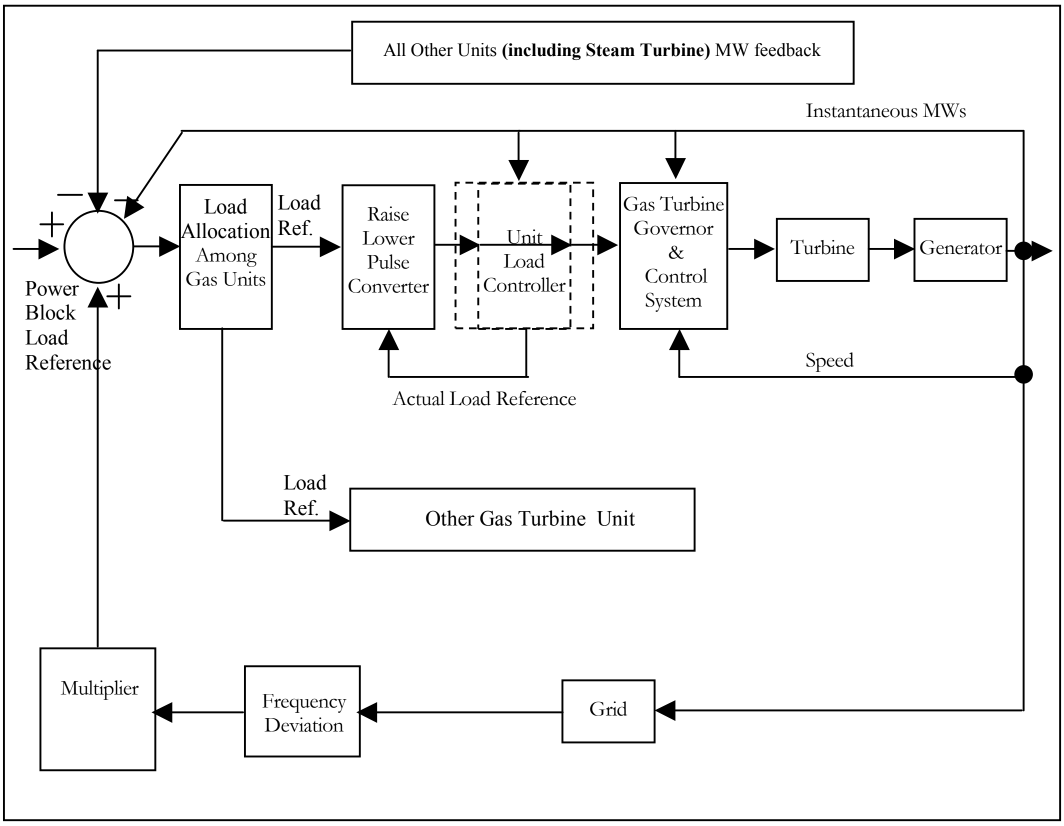

Figure 2.1.2-1: Combined Cycle Power Plant Fundamental Load Control Loop

Critical point is to allocate the block set point to operating gas turbines accordingly, "creating necessary reserve power", and not to hinder their primary regulating response for the sake of following steady state power block load reference. This can be achieved by updating the load set point depending on the frequency shift, multiplying by a constant. (See figure 2.1.2-1)

If load set point for the power block is not updated through the mentioned multiplier, then closure of total plant/block MW feedback loop will behave in such a way that instantaneous output changes in response to instantaneous frequency deviations will be overridden by the plant load controller for the sake of following the steady state load reference signal.

On Figure 2.1.2-1, symbolic representation of a combined cycle power plant load control loop is given. See that interfacing to a gas turbine unit by an outside controller is realized through the application of raise/lower pulses to the load reference input.

Another important point is that “Unit Load Controllers” (detailed discussion on unit load controllers are given on Chapters 5 and 6) must never be active, otherwise primary response of the unit is overridden by the unit load controller.

The above depicted control loop was realized in the Simulink model that accompanies this thesis study.

Reserve power management, although very important and very difficult for gas turbine based power plants will not be detailed in this study. But modeling approach of this thesis work offers possible means for controlling reserve power, to be implemented in “Load Allocation Among Gas Units” block.

CHAPTER 3

GAS TURBINE AND CONTROL SYSTEM MODEL

Throughout this chapter, fundamental concerns related to gas turbine units and their main operating characteristics will be given. Previous studies on gas turbine and control system modeling will be put forward. Main modeling approach of this thesis study and methods used will be explained.

3.1 Gas Turbine Basics and Key Differences

Gas turbines and their control systems are significantly different from steam turbine based power units. One major fact is that their thermal cycle is an "open cycle" using atmospheric air.

While steam turbines have a closed thermal cycle (steam having been produced in the boiler, does its work through the turbine and then repressurized back to the boiler which is a closed volume), in the so called Brayton Cycle of gas turbines, air compressed through the compressor goes through the combustor and gets burnt with fuel resulting in high temperature, high pressure exhaust gas. This exhaust gas does its work while expanding through the turbine (and through the heat recovery steam generator in combined cycle applications) back to the atmosphere.

This above explained major difference makes maximum power output of the gas turbine highly dependent on environmental conditions especially to the ambient temperature and pressure, and determination of this maximum power level is a great challenge. Another important parameter that affects gas turbine's maximum continuous output is the network frequency due to the fact that axial flow compressor is directly coupled to the shaft of the turbine and airflow changing with the speed results in a change in the output.[11]

So a gas turbine is a complex, multivariable and dynamic system whose modeling requires intimate study and analysis.

3.2 Modeling the gas turbine

Modeling of gas turbines has been a great concern of the academic environment. Although these engines were originally designed for aircraft propulsion they are also widely used in industrial applications for decades especially in power generation area.

Together with their growing usage and increasing number in power systems, analysis of their dynamic effects, which are very different in nature compared to steam and hydro units, became one of the most important research subjects of power system engineers.

Main concerns in power system area that require generation system modeling are voltage control, power system stability and generation-load unbalance phenomena, that is, frequency control. So main concern of my modeling approach will be to reach a gas turbine simulation model that can represent its response to frequency deviations accurately. This will require a turbine and a governor model but not a generator and voltage regulator model.

Gas turbine engines used in power generation applications are divided into two categories namely being *aero-derivative* and *heavy duty.* Aeroderivative gas turbines being derived by modifying aircraft engines are used for small power generation applications in 10 to 50 MW ranges, whereas heavy-duty gas turbines having different mechanical construction characteristics are mainly designed for power generation levels up to 300 MWs. On this thesis study, modeling details of a heavy-duty gas turbine will be given.

3.2.1 Gas turbine modeling approaches and justification of the approach used.

There are different approaches in gas turbine modeling. First approach is thermodynamic modeling by using fundamental heat and mass balance equations for the compressor, combustor and turbine which are the three main components of a gas turbine engine. Such kind of modeling require going into the deep insides of the machine internal physics (enthalpy equations, efficiency calculations…), especially to reach accurate steady state and dynamic response characteristics.[9]

The other end of approaches in modeling is the use of generic gas turbine models subject to some parameter and predetermined function modifications like maximum power level and frequency dependency of the output. Dynamic effects are included with simple time constants. Through this approach, a general representation of a gas turbine unit can be reached. This approach is mostly preferable for system-wide analysis that will require the models of hundreds of units.[10]

Modeling for power system studies does not require a detailed thermodynamic model since principal concern is to analyze unit's dynamic behaviour with respect to frequency deviations around the main operating point, the nominal grid frequency. So machines' internal variables or start-up details, in other words, internal behaviour of the engine, although might be important for the engine performance studies, are not required for most power system studies. This does not mean that generic models are perfectly suitable for all power system analysis purposes. Once reserve power for gas turbines is under concern, modeling requirements cannot be that much away from physical principles. Also inclusion of various dynamic effects is a must. That's why Rowen's model [5] and its derivatives [6], [11] which stand in between generic models [10] and highly detailed thermodynamic models [9] are good starting points for analysis on gas turbines.

As the modeling approach of this thesis work, static components which reflect the steady-state input-output relationships of fundamental physical variables were obtained by some detailed input-output data analysis as a result of the experiments carried out [3], [11]. For the dynamic characteristics, various papers [5], [6], were analyzed modified and tuned accordingly to enlarge on the load control loops that are important for interconnected grid operation.

3.2.2 General physical structure of a gas turbine unit

A gas turbine is comprised of three major components, which are the axial compressor, combustor(s) and the turbine. Air compressed through the compressor goes through the combustor and gets burnt with input fuel resulting in high temperature, high pressure exhaust gas. This exhaust gas does its work while expanding through the turbine (and through the heat recovery steam generator in combined cycle applications) back to the atmosphere.

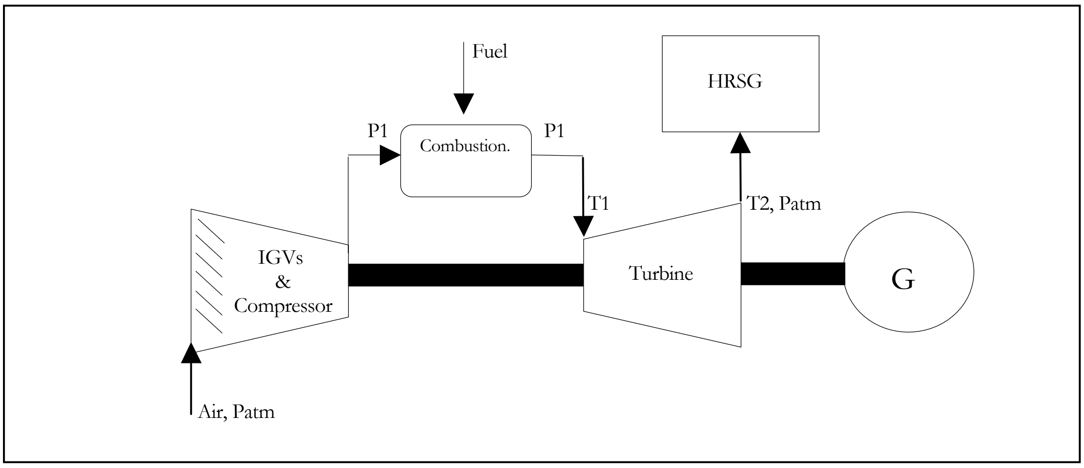

Figure 3.2.2-1: Heavy duty gas turbine

Primary input variables are "fuel flow" and "airflow". Fuel flow is a completely controllable parameter whereas airflow which is a function of ambient temperature together with shaft speed (or power system grid frequency) can be regulated with the help of compressor inlet guide vanes (IGVs) up to a certain degree. Airflow and its interaction with shaft speed is one of the key points to be solidly understood in gas turbine operation.

Primary output variables are electrical power from the generator, exhaust temperature of the turbine (T2) and pressure ratio across the compressor (P1/Patm), all of which can be measured with no difficulty.

Fuel and air flows are adjusted such that required output power is attained, while keeping the exhaust temperature (T2) as high as possible taking into account the maximum permissible turbine inlet temperature (T1).

3.2.3 Gas Turbine Control System

Gas turbine control system is comprised of two important closed control loops namely, "speed-load controls" and "exhaust temperature controls". These two control loops are always in action during operation connected to the grid.

Figure 3.2.3-1 is a block diagram representation of a gas turbine together with its critical control loops, which is a basis for the simulation model. [5] Overall, as dynamic elements, there exists simple time constant blocks and transport lags. There are four fundamental equations that put forward the main thermodynamic characteristics, which are to be obtained based on input-output data analysis.

3.2.3.1 Governor

On this model, a “droop governor” will be utilized. In large utility systems where the speed of the turbine is dictated by the grid frequency, droop governor maintains a proportional action on the error between setpoint, and system frequency. For a detailed discussion please refer to Chapter 2.

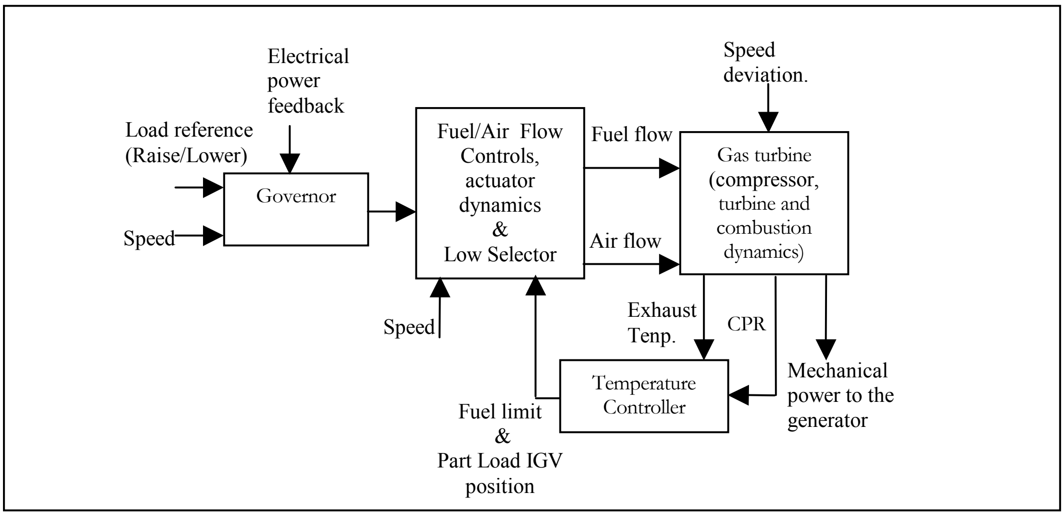

Figure 3.2.3-1: Gas turbine control system

3.2.3.2 Temperature Controller

Temperature controller which is a peculiarity of a gas turbine is the most vital to understand part of the control system. The main task of the temperature controller is to limit the fuel flow and/or regulate the airflow (and in turn determine maximum output together with available reserve power) such that maximum allowable turbine inlet temperature (T1) is not exceeded in any case. Measurement of turbine inlet temperature although possible is not practical due to the temperature levels involved (approximately 1400 deg C). As a solution, exhaust temperature (T2) is measured and controlled accordingly at a calculated setpoint, such that firing temperature is kept at an optimum level.

3.2.3.2.1 Air flow control (Inlet guide vane control)

In combined cycle applications of gas turbines, exhaust temperature is to be kept at maximum allowable levels even at part load conditions so that optimum performance on the overall cycle is attained. This can be achieved by closing IGVs under a controlled manner until exhaust temperature is at maximum allowable level even at part load.

While IGV control is active, if there is a demand of fuel flow increase, there must be also a proportional increase in the IGV opening so that temperature limit is not violated. Temperature Control –IGV- was implemented as a PI controller in the simulation. (Please refer to Figure D-2 of Appendix D)

3.2.3.2.2 Fuel flow control

Once IGVs are fully open, the machine is on its temperature limit. Further increase on fuel flow (or further decrease on air flow) must be limited or reversed so that machine's internal maximum allowable limit is not offended. This task is achieved by fuel flow controller of the temperature control system. Temperature Control -Fuel- was implemented as a PI controller in the simulation. (Please refer to Figure D-2 of Appendix D)

3.2.3.3 Fuel Control and Combustion System

Fuel control and combustion system consist of fuel control valves, combustors and volume of the expansion path, all of which introduce a real dynamic characteristic to a gas turbine unit. [5]

The dominant time constants for fuel control belong to the control valves and storage volume of the gas piping [5]. There exists a definite time for the combustion to occur which can be expressed as a transport lag. Once combustion reaction takes place, it takes some time to transport the gas from the combustors through the turbine which is again to be represented as a transport lag. The continuous displacement of airflow through the compressor is to be modeled as a simple time constant.[5]

3.2.3.4 Fundamental Equations

Whatever efforts are taken to simplify the mathematical model of the gas turbine, its inborn thermodynamic characteristics cannot be overlooked in order to reach a satisfactory model that will mimic the actual behaviour.

Physical principles of the machine are introduced via four fundamental multivariable "algebraic" equations, which will comprise the "steady-state characteristics" of a gas turbine unit.

Throughout the modeling process, key point is to reach the structures of the functions and relationship between independent and dependent variables.

"Slowly varying" input variables and corresponding output data analysis can give information about steady state characteristics of the machine "free from control system dynamics". As a result of purposely slow variation of input variables and based on above reasoning, the relation between inputs and outputs can be assumed to be ALGEBRAIC (not dynamic, but static). So, in order to reach the physical equations that will give information about machine thermodynamics, "curve fitting" approach can be used on the observed data.

As it can also be seen from gas turbine unit start up and loading data curves in Appendix A (please see the detailed explanation of each graph), the variables in concern for each equation and their physical interpretations are as follows.

Output Power (P_{out}): Output power is a linear function of fuel flow.

$$P_{out} = F(W_f)$$

Airflow (W_a): Airflow, the most critical equation to be obtained in order to reach other dependent variables, is a function of speed, Inlet Guide Vane (IGV) position and ambient temperature. Apparent nonlinearity in the speed dependency of airflow can be observed on Figure A-2, in Appendix A.

$$W_a = F(N, \theta_{IGV}, T_a)$$

In addition, temperature dependency of airflow must not be ignored. This calls for similar output data analysis at different temperatures and then temperature dependency of airflow can be included accordingly.

Compressor Pressure Ratio (CPR): Compressor Pressure Ratio, which is the limiting factor of maximum output available for the unit and which is used in the exhaust temperature reference calculation is a function of both airflow and fuel flow. Refer to Figure A-4 in Appendix A.

$$CPR = F(W_a, W_f)$$

Exhaust Temperature (T_x): Exhaust temperature which is the main controlled variable for the temperature controller is a function of fuel flow and airflow. Refer to Figure A-5 of Appendix A.

$$T_x = F(W_f, W_a)$$

As it can apparently be seen, airflow equation is the most critical equation to be reached in order to model the physical characteristics of a gas turbine unit.

To underline once more, the critical point in input-output data analysis and curve fitting, is to get a function structure that will reflect the physical nature and then reach the relationship between variables. Following chapter will put forward the data analysis approach used and how to reach a satisfactory function structure together with the relationship between variables.

CHAPTER 4

EXPERIMENTAL STUDY AND DATA ANALYSIS

Throughout this chapter, data analysis that was carried out to reach four fundamental equations of the gas turbine simulation model, which were mentioned in the previous chapter, will be explained. The loading and start-up experiments together with step response test will be detailed.

"Least Square Regression Analysis" will be carried out on the observed unit start up and loading data to reach curve fits of what is observed, so that obtained algebraic equations can be combined with the dynamic model to reflect both dynamic and steady-state nature of the machine.

Just to underline once more, "slowly varying" input variables and corresponding output data analysis can give information about steady state characteristics of the machine "free from control system dynamics". As a result of purposely slow variation of input variables and based on this reasoning, the relation between inputs and outputs can be assumed to be ALGEBRAIC (not dynamic, but static). So, in order to reach the physical equations that will give information about machine thermodynamics, "curve fitting" approach can be used on the observed data.

The fundamental relationships to be detailed here are;

- Relationship between output power and fuel flow
- Relationship between airflow, speed, IGV angle and ambient temperature
- Relationship between "compressor pressure ratio", fuel flow and air flow
- Relationship between exhaust temperature, fuel flow and air flow.

4.1 Relationship Between Output Power and Fuel Flow

If we leave aside all the complex and multivariable nature of a gas turbine, it is basically a device that converts chemical energy of the natural gas to mechanical energy. Mechanical energy is then converted to electrical energy at the generator.

So, just by intuition, it is easy to say that output power of a gas turbine unit is directly proportional to input fuel flow but nothing else. The only thing under concern is whether this relation is linear or not. Closer look to Figure A-1 of Appendix A indicates an apparent linearity between fuel flow and output power. So the function structure for this relationship can be proposed to be,

$$P_{out} = aW_f + b \qquad \text{(Eqn 4.1-1)}$$

where *a* and *b* are the parameters to be determined.

Once data of the linear region of Figure A-1 is transferred to the Matlab for Least Square Analysis with the proposed function structure of equation 4.1-1, the linear fit seen on Figure C-1 of Appendix C is obtained.

As it can be seen on Figure C-2, assumption of a linear function like a first order polynomial results a successful fit of error less than 1% for the analyzed range of power-fuel flow relationship.

4.2 Relationship between Airflow, Speed, IGV Angle and Temperature

Airflow equation is the most critical function to be reached in order to be able to calculate the remaining variables of the machine and to establish the proposed gas turbine model.

Airflow through the machine is created by the compressor coupled to the turbine. (Refer to Figure 3.2.2-1 in chapter 3.). Simple mechanics tell us that with constant pressure difference, flow through a restriction is proportional with the area of the restriction. So IGV angle or in other words, the amount that the inlet guide vanes are open, has a definite effect on the airflow through the machine. Also, if we are talking about an axial flow compressor like the case we have on a gas turbine unit, basic mechanics again say that the amount of airflow is directly proportional with the speed of rotation. Once we also take into account that we are speaking in terms of mass flow and air is a gas, then temperature is also important for us due to the fact that density of air changes with temperature resulting a change in total mass flow, although speed and IGV angle remains constant.

Please refer to Figure A-2 and A-3 in Appendix A. Figure A-2 was obtained during a unit start-up, i.e. while the unit is brought to 3000 rpm from standstill. On this graph dependency of airflow both to the speed and IGV angle can be clearly seen. See that while IGV angle is constant at around 27 degrees, airflow increases with the increasing speed. When IGV angle changes from 27 to 55 degrees, a very apparent change in airflow can be observed. Once IGVs were settled at 55 degrees, further increase in airflow is due to speed only.

Figure A-3 was obtained after synchronization to the grid and while the unit is being loaded up. The change in speed is only due to random oscillations of the power network. Also on this graph, apparent dependency of airflow to the IGV angle can be seen while IGVs are opened up from 55 degrees to 86 degrees.

Being a multivariable function with independent uncontrollable variable speed or namely the grid frequency and controllable variable IGV angle, it is not an easy task as the previous power-fuel flow relation to reach a successful fit of the airflow. Also temperature dependency requires long-term observation while the other parameters are constant.

4.2.1 A proposal for the function structure of airflow equation

All the above explanation and reasoning prove that airflow is a function of speed, IGV angle and temperature.

$$W_a = F(N, \theta_{IGV}, T_a) \quad \text{...(Eqn 4.2.1-1)}$$

The preceding function of several variables can have any structure like division, multiplication, addition or even square root of the variables under concern. In order to carry out a curve fit, first of all a function structure is necessary. Then necessary parameters of the determined function structure can be searched through the minimization of the error.

For example in the case of power-fuel flow fit, being a function of only one variable and apparent observable linearity could easily let us use a first order polynomial as the function structure for the fitting process. But in this case, we have a function of many variables and determining the right function structure that will contain all the variables while also reflecting the physical reality is not an easy task.

All the data trends that were collected from the machine have some period of time where only one variable under concern changes while the other is constant. (Please refer to Figure A-2 and A-3 in Appendix A.) So with this nature of the variables, initial function structure is considered to be;

$$W_a = K.F_1(N).F_2(\theta_{IGV}).F_3(T_a) \quad \text{.........................(Eqn 4.2.1-2)}$$

where K is the scaling constant, F_1 is the speed dependency function, F_2 is the IGV opening dependency and F_3 is the temperature dependency function for the overall airflow equation. The next step was then to determine the structure of the functions of only one variable while keeping the other variable constant and reaching the necessary equation of airflow.

4.2.2 A closer look to Figures A-2 of Appendix A

It is obvious that we do not have the control of speed once the machine is connected to the network. But if the machine is separated from the grid, speed of the machine can be controlled just as it is done during start-up. So the last region of Figure A-2, where IGV angle is constant at 55 degrees and speed is increasing to 3000 rpm, can be a beneficial region to find out the speed dependency of airflow. This is useful to understand the dependency of airflow to severe grid frequency changes during daily operation.

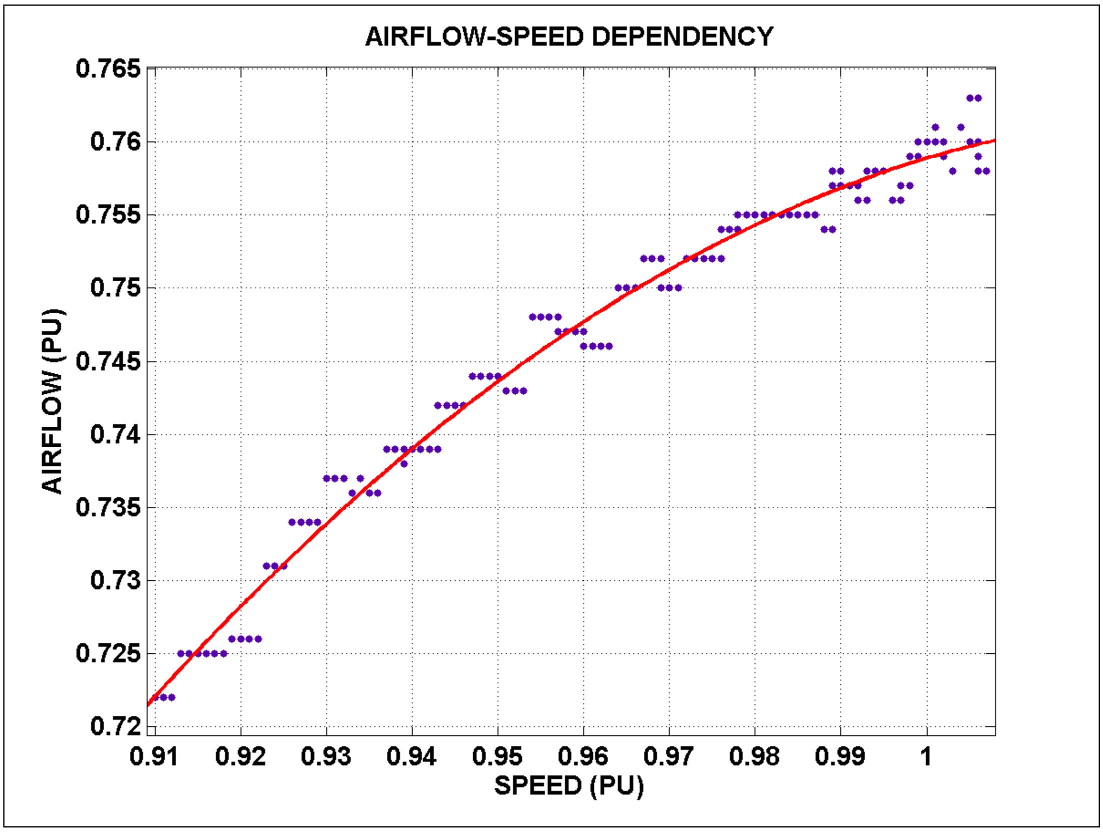

Figure 4.2.2-1 A different look at Figure A-2 for 90-100 % speed range. Airflow vs. speed and second order polynomial fit.

Figure 4.2.2-1 is obtained by using the data of Figure A-2 for the speed values over 90%. The apparent non-linear relationship between airflow and speed can be readily seen. The first proposal to this non-linear relation is a second order polynomial as,

$$Wa_{(speed)} = F_1(N) = cN^2 + dN + e \quad \text{.........................(Eqn 4.2.2-1)}$$

Second order polynomial (red line) drawn over the observed data proves to be a successful fit for the speed function of the airflow equation.

4.2.3 A closer look at Figures A-3 of Appendix A

When the unit is connected to grid and if system frequency does not have significant oscillations it can be accepted that the speed function of airflow is constant. Then the change of IGV angle introduced at Figure A-3 will be useful to determine the IGV opening dependency of airflow.

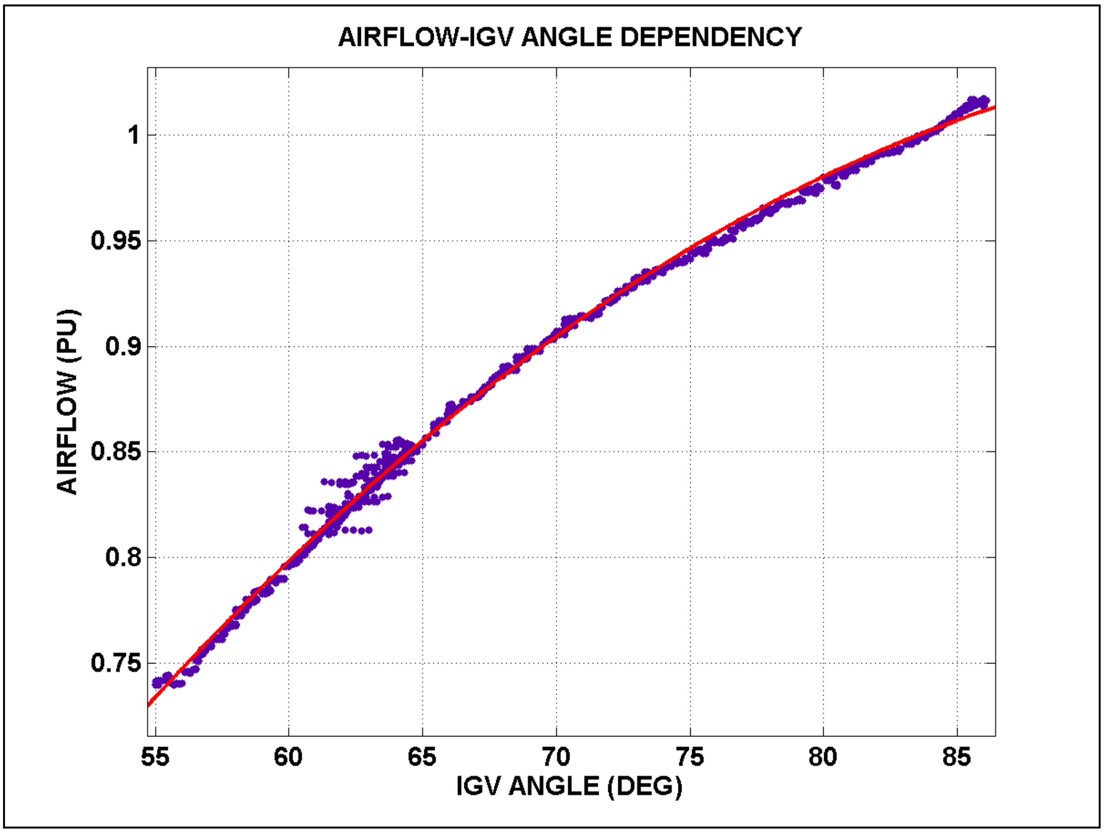

Figure 4.2.2-2 Data of Figure A-3 of Appendix A drawn for IGV angle range of 55 to 86 degrees. Airflow vs. IGV angle and second order polynomial fit.

Figure 4.2.2-2 is obtained by using the data of Figure A-2 for IGV angle range between 55 and 86 degrees. The non-linear relationship between airflow and IGV opening can be observed. The first proposal to this non-linear relation can be again a second order polynomial as,

$$Wa_{(IGV)} = F_2(\theta_{IGV}) = f\theta_{IGV}^{2} + g\theta_{IGV} + h \text{ ....(Eqn 4.2.3-1)}$$

Second order polynomial (red line) drawn over the observed data proves to be a successful fit for the IGV angle dependency of the airflow.

4.2.4 Temperature Dependency of Airflow and final Airflow equation

As it was previously mentioned, our main concern is the air mass flow through the machine and its effects on other variables like temperature. So air being a gas, temperature and also pressure have certain effects on its density, resulting a considerable effect on the air mass flow.

Unlike other variables, ambient temperature and pressure are totally uncontrollable variables and have a comparably very slow rate of change. Temperature and pressure dependency of airflow require very long term observation of unit behaviour while all the other parameters that affect airflow are constant which is a hard to attain experimental condition. Hence, previous approach is not applicable.

For this study, pressure dependency of airflow was ignored and it was tried to reach temperature dependency, through long-term observation of different units at nearly equal parameters that affect airflow as far as those situations could be caught. That's why there is not a parametric way of representing temperature dependency of airflow in this thesis. Though, intuitively, following equation can be used, knowing the fact that air mass flow is inversely proportional with temperature since density of air decreases with increasing temperature.

$$Wa_{(Ta)} = F_3(T_a) = T_{design}(15\deg C)/T_a \quad \text{.....(Eqn 4.2.4-1)}$$

So the final airflow equation becomes,

$$W_a = K.(cN^2 + dN + e).(f\theta_{IGV}^{\ 2} + g\theta_{IGV} + h).T_{design}(15\deg C)/T_a \quad \text{.....(Eqn 4.2.4-2)}$$

Figure C-3 in Appendix C contains both the airflow data of Figure A-3 of Appendix A and obtained fit on the same graph together with error analysis.

4.3 Relationship between "compressor pressure ratio", fuel flow and air flow

Again let's have a closer look at Figure 3.2.2-1 of Chapter 3. Basically what the compressor does is, maintaining continuous airflow by supplying compressed air to the combustor which is necessary for the combustion to take place. The more compressed air provided for the combustion, the more fuel can be input and in the end, more power can be produced. But just like every machine a gas turbine unit has certain limits and measurement of pressure ratio across the compressor is necessary to determine and not to offend those limits.

In the end, the philosophy of gas turbine control is to keep the turbine inlet temperature (T1) at its limit despite the fact that it cannot be measured. Measuring exhaust temperature, which is at acceptable levels, together with the compressor pressure ratio, one can calculate turbine inlet temperature and do not offend the limit.

Please refer to Figure A-4 of Appendix A. See that there is a period of time where fuel flow changes but air flow is constant. There is also a longer period of time where both air and fuel flows change. It is apparently seen that compressor pressure ratio increases at a certain rate with the increase of fuel flow but this rate goes higher in the region where both fuel and air flows increase together. So it is obvious to say that compressor pressure ratio is a function of both fuel and air flows.

While curve fitting, in order to benefit from the region where fuel flow changes only, with the same reasoning that was used for airflow equation, it can be assumed that,

$$CPR = F(W_f, W_a) = F_1(W_f).F_2(W_a) \quad \text{.....(Eqn 4.3-1)}$$

Next step becomes to reach $F_1(W_f)$, in the region where airflow is constant. As,

$$CPR = K.F_1(W_f) \quad \text{.....(Eqn 4.3-2)}$$

Then using the mathematical fact that,

$$\frac{CPR}{F_1(W_f)} = F_2(W_a) \quad \text{.....(Eqn 4.3-3)}$$

$F_2(W_a)$ is the curve fit to reach for the data values obtained as a result of the division. CPR is measured and $F_1(W_f)$ is obtained in the preceding step.

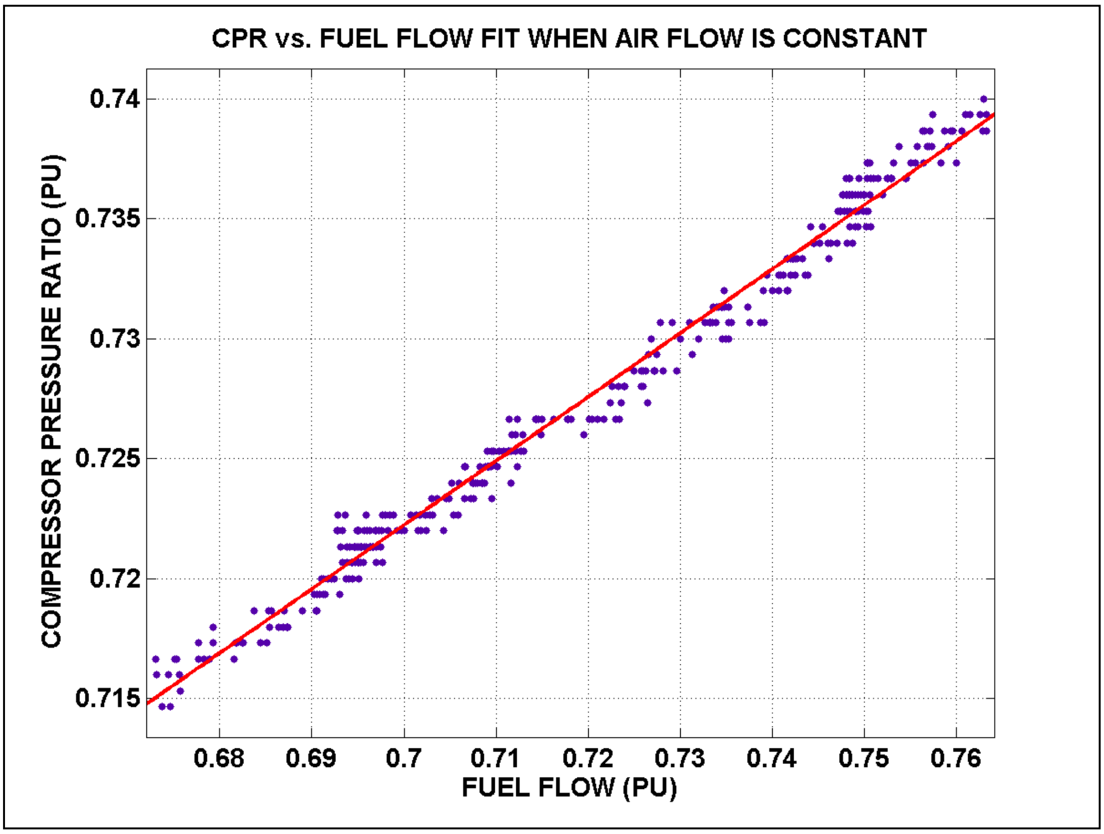

Figure 4.3.1 Curve fit to data of Figure A-4 of Appendix A for the period where airflow is constant. $F_1(W_f)$ is obtained as a result of the fit.

As a result CPR equation can be reached with the multiplication of F_1 and F_2. Please take a look at to linear fits obtained for $F_1(W_f)$ and $F_2(W_a)$ on figures 4.3.1 and 4.3.2.

Final CPR equation becomes as,

$$CPR = F_1(W_f).F_2(W_a) = (kW_f + l).(mW_a + n) \qquad \text{(Eqn 4.3-4)}$$

Please see Figure C-4 of Appendix C for the overall CPR fit and corresponding error analysis.

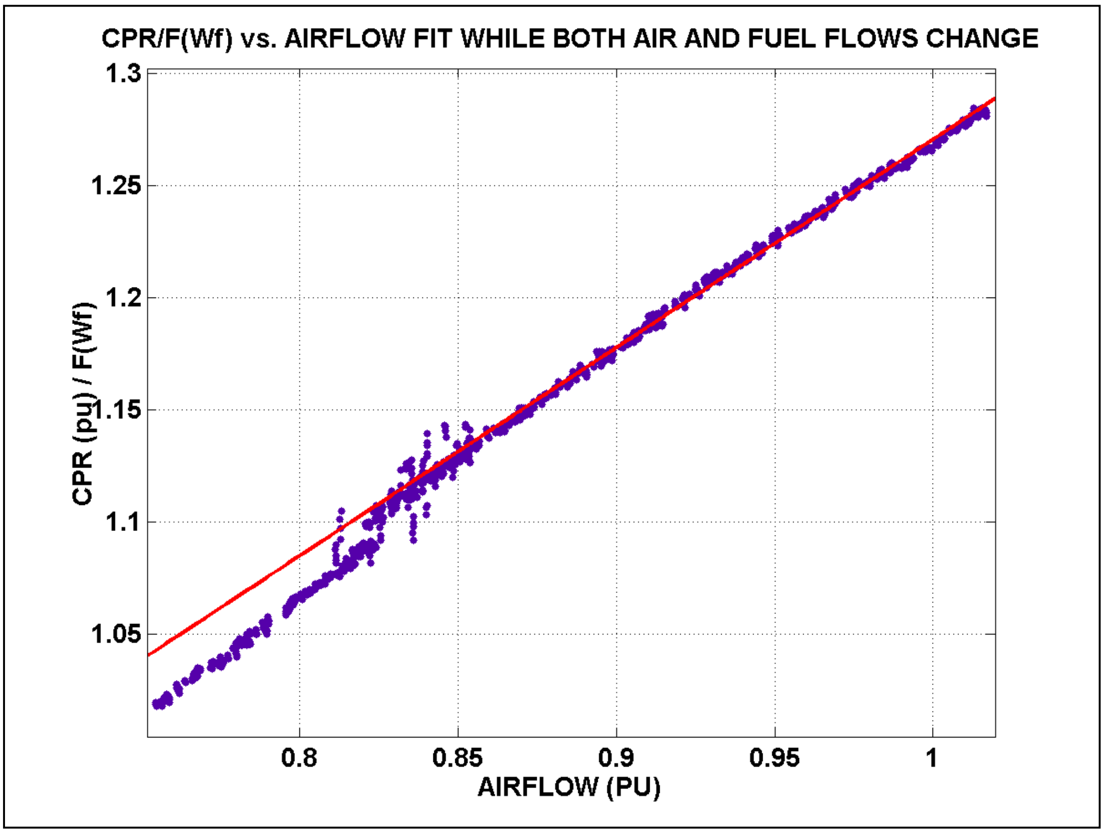

Figure 4.3.2: $F_2(Wa)$ is obtained as a result of the linear fit obtained for the CPR/$F_1(Wf)$ vs. Airflow data. See that data values below 0.82 of airflow were intentionally excluded from the fit. This is to increase accuracy at higher air and fuel flow operation region i.e. higher MW operating point where we operate the machine normally.

4.4 Relationship between exhaust temperature, fuel flow and air flow

Another glance at Figure 3.2.2-1 of Chapter 3 might be helpful to understand what is mentioned by the exhaust temperature (T2).

Measurement of exhaust temperature is the controlling means of a gas turbine unit since the main variable to be kept under a limit, that is, turbine inlet temperature (T1) cannot be measured.

It is easy to accept that exhaust temperature is dependent on two factors that are air flow and fuel flow, this apparent dependency can also be observed by looking at Figure A-5 of Appendix A. See that there is a period of time where air flow is constant but fuel flow is increasing accompanied with a corresponding increase on the exhaust temperature. After a certain temperature value, increase in airflow together with an increase in fuel flow results in a decrease at exhaust temperature.

With the same reasoning that was followed at the preceding curve fit trials e.g. CPR equation, in order to benefit from the region where airflow is constant but fuel flow is changing, the relationship can be established as;

$$T_x = F(W_f, W_a) = F_1(W_f).F_2(W_a) \quad \text{(Eqn 4.4-1)}$$

Next step becomes to reach $F_1(W_f)$, in the region where airflow is constant. As,

$$T_x = K.F_1(W_f) \quad \text{(Eqn 4.4-2)}$$

then using the fact, just like it was used for reaching the CPR equation,

$$\frac{T_x}{F_1(W_f)} = F_2(W_a) \quad \text{(Eqn 4.4-3)}$$

where $F_2(W_a)$ is the curve fit to reach for the data values obtained as a result of the division. T_x is measured and $F_1(W_f)$ is obtained in the preceding step.

As a result exhaust temperature equation can be reached with the multiplication of F_1 and F_2. Please take a look at to linear fits obtained for $F_1(W_f)$ and $F_2(W_a)$ on figures 4.4.1 and 4.4.2.

Final exhaust temperature becomes,

$$T_x = F_1(W_f).F_2(W_a) = (pW_f + q).(rW_a + s) \quad \text{(Eqn 4.4-4)}$$

Please see Figure C-5 of Appendix C for the overall exhaust temperature fit and corresponding error analysis.

Consequently, all the critical variables of a gas turbine unit are reached for the simulation.

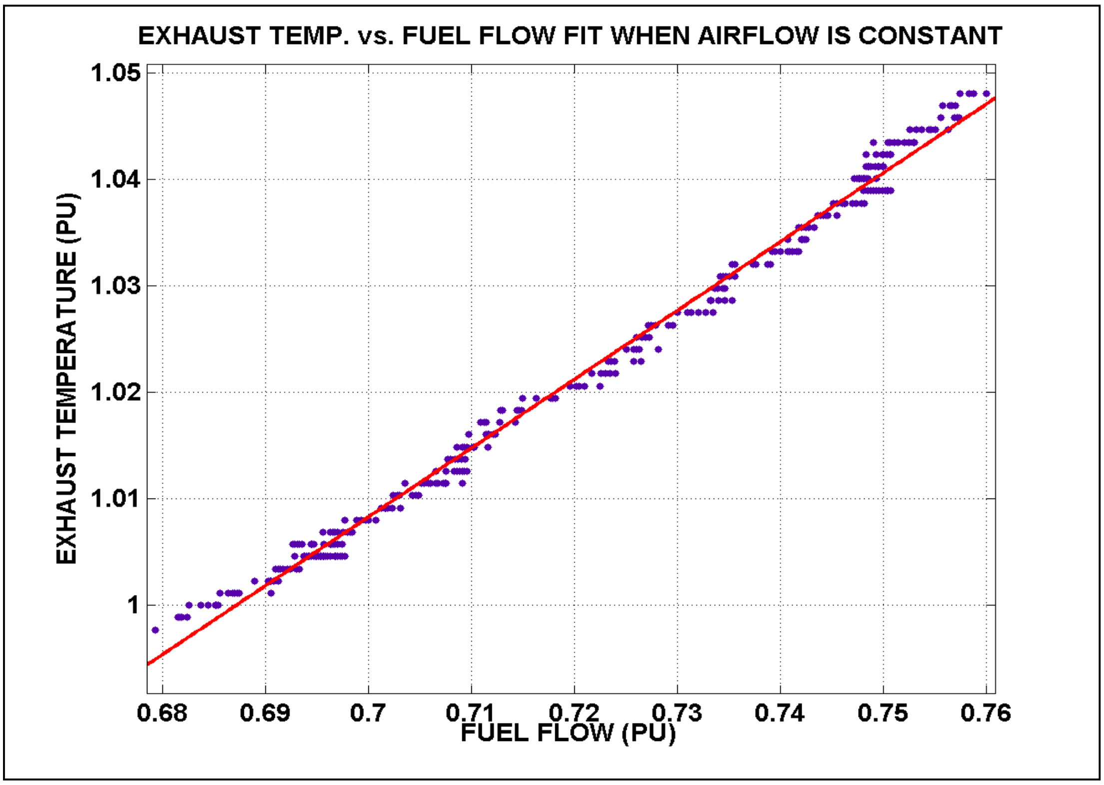

Figure 4.4.1: Curve fit of measured exhaust temperature to measured fuel flow when airflow is constant. $F_1(W_f)$ of equation 4.4.2 is obtained as a result of the fit.

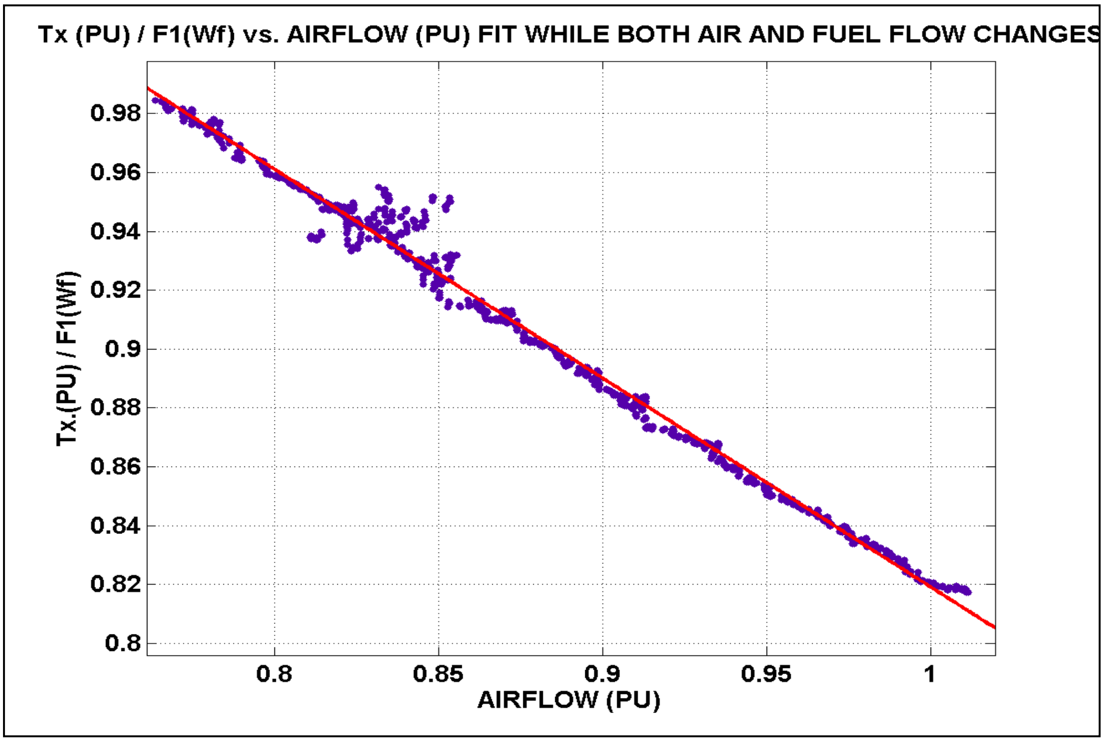

Figure 4.4.2: Curve fit of $Tx/F_1(Wf)$ obtained in the preceding step to measured airflow data. As a result $F_2(W_a)$ is obtained as a linear fit.

C H A P T E R 5

SIMULATION MODEL

Plant load control system for a combined cycle power plant, proposed in Chapter 2 was realized as a MATLAB Simulink model, details of which will be given in this section. Applied simulation model can be seen in Figure D-1 of Appendix D.

Proposed model consists of two dynamic gas turbine models, a simple dynamic steam turbine model and a plant load controller model.

The steady state characteristics and fundamental variables of the gas turbine model were obtained as a result of experimental studies and curve fits detailed in Chapter 4. For the dynamic model, as the starting point, the "structure" provided by Rowen of General Electric [5], [6] was used. As both of those models are for some other specific applications, in order to reflect the dynamic nature called for this study, that is, power system frequency control point of view, modifications were carried out in the model structure. A "load controller" for the gas turbine unit was also integrated to the model. Applied gas turbine simulation model in block diagram configuration can be seen in Figure D-2 of Appendix D.

Steam turbine was modeled as the cascade combination of steady-state characteristics obtained as a result of "gas turbine power" to "steam turbine power" curve fit and apparent dynamics. For simplicity, steam turbine dynamics were just introduced as a single "time delay" element. Justification of this approach is straightforward. By experience, it is common practice to operate combined cycle power plants close to their maximum operating point. So at this stage, whatever steam produced at the HRSGs *(Heat Recovery Steam Generator)* of the gas turbines, are let through the fully open control valves of the steam turbine. That's, basically there is no dynamic control at the steam turbine when it is operating under this “sliding pressure control mode”. In sliding pressure control mode, whatever steam delivered to the steam turbine is let through without any active control at the valves by neither the governor nor the pressure controller. This is actually the turbine-follow mode [3].

Plant load controller is the interfacing means to the gas turbine units. Load reference to be followed by the power block is introduced only to the gas turbine units, since steam turbine is just a follower. As it was also mentioned in Chapter 3 while detailing on the gas turbine control system, interfacing to the gas turbine can be via raise/lower pulses to the load reference. So conversion of analog set point to raise/lower pulses together with required control system dynamics are carried out on this block.

To underline once more, in case of an “analog set point” but not raise/lower pulses sent to the power block, total plant load control loop must be closed to follow up the reference. When total plant/block MW feedback loop is closed, instantaneous output changes in response to instantaneous frequency deviations will create an error for the plant load controller if its input is a steady state reference signal. Then this error is to be eliminated by the plant load controller for the sake of following reference signal, “which means hindering the primary response”.

Gain “K” (please refer to Figure D-1of Appendix D) that acts on frequency error and adds up to the load reference, updates the set point as long as there is a frequency deviation. As a result, instantaneous megawatt output change is not considered as an error by the plant load controller having a steady state load reference.

5.1 Gas Turbine Big Block

Gas turbine big block expects a load control set point when load controller of this specific unit is activated via “1” through the “Activate Load Controller” input.

When load controller of the specific unit is not activated and “plant load controller” is active, raise/lower pulses that increment/decrement the load reference are input through the “Raise/Lower Pulses from Plant Control” input.

The speed or "frequency" information for the governor and airflow calculation model is applied to the corresponding input.

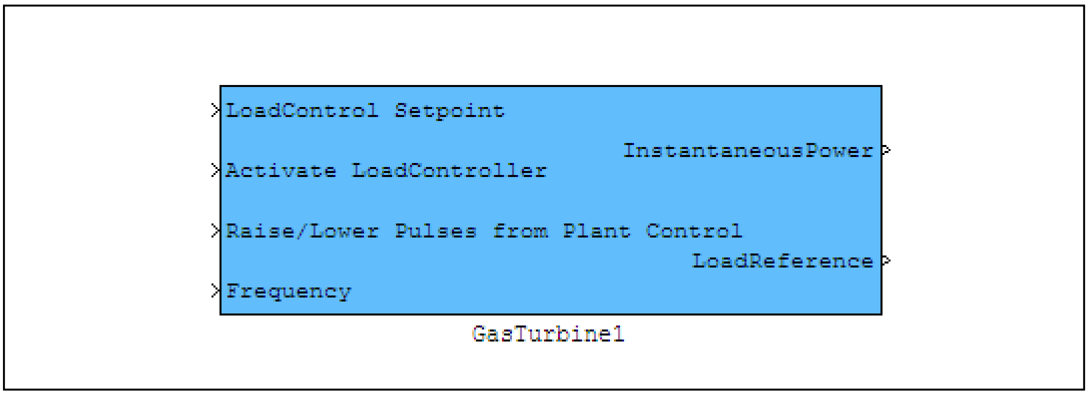

Figure 5.1: Gas turbine big block of the simulation model

"Instantaneous Power" is the output power in megawatts. "Load Reference" is an internal variable but used as a feedback signal to the "Plant Load Controller"

5.1.1 Unit Load Controller

Unit load controller is a PI controller that acts on the outer megawatt control loop of the gas turbine unit such that when activated, it always maintains the input MW set point. *(Please refer to UnitLoadController block of Figure D-2 of Appendix D)*

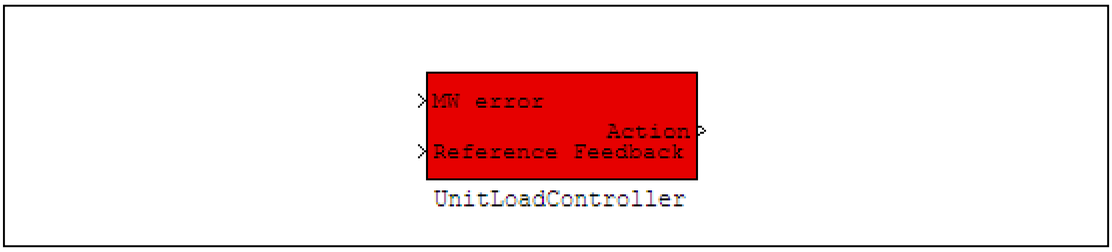

Figure 5.1.1-1: Unit Load Controller big block of the gas turbine big block

"Load Controller" module of the "Unit Load Controller" is a PI controller. MW error is introduced to a PI controller and the corrective action is a "Load Reference" signal for the gas turbine unit. Since interfacing to a unit is via raise/lower signals; "Load Reference Feedback" from the unit is compared with the output of the PI regulator, to generate raise/lower signals at the "Pulse Converter". In this model, raise/increment signal is a "1" and lower/decrement signal is "-1". Since both raise and lower can not

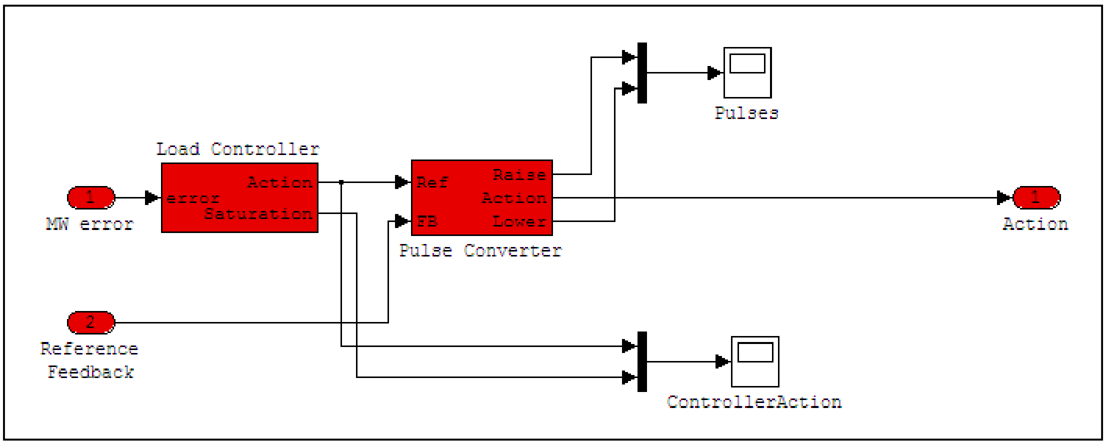

Figure 5.1.1-2: Building blocks of the Unit Load Controller

be active at the same time, appropriate integration of those raise/lower signals at the "PulsetoRefConverter" block give the total amount of change to be made on the load reference.

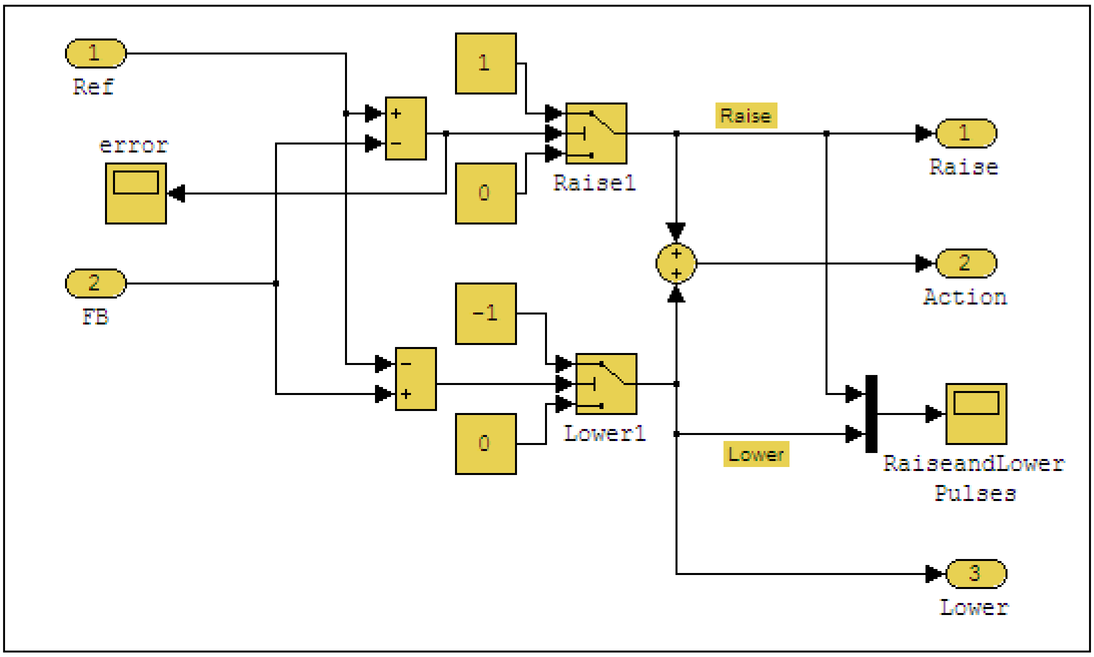

Figure 5.1.1-3: Building blocks of the *PulseConverter*

For the discussion of other internal blocks used in this gas turbine big block, please refer to Figure D-2 of Appendix D and detailed information given in Chapters 3 and 4.

5.2 Steam Turbine Big Block

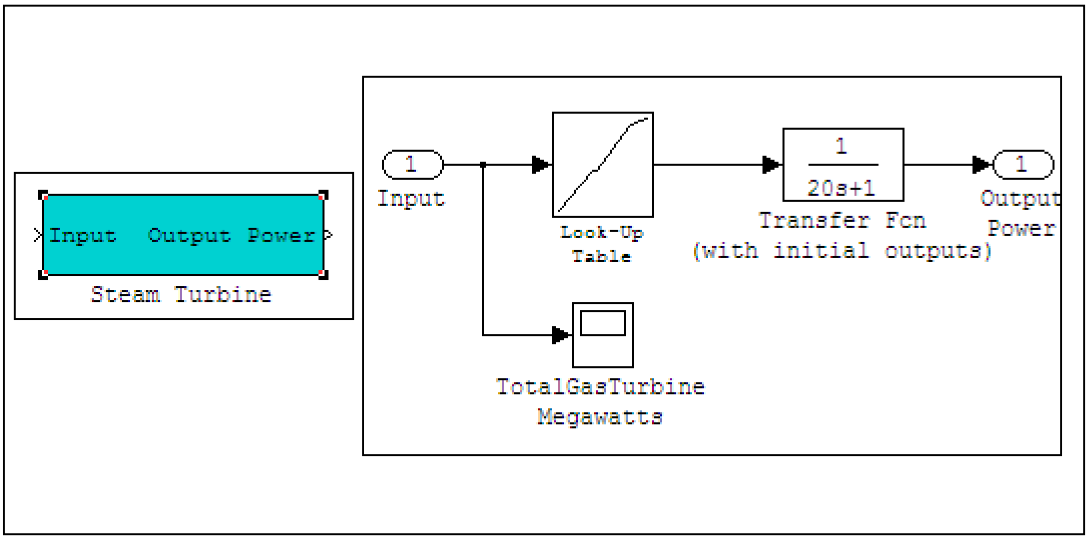

Figure 5.2: Steam Turbine Big Block and internal building blocks.

“Steam Turbine” big block expects the summation of both gas turbine output megawatts as input and calculates the corresponding steady state steam turbine output power via a look-up table. Dynamics are introduced as a simple time delay function.

5.3 Plant Load Controller Big Block

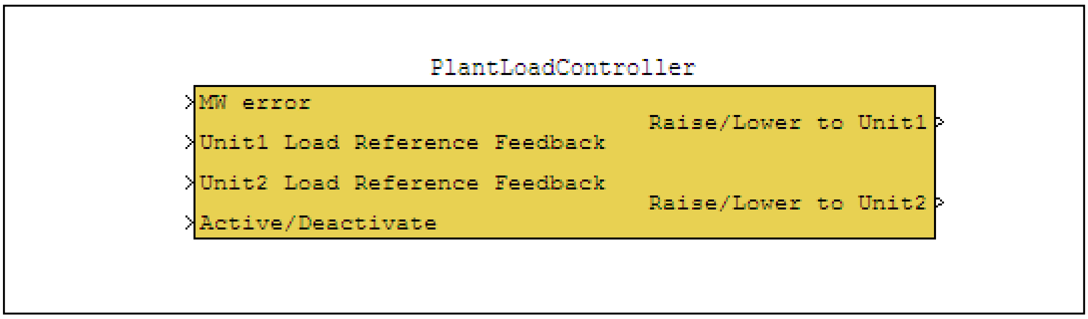

Figure 5.3-1: Plant Load Controller Big Block

Just like unit load controller, “Plant Load Controller”'s primary input is the error between the reference megawatts and instantaneous feedback, namely as “MW error”. MW error is introduced to a PI controller and the corrective action is a Load Reference signal for both gas turbine units. Since interfacing to a unit is via raise/lower signals, "Load Reference Feedback" from the units is compared with the output of the PI regulator, to generate raise/lower signals at the "Pulse Converter" same as used in Unit Load Controller. Similarly integration of those raise/lower signals gives the total amount of change to be made on the load reference.

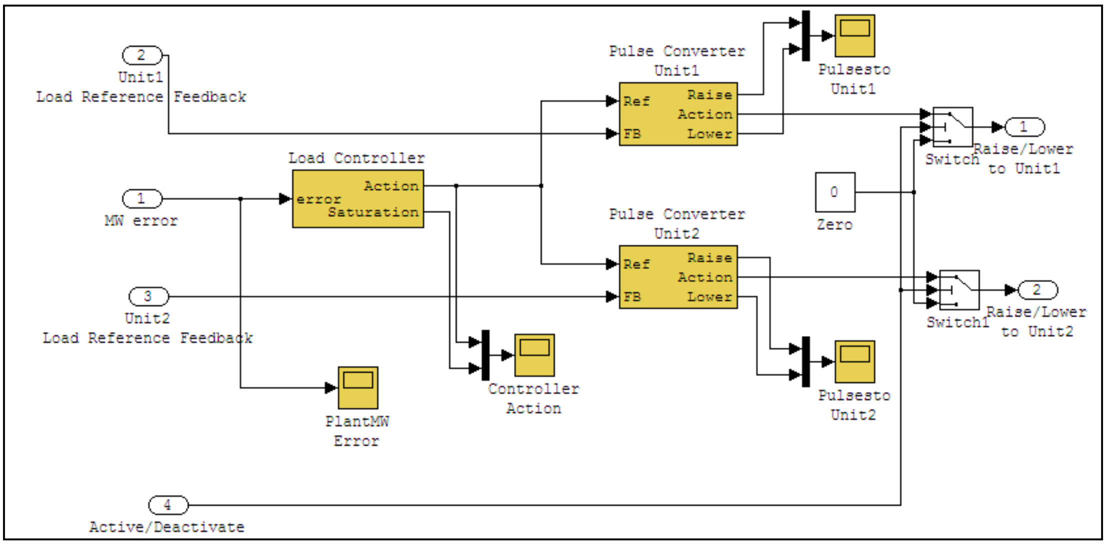

Figure 5.3-2: Building blocks of the *PlantLoadController* big block

CHAPTER 6

SIMULATION RESULTS AND DISCUSSION

6.1 Primary Response of a Gas Turbine Unit to a Frequency Deviation

Let us assume that the gas turbine unit whose primary response will be simulated here is a 250 MW rated output machine and has 5% governor droop characteristic.

When we analyze the case of gas turbine units that have droop governors and are connected to a large interconnected system, participation of one or two units to a frequency deviation by changing their output, can not make the system reach an equilibrium point but can only play a minor role to reach the mentioned equilibrium. Only the contribution of many units can make the system reach a quasi-steady state equilibrium in an interconnected network.

For the purpose of this study, in analyzing the response of a gas turbine and a combined cycle power plant's response to a frequency change; quasi-steady state deviation in the frequency will be a controllable disturbance that is intentionally introduced and removed to the frequency or speed inputs of the turbines. So the grid behaviour, which can only be represented by the collection of many power plants will be "assumed" to be a frequency disturbance that can be manipulated on, just for the ease of applicability. In the end whatever the response of the unit that is being simulated, grid will behave with its own distributed characteristic, which can only be assumed to be in accordance with our desires for the simulation case of a power block of a combined cycle power plant.

6.1.1 CASE 1: Response of a Single Gas Turbine Unit to 200 mHz Frequency Deviation (Disturbance ON= 70.sec, Disturbance OFF=184.sec)

Figure 6.1.1 is the simulation of a gas turbine unit's primary response to 200mHz frequency deviation. According to the UCTE standards [1] maximum allowable quasi-steady state frequency deviation is 180 mHz. For the ease of calculations, our case study will be for a 200mHz frequency deviation. See that with the droop value of 5%, a machine with rated output of 250MWs should respond with 20MWs since,

$$Droop = -\frac{\%\Delta f}{\%\Delta P} \Rightarrow \%5 = \frac{200mHz/50Hz \times 100}{\%\Delta P} \Rightarrow \%\Delta P = 8\% \Rightarrow 250 \times 0.08 = 20MWs$$

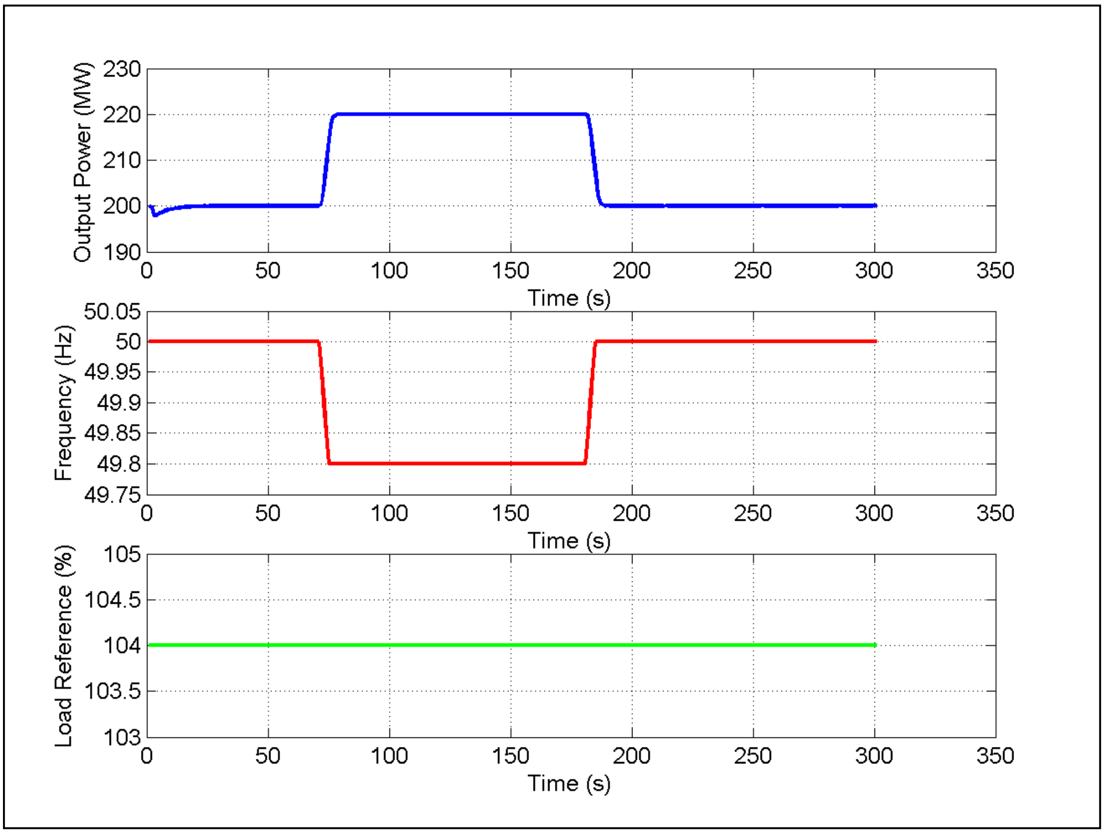

Figure 6.1.1: Primary response of a gas turbine unit to 200 mHz frequency deviation

Initial steady state output of the machine set to 200MWs maintains a reserve power level of more than 20 MWs for the ambient conditions under concern, so that output limitation due to acting temperature controller is not observed.

Please see that machine responds to the frequency deviation naturally without any interference to the "steady state load reference" (%Load reference is constant at %104 through the simulation period as on Figure 6.1-1. Signal depicted as Point "A"on Figure D-2 of Appendix D.). Hence, when frequency deviation is over, machine is back at its initial, steady state operating point.

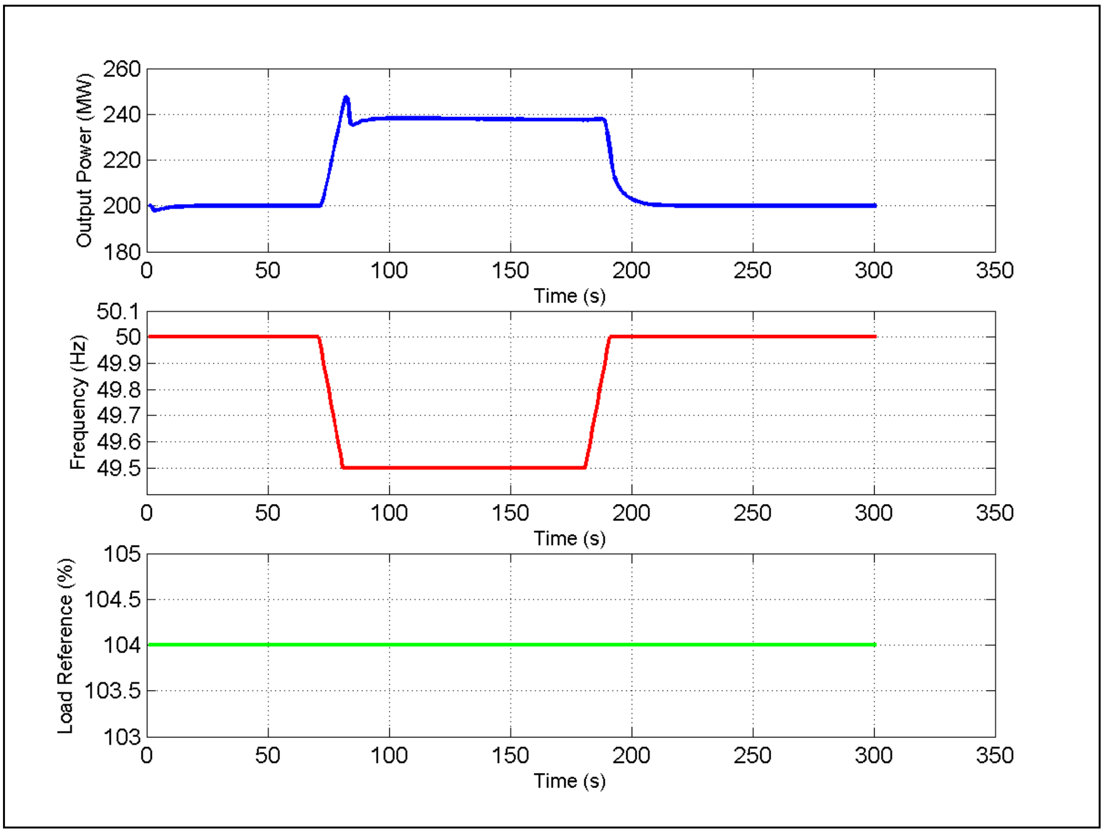

Figure 6.1.2-1: Primary response of a gas turbine unit to 500 mHz frequency deviation.

6.1.2 CASE 2: Response of a Single Gas Turbine Unit to 500 mHz Frequency Deviation ($T_{ambient}$=T, Disturbance ON=70.sec, Disturbance OFF=190.sec)

As any other machine, a gas turbine unit also has a maximum output controlled by the temperature controller. (Temperature Control -Fuel- on Figure D-2 of Appendix D). Peculiarity of the gas turbine unit is the dependency of this maximum output to many factors, especially to the ambient temperature. 500 mHz deviation in the frequency will force the machine to reach its maximum output as a result of its primary response. On Figure 6.1.2-1, output power limitation due to acting temperature controller can be seen. Although, governor increases the output to 250 MWs, required for 500mHz frequency deviation, temperature controller pulls back the output to the maximum allowable operating point that machine can withstand continuously at these ambient conditions. Please refer to Figure 6.1.2-2 where fuel flow reference from the governor and fuel flow reference from the temperature controller can be observed (signals depicted as "B" and "C" respectively on Figure D-2 of Appendix D). Once the fuel flow reference signal from the Temperature Controller is less than the fuel flow signal generated by the Governor, it passes through the "minimum value selector" and applied to the gas control valves.

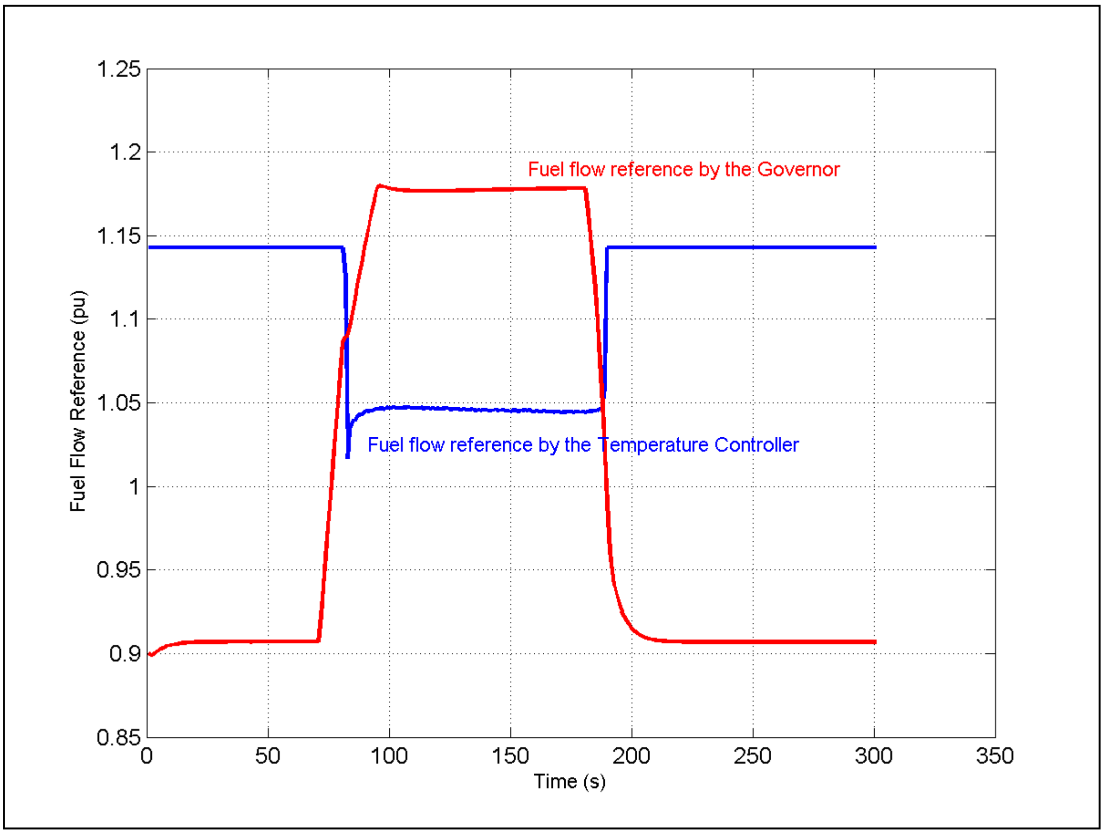

Figure 6.1.2-2: Fuel flow reference signals both from governor and temperature controller. See that, while governor increases the fuel reference in order to respond to frequency deviation, temperature controller decreases its output. The lower value becomes valid due to the minimum value selector.

6.1.3 CASE 3: Response of a Single Gas Turbine Unit to 500 mHz Frequency Deviation ($T_{ambient}$=T+10, Disturbance ON=70.sec, Disturbance OFF= 190.sec)

This simulation case is identical with the preceding case except that ambient temperature was taken 10 degrees higher. Please see on Figure 6.1.3-1 that the maximum allowable point is much lower, due to the fact that increase in temperature will result in reduced airflow through the machine.

Figure 6.1.3-2 summarizes the simulation of the IGV angle (signal "D" of Figure D-2 of Appendix D) and corresponding airflow change (signal "E" of Figure D-2) for Case 2 and Case 3, where $T_{ambient}$=T and $T_{ambient}$=T+10 respectively.

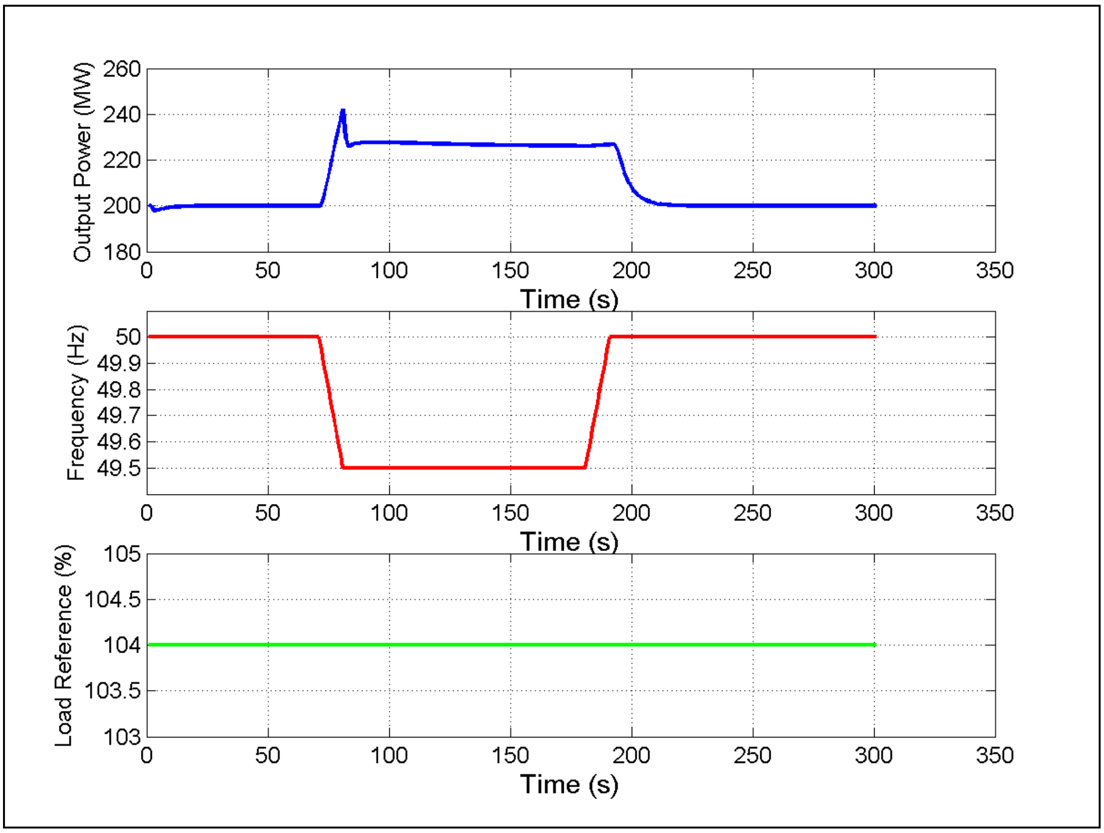

Figure 6.1.3-1: Primary response of a gas turbine unit to 500 mHz frequency deviation. Please see that due to higher ambient temperature unit cannot respond with the same reserve power as it provided for the preceding case.

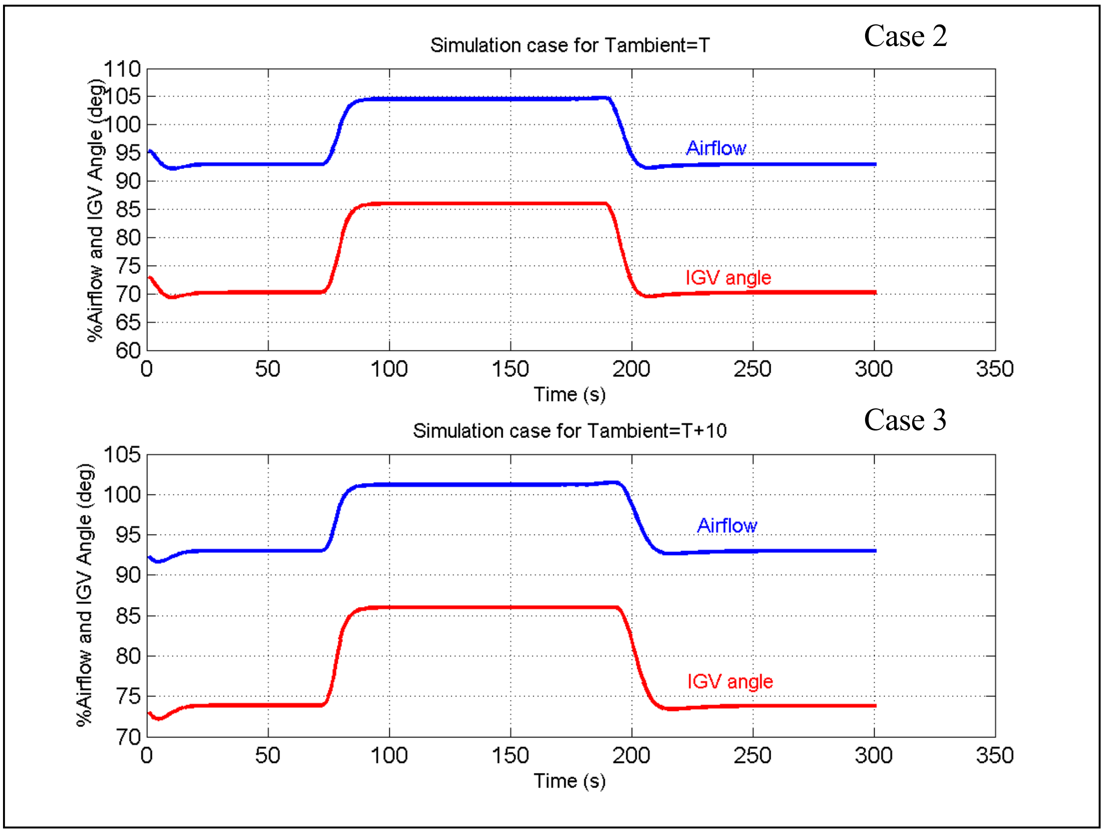

Figure 6.1.3-2: See that increase in fuel flow requires an increase in airflow to keep the turbine inlet temperature constant. Although for both cases, IGVs reach to their maximum limits, airflow is lower in Case 3 due to higher temperature.

6.2 Effect of Unit Load Controllers on Primary Response

Unit load controller is in the form of a simple PI controller that expects a MW set point and a feedback from the instantaneous output power so that integral action is applied on the error. (Please see UnitLoadController Block on Figure D-2 of Appendix D and the discussion in Chapter 5.)

Unit load controllers should only be considered to be used during start-up conditions and must never be activated during normal frequency regulating duty. Otherwise, primary response of the unit will be overridden by the load controller as a result of modifying the load reference. Please see how load reference is modified by the load controller on Figure 6.2-1. This figure depicts the result for the simulation of Case 1, like Figure 6.1-1, but this time unit load controller is active.

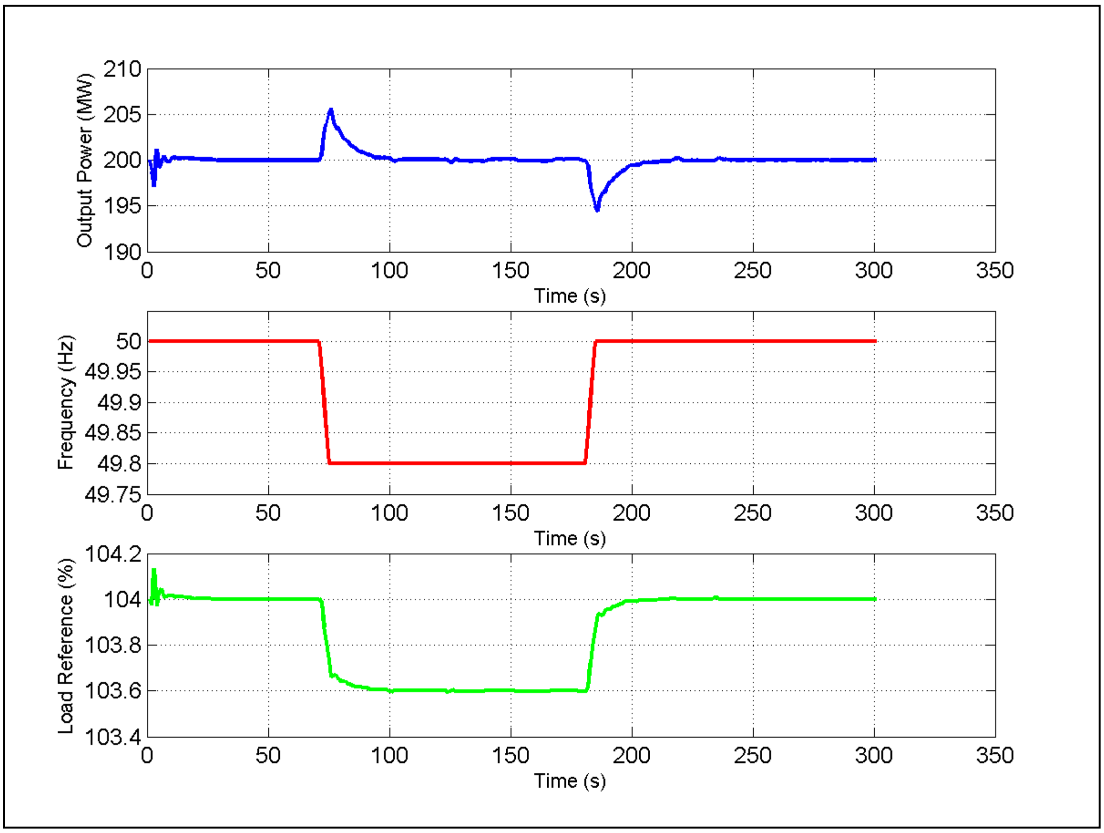

Figure 6.2-1: Simulation of a gas turbine unit's primary response to frequency deviation when unit load controller is active with a set point of 200MWs.

First of all see that initial transient in the output is just because of the initialization of the model. Primary response expected to be seen on Figure 6.1-1 is overridden by the acting unit load controller, which is strictly not desired for normal operation.

Explanation of the major difference on the above figure is straightforward. Please refer to Figure D-2 of Appendix D. Once there is a frequency deviation, an error is generated to be zeroed by the governor since it is a PI controller. Output of the governor is transferred to the gas control valves, resulting an increase in fuel flow (signal "G") to the turbine and in turn increase in MWs. (signal "H") Power being filtered and scaled, is fed back to the governor so that error for the governor is eliminated. It is also fedback directly to Unit Load Controller. If unit load controller is active, output (signal "H") being different than steady state MW set point (signal "J") of the load controller will introduce a non-zero error (signal "K"), which will in turn introduce non-zero controlling action (signal "L") on the load reference (signal "A").

As can be seen on Figure 6.2-1, contrary to preceding cases, even when the frequency is at temporary equilibrium with a certain deviation from initial steady state, output power of the unit is maintained again at 200MWs by modifying the load reference.

6.3 Primary Response of a Power Plant Following an External Plant Load Reference Signal

In the end, a power plant is a collection of generating units and it is natural to expect that the same behaviour will be observed on a larger scale. The main point here is, to implement the required interfacing system so that expected plant behaviour can be achieved.

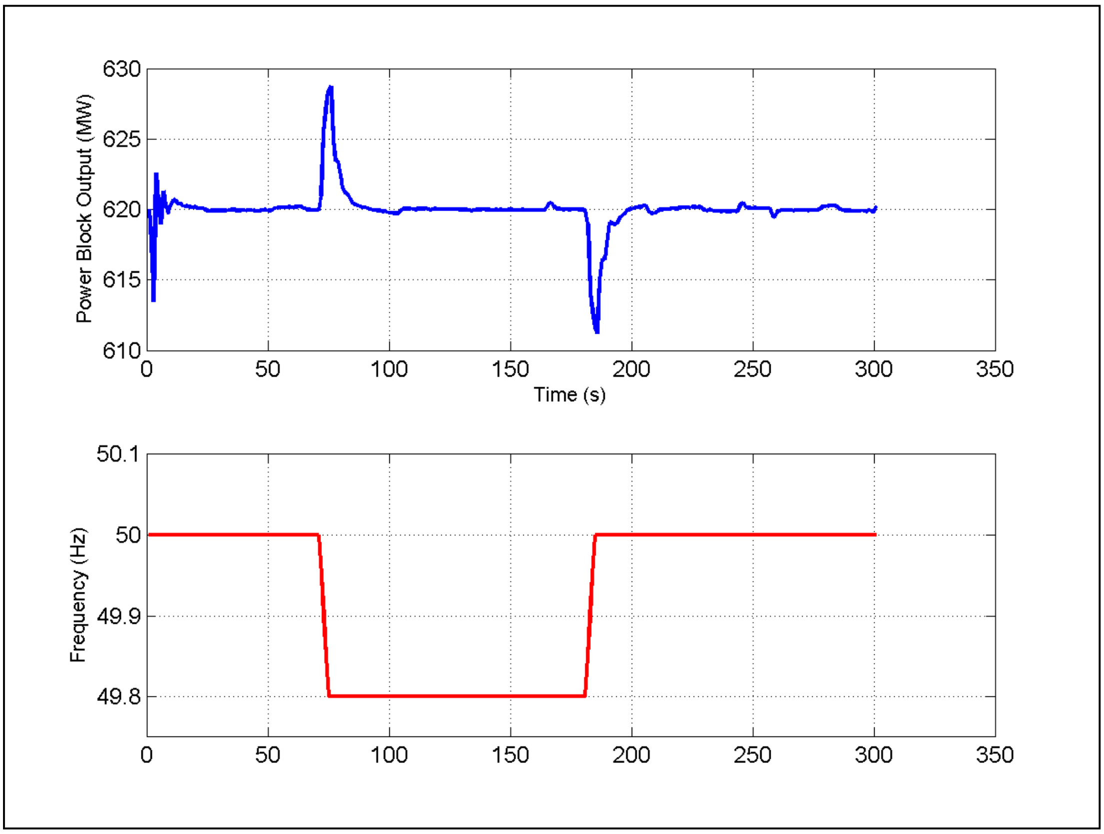

Figure 6.3-1: If a load reference signal which is not updated accordingly to be considered as an instantaneous reference signal is introduced to the system as if it were an instantaneous reference signal, then primary response of the power plant will be overridden.

Figure 6.3-1 is the behaviour of a power block to a 200 mHz frequency drop, without an external droop characteristic for the whole block being implemented. In the figure the MW output curve clearly indicates the increase in power output due to the frequency drop. However, as explained in Chapter 2, primary response is overridden for the sake of following the load set point reference perceiving this signal as an instantaneous reference. Therefore primary control has not been achieved in this case.

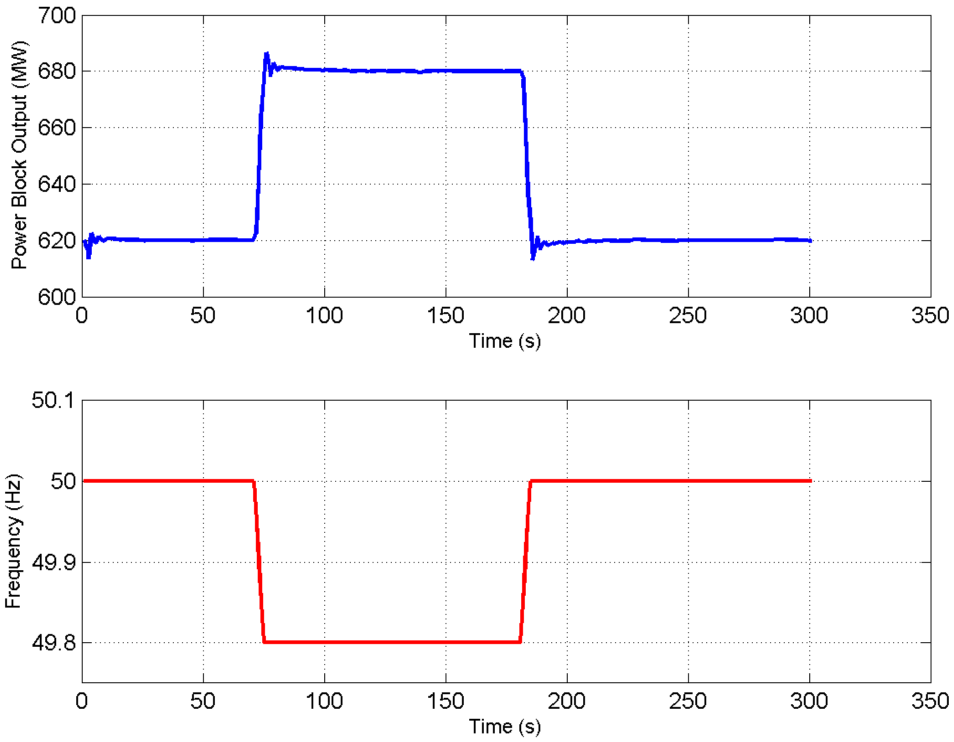

Figure 6.3-2: Expected output response of the power plant when droop characteristic is externally introduced to the system as can be seen on Figure D-1 of Appendix D.

Once the plant load reference is perceived as a steady-state reference, then with the addition of frequency deviation multiplied by the inverse of power plant droop, we can control the power plant output as a whole with respect to a desired frequency-power output characteristic as can be seen on Figure 6.3-2.

CHAPTER 7

CONCLUSION

In this thesis, load frequency control model for a combined cycle power plant is analyzed and a simulator is developed to investigate the behaviour of the power plant under various disturbances.

Principle building blocks of combined cycle power plants are the gas turbines. Due to this fact, a gas turbine model that will also include the physical nature of the machine was obtained in this study. As the modeling approach, static components that reflect the steady-state input-output relationships of fundamental physical variables were reached by curve fitting to the input-output data collected through the experiments performed [11]. For the dynamic characteristics, various papers [5], [6], were analyzed, modified and tuned accordingly to enlarge on the load control loops that are important for interconnected grid operation.

Generating units in a power plant, which are nominated to participate in system frequency control must not be at their maximum operating point (base load), but must be operated under "governor control" with certain, predetermined amount of reserve power. Of course in addition to these facts, proper control over correctly configured load control loops is a must. Otherwise, units will either not respond to frequency deviations or; those respond, will have to load/unload in large magnitudes to an extent, frequency occurrences of which might not only worn out the machines but also result in inadvertent grid behaviour. So appropriate control loop implementation is a must for all power plants.

Reserve power control, although very important and very difficult for gas turbine based power plants was not analyzed in this study. But modeling approach of this thesis work offers possible means for controlling reserve power at gas turbines so that predetermined amount of frequency responsive reserve power can continuously be maintained.

Regarding the testing work done on the gas turbines, step change tests in speed feedback of the turbine governors can only give information on the droop value and loading rates of those machines. Such kind of tests will not and cannot give "definite" information about performance of reserve power provision.

Primary reasoning for the above claim is that frequency and in turn air flow which is critical for the temperature limiter does not actually change during such a test; secondly those tests are carried out at "test" conditions of vast reserve power levels at which machines are hardly operated. So success on those tests does not have to reflect the reality. That's why, "15 minutes of continuous operation at the quasi-steady state frequency requirement of UCTE" [1] must be understood in terms of above claim especially for gas turbines, because hydro and steam units do not have the previously explained characteristic of the gas turbines.

There is one more concept that needs to be clarified in the evaluation of primary control response. Turbine rotors of 40 to 60 tons of 3000 rpm rotating masses are huge kinetic energy reservoirs with their high inertias. So instant and short load pickups (1 to 3 seconds of 2-4 MWs at falling frequency) are purely because of the inertial response and cannot be continuously maintained without increasing the power generation.

Analysis of Turkey's grid frequency (please refer to Figure E-1 of Appendix E) reveals the fact that there are continuous oscillations on the frequency even though there is no major disturbance in the system. One of the key reasons for this apparent problem might be the primary response being overridden by the inappropriately acting load controllers or improperly configured power plant secondary control interfaces.

Recently, units with digital controllers and that have a Remote Terminal Unit (RTU) at their sites, are controlled on secondary control by the transmission system operator. It must not be forgotten that understanding of the reference signal sent to the plant is of utmost importance. It can either be perceived as a "steady-state reference" signal or an "instantaneous reference" signal. Correct interpretation of this signal has to be established among transmission system operator and power plant crew. In any case, units must be free to utilize their reserve power for the primary response. If a solid understanding on this issue is not reached, interconnected system's overall frequency behaviour can not be improved. Units under the effect of secondary controller's outer megawatt control loop, if not controlled correctly, once having given their governor response to a frequency deviation, will have to respond in an opposite way due to slowly acting secondary controller. A solution to this problem was presented and proved in a simulation program in this thesis.

A system-wide simulation, that will not only include the certain dynamic effects of different power plant types but also the electrical characteristics of generators and transmission system is probably the best next step to move forward following this thesis. Once also the AGC algorithm is integrated to this simulation, underlying principles to have an accurate control on Turkey's grid frequency can be established. But it must never be forgotten that, thorough understanding of power plant control loops is the fundamental concern to achieve a satisfactory solution.

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A P P E N D I X   A

GAS TURBINE UNIT START-UP AND LOADING DATA SAMPLED AT 4 SAMPLES/SECOND

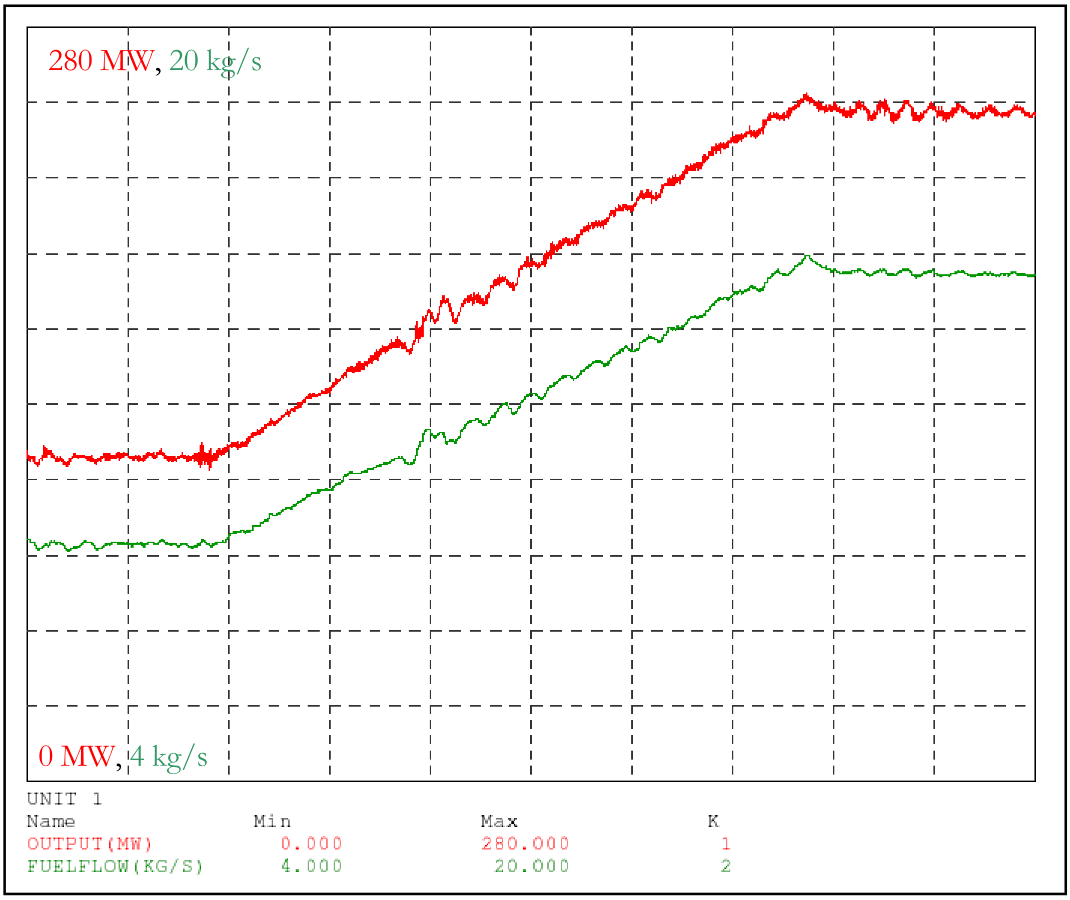

Figure A-1: Gas turbine unit output and fuel flow relation real time plot with respect to time. *(860 seconds total observation time, 4 samples/sec)*

This trend was obtained while loading the unit up. Linear relationship between output power and fuel flow can be clearly seen.

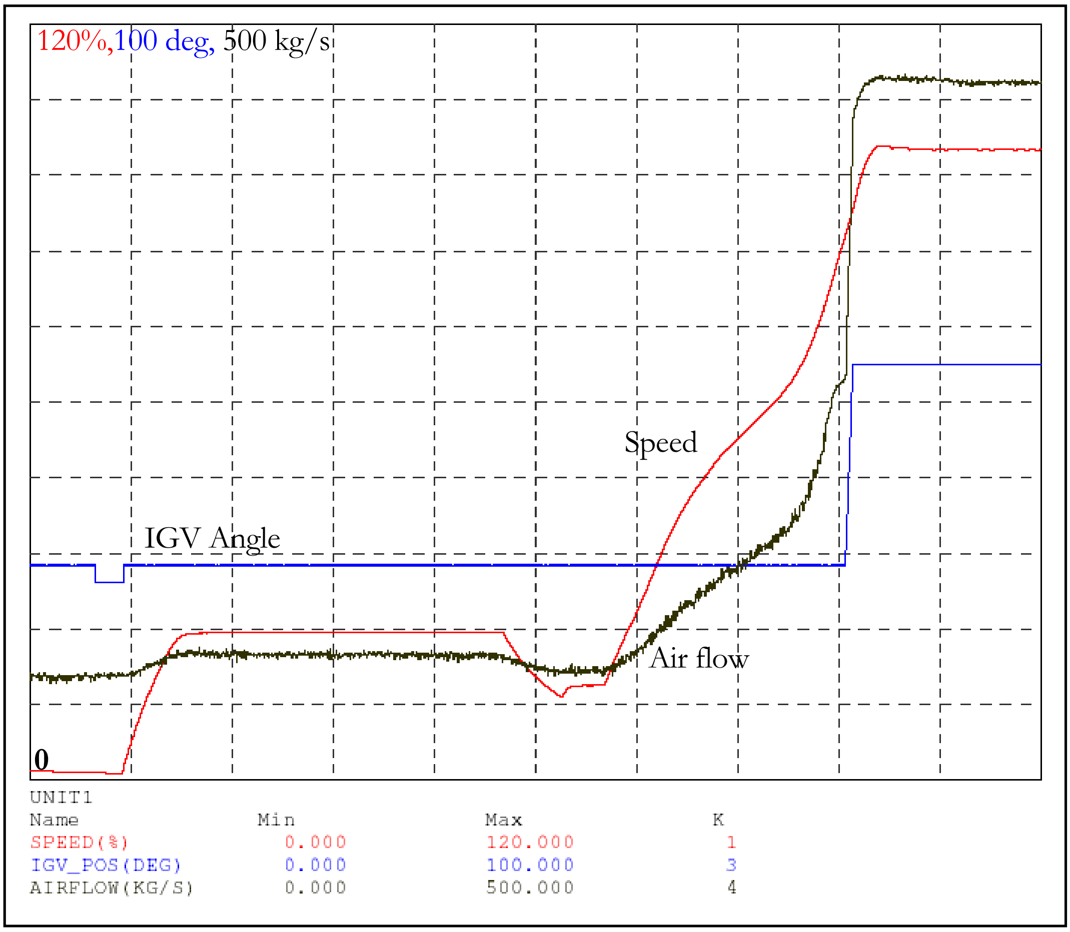

Figure A-2: Gas turbine unit start-up real time plot (0 to 100% speed). Dependency of airflow to speed and IGV (Inlet Guide Vane) position. *(1802 seconds total observation time, 4 samples/sec)*

This trend was obtained during a unit start-up ie. while the unit is brought to 3000 rpm from standstill. Dependency of airflow both to the speed and IGV angle can be apparently seen. See that while IGV angle is constant at around 27 degrees, airflow increases with the increasing speed. When IGV angle changes from 27 to 55 degrees, a very apparent change in airflow can be observed. Once IGVs are settled at 55 degrees, further increase in airflow is due to speed only.

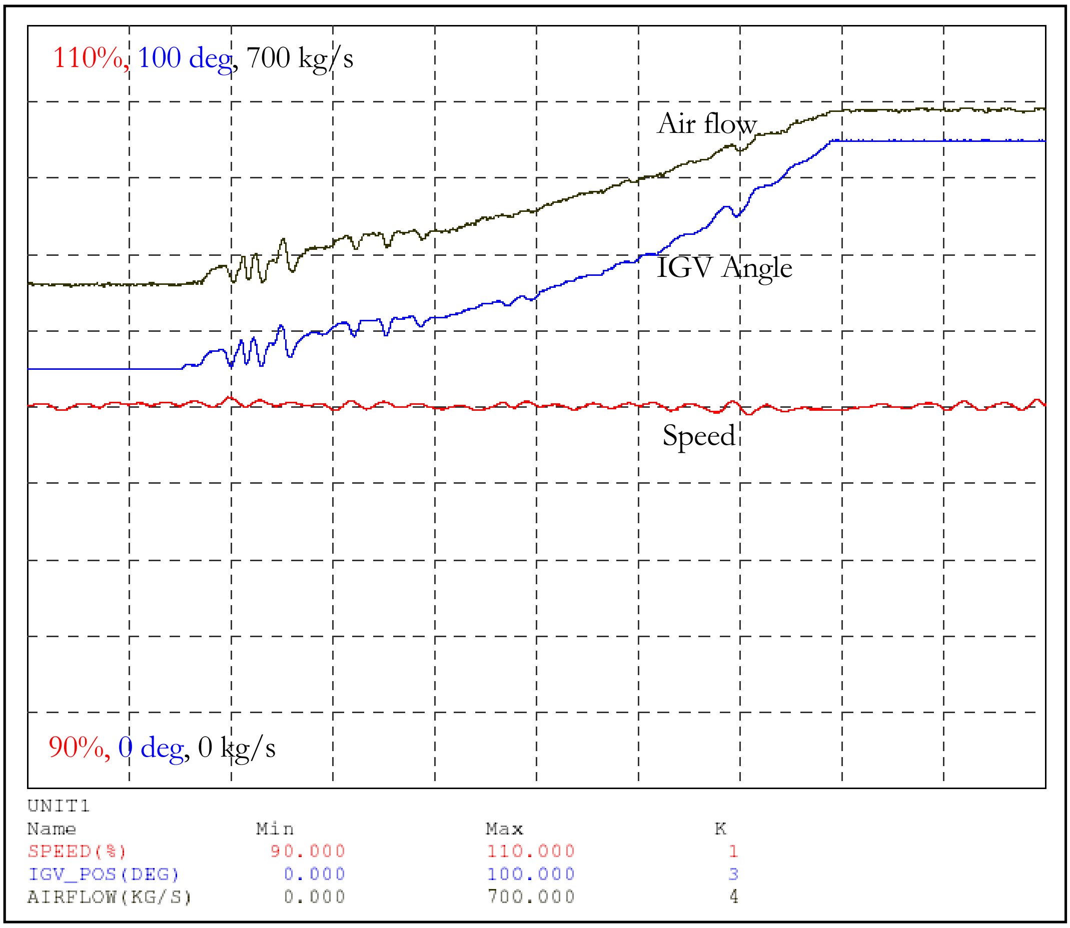

Figure A-3: Dependency of air flow to IGV angle at *constant* speed. *(647 seconds total observation time, 4 samples/sec)*

This trend was obtained after synchronization to the grid and while the unit is being loaded up. The change in speed is only due to random oscillations of the power network. Also on this graph, apparent dependency of airflow to the IGV angle can be seen while IGVs are opened up from 55 degrees to 86 degrees.

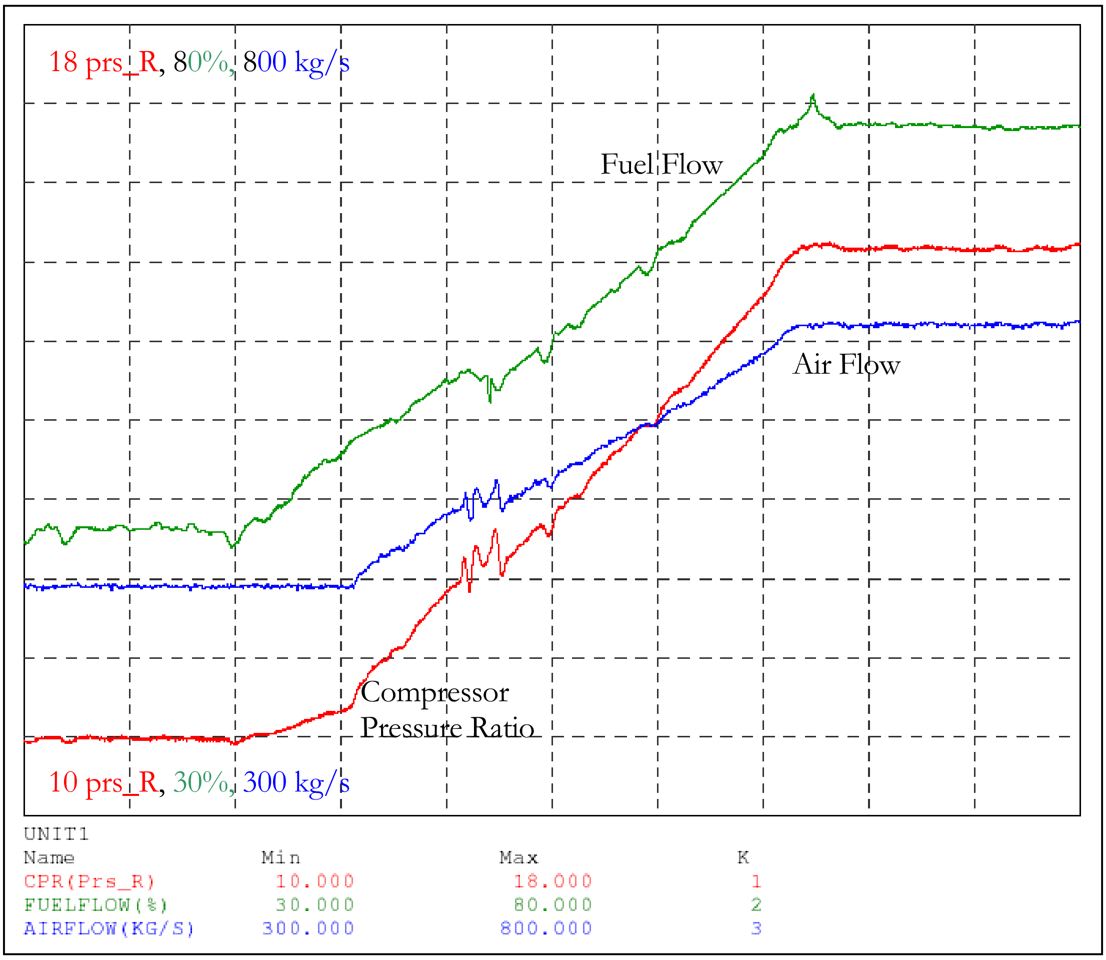

Figure A-4: Dependency of Compressor Pressure Ratio, to airflow and fuel flow. *(840 seconds observation time, 4 samples/sec)*

This trend was obtained while the unit is being loaded up. Increase in fuel flow proves this fact. While loading up, after a certain limit, control system increases airflow to compensate for the temperature increase that will result due to increasing fuel flow. As a result, apparent dependency of compressor pressure ratio both to air and fuel flows is observed.

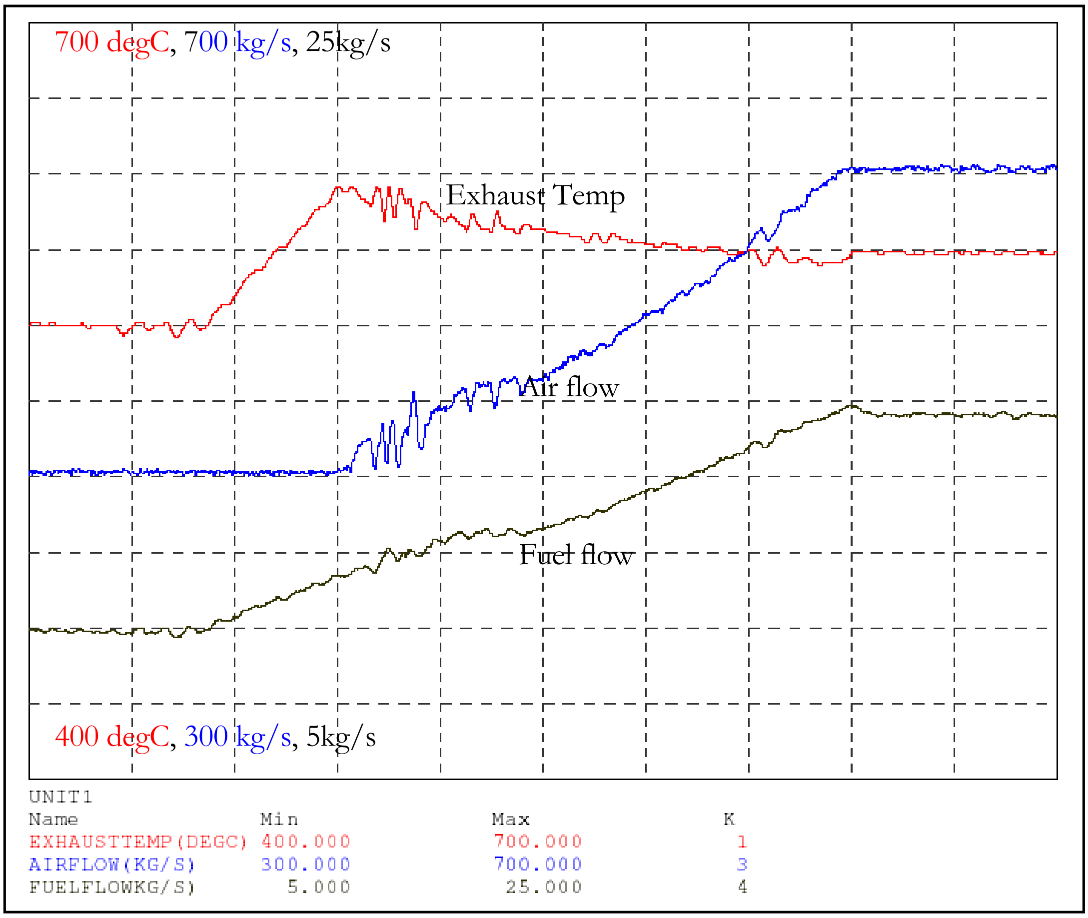

Figure A-5: Dependency of exhaust temperature to airflow and fuel flow. *(842 seconds total observation time, 4 samples/sec)*

This trend was also obtained during a unit load-up. See that there is a period of time where air fuel flow is constant but fuel flow is increasing accompanied with a corresponding increase on the exhaust temperature. After a certain temperature value, increase in airflow together with an increase in fuel flow results in a decrease at the exhaust temperature. Hence proving the fact that exhaust temperature is directly proportional with fuel flow and inversely proportional with airflow.

A P P E N D I X   B

STEP RESPONSE TEST

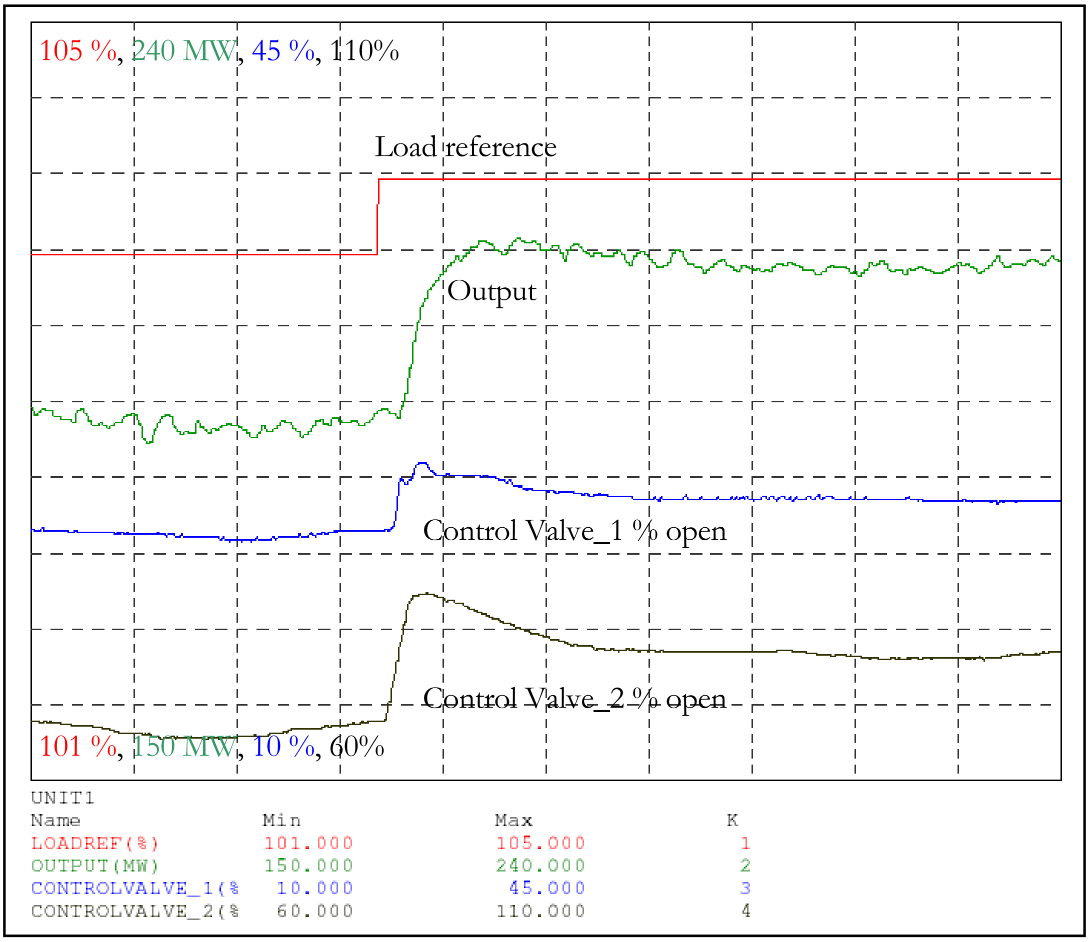

Figure B-1: Step response of a gas turbine unit to 0.4% change in the load reference. *(133 seconds total observation time, 16 samples/sec)*

APPENDIX C

OBTAINED CURVE FITS FOR THE OBSERVED INPUT-OUTPUT DATA

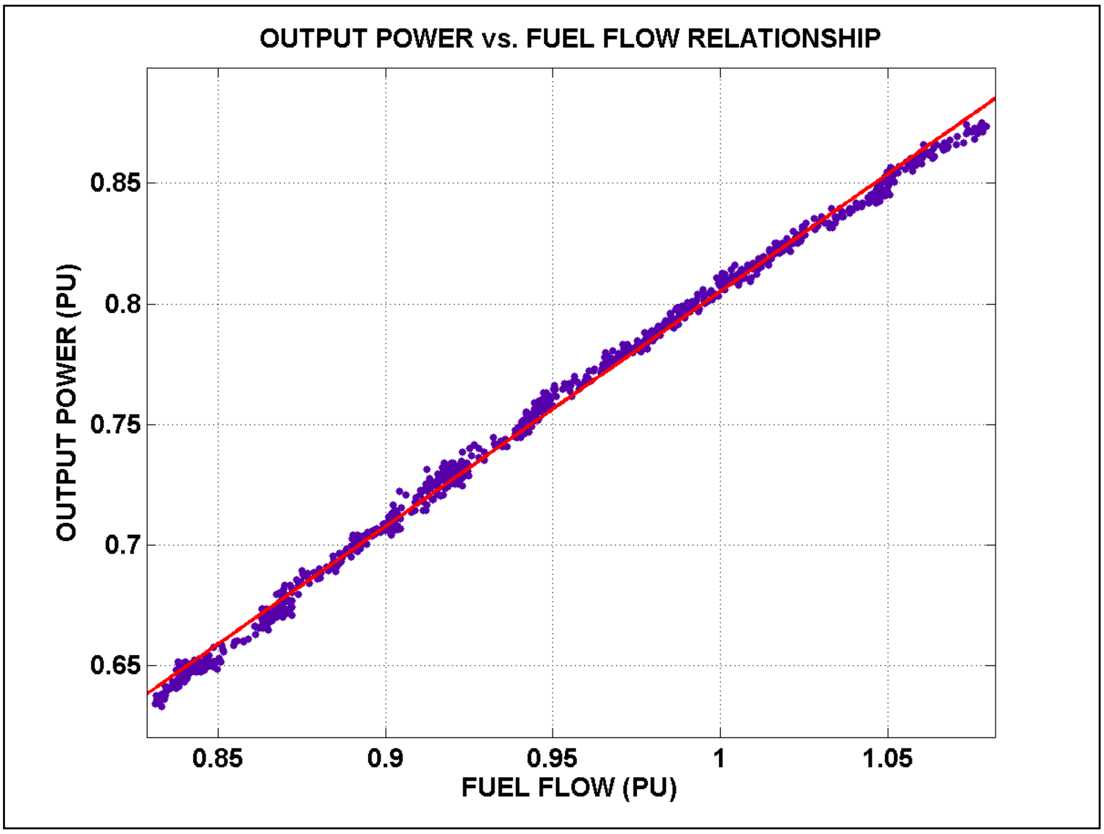

Figure C-1: Output power vs. fuel flow relation. Apparent linearity between fuel flow and output power together with the first order polynomial fit can be observed.

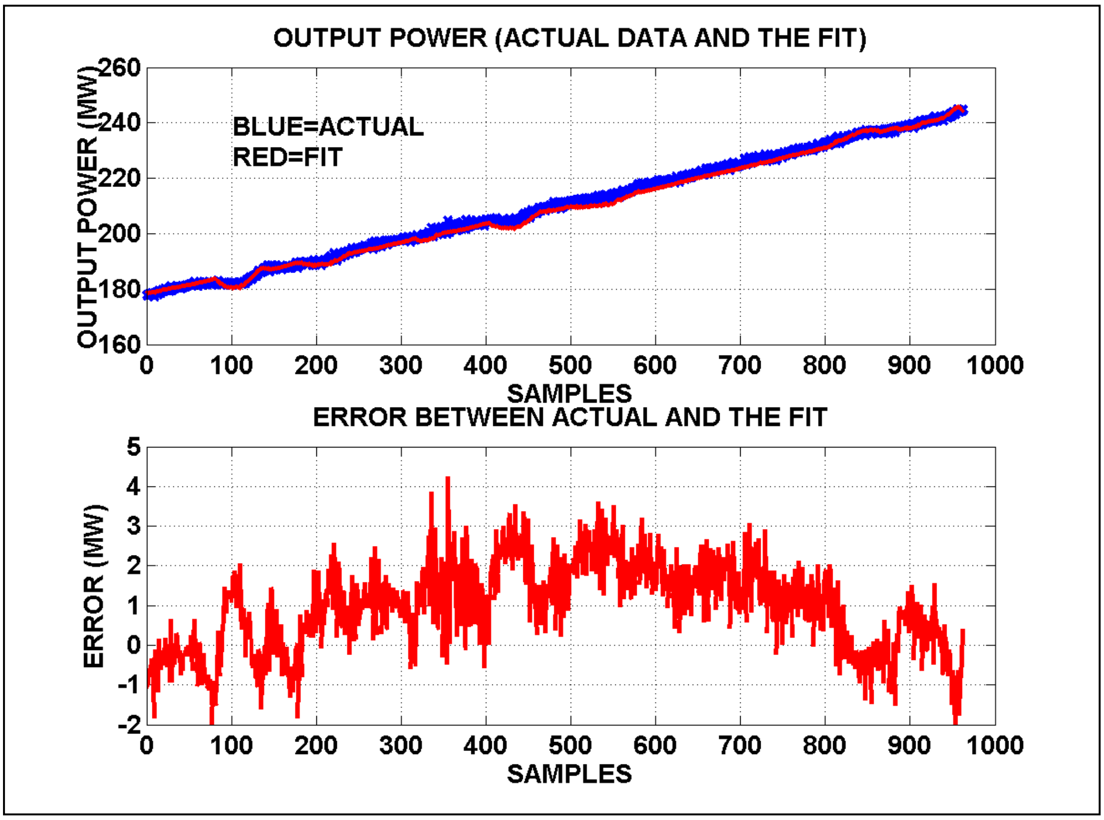

Figure C-2: Analysis of the actual data and the fit for the output power- fuel flow relationship. Error less than %1 can be accepted as a successful fit.

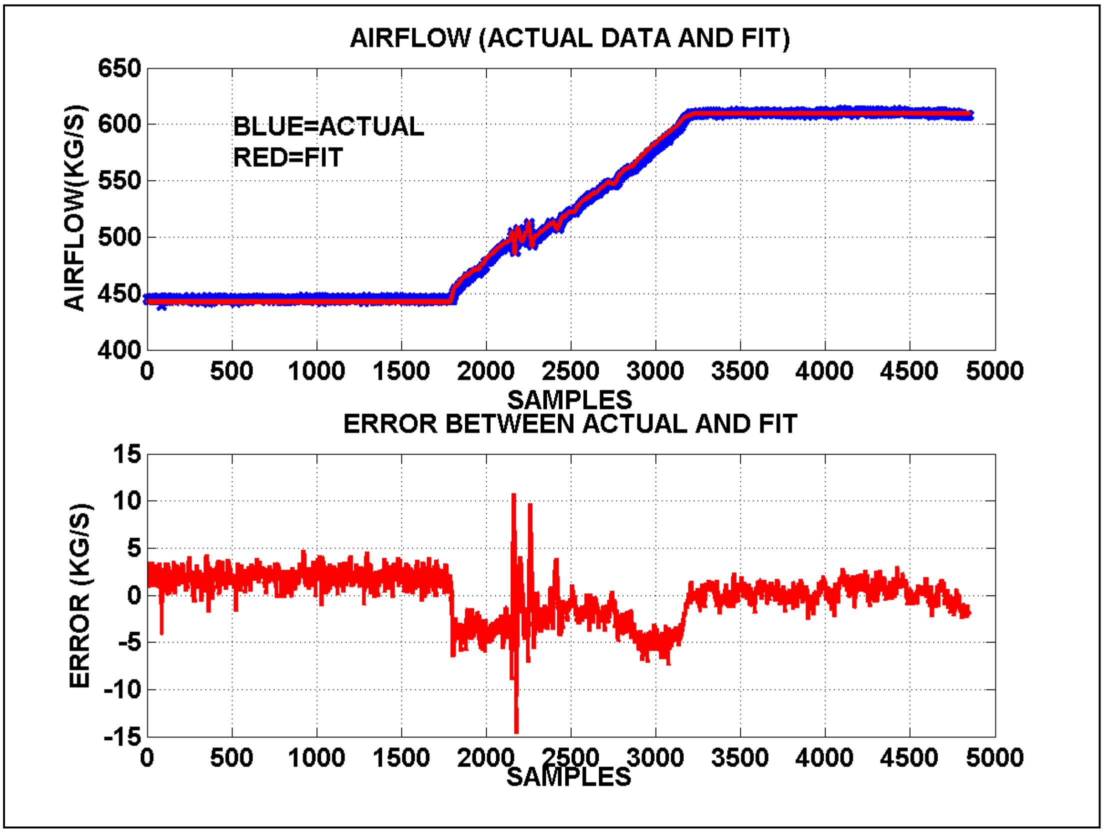

Figure C-3: Analysis of actual data and the fit for the Airflow Calculation.

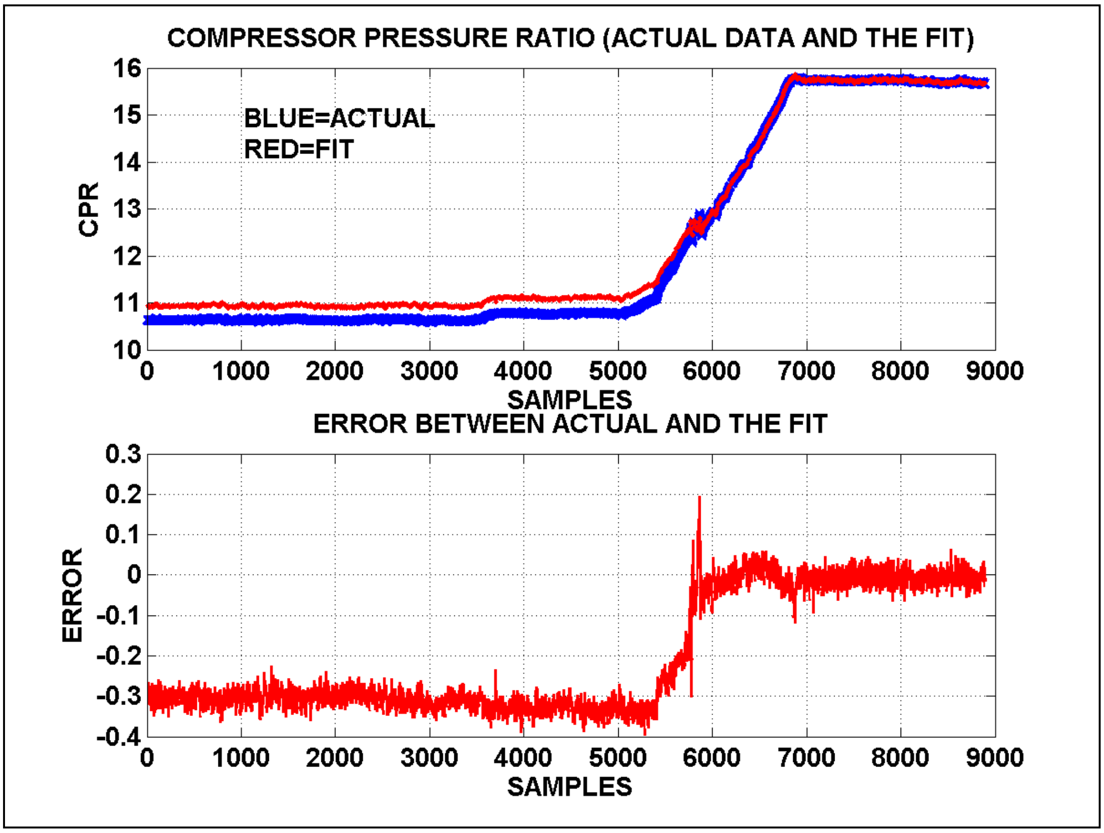

Figure C-4: Analysis of actual data and the fit for Compressor Pressure Ratio Calculation. See that apparent error for data samples before 5500 is because of intentional elimination of curve fit for those points in order to increase the accuracy for higher operating points. Please see Figure 4.3.2 of Chapter 4 for the discussion.

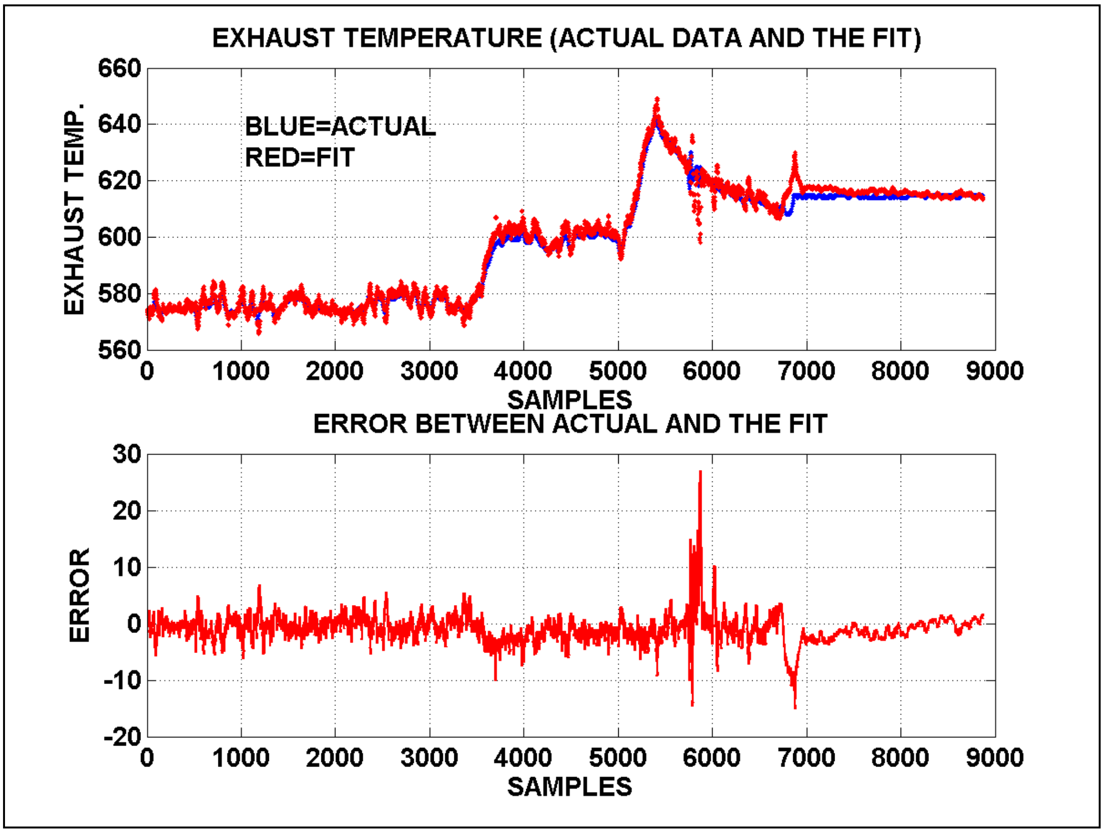

Figure C-5: Analysis of actual data and the fit for Exhaust Temperature Calculation.

A P P E N D I X D

SIMULATION BLOCK DIAGRAMS

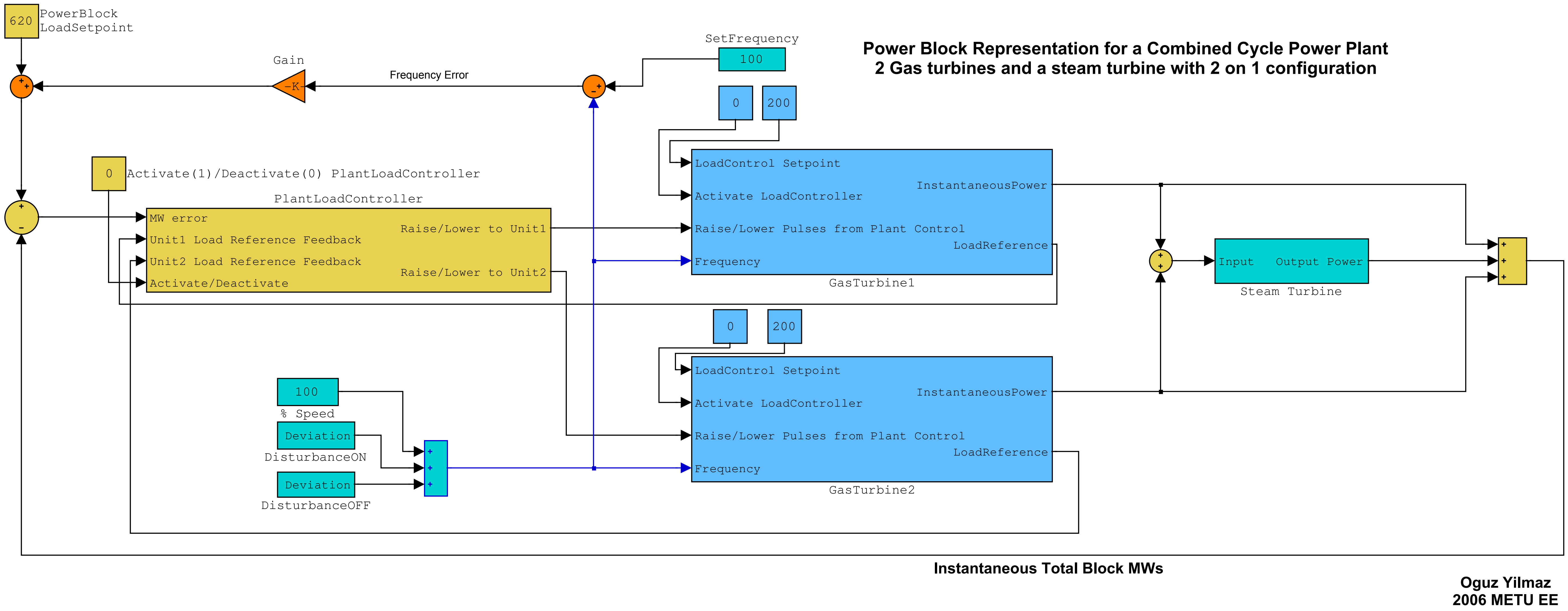

Figure D-1: A combined cycle power block representation with a Plant Load Controller

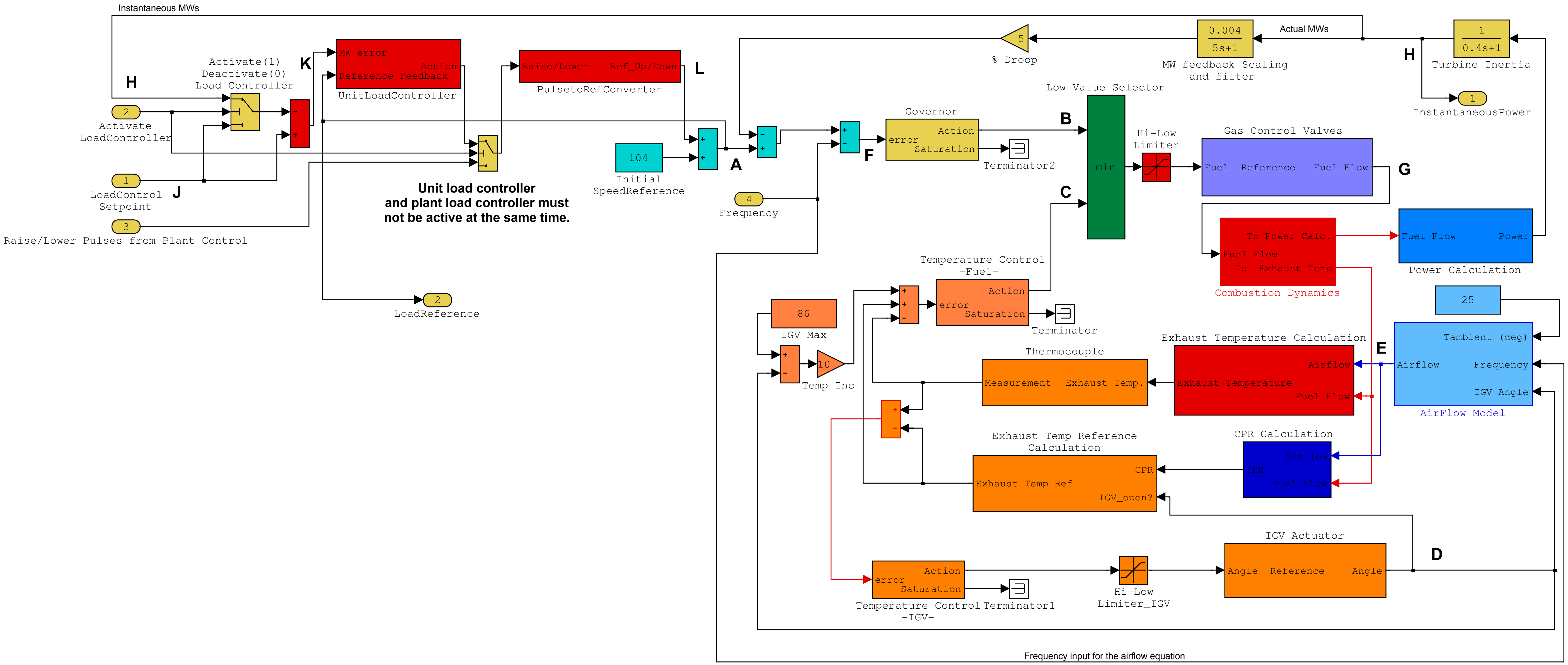

Figure D-2: Dynamic Gas Turbine Model

Oguz Yilmaz
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A P P E N D I X   E

NETWORK FREQUENCY TREND

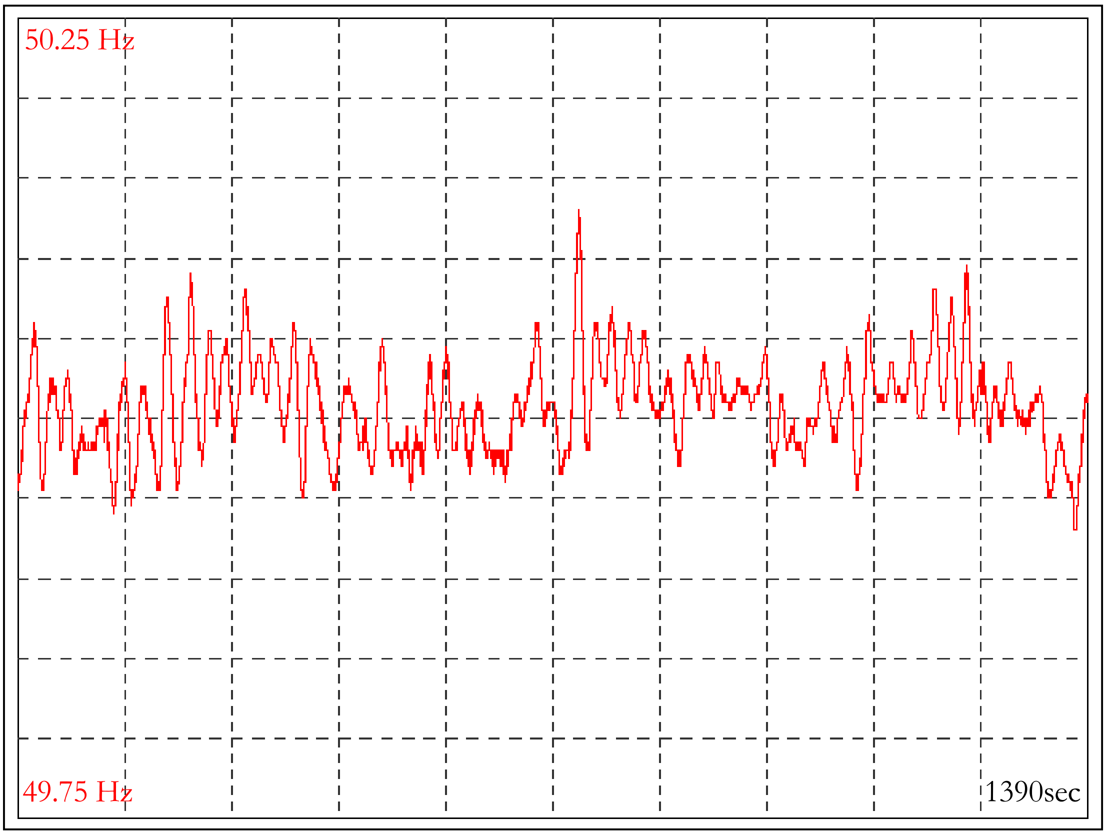

Figure E-1: Frequency trend of Turkish grid.*(1390 seconds of total observation time, 8 samples/sec)*