The Simplified Simulation of a Boiler, Turbine and related Control Systems

Antony John Turner

A dissertation submitted to the Faculty of Engineering, University of the Witwatersrand, Johannesburg in fulfillment of the requirements for the degree of Master of Science in Engineering.

Johannesburg, 1998

DECLARATION

I declare that this dissertation is my own, unaided work. It is being submitted for the degree of Master of Science in Engineering at the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

A.J. Turner

Signed on this 7th **day of** August **199**8

DECLARATION

I declare that this dissertation is my own, unaided work. It is being submitted for the degree of Master of Science in Engineering at the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

A.J. Turner

Signed on this 7th **day of** August **199** 8

ABSTRACT

The simple simulation of a power generating unit has numerous advantages in the fields of engineering, testing and training. This dissertation investigates the effectiveness of a simple reduced boiler and turbine model as an engineering tool. The implementation of a simple model of a physical boiler an turbine is described, as well as the implementation of the unit controls.

Results of actual tests performed on a generating unit are compared with the simulated data from identical tests. These comparisons show good correlation between the simulation and actual test data, leading to the conclusion that a simplified model is adequate as an engineering tool.

DEDICATION

To my Wonderful Wife
Natalie,
whose constant encouragement,
motivation, and support
gave me the determination
to complete this research.

ACKNOWLEDGEMENTS

The author would like to thank Doctor Simon Higgins who was instrumental in the initial suggestion as well as early motivation for this research project.

Thanks are also due to Professor I. MacLeod from the Electrical Engineering Department of the University of the Witwatersrand as well as the following people at Duvha Power Station:

Dave Taylor
Don Lawlan
Tienus Kuiser
Pieter Wessels

TABLE OF CONTENTS

Page

LIST OF FIGURES

Page

LIST OF TABLES

Page

LIST OF CHARTS

TERMINOLOGY AND SYMBOLS

Att .... Attemporator
Att SW .... Attemporator Spraywater
Autocontrols .... Automatic Control System for the Boiler, Turbine, Generator and Auxiliaries
BFP .... Boiler Feed Pump (includes the SFP and EFP's)
CW .... Cooling Water
EFP .... Electric Feed Pump
Generating Unit .... Boiler, Turbine, Generator, Auxiliaries, and all Autocontrols which together form a single power generating unit
HP Turbine .... High Pressure Turbine
IP Turbine .... Intermediate Pressure Turbine
Load .... Refers to the electrical power generated (MW) by the Generating Unit
LP Turbine .... Low Pressure Turbine
P. .... Pressure OR Proportional Control Element
PC .... Personal Computer (IBM compatible)
PD .... Proportional plus Differential Controller
PID .... Proportional plus Integral plus Differential Controller
PI .... Proportional plus Integral Controller
PF .... Pulverized Fuel (pulverized coal from the mills)
Plant .... Refers to the electrical, mechanical and other equipment and systems that together make up a generating unit
P., Press. .... Pressure
RH .... Reheater
SFP .... Steam Feed Pump
SH .... Superheater
SW .... Spraywater (Attemporator)
T. .... Temperature (in Degrees Centigrade)
Temp. .... Temperature (in Degrees Centigrade)
Turbine .... Unless otherwise specified, this refers to the set of HP, IP and LP 1 and 2 Turbines
Turbogenerator .... Turbine and Generator Unit
Unit .... Generating Unit

CHAPTER 1

1. Introduction

1.1 Background

A power generating unit is a complex system, involving all spheres of engineering. The combustion process involves both chemical and mechanical engineering, whereas the mass flows and thermodynamics require mechanical engineering knowledge in order to understand them completely.

The high fluctuations in flow rates, pressures, temperatures and differential temperatures encountered in a boiler and turbine cause a great deal of stresses and fatigue which may, over time, lead to the failure of critical plant. Coulman[1] states that "Fluctuations of the steam temperature during a transient resulting from some perturbation of the operating conditions may cause excessive wear in the tube circuitry and in the turbine, as well as reduced efficiencies". The subject of stress and fatigue is part of the engineering field of metallurgy, and to a lesser extent mechanical engineering. The theory behind the boiler controls, as well as the implementation thereof, lies in the electrical and process engineering field while the operation of the generator falls in the heavy current electrical engineering field.

In addition to the engineering disciplines, the people who best know the operation of a generating unit are the operators. Although they may be unaware of the theory or logic behind certain procedures, they are equipped with the experience with which to operate the unit, even under exceptional circumstances.

The complexity of the generating unit is further emphasized by Maffezzoni[2] who states the reasons as being due to " structural complexity, strong nonlinearity, interaction among subsystems, relevance of distributed parameter phenomena, considerable uncertainty of process parameters at macroscopic level (heat transfer constants, friction, flame radiation, two-phase interaction etc.)".

As can be seen, the knowledge at any power station is spread between many people in many disciplines, making problem solving and engineering difficult without the cooperation of many people. One of the drawbacks of splitting engineering departments according to discipline is that, for instance, a purely electrical modification to the controls may have serious mechanical implications which the electrical engineer is not aware of. This is overcome, to a certain extent, by close cooperation between the different departments (electrical engineering, mechanical engineering, maintenance, operating) when working on problems or design modifications.

One critical area where the problems outlined above are very apparent is that of the optimization of the generating unit controls. Although control optimization could be considered a pure electrical engineering function, optimization on a power station unit requires knowledge and understanding of every field of boiler and turbine engineering, as well as operating experience.

One possible solution to this problem is the use of a simulation of the relevant plant when attempting to provide engineering solutions or fault finding on a generating unit. Such a simulation, which would typically be carried out on a digital personal computer, may be used to model the behavior of the unit both before and after a modification in order to assess the overall effect of certain actions or modifications.

1.2 Analysis of the Problem

At present no simulation tool exists at Duvha Power Station for use as an engineering tool. As discussed previously, there is a substantial benefit that may be gained from the development of such a tool, provided that it fits certain user requirements.

One such simulation which is already available off-site at the Eskom training center in Midrand, is the operator training simulator. Although extremely useful in the training of operators, it has no further use in the engineering field. The main reason for this is that it is very difficult to perform even minor modifications to the simulation, and even these need to be done by a specialist (on that particular simulator).

An important requirement of any simulation is that it is easy enough to understand to enable an engineer to make modifications without spending months learning the simulation program or language. An engineer would still require a certain amount of background knowledge on the workings of the simulation package as well as the rationale of the implementation itself, before being able to make full use of the simulation. This approach also ensures that use of the simulation is not restricted to one or two specialists, but may be used by anyone needing to perform plant simulations.

Although an ideal model would include every system in the generating unit, many of the auxiliary systems have negligible influence on the overall operation of the generating unit. An example of such a system is the condenser vacuum control system which includes a physical system as well as autocontrols. Under normal circumstances the vacuum is maintained at a setpoint independently of the operation of the rest of the generating unit. Under normal circumstances this system could be modeled as a constant pressure condenser.

Another area which would benefit from the use of a power plant simulation is that of optimization of the unit autocontrols. Presently this is a long tedious process involving a lot of guess work which must be done while the unit is on-line. This introduces an additional risk of mistakes causing inefficient operation or even a unit trip. The optimization procedures also place extra stresses on the plant which negatively impact its efficiency and life-span.

A Simulation would be a useful tool for the optimization of a control system. This would be achieved by using a simulation to replace the physical plant as shown in Figure 1-1. The existing control system would be interfaced to the plant simulation, thus allowing extensive optimization without the negative effects of excessive stresses on the plant.

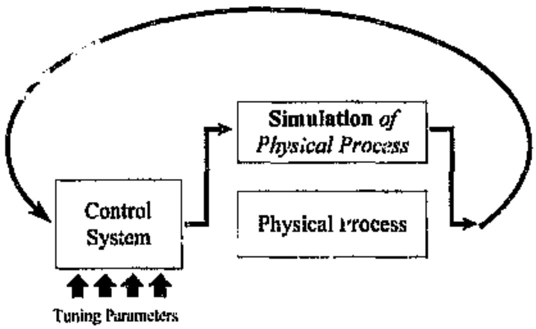


Figure 1-1: Simulation for Control System Optimization

The remainder of this dissertation addresses the problem of develop[illegible] that is both simple enough for easy understanding, while maintaining the [illegible] realism that is required of an engineering simulation tool.

This problem may be summarized as follows:

Is a relatively simple or reduced model of the boiler, turbine and related autocontrols adequate for use as an engineering tool?

The specific objectives of this research dissertation are:

a) *To develop a* *<u>simple</u>* *dynamic simulation of a boiler and turbine in terms of fuel input, boiler pressure and megawatts (MW) output. This excludes simulation of some of the less important systems such as the internal operation of the steamfeed pumps, blowdown vessel, condenser, mills etc.*
b) *To simulate the physical processes through the use of simple mathematics while confining the use of complex thermodynamic equations to a minimum.*
c) *To develop a detailed simulation of all relevant autocontrols which form part of the physical systems.*
d) *To develop an operator interface for the simulation.*
e) *To asses the question as to whether the simulation adequately fulfills the requirements of an engineering tool. This will be done by means of comparisons between live data and simulated data.*

1.3 Overview of the Dissertation

The dissertation begins by introducing the concept of simulation of a power generating unit. This chapter describes the problems which are to be addressed by simulation of the plant, as well as formulating the main objectives of the dissertation.

Chapter 2 discusses the application of simulations in general, as well as dealing with the issues of stability, speed and accuracy.

Chapter 3 provides a detailed description of the autocontrols which are relevant to this simulation.

Chapter 4 deals with the design and implementation of the simulation. It begins by describing the advantages of using a graphical simulation "language" as opposed to a non-graphical language. The design methodology is discussed next, followed by the detailed description of both the simple simulation and the alternative more complex simulation of the boiler and turbine. Lastly, the simulation of the autocontrols is described.

Chapter 5 describes the results of the tests done to determine the success of the simulation in achieving the goals outlined in chapter one. Comparisons are made between the live data captured during tests performed on a power generating unit and simulations of the same tests.

Chapter 6 concludes the dissertation by assessing whether th[illegible]'s outlined in chapter one have been achieved. The accuracy and limitations of the sim[illegible]on are discussed, as well as the significance of the simulation. Further research possibilities are briefly described at the end of the chapter.

The dissertation contains five appendices. Appendix A gives a brief description of Duvha Power Station. Appendix B tabulates a comparison between three simulation software packages, outlining the strengths and weaknesses of each. Appendix C quantifies the approximation errors which result from the simplification of the turbine and boiler. Appendix D contains the VisSim source code which makes up the complete alternate model. The source code for the simplified model has not been included.

CHAPTER 2

2. Simulation Concepts

Modeling and Simulation are disciplines that are becoming extremely important in almost all spheres of engineering. A model may exist either as a conceptual design or as a detailed mathematical model. Simulation of a system or process provides numerous advantages, especially in terms of resources used during the design phase.

2.1 Why Simulate?

Although there are an abundance of systems or processes that warrant simulation, the requirements can be classified into four main types (Debelle[3]). Although some simulations are designed with more than one of these objectives in mind, for clarity they may be identified as follows:

2.1.1 Knowledge Models - For Design and Understanding

In this instance, the simulation is constructed to be of assistance in the design of a system or solution. Although many examples may be cited, an appropriate example would be the simulation of a boiler to determine the optimal design based on physical limitations as well as operating requirements.

The required accuracy of design parameters is generally proportional to the complexity and criticality of the system to be designed. It is quite possible to design a simple item, for example a brick, without doing a single calculation or drawing. On the other hand, the design of a system of marginal complexity, such as a bicycle, requires a certain amount of calculation and drawing.

In the case of the power station mentioned above, an enormous amount of time, effort and resources are needed during the design. The following points are of paramount importance and have a significant influence on the final design.

- Available Capital
- Required Operating Efficiency
- Safety of plant and persons
- Required Reliability
- Stability of the Controls
- Controllability
- Expected Lifespan

The utilization of complex models as design aids, has brought about a significant decrease in both the cost and duration of the design process, while also increasing the performance of the plant. This has been further enhanced over the last decade by the increase in power of the personal computer and mainframe computers.

Consider the design of an aircraft or spacecraft. The design is based on physical principles, experimental data from similar models, past experience, and the various specifications imposed on the system. According to the design, the system should perform as intended. However, before the pilot is allowed to take the first test flight, it is preferable to verify the operation of the aircraft and controls without risking the pilot's life and the craft itself. Even before the aircraft is built, the manufacturer would like to verify the design before spending a large amount of time and money in its manufacture.

These models are generally built up from mathematical equations which define the behavior of the system's constituent elements. These equations are normally those describing mass and energy balances, the thermodynamic properties of fluids and gasses (water and steam, for example), and the static and dynamic characteristics of items of equipment (pipes, pumps, valves, etc.).
The parameters used in these calculations are normally precise and relate to a specific plant.

The models obtained in this way are normally very complex, but may be simplified by linearization at specific operating points.

2.1.2 Control Model

This model is used for the design, redesign and optimization of automatic controls. These models are normally obtained by a more comprehensive approach to process representation. The parameters used in these models do not necessarily relate to specific physics of the plant but normally describe plant functionality. These models are generally fairly simple and are mainly used for control studies. They must also take into account the behavior of the instrumentation and control actuators.

2.1.3 Training Models

Training models are used to simulate the behavior of a process for the training and practice of operating personnel. These models are normally based on the specific designs of the unit.

This type of simulation is employed extensively in the aircraft and spacecraft industry where it is far cheaper and safer to train pilots in a flight simulator than in a real flight. It is also possible to simulate a number of situations which would not normally be encountered during a routine training flight.

2.1.4 Operation Models

This type of model is used to study the behavior of the process and constituent parts in entirety. These models are normally described in a global manner, being less specific to individual systems. As a general rule, it can be said that dynamic aspects are represented in a simple way whilst the system's non-linear characteristics must be modeled in a more detailed manner.

An application of this type of model is for the verification of the physical operation of a plant compared to the required (simulated) results. This is only possible when the desired system has been modeled to the required accuracy. This is useful in identifying maintenance requirements or plant problems.

The last type of simulation differs from the previous three in one crucial way; it is the only model which requires on-line plant data. In order to compare actual plant operation with simulated plant operation, it is necessary to provide the same inputs to both the simulation and the simulated plant. This includes operator and environmental inputs as well as all inputs from associated plants.

Two possible ways exist to provide this information for the simulation.

a) The first method involves manual input of data into the simulation in parallel with operation of the plant. The disadvantage of this is the extra work required as well as the possibility of mistakes. This is not practical in the case of a power station such as Duvha due to the concentration and quick response needed by the operator during unit light-up, and especially when handling emergency situations.

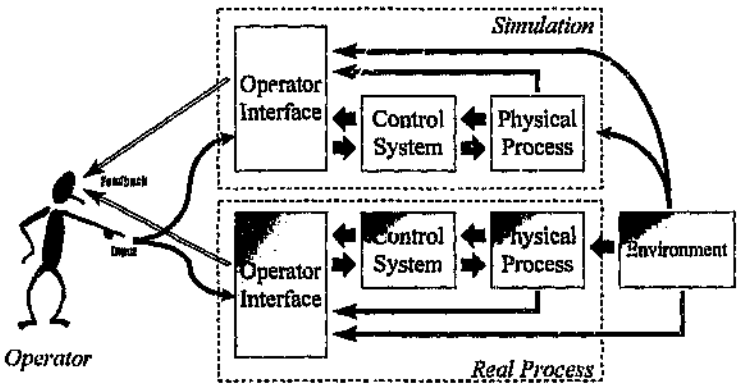


Figure 2-1: Online Simulation for Plant Verification - Manual Data Input

b) The second method involves online capturing of relevant operating data from the plant to be used as input for the simulation. This has the advantage of being automatic and not subject to human errors or manipulation.

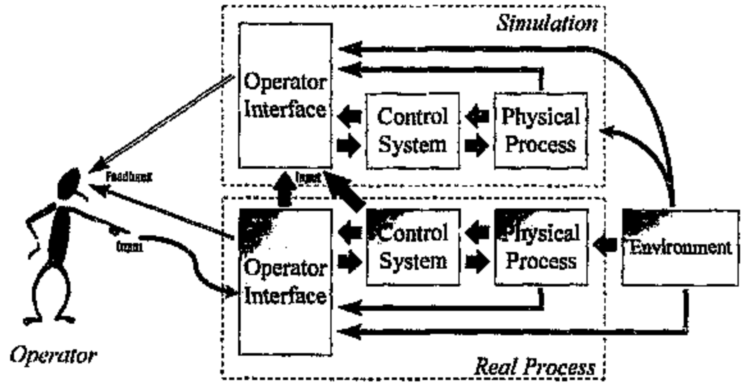


Figure 2-2: Online Simulation for Plant Verification - Automatic Data Input

2.2 Advantages of Power Plant Simulation

The simulation of a generating unit brings together the combined experience and knowledge of every separate engineering field in such a way that an individual or group has access to this pool of information. A well developed model reduces to some extent the need for an engineer to fully understand the entire boiler and turbine before assessing the overall effect of any plant modifications. A simulation also allows testing of adverse operating conditions, control modifications or plant modifications without the risk of damage to the plant.

Another benefit of generating unit simulation is the possibility of reduced downtime, which results in a decrease in lost revenue. Modifications are usually performed on a unit during periods of routine maintenance. Although tested in theory and passed by all the engineers and operators, there is always the possibility that these modification will produce unexpected effects. This could result in a unit "trip" and subsequent downtime, or even possibly damage to plant. Depending on the complexity and scope of the simulation, it may be possible to detect these problems during simulation before they manifest themselves on live plant.

Another area where a great deal of benefit could be gained from a model of the generating unit, is during startups of the boiler and turbine. Although there are countless problems that may be encountered during a unit startup, a particular problem that has occurred is the over-pressurization of the boiler. This is extremely detrimental to the long-term health of the plant, and is thus part of the boiler trip circuit. Although damage to the plant is averted by tripping the unit, this still results in a large amount of lost revenue as well as the extra cost and risk of another startup.

In view of the great technical and economical importance in the power industry, the dynamics of boiler-turbine processes have been the subject of considerable research work since the late fifties. Pioneers in this field include Chien et al.[4] and Profos[5]. These early attempts at boiler-turbine simulation were further complicated by the lack of digital computers. The available analogue computers were awkward to use, and even more difficult to "program". A further problem was the high cost of these analogue computers, which restricted their use to a small number of institutions and large companies.

With the availability of digital computers from the mid 1960's, the task of simulation became far easier. The further exponential increase in processing power of these computers also saw a rapid increase in the complexity of the models. The later advent of the personal computer as well as advanced modeling software has reduced the costs and expertise required to embark on a simulation of even complex systems.

There is an abundant amount of literature dealing with the simulation of a boiler-turbine system using complex thermodynamics and mathematics to describe the system characteristics. One example of this approach is taken by Dymek[6] where the heat exchange surfaces are broken down into 1-dimensional cells which are modeled according to compressible, adiabatic and lossy flow. The mathematical description of the change of state of the fluid flow process depending on the displacement x and the time y is given by the laws of conservation of energy, mass and momentum. This model proceeds to approximate the working fluid and tube wall to produce a simplified heat exchange model.

Numerous books are also available on this subject, providing extensive information on all aspects of boiler-turbine simulation. For example, Knowles [7] investigates the mathematical equations used to model the boiler-turbine, while Dukelow [8] provides an in-depth discussion of the boiler controls for various types of boilers.

Extensive research has revealed that the majority of research into power plant simulation only attempts to model the mechanical systems without including the control systems. Goeminnie[9], Schuck[10], Cheres[11] (drum boiler), De Mello[12] (drum boiler), Coulman[13] and Kwan[14] use complex mathematics to model the physical boiler and turbine without including the control systems. Beato[15] takes a more simplistic approach to the simulation of the physical systems, as well as simulating the main control loops. Klefenz[16], on the other hand, breaks down the boiler, turbine and control system into simulated blocks in an attempt to describe the control aspects of the boiler and turbine.

A possible reason for the lack of "complete" generating unit simulations in the sixties and seventies was the high cost versus performance ratio of the available computers. With the advent of the PC and its subsequent large increases in performance, the number of "complete" simulations has been seen to increase.

The major constraints in a simulation, namely stability, accuracy and simulation time, are overcome to a certain extent by the use of more powerful computer hardware. These issues, together with implementation problems specific to power stations, are discussed in the next section.

2.3 Simulation Implementation Issues

Three major issues that must constantly be considered when designing a simulation are those of stability, accuracy and simulation time. These phenomenon depend largely on the following

- model complexity
- required simulation speed (faster than real-time, real-time or slower)
- dynamics of the simulated plant/process

Depending on the objectives of a simulation, one or more of these problems may be alleviated to a certain extent by careful design. Knowledge models (Section 2.1.1) are not usually required to run in real time, and are thus affected less by the problems of stability and accuracy. In this case simulation speed is compromised for increased stability and accuracy.

When deciding on the most suitable simulation software, these three issues play an important part when deciding between packages. Refer to Appendix B for a comparison of the available software packages.

These issues have also been investigated thoroughly in many publications. Hartley[17] states "Often, trade-offs must be made because of the conflicting requirements of accuracy, stability, and computation time."

2.3.1 Stability

Hartley [18] suggests that "A simulation becomes unstable when the numerical solution of the simulation does not converge to the exact solution, but produces an error which is unbounded.". It can, however, be expected that the simulation of an unstable system would also produce an unstable simulation.
The stability of a digital simulation is very dependent on:

- the integration method
- the characteristics/dynamics of the simulated system/plant
- the simulation time-step (time between iterations)

During the preparation of the simulation for this project, it was found that the relationship between each of these parameters was very critical. Due to the increasing complexity of the simulation, it became necessary to increase the length of the time-step in order to enable the simulation to operate in real-time rather that at a slower pace. It was found that the simulation became unstable if the time-step was increased above a certain value. One solution to this problem is to change the integration method as well as the time-step until the required stable real-time simulation is achieved. This problem was later alleviated slightly by the upgrade of the simulation software to a new 32 bit version which proved to be about 30% faster.

2.3.2 Accuracy

The accuracy of a simulation may be assessed in two ways, the first being an assessment of how closely the simulated results compare with the physical system being modeled. The second measure of accuracy relates to how closely the numerical results match the values that would be obtained if the system equations were solved exactly. The major sources of inaccuracy in a digital simulation are truncation errors, integration approximation errors, sampling errors and rounding off errors. The accuracy is influenced by, among other things, the integration method and the time-step.

Hartley[19] also states that "Higher accuracy algorithms also generally require a smaller simulation step size to maintain stability. This also conflicts with computation time needs."

2.3.3 Simulation Time

The simulation time is directly related to the complexity of the integration method as well as the simulation time-step.

This project's simulation is required to run in real time, or faster, for it to be useful. The simulation was originally designed to run at 28 times faster than real time. This would allow a complete 8 hour boiler startup to be simulated in just over 16 minutes. This was never achieved due to the increased complexity of the model, and has now been decreased to 10 times real-time.

2.4 Integration Methods

Numerous integration methods have been developed for digital simulation of an analog integrator. Each method has its particular niche where it is ideally suited to the task at hand. A description of the integration methods may be found in many simulation publications, including Hartley[20], Schwarzenbach[21], Matko[22]

Each method has a tradeoff in terms of either stability, accuracy or computation time.

The algorithms available in VisSim are:

Algorithm	$\int_0^t error^2 .dt$	Simulation Time (s)
Euler	24.08	9
Trapezoidal	14.27	9
Runge Kutta 2nd Order	2.47	9
Runge Kutta 4th Order	0.83	9
Adaptive Runge Kutta 5th Order	0.0444	16
Adaptive Bulirsh-Stoer	0.0041	13
Backward Euler (Stiff)	65	12

Table 2-1: VisSim Integration Methods

Table 2-1 shows the results of a series of tests performed using VisSim in order to asses the accuracy and computation time for each of the simulation algorithms. The test simulation consists of two sine waves, one with a frequency ten times quicker than the other. These waves are both integrated (the integral of a sine wave results in a cosine wave), and the results compared with cosine waves of the same frequencies. The error signals for both frequencies are then added, squared and then integrated for the duration of the simulation. The real time duration of the simulation was set to 200 seconds, the time step set to 0,03 seconds, while the two frequencies were 5Hz and 0,5Hz. The simulation was then run once for each integration method, and the simulation time recorded.

The ratios of simulation time for each algorithm were found to change greatly with an increase in the complexity of the simulation, so much so that the last four algorithms were found to be far too slow for the simulation developed during this dissertation. Although the Runge Kutta 4th Order algorithm appears to be the best choice, the Runge Kutta 2nd Order algorithm was found to run quicker while still producing a stable simulation. This was chosen as the method to be used for this simulation.

2.4.1 Stiff Systems

A stiff system is one that has dynamics operating on two different time scales. A power plant consists of extremely fast systems (e.g. the governor valves - 2 seconds) as well as extremely slow systems (fuel demand - up to 200 seconds). This is confirmed by Rossiter[23] who states that "The presence of some fast and slow dynamics implies that the boiler-turbine system is in essence a stiff problem.".

The problem with the simulation of stiff systems is that, to ensure a stable simulation, the simulation time-step is determined by the response time of the fastest system while the time necessary to observe the full response is determined by the slowest system.. The effect of this is that the simulation speed must be decreased in order to ensure a stable response. In the case of this simulation the time step needs to be below 0,008 seconds for stable operation.

2.5 Importance of Simulation Objectives

Knowles[24] argues that "In any given problem, modeling has an objective, and the effective modeller will create a model capable of providing reliable results in the context of the engineering objective." That is, a model may only be evaluated in terms of the engineering objectives set for the model. This is especially true in large, highly complex systems where very specific models are created to simulate specific phenomenon. Such a model may prove extremely accurate in predicting one phenomenon, while being inaccurate in terms of other simulated variables.

Numerous models may exist for any one system, each tailored to achieve a particular objective. In the case of a motor vehicle, for example, one simulation may have the objective of simulating the drag coefficient based on body shape, temperature, wind velocity and vehicle speed. Another simulation on the identical car body may be used to determine noise levels inside the car at various speeds, and with different body shapes. Both models would differ due to the different objectives.

Knowles[24] also states that "A model has little value in an engineering environment unless the precise objective(s) of that model are clearly stated and understood!" The advantage of this is twofold: Firstly, it is important that the objectives be set before design begins as this focuses the design process on the specific goals rather than proceeding in a "lets see what we can achieve" type of way. Secondly, a thorough statement of the objectives is very useful to anyone else using the model. A knowledgeable user may be able to gain a lot of insight into the workings of the model as well as its strengths and weaknesses from the design objectives, without requiring any further explanations or information.

CHAPTER 3

3. Boiler and Turbine Autocontrols

This chapter briefly describes the operation of the autocontrols at Duvha Power Station. The simulation of these controls is discussed in the following chapter (Section 4.6). For further information on Duvha Power Station refer to Appendix A.

In order to create a model of the physical boiler and turbine with the relevant autocontrols, it was necessary to develop an understanding of the controls. The reasons for this are:

- To enable simplification where and when necessary
- To determine which controls may be approximated or omitted entirely
- To allow tuning of the control parameters to approximate those found on the simulated plant

Although the autocontrols consist of a large number of components only the main control loops were analyzed and implemented in this dissertation.

The boiler and turbine controls at Duvha Power Station are extremely complex, consisting of numerous control loops inter-linked to form single-loop feedback, cascade feedback and feedforward control configurations. Another degree of complexity is added by the number of different operating modes that exist. The following distinct modes of operation (described in Section 3.2.2 and 3.3) have been implemented in the simulation:

- Fixed Pressure
- Sliding Pressure
- Valve Position
- Boiler Follow
- Turbine Follow
- Coordinated
- Pressure Control On
- Fuel Control Auto / Manual

3.1 Capability Computation

The main function of the unit capability controls are to determine the maximum load capability of the generating unit based on the state of certain auxiliaries. An example of this is the limitation of unit load to 51% if one of the two air heaters fail. It also determines the ramp rates, startup setpoints and makes the necessary adjustments to the unit setpoint to ensure safe operation within the acceptable limits of plant stress.

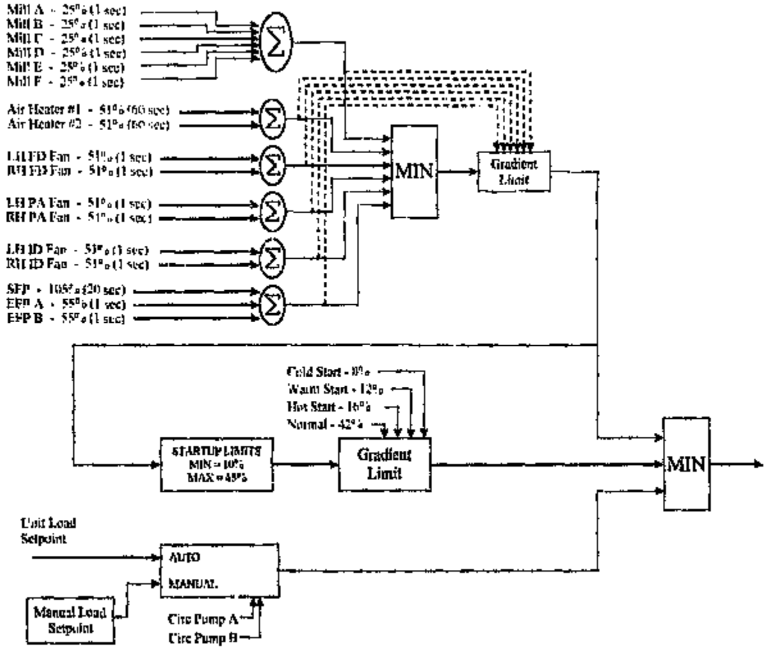


Figure 3-1: Boiler Capability Control

The auxiliaries are shown with their percentage load effect as well as a time delay. The time delay is important to ensure that any signal spikes are not passed on to the capability limitation. On some auxiliaries the time delay allows sufficient time for the plant to be restarted after a trip before the controls react by decreasing the capability. In the case of the airheater, one of the heaters must be off for a continuous period of 1 minute before the capabilities are reduced to 51%. The reason for this reduction in capability is that the generating unit is not able to operate above 51% load without both airheaters in service.

During startup the boiler demand is limited to between 10% and 45%, while the rate of change is limited according to the type of startup. The rate-of-change limits are quoted as a percentage of the maximum possible rate of change achievable by the control system.

In the event that both circulating pumps are not operating, the controls ensure that boiler load does not drop below 45%.

3.2 HP Bypass Controls

3.2.1 Safety Function and Ramp Rate Limitation

The HP bypass valves operate as safety valves by tripping open in the event of excessive boiler pressure. The HP bypass valves also operate when the rate of change of the boiler pressure exceeds a setpoint. The rate of change is prevented from exceeding the setpoint by opening the HP bypass valves.

These safety functions are active at all times, and may override the active control mode if necessary.

3.2.2 Pressure Control Modes

The HP bypass valves provide pressure control in three modes:

3.2.2.1 Valve Position Mode

The controller receives a signal (which represents the desired valve opening position) from the control room control desk. It compares this signal to the actual valve position feedback and adjusts the valve position accordingly.

3.2.2.2 Fixed and Sliding Pressure Modes

When sliding pressure mode is active, the controller continually adjusts the pressure setpoint to that of the actual pressure. This ensures that the HP bypass valves remain stationary except during ramp rate limitation described in Section 3.2.1.

When the controller is changed to fixed pressure mode, the pressure setpoint stops tracking the actual pressure. This results in the actual pressure being controlled at the setpoint by the action of the HP bypass valves. When in fixed pressure mode it is possible for the operator to manually increase or decrease the fixed pressure setpoint.

3.3 Fuel Master Controller

3.3.1 Pressure Setpoint Formation

The purpose of this control loop is to determine the pressure setpoint at which the boiler must be controlled, according to the boiler conditions, operating modes and operator interaction.

The control philosophy depends on the boiler operating mode as described briefly below. A more detailed description may be found in literature by Dukelow[25].

3.3.1.1 Boiler Follow

During boiler follow mode, boiler loading is set according to the turbine steam demand. The effect of this is that the boiler load (steam generated) follows the turbine load (MW). This mode is normally enabled when problems occur on the turbine.

In this mode, fuel demand is calculated as follows:

$$\text{SteamDemand} = \frac{\text{Pressure}_{\text{Setpoint}}}{\text{Pressure}_{\text{Actual}}} \times \text{Flow}_{\text{Steam}} \qquad \text{Eq 1}$$

3.3.1.2 Turbine Follow

During turbine follow mode, the turbine governor valves operate to control the boiler pressure at the setpoint. The effect of this is that the turbine load (MW generated) follows the boiler load (steam production). This mode is useful when problems are encountered on the boiler.

3.3.1.3 Coordinated Mode

During coordinated control mode the boiler pressure setpoint is set manually by the operator or remotely by the electricity grid load dispatch center. The load of both the boiler and turbine is coordinated according to this setpoint as well as the unit capability.

3.3.2 Fuel Master Control & Load Control

The main function of the fuel master controller is to regulate the boiler main steam pressure according to the operating mode (see below) as well as the boiler setpoint from the Pressure Setpoint Controller. This is achieved by controlling the fuel flow to the furnace.

This controller provides both a feed-forward path as well as a feedback path, both of which are discussed below.

3.3.2.1 Pressure Control OFF (Feed Forward Control)

With pressure control disabled, the fuel demand is derived from the load demand signal (i.e. fuel demand is proportional to load). No feedback signal is used to fine tune the fuel demand setpoint in order to achieve the correct pressure.

3.3.2.2 Pressure Control ON (Feed Forward plus PID Control)

Any deviation of the actual pressure from the pressure setpoint is fed into a load dependent PID controller, which adjusts the fuel demand accordingly. The two main inputs to this PID controller are the boiler load demand and pressure deviation signals, both from the Pressure Setpoint Formation controller.

The pressure deviation is fed into a PID controller, the P and I being load dependent. The fuel factor is determined by comparing the output of the PID controller with the boiler demand input. This fuel factor, which is a measure of the quality of the coal, is used as a multiplication factor on the demand signal to each coal feeder.

Under ideal design conditions, the feed-forward loop would be exact in determining the optimum amount of coal required to maintain the pressure setpoint at each load level. This is not, however, the case as the PID controller is constantly required to fine-tune the fuel demand. Any number of normal and abnormal plant conditions may affect this PID controller, including the use of bled steam for feed-heating, or an increase in the fouling of the boiler pipework.

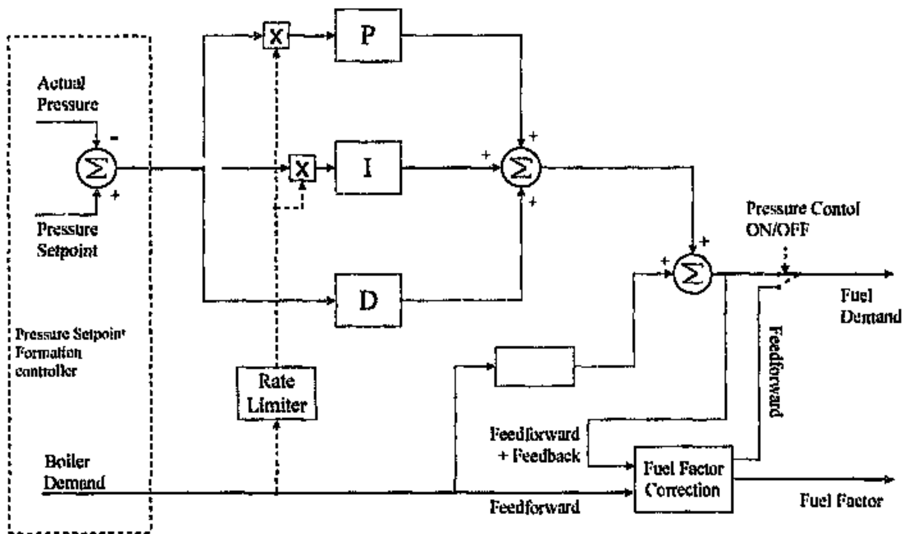


Figure 3-2: Fuel Demand Controller

3.4 LP Bypass Controls

The main function of the LP bypass valve is to control the hot reheat steam pressure within safe limits.

A simulated hot reheat pressure is derived from the HP loop pressure (multiplied by a gain factor). This simulated pressure is used as a reference due to the fact that it is always the same as the actual hot reheat pressure during normal turbine operation (with HP and IP bypass valves closed) at any given load and will not vary with the operation of the HP and LP bypass valves as will the actual hot reheat pressure.

The unit operator is able to set the hot reheat pressure setpoint to any value over the complete range of the reheat pressure. The controller takes the maximum between this setpoint and the simulated hot reheat pressure, and uses this as the control setpoint. This ensures that the operator is only able to limit the hot reheat pressure to a value higher than that which would be experienced under normal circumstances.

3.5 Combustion Control

The combustion controller is primarily responsible for ensuring, among other things, the correct fuel to air ratio in the boiler. This is very important to ensure that the boiler operates at the optimum efficiency at all times. The importance of this ratio is discussed further in Section 5.6.1. The furnace pressure is also controlled to a setpoint slightly below atmosphere by the correct use of the forced draught (FD) fans and induced draught (ID) fans.

3.6 Superheater Outlet Temperature Control

Depending on the location of a heat exchanger it may be classified as either convective or radiative depending on its location inside the furnace . If the superheater bank is able to "see" the combustion flame, it is considered to be of the radiation type, whereas the superheaters which rely on the passing flue gas for heat exchange are of the convection type. Figure 3-3 below shows the opposite temperature effect of a change in steam flow for the two types of superheaters. The advantage of these opposite effects is that, if designed correctly with the convective and radiative superheaters in series, the temperature of the main boiler steam remains relatively load independent.

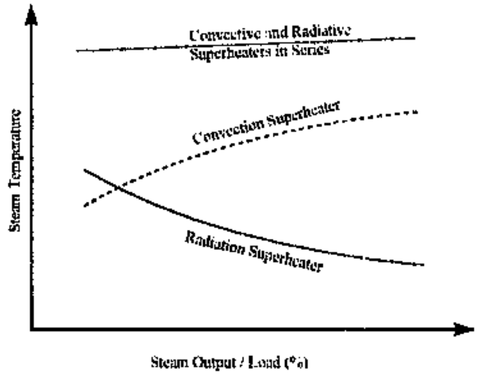


Figure 3-3: SH Typical Steam Temperature Characteristics

Even with optimal design there is a slight increase in temperature with an increase in boiler load. The superheater discharge temperature is controlled by means of 3 groups of attemporators, one situated before SH 1, the second before SH 3 and the last before SH 4.

The SH temperature controller is a cascade controller, configured as follows:

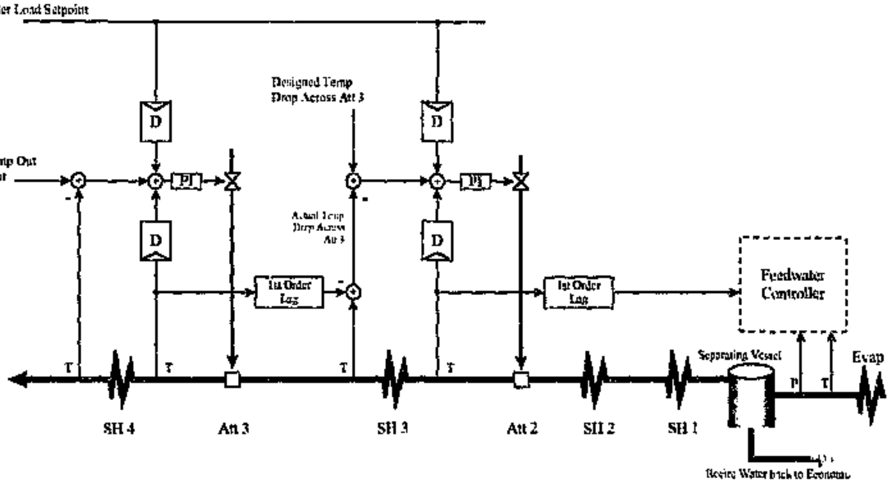


Figure 3-4: Superheater Temperature Controls

The primary controlled variable is the SH 4 outlet temperature. Any deviation from this setpoint results in a change in attemporation at Att 3. The derivatives of both boiler load and SH4 inlet temperature are also used as deviation signals to the Att 3 PI controller. This serves as a feed-forward signal for load changes or changes in SH 4 inlet temperature. The deviation signal for Att 2 PI controller is calculated as follows:

$$\text{Deviation}_{\text{Att2PI}} = \text{Setpoint}_{\text{Att3TempDrop}} - \text{Actual}_{\text{Att3TempDrop}} \qquad \text{Eq 2}$$

As with the att 3 controller, the load SH3 Inlet temperature are fed forward via derivative blocks.

3.7 Feedwater Master Control

The feedwater master controller is primarily responsible for the control of the evaporator discharge enthalpy at the required setpoint. This is achieved by regulating the feedwater flow through the evaporator by means of the boiler feedwater pumps. A deviation signal from the superheater temperature controller also forms part of the control signal to the boiler feedwater pumps. The feedwater controller thus also contributes slightly to the control of the SH temperatures.

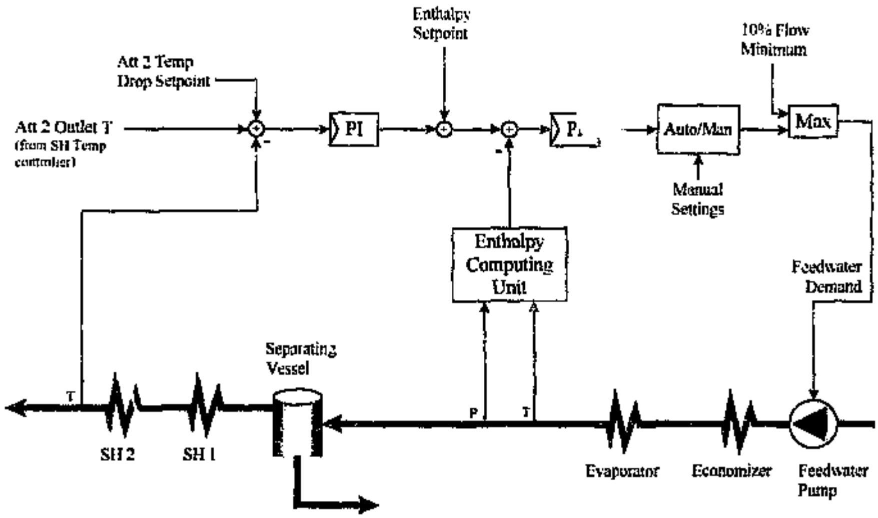


Figure 3-5: Feedwater Master Controller

3.8 Turbine Governor Control

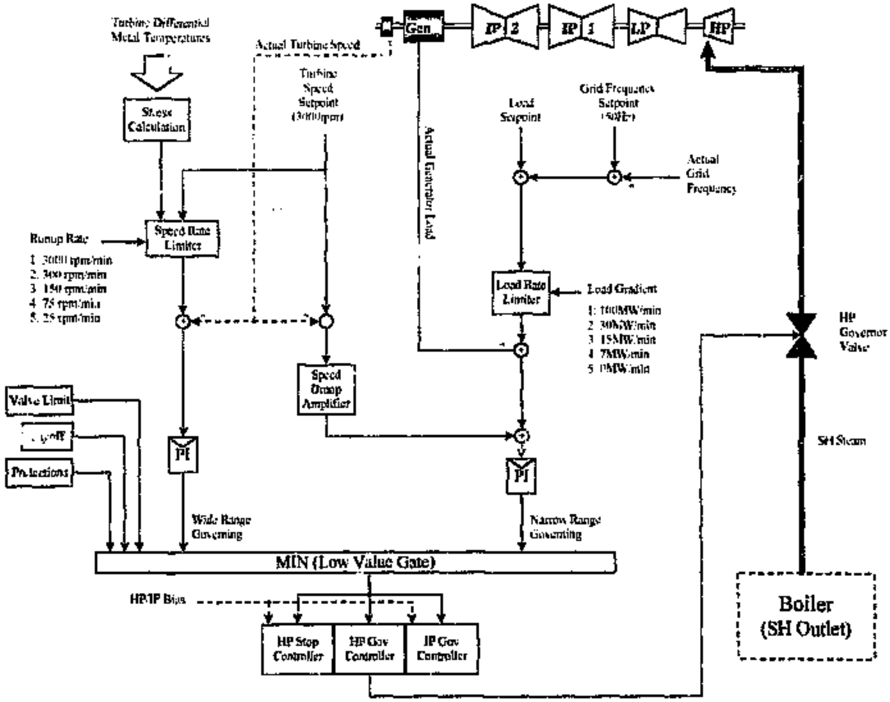


Figure 3-6: Governor Controls

With reference to **Figure 3-6** the HP/IP Governor and HP Stop valve controls consist of 5 separate signals, all of which pass into the "low value gate" which selects the lowest of the signals.

3.8.1 Turbine Protections

This signal (normally high) goes low in the event of a turbine protection trip, or when the "Turbine Trip" button is pressed. The turbine and boiler trips have not been included in the simulation as they are not part of the normal operation of the unit. It would be a relatively simple matter to include in the simulation the action of these protection circuits.

3.8.2 Governor Valve Limit

This setting on the control desk limits the opening of the governor valves to a maximum value. Under normal circumstances this would be set at 100%.

3.8.3 Wide and Narrow Range Governor

Once the steam conditions are correct the operator starts the process of running the turbine up from standstill to a speed of 3000 rpm. This speed setpoint, which determines the output frequency of the generator, is never changed. Initially the turbine run-up controls are dominated by the wide range governing signal which limits the run-up rate depending on calculations on plant stress levels. The run-up is frequently halted a extended periods to allow turbine stresses to decrease to acceptable levels. These stresses are calculated by comparing temperatures throughout the inner and outer turbine casings.

Once the turbine speed reaches 2800 rpm, the controls switch to narrow range governing. The narrow range governor PI controller is slower than that of the wide range governor, thus allowing more precise speed control. This PI controller is affected by the following process variables:

- speed deviation (= speed setpoint - actual turbine speed)
- load deviation (= load setpoint - actual turbine load)
- grid frequency deviation (= grid frequency setpoint - actual grid frequency)

The grid frequency correction ensures that each separate power generating unit connected to the national electricity grid plays a role in controlling the grid frequency. This is an important function of the narrow range governing circuit as the frequency of the electricity grid must remain at 50 Hz ± 1%. This circuit is responsible for the control of the generator load by regulating the steam flow through the turbine.

3.8.4 Unloading (Payoff) Control

This signal causes unloading of the turbine in the event of one of the following:

- Low Stator Water Flow
- High Condenser Pressure (i.e. Low Vacuum)
- Low Boiler Main Steam Pressure

CHAPTER 4

4. The Simulation

4.1 Introduction

One of the main objectives of this dissertation is to limit the amount and complexity of the mathematics used in the simulation in order to make it relatively easy to understand. In order to achieve this it was decided to use a non-mathematical approach wherever possible. This is especially true of the field of thermodynamics which is, in itself, an extremely complex subject heavily populated by complicated mathematical formulae. A graphical simulation language is ideal for this purpose while also having the advantage of being self explanatory and easily changeable.

In order to illustrate this concept, consider the simple example of a tank which must be simulated in terms of inlet flow, outlet flow and water level (Figure 4-1).

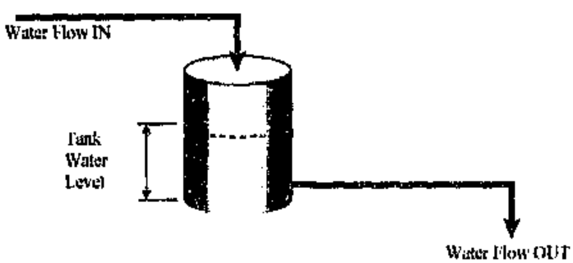


Figure 4-1: Water Tank

The normal approach would be to construct the following equations to describe the water level in terms of inlet and outlet flows.

$$\text{Level}_{\text{Water}} = \text{Area}_{\text{TankSurface}} \times \int \left(\text{Flow}_{\text{IN}} - \text{Flow}_{\text{OUT}} \right) \cdot dt \qquad \text{Eq 3}$$

$$\text{Area}_{\text{TankSurface}} = \Pi \times \text{Radius}^2 \qquad \text{Eq 4}$$

$$\text{Flow}_{\text{OUT}} = k \times \text{Level}_{\text{Water}} \qquad \text{Eq 5}$$

Combining these equations, and eliminating Flow_{OUT}, gives

$$\text{Level}_{\text{Water}} = \Pi \times \text{Radius}^2 \times \int \left(\text{Flow}_{\text{IN}} - k \times \text{Level}_{\text{Water}} \right) \cdot dt \qquad \text{Eq 6}$$

As can be seen in equation 6, the water level is evaluated as a function of itself. Although this is perfectly understandable to the average engineer, it is difficult to interpret the exact function of such an equation without sufficient explanation. Figure 4-2 is the graphical representation of this same model. This model is far easier to interpret, even for the casual reader. In terms of Duvha Power Station, this graphical approach has the added advantage of being similar to the majority of the autocontrol drawings used on the station.

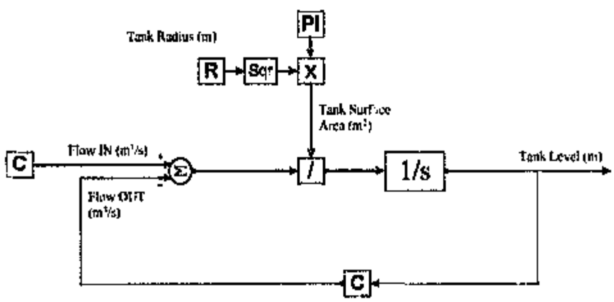


Figure 4-2: Graphical Simulation of Water Tank

This example fails to illustrate how complex a mathematical representation of the entire generating unit would be, mainly due to the large number of cross-couplings that occur between the different systems and control loops. The resulting equations would be extremely difficult to interpret and understand, thus limiting the use of such a simulation to a handful of experts.

4.2 Design Decisions

Before beginning with the simulation it is important to make certain preliminary design decisions in terms of the simulation approach in order to ensure that a single standard is used throughout.

Although the Siemens Teleperm C control system (used at Duvha for the majority of the autocontrols) generally uses a voltage range of 0-10v DC, the existence of certain modules operating outside of this range (e.g. -10 to +10v) makes it difficult to mimic these controls exactly while still using one standard throughout the simulation. It was decided that certain of the process and control signals would be unitized (i.e. scaled for the range 0-1).

In order to choose the optimum scaling for each measurement, the normal operating position as well as the absolute maximum possible values were considered.

The following scaling was used throughout the simulation:

Variable Type	Simulation Range	Plant Range	Normal Operating Point / Range
Water / Steam Flow:	0 → 1	0 → 700 kg/s	228 → 507 kg/s
Fuel Flow:	0 → 1	0 mills → 6 mills in service	3 → 5 mills
Power Generated:	0 → 1	0 → 700 MW	300 → 620 MW
Valve Position:	0 → 1	0 → 100% valve open	n/a
SH Attemporation Flow	0 → 1	0 → 80 kg/s	40 kg/s
RH Attemporation Flow	0 → 1	0 → 24 kg/s	12 kg/s
Steam Temperatures	0 → 700 °C	0 → 700 °C	up to 540 °C
Furnace Temperatures	0 → 1700°C	0 → 1700°C	900 →1300 °C
Enthalpy	0 → 4000 kJ/kg	0 → 4000 kJ/kg	100→ 3550 kJ/kg

Table 4-1: Simulation Variables

4.3 Methodology used for Simulation of the Physical Processes

Two separate models have been developed for the simulation of the boiler and turbines. The first model discussed is a highly simplified model of the boiler, and attempts only to approximate the superheater and reheater outlet pressures without any concern for steam temperatures and enthalpy. The second model, although developed initially for the purpose of testing the original model, is a much more realistic model which is based on physical properties of flow, enthalpy, pressure and temperature. Both physical models use exactly the same simulation for the autocontrols and user interface as illustrated in Figure 4-3.

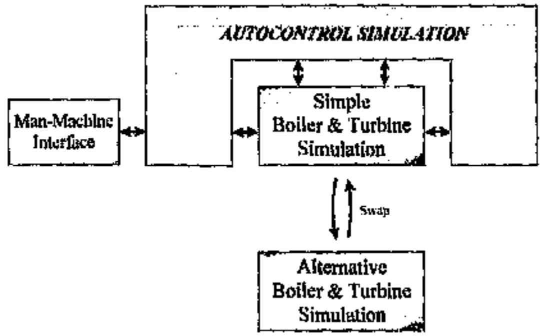


Figure 4-3: Re-use of Autocontrols and Simulation Interface

4.4 Simple Simulation of the Physical Boiler and Turbine

Figure 4-4 below illustrates the main path of energy starting with the raw coal in the bunkers and finally ending in the electrical output from the generator. The boiler/turbine unit has been divided into sections which will now be described in detail:

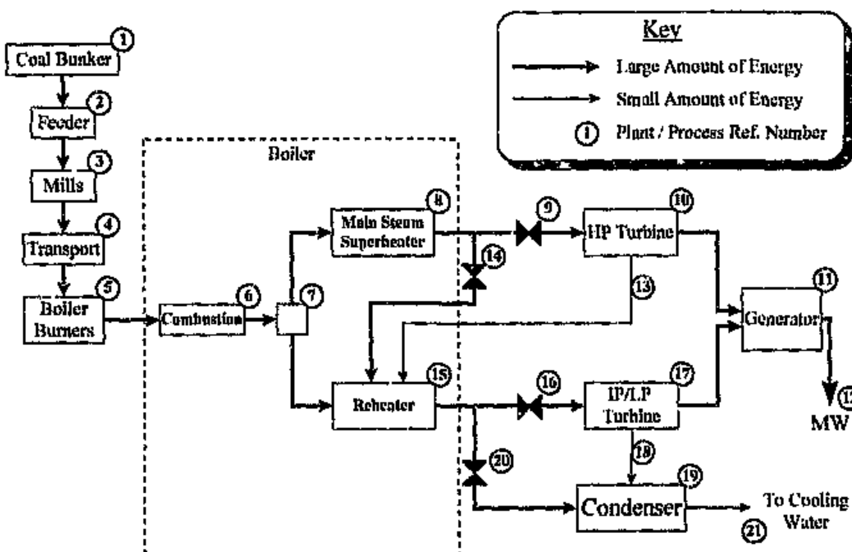


Figure 4-4: Overview of Energy Flow Through Boiler and Turbine

Plant / Process	Description	Section Reference Number
1.	Coal Bunker	4.4.1, page 28
2.	Feeder	4.4.2, page 28
3.	Mills	4.4.3, page 29
4.	Fuel Transport	4.4.4, page 29
5.	Boiler Burners	4.4.5, page 30
6.	Combustion	4.4.6, page 30
7.	SH and RH Heat (Energy) Exchange	4.4.7, page 30
8.	Main Steam Superheaters	-
9.	HP Turbine Governor Valves	-
10.	HP Turbine	-
11.	Generator	-
12.	Electrical Power to Grid	-
13.	Energy Content of HP Exhaust Steam	-
14.	HP Bypass Valves	-
15.	Reheaters	-
16.	IP / LP Governor Valves	-
17.	IP / LP Turbine	-
18.	Energy Content of LP Exhaust Steam	-
19.	Condenser	-
20.	IP / LP Bypass Valves	-
21.	Waste Energy Transferred to Cooling Water	-

Table 4-2: Key to Figure 4-4

4.4.1 Coal Bunker

Six coal bunkers act as an intermediate store for coal on its way to the mills. No delays have been associated with the bunkers which do not form part of the simulation. The only instance where a delay may occur would be in the event of either a blockage of the chute or if the bunker becomes empty. These two scenarios have not been accounted for in the simulation but may be approximated by stopping/tripping one of the mills.

4.4.2 Feeder

The coal falls from the bunkers directly onto six mechanical coal feeders. The rate of coal fed to the boiler is controlled by varying the drive speed of the feeders. An increase in the fuel demand does not directly increase the feeder speed, but increases the primary air (P.A.) flow through the mills. This increased flow is fed back to the controls which results in an increase in feeder speed setpoint to the feeders (see Figure 4-5).

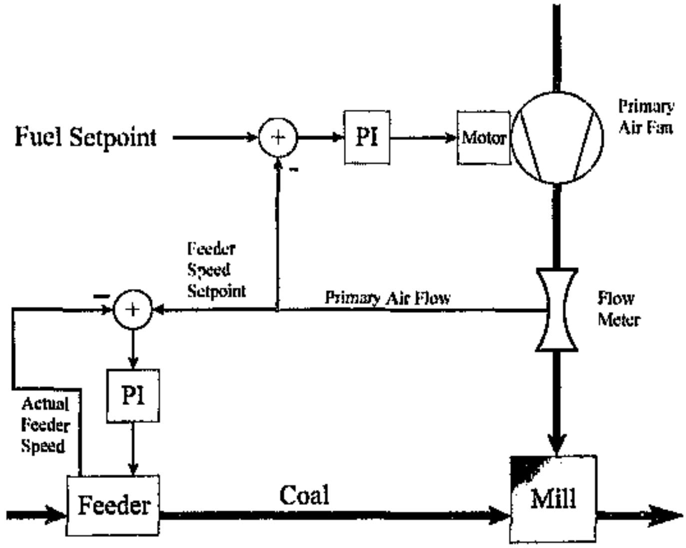


Figure 4-5: Feeder Speed Controller

Two PI controllers in series are used to approximate the response delay inherent in the feeder speed controls. The first simulates the response of the PA setpoint control system, and the second approximates the Feeder Speed controller. Other mechanical non-linearities in this system are the response of the primary air fan, the response of the flow meter and the response of the feeder speed to a step input. These mechanical non-linearities have been omitted from the simulation due to their relatively fast time constants.

4.4.3 Mills

The mill is a complex mechanical system on its own. Due to the minimal response times of the processes inside the mill, as well as the complexity of the milling process, this item is approximated with a first order delay element (I controller). This incorporates the control system response lag as well as the delays due to the crushing of the coal into pulverized fuel (P.F.). The issue of mill simulation is discussed in depth by Dolezal[26].

4.4.4 Fuel Transport

This phenomenon has been labeled 'transport' on the block diagram. This encompasses all fuel transport delays from the mills to the inside of the furnace via the boiler burners. The delay depends on the distance as well as the flow velocity of the pulverized fuel, which in turn depends on the boiler load. This load dependence has not been implemented in the simulation due to the fact that the primary air flows are not known as well as the fact that the time constants are relatively quick.

4.4.5 Boiler Burners

As the pulverized fuel and secondary air is blown through the burners into the furnace, the vanes inside the burners cause a swirling action which helps to ensure fast and complete combustion inside the furnace. In terms of this simulation, this swirling effect has been omitted completely due to its complexity. The only possible influence which could be determined from the burners, would be the influence of the swirl angle on the combustion rate. This calculation is far beyond the scope of this research, and has not been included in any other simulations that were reviewed.

4.4.6 Combustion

Combustion is a complex process involving thermodynamics, conservation of momentum and mass, radiation, and is dependent on particle size, composition, mass, temperature as well as the oxygen content and temperature of the air. This process is very fast compared to the overall time response of the boiler, and has thus also been approximated with a simple lag circuit (I controller). A detailed description of the complex process of combustion may be found in literature by Dolezal[27].

Although provision has been made in the simulation to model the influence of the oxygen content of the flue-gas as well as the energy in the secondary air, this feature has not been completed due to the lack of input data. The input energy (in secondary air) requires air temperatures and pressures (which cannot be calculated in the simulation) and simulation of the oxygen content of the flue gas is beyond the scope of this dissertation.

Dymek[28] treats the combustion chamber as a discrete number of adjoining 1-dimensional cells, each with its own properties. Mass, momentum and energy flows entering and leaving each cell are calculated, allowing for an accurate calculation of flows and temperatures throughout the combustion chamber. This approach is also extended to model the rest of the flue gas path through the superheaters, reheaters etc.

4.4.7 SH and RH Heat (Energy) Exchange

The first six areas described above were all modeled using a simple combination of first order lag circuits and pure time delays. This next section is more complex, and requires a detailed explanation.

On a live boiler, the energy liberated during the combustion process is not all transferred to the water/steam but is lost to the environment. The following summarizes the heat transfers that take place in a boiler:

- Heat transfer to the boiler tubes by means of radiation. This occurs mainly in the lower levels of the boiler, in and around the combustion chamber.
- Heat transfer to the boiler tubes by means of convection. This occurs throughout the flue-gas path
- Heat lost to the environment due to conduction through the boiler walls
- Heat lost to the environment in the flue-gas.

In a reheat boiler, a certain percentage of the energy is absorbed to produce superheated steam, while most of the rest is used to reheat the steam before the IP and LP turbines. During startup the main influence on these percentages is the sequence in which the burners are ignited. This sequence determines the height of the combustion area, which intern influences the proportion of energy absorption by the superheat and reheat pipework.

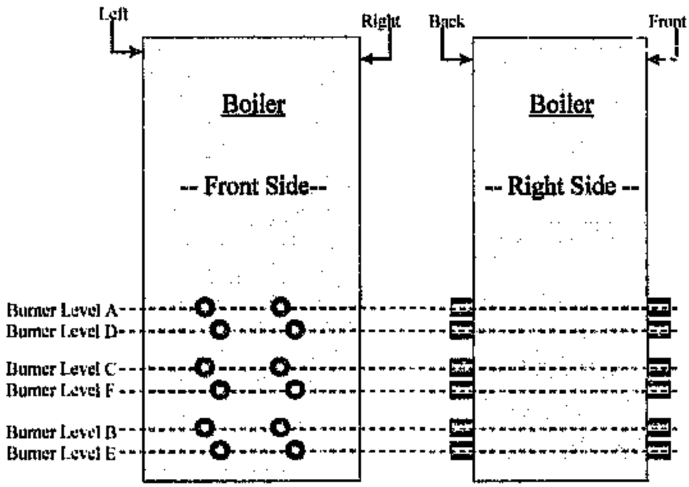


Figure 4-6: The Boiler Burners

The boiler model calculates the relative percentages by assigning a weighting to each ring of boiler burners, depending on their position (height). The relative weight of a particular row is equal to the percentage of energy that is absorbed by the SH, with the rest being absorbed by the RH. A value of 0.8 indicates 80% to the SH and 20% to the RH.

This is a simplification of the actual boiler heat exchange processes where the flame height affects the ratic of radiated heat absorbed by the evaporator and superheater. This is further described in section 4.5.2.2.

4.4.8 The Main Steam Pressure Model

Block number 8, labeled the "Main Steam Superheater" includes the economizer, the separating wall, the screwtubes, the evaporator, and superheaters 1,2,3 and 4.

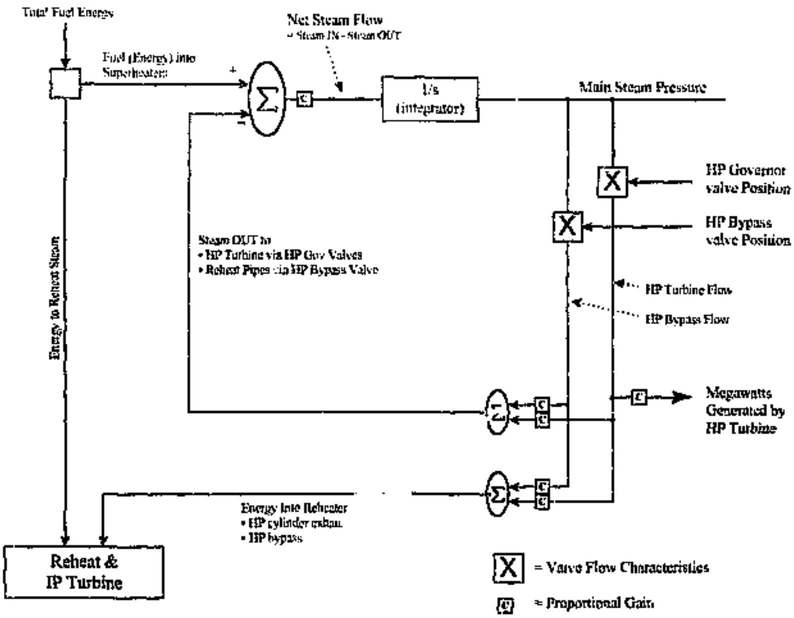


Figure 4-7: Main Steam Pressure Model

$$\text{Energy}_{\text{ToSteam}} = \text{Energy}_{\text{FromCombustion}} \times \text{Function}_{\text{AccumulationEffect}} \qquad \text{Eq 7}$$

Accumulation is a phenomenon which occurs as a result of the stored energy in the metal pipework of the boiler and in the steam. Modern boilers have a very high accumulation value due to the large surface areas of the heat exchangers, especially the economizer. This is also compounded by the necessity for thick walled tubes to withstand the additional pressures required of modern power stations.

In a once-through supercritical boiler rated at 900MW, a change from 66% to 100% load results in an increase in steam flow rate to meet the additional steam demand. This increased flow rate, coupled with the relatively slow response of the furnace, leads to a temporary decrease in the steam temperatures. This effect is counteracted by a heat exchange from the high temperature metal pipework to the cooler steam. The amount of heat that is released from the pipework is, in this example, equivalent to the full heat output of the boiler for 30 seconds. This process is further described in the book "PROCESS DYNAMICS "by Dolezal[29].

The accumulation effect has not been modeled in this simple simulation due to the complexity of the phenomenon as well as the need for temperature simulation. The alternative simulation, described in the following section, takes this accumulation effect into account by approximating the heat exchange to a low order filter.

Referring back to Figure 4-7, the boiler has been modeled as a pressure vessel with steam entering the vessel as a result of the combustion process and steam leaving the vessel to flow through the turbine. This is the most significant simplification in this simulation.

$$\text{Pressure} = \int(\text{SteamFlow}_{IN} - \text{SteamFlow}_{OUT}).dt \qquad \text{Eq 8}$$

This simplification is also used by Renze[30] in a simplified boiler pressure model.

Then,

$$\text{SteamFlow}_{IN} = k \times \text{Energy}_{IN} \qquad \text{Eq 9}$$

$$\text{SteamFlow}_{ToRH} = \text{SteamFlow}_{HPTurbine} + \text{SteamFlow}_{HPBypass} \qquad \text{Eq 10}$$

The flow through the valves was calculated according to the equations for flow through an orifice. The non-linearity of valve area versus valve lift was implemented by means of a lookup table, and the flow calculated as follows:

$$\text{Flow}_{Valve} = \text{Area}_{Valve} \times \sqrt{P_{IN} - P_{OUT}} \qquad \text{Eq 11}$$

The HP Governor and HP turbine were treated as a single unit, consisting of a variable flow orifice (governor valve) leading to a fixed flow orifice (turbine):

An alternative flow calculation is suggested by Knowles[31] who uses the following simplified approximation to calculate the flow through a valve:

$$\text{MassFlow}_{Valve} \cong \text{Constant} \times \text{ValveLift} \times \text{Pressure}_{IN} \qquad \text{Eq 12}$$

This is similar to the equations for critical flow through a nozzle, which depends only on the upstream pressure and not the discharge pressure.

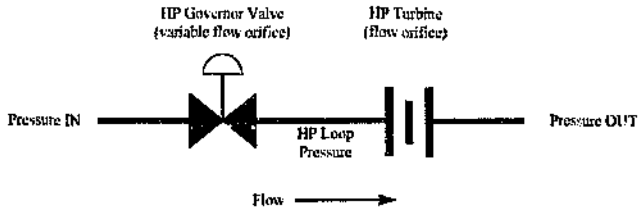


Figure 4-8: HP Governor and HP Turbine

4.4.8.1 HP Turbine Power Generation Theory

Throttling of the governor valves results in a decrease in flow through the turbine. Increased throttling also causes an increase in the energy (enthalpy) loss across the valve, leading to a decrease in the amount energy available to convert into rotational energy in the turbine. These two effects together result in decreased power generation from an increase in governor valve throttling. The exact details of this energy conversion process is covered in Section 4.5.1.1.

It has been shown through experimentation that the following linear relationship (Willans[32] Line) exists between mass flow rate and output power[33]

$$\mathrm{SteamFlow} = \left(\mathrm{SteamFlow}_{MCR} - \mathrm{SteamFlow}_{ZeroLoad}\right)\left(\frac{\mathrm{Power}}{\mathrm{Power}_{MCR}}\right) + \mathrm{SteamFlow}_{ZeroLoad}$$

Eq 13

where

MCR = Maximum Continuous Rating (MW)

4.4.8.2 Simulation of the HP Turbine

The simplified simulation does not take into account any enthalpy or entropy changes during throttling (governing), but calculates the power generated to be proportional to the mass flow rate.

$$\mathrm{Power} = \mathrm{Const} \times \mathrm{SteamFlow}_{MASS} \qquad \text{Eq 14}$$

This assumes a constant energy drop across the turbine per kg of steam.

This turbine-generator model is supported by Renze[30] who also makes use of similar simplifications in the implementation of a boiler-turbine model. Suzuki[34] also states that "In the system, generated electricity is approximately proportional to the steam flow to the turbine".

4.4.8.3 Sub-Critical vs Critical Flow

All calculations above as well as in the simulation have assumed sub-critical flow through the turbine governor/bypass valves and the turbines. The critical flow point is the pressure ratio P_{OUT} / P_{IN} at which the flow is a maximum for the particular area and inlet pressure. For superheated steam this ratio is 0.57. Any further reduction of outlet pressure has no effect on the flow. In the critical flow area, flow is not related to discharge pressure, and may be approximated by the equation[35]

$$\text{Flow} \approx \text{Constant} \times P_{IN} \qquad \text{Eq 15}$$

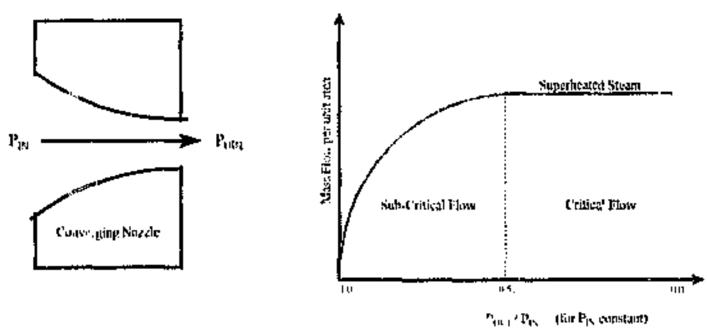


Figure 4-9: Critical (Choked) Flow Through a Nozzle

4.4.8.4 Pressure and Temperature Flow Correction

The flow equation used in the simulation (equation 15) is only accurate at a specific pressure and temperature. All other operating points should be corrected for pressure and temperature variations according to the formula:

$$\text{MassFlow}_{\text{Corr}} = \text{MassFlow}_{\text{Eq}} \times \sqrt{\frac{\text{Density}_{\text{Actual}}}{\text{Density}_{\text{Design}}}} \qquad \text{Eq 16}$$

Due to the fact that the simplified simulation does not calculate any temperatures or volumes, it is not possible to perform temperature and pressure correction.

Although not implemented in either simulation, this correction could be performed in the advanced simulation as enough data is available to allow calculation of the density from steam tables. It was decided to ignore this due to the additional computational time that would be required in order to calculate the corrected mass-flow.

4.4.8.5 HP Bypass Flow

The HP bypass valve flow is calculated in the same way as the HP governor valve. The flow through the bypass is not influenced in any way by the flow through the governor, and vice versa. This is a shortcoming of the simulation, and has been addressed in the alternative simulation presented in Section 4.5.

Both the HP bypass and the HP governor are able to carry 100% flow, up to the maximum design flow of 507kg/s. In the configuration as shown in Figure 4-10, the flow through each individual valve would, in practice, be related to the flow through the other valve. This relationship has been ignored in the initial simulation, but has been implemented in the more advance alternative simulation.

Analysis of the simple simulation shows that the flow through the governor is only a function of inlet pressure, discharge pressure and valve position. Opening the simulated bypass valve quickly will not have an initial effect on the inlet pressure, but will instead cause a gradual decline in boiler pressure, which would then lead to a reduced flow through the governor valve. On the real plant, however, opening the bypass valve has an immediate effect on the pressure directly upstream of the valves, which causes an immediate drop in flow through the governor valve.

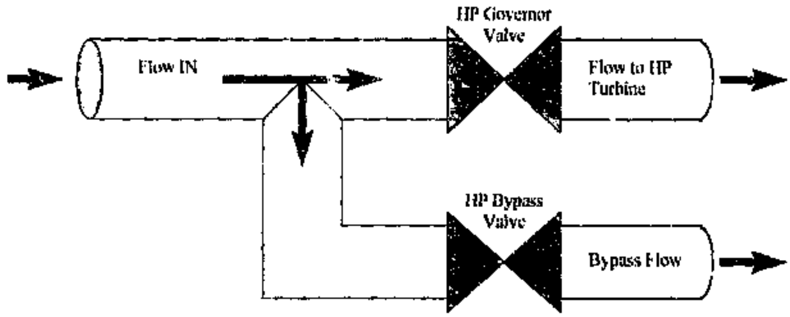


Figure 4-10: HP Governor and Bypass Valves

4.4.9 Reheat Steam Pressure Model

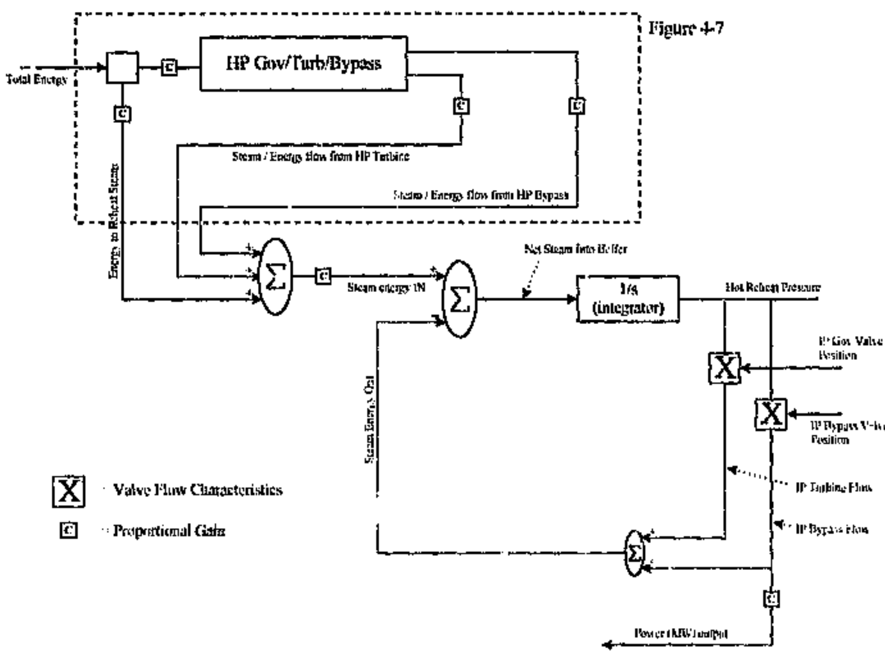


Figure 4-11: Reheat Steam Pressure Model

Three sources of energy exist for the enthalpy of the steam entering the IP/LP turbines. The first source is the boiler furnace where energy is transferred from the combustion process to the steam flowing through the pipes. Refer to Section 4.4.7 for a description of the simulation in terms of energy transfer from the combustion process to the superheaters and reheaters.

During expansion in the HP cylinder only about 30% of the available energy is extracted from the steam. In practice this residual energy is inversely proportional to the turbine load, due to the reduced enthalpy drop across the turbine at part loads.

Steam entering the cold reheat from the HP bypass also contains a large amount of energy. This energy is almost equal to the energy contained in the superheated steam. In practice, as the HP bypass valve closes, the entropy of the steam increases, which reduces the useable energy content in the steam. For the reason of simplification, the simulation neglects this reduction in (useful) energy and assumes a constant energy content per kg of bypass steam.

4.4.9.1 IP and LP Turbines

For simplification, this simulation treats the IP and LP turbines as a single unit

The IP turbine has been modeled in exactly the same way as the HP turbine, with the same simplifications as well as limitations (Refer to Section 4.4.8.2).

4.5 Alternative Physical Boiler and Turbine Simulation

4.5.1 Introduction

This chapter deals with the simulation of the physics of the power generating unit. The first sub-section describes the Rankine Cycle, which is a common cycle used to illustrate the thermodynamics of a power plant. The following sections describe the implementation of the more advanced power plant simulation taking which takes into account the basic thermodynamic processes within the boiler and turbine.

Although the implementation of the advanced simulation is completely different from the simple simulation, certain elements have been reused to save development time. The simulation of the fuel flow from coal bunker, through the milling process, until it reaches the boiler burners, has been reused and will thus not be discussed again. The model of the flow characteristics of the governor and bypass valves has also been based on the previous simple simulation.

4.5.1.1 The Rankine Cycle

In order to develop this alternative simulation it was necessary to understand the relationships between temperature, enthalpy and entropy during the water / steam cycle. This is best described by means of the following two diagrams.

The Temperature-Entropy (T-S) diagram below is commonly used to explain many different thermodynamic cycles, including the Rankine cycle.

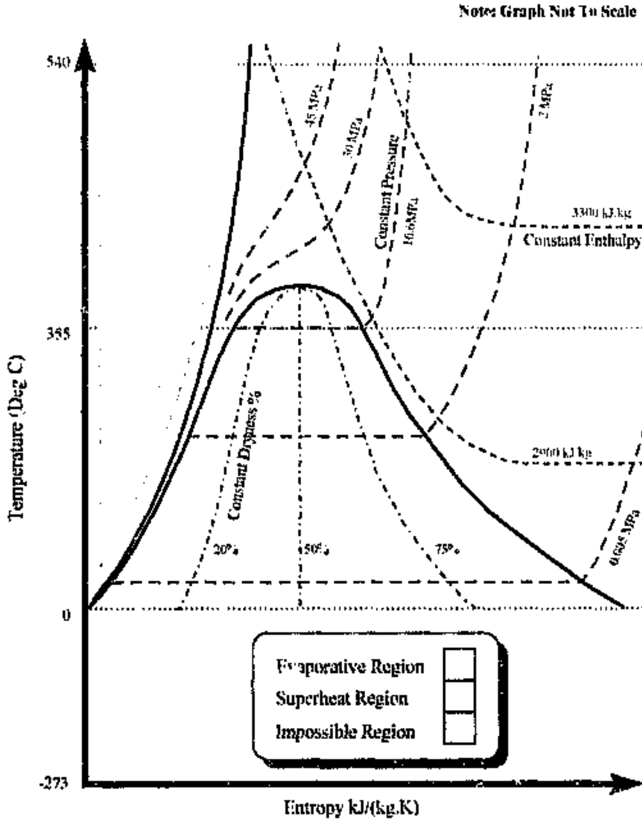


Figure 4-12: T-S Diagram for Water/Steam

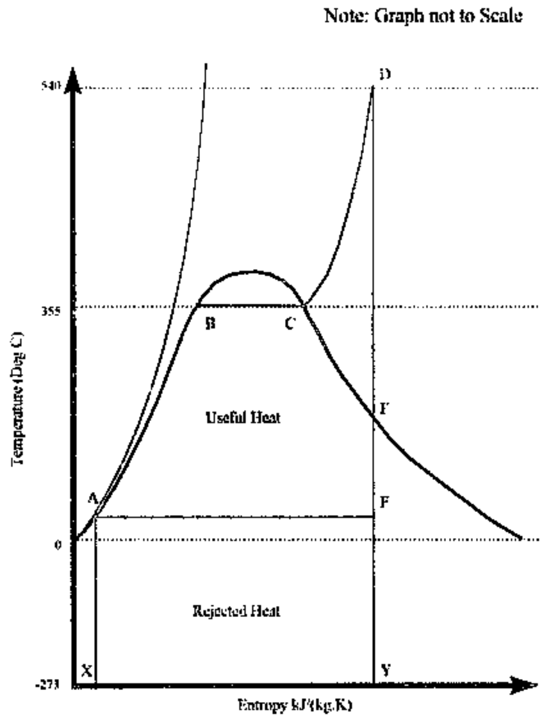


Figure 4-13: Ideal Steam Cycle

Figure 4-13 depicts the ideal steam cycle in the r[illegible]tion (Rankine Cycle).

Starting at point A, the condensate is pumped fr[illegible] condenser through the various low pressure heaters, feedwater pump, high pressure heaters and economizer until it reaches boiling point somewhere in the evaporator. The point where the water begins to evaporate is indicated at B. As further latent heat is added to the water more evaporation takes place, which increases the Entropy and enthalpy of the water/steam. The Section B→C, indicating the evaporation phase, occurs at constant pressure. Once the steam is 100% dry (completely evaporated) it enters the superheaters where it is heated to its final temperature (540 °C in this case) at constant pressure (almost) indicated by point D.

The line D→E→F indicates the work done in the turbine during the expansion of the steam. In the diagram the expansion is assumed to be frictionless and adiabatic, which results in no change to the entropy of the steam during the process. The final line F→A indicates the removal of heat in the condenser. This wasted heat is then released into the atmosphere by way of the cooling towers.

Point E is significant in that it indicates where in the expansion the steam becomes wet. The water droplets in wet steam cause excessive wear and damage to the blade tips, and should be minimized as much as possible.

The area A-B-C-D-E-F indicates the useful heat that is converted into rotational energy in the turbine, while area A-F-Y-X is heat/energy that is rejected to the condenser. It can be seen that decreasing the condenser pressure (and thus temperature) results in an increase in useful heat and a decrease in rejected heat, leading to an overall increase in thermodynamic efficiency.

The reheat boiler (as used on Duvha Power Station) was designed to reduce the damage caused by wet steam, as well as increase the efficiency of the cycle.

Note: Graph not to Scale

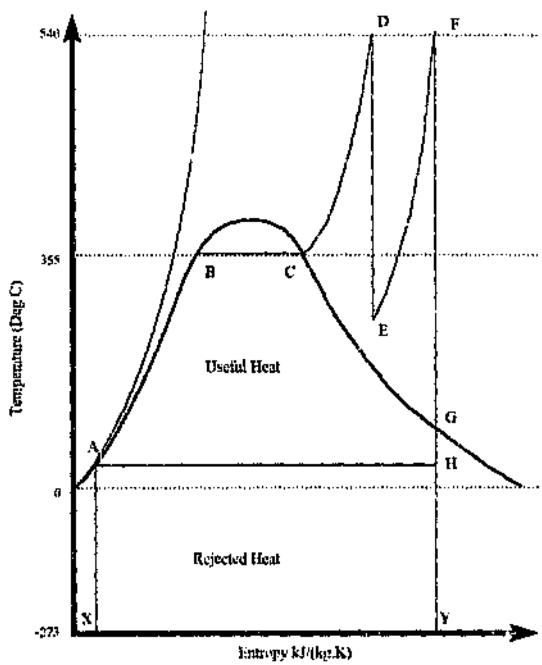


Figure 4-14: Water / Steam Cycle with Reheat

In a reheat turbine the steam passes from the HP turbine exhaust (point E) into the reheater which heats the steam up to point F at constant pressure. The line F→G→H indicates the adiabatic expansion in the IP and LP turbines. The line H→A indicates, once again, the unavailable heat which is removed in the condenser.

It is evident that in the reheat boiler the steam flowing through the last few stages of the turbine before the condenser is far dryer (approx. 90% dry) than the non-reheat (approx. 75%) operating at the same pressures and temperatures. The reheat turbine is also more efficient than a similar non-reheat turbine due to the increase in the ratio of useful heat to rejected heat.

4.5.2 *Lumped Elements*

This section describes the fundamental equations of the boiler. Although a single superheater consists of many thousands of parallel, it was decided to consider them lumped into a single equivalent tube displaying similar characteristics. This has been done for each of the heat exchangers. This process of lumped parameters is commonly used in the simulation field, especially in boiler simulation (Suzuki[36]) where it is mathematically impractical to simulate the exact tube layout inside the boiler.

4.5.2.1 Pressure and Flow

It was decided to model the various sections of the boiler pipework by breaking it down into equivalent elements (see Figure 4-15). This is similar to the approach taken in order to simulate changes in flue gas temperature as it passes through the various sections of the boiler (refer to Section 4.5.4.1) and is also the approach taken by De Mello[37], Schuck[38], and other well known authors in this field. The accuracy of this lumped parameter method is investigated in Appendix C.

Before embarking on the design of this lumped element block, it is important to determine what variables are known, and which are unknown. If one considers the boiler to include all pipework, turbines and valves from the boiler feedwater pump discharge to the condenser, then the following holds true:

Known:

- Condenser Pressure (this is controlled independently at a given pressure setpoint)
- Feedwater Pump Flow (the SH temp. controller controls this flow)

Unknown:

- Feedwater Pump Delivery Pressure
- Flow into Condenser

This means that, from an overall point of view, given the known variables it must be possible to calculate the feedpump pressure and condenser flow. In order to satisfy this requirement, the lumped element must have the same structure as the boiler as a whole. Thus, for each "fundamental block":

Known:

- Flow IN
- Pressure OUT

Unknown:

- Flow OUT
- Pressure IN

From these requirements, the block shown in Figure 4-15 was designed.

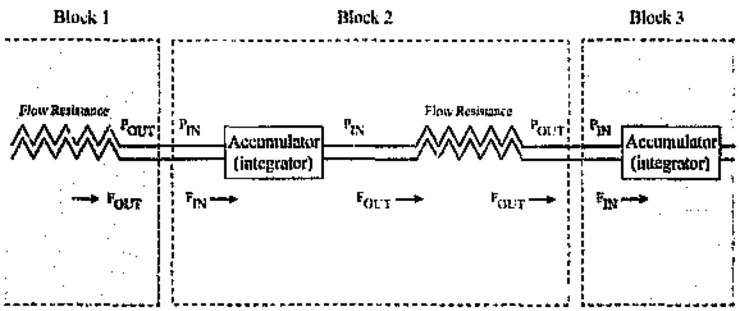


Figure 4-15: Lumped Parameters: Pressure and Flow

$$P_{IN} = k_1 \times \int \left(F_{IN} - F_{OUT}\right) \cdot dt \qquad \text{Eq 17}$$

$$F_{OUT} = k_2 \times \sqrt{P_{IN} - P_{OUT}} \qquad \text{Eq 18}$$

Where $k_1 = 1$ / Volume and $k_2 = 1$ / FlowResistanceCoefficient

Combining equations 17 and 18 (eliminating F_{OUT}) gives

$$P_{IN} = k_1 \times \int \left(F_{IN} - k_2 \times \sqrt{P_{IN} - P_{OUT}}\right) \cdot dt \qquad \text{Eq 19}$$

As can be seen P_{IN} is equated in terms of itself. This would normally make this equation difficult to solve. VisSim, however, is ideally suited to solving this equation using the following simulation block.

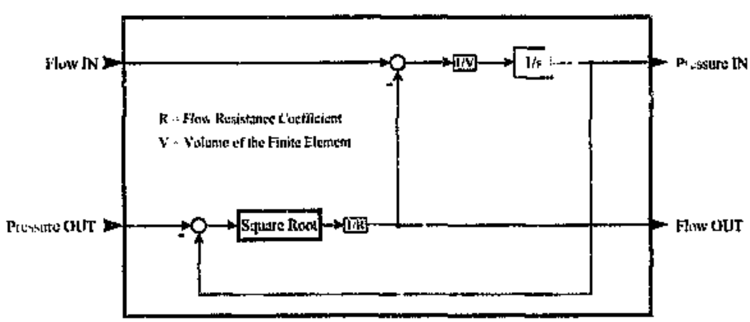


Figure 4-16: Implementation of Equation 19

The square root block in Figure 4-16 is implemented as show in Figure 4-17 in order to allow for a reverse flow through the element.

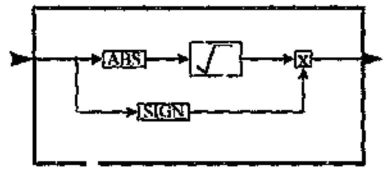


Figure 4-17: Square Root Block

Substituting Equation 17 into Equation 18 (eliminating P_{IN}) results in a slightly different equation:

$$F_{OUT} = k_1 \times \sqrt{\left[k_2 \times \int\left(F_{IN} - F_{OUT}\right)\cdot dt\right] - P_{OUT}} \qquad \text{Eq 20}$$

Although this yields almost identical results when implemented in VisSim, Equation 20 was found to simulate slower than Equation 19.

Using the lumped element block in Figure 4-16, the pressures and flows throughout the boiler are simulated as follows:

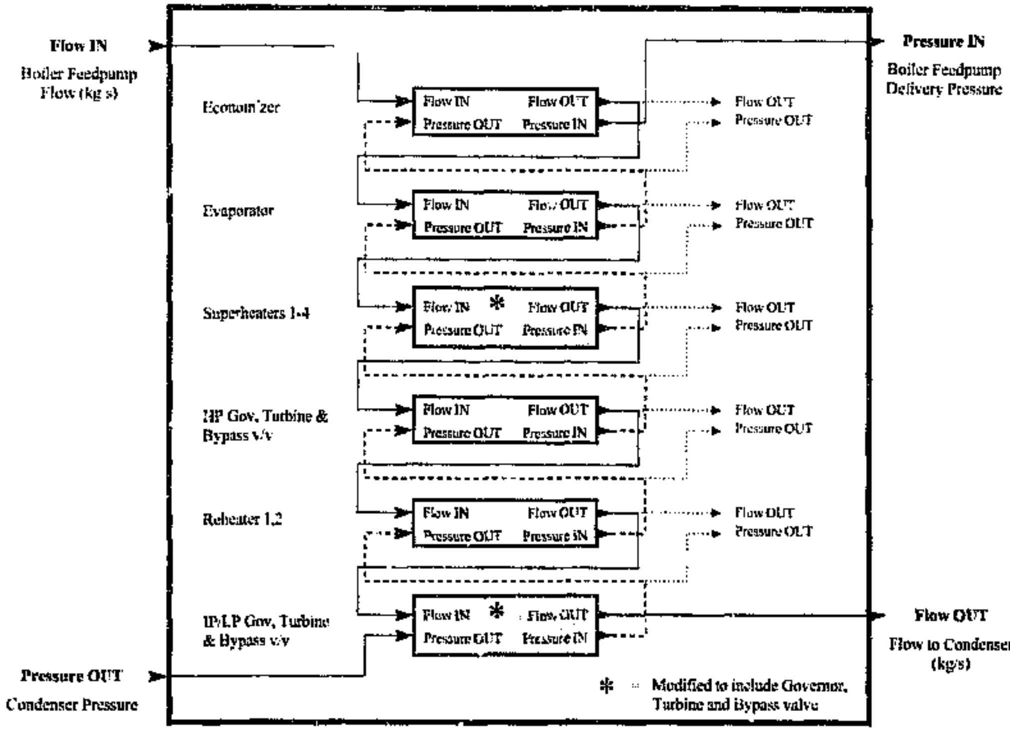


Figure 4-, 8: Pressure and Flow Implementation (whole Boiler and Turbine)

Each heat exchange bank has been simulated using a single lumped block. In the case of the HP and IP/LP turbine (and associated governor and bypass valves), a combination of three lumped parameter blocks was used for each turbine. Section 4.5.3 describes the design of the simulation for these plant items.

4.5.2.2 Heat Exchangers

The boiler consists of 9 heat exchangers in total. These are, in order of water/steam flow, the economizer, dividing wall, evaporator, superheaters 1,2,3,4, and reheaters 1 and 2. For the purpose of this simulation the economizer and dividing wall have been grouped together, as well as all four superheaters and both reheaters. This was done to conserve computation power.

Each heat exchanger has an associated volume, resistance, and coefficient of conduction, all of which remain constant throughout the simulation. In reality, the conduction coefficient would change depending on the amount of soot covering the heat exchange surfaces, which is influenced by the frequency of sootblowing of the boiler tubes.

Due to the simplifications to the furnace combustion model, the heat exchange simulation has been greatly simplified. The fluegas is assumed to have a constant temperature at each heat exchanger. The calculation of the heat exchange from the fluegas to the water/steam is based on the temperature difference between the two media as well as the contact time (total time that the water/steam takes to flow through the particular heat exchanger).

It is assumed that, after an infinite time, the water/steam will have reached the temperature of the flue-gas. The enthalpy is then calculated for the water/steam at this maximum temperature which corresponds to the fluegas temperature .

$$\text{Enthalpy}_{\text{MAX}} = \text{LookupTable}\left(\text{Pressure}_{\text{Steam}}, \text{Temperature}_{\text{FlueGas}}\right) \qquad \text{Eq 21}$$

An iterative calculation of the enthalpy at each position as the steam flows through a pipe would be calculated as follows:

$$\text{Enthalpy}_{n+1} = \text{Enthalpy}_{n} + \left(\text{Enthalpy}_{\text{MAX}} - \text{Enthalpy}_{n}\right) \times k \qquad \text{Eq 22}$$

where $k =$ constant heat conduction coefficient.

Then n would be iterated until n_{Final} which would give the total enthalpy exchange throughout the length of the pipe.

$$n_{\text{final}} = \frac{\text{TubeLength}}{\text{Flow}_{m/s} \times \text{TimeStep}} \qquad \text{Eq 23}$$

This equation, although being accurate, is iterative and thus very difficult to implement in VisSim as part of the main simulation. Figure 4-19 gives a comparison between the actual heat exchange curve (from equation 22) and two other simplified mathematical heat exchange models which do not require iteration for their solution. The two equations are given below:

$$\text{EnthalpyExchange}_{\text{FlueGas-To-Steam}} = \frac{\left(\text{Enthalpy}_{\text{MAX}} - \text{Enthalpy}_{\text{IN}}\right)}{1 + k \times \text{Flow}} \qquad \text{Eq 24}$$

$$\text{EnthalpyExchange}_{\text{Flue-to-Steam}} = \left(\text{Enthalpy}_{\text{MAX}} - \text{Enthalpy}_{\text{IN}}\right) \times k^{\text{Flow}} \qquad \text{Eq 25}$$

assuming $k < 1$

The comparison of the two simplified equations show the first to be more accurate. This was used throughout the simulation to approximate the heat exchange from the flue gas to the steam/water.

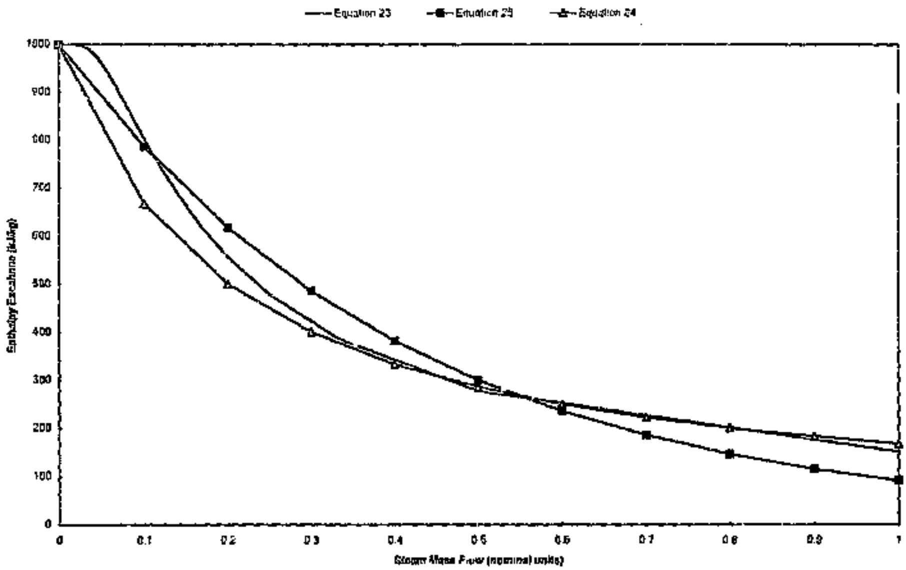


Figure 4-19: Enthalpy Exchange vs Steam Flow

The temperature and enthalpy of the steam was simulated in a similar way to the flow and pressure by treating each heat exchanger as a discrete finite element with lumped parameters.

Using the same approach as before, considering the generating unit as a whole the following characteristics exist:

Known:

- Temperature IN (temperature of water from boiler feedpump)
- Enthalpy IN (enthalpy of water from boiler feedpump)

Unknown:

- Enthalpy OUT (enthalpy of LP turbine exhaust steam)
- Temperature OUT (temperature of LP exhaust steam)

Using a steam-table lookup block, the Enthalpy out of the BFP may be calculated using the BFP outlet temperature and pressure (derived from pressure/flow lumped element block).

The temperature/enthalpy block for the heat exchangers is designed as follows:

Known:

- Temperature IN
- Enthalpy IN
- Fluegas Temperature (from fluegas temperature simulation)

- Flow (from pressure/flow lumped element block)

Unknown:

- Temperature OUT
- Enthalpy OUT

The heat (enthalpy) exchange from the flue gas to the steam/water is calculated in the figure below according to equation 24. Although the enthalpy of the inlet steam/water is know. it was decided to recalculate this from the inlet pressure and temperature. This was done to minimize the accumulation of errors which results from the inaccuracies of the lookup tables.

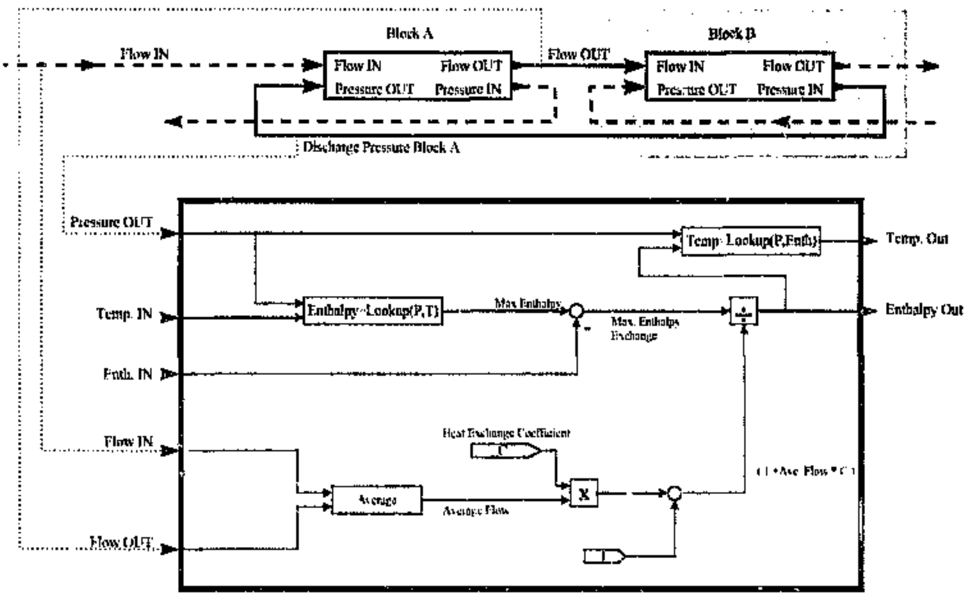


Figure 4-20: Temperature / Enthalpy Lumped Element Block

4.5.3 HP and IP/LP Turbines, Governors and Bypass Valves

The HP and IP/LP turbines consist each of a governor valve. a turbine (s) and a bypass valve. The governor and bypass valves were simulated using a fundamental block, with a very small volume and a resistance proportional to valve position. This proportional relationship was later replaced by a lookup table relating valve position to valve area. which gives a better representation than before. The volume of the bypass is set slightly higher than that of the governor valve, due to the extra pipework involved.

The turbine is represented by a fundamental block with a larger volume than the valves and with a high (constant) resistance to flow.

4.5.3.1 Pressure and Flow

Initially, the HP turbine, governor and bypass was configured as follows:

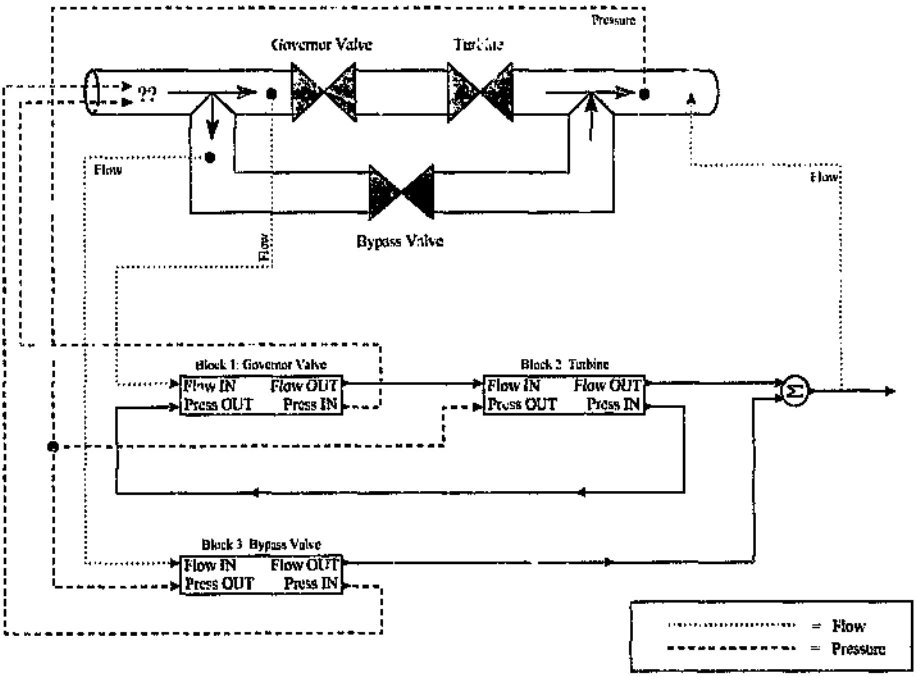


Figure 4-21: HP Governor, Turbine and Bypass Simulation

The main problem encountered with the above configuration is that the pressure at the inlet to the governor and bypass valves must be equal. Although the total inlet flow is known, the relative flows depend on the governor and bypass valve positions as well as the turbine resistance. Although the flows are not necessarily the same, the pressure differential across each of the flow paths are always equal. This fact was used in the simulation to calculate the relative flows using the following model.

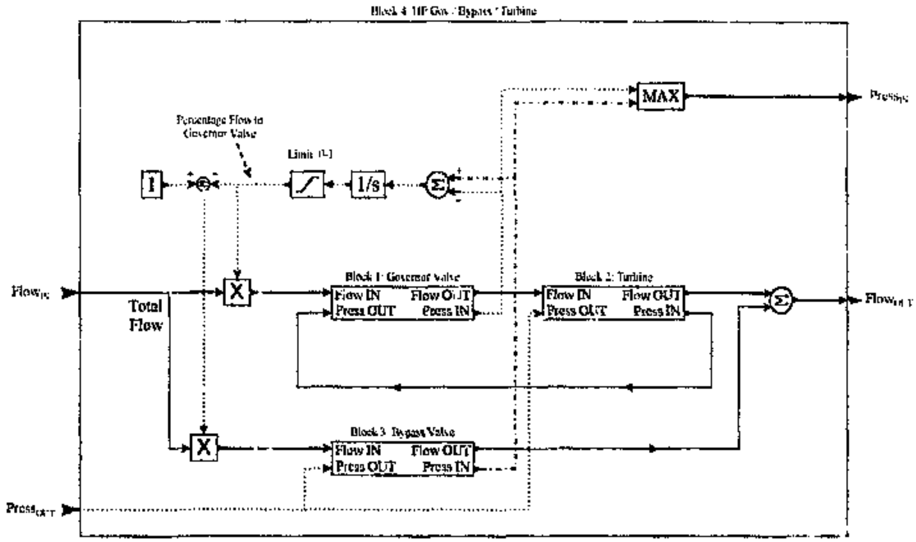


Figure 4-22: Flow Percentages to Governor and Bypass Valves

In reality, the flows through the two legs reach an equilibrium at which point the pressure drop across each leg is the same.
This equilibrium flow through the two different legs is calculated by integrating the difference between P_{IN} block 1 and P_{IN} block 2, and using this to determine the relative proportion of flow through the two legs.

The MAX gate was incorporated to prevent unpredictable results when one of the valves is fully closed.

4.5.3.2 Temperature and Enthalpy

Although the pressure drop across the valve increases as the governor/bypass valve is closed, it can be shown that the enthalpy drop is insignificant (refer to Knowles [39]) and may thus be ignored in the simulation.

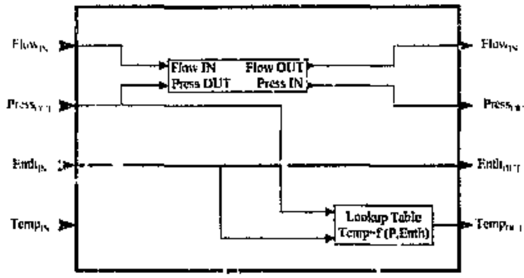


Figure 4-23: Temperature and Enthalpy Changes across a Valve

In an ideal turbine the expansion of the steam is completely adiabatic and frictionless, leading to a drop in steam enthalpy without any increase in entropy. In reality a certain inefficiency may be associated with this expansion, typically 8-12%, which is caused by a slight increase in entropy during the expansion process.

The simulation assumes that the expansion of steam through the turbine is completely isentropic (100% efficient), which allows the calculation of exhaust enthalpy and temperature given that the inlet enthalpy is know, and that both inlet and outlet flows and temperatures are known. This is calculated as follows:

At Turbine Inlet:

$$\text{Entropy}_{\text{INLET}} = \text{Function}_{\text{LookupTable}}\left(\text{Pressure}_{\text{IN}}, \text{Enthalpy}_{\text{IN}}\right) \qquad \text{Eq 26}$$

At turbine exhaust:

$$\text{Enthalpy}_{\text{OUT}} = \text{Function}_{\text{LookupTable}}\left(\text{Pressure}_{\text{OUT}}, \text{Entropy}_{\text{OUT}}\right) \qquad \text{Eq 27}$$

where, for isentropic expansion through the turbine

$$\text{Entropy}_{\text{OUT}} = \text{Entropy}_{\text{IN}} \qquad \text{Eq 28}$$

4.5.4 Separating and Collecting Vessels

During light-up and low load conditions the boiler operates in Sulzer mode whereby the steam leaving the evaporator is not 100% dry. The water is then separated from the steam as it passes through the separating vessel (refer to Figure 4-33) and flows to the collecting vessel. From the collecting vessel the water is pumped back to the economizer inlet.

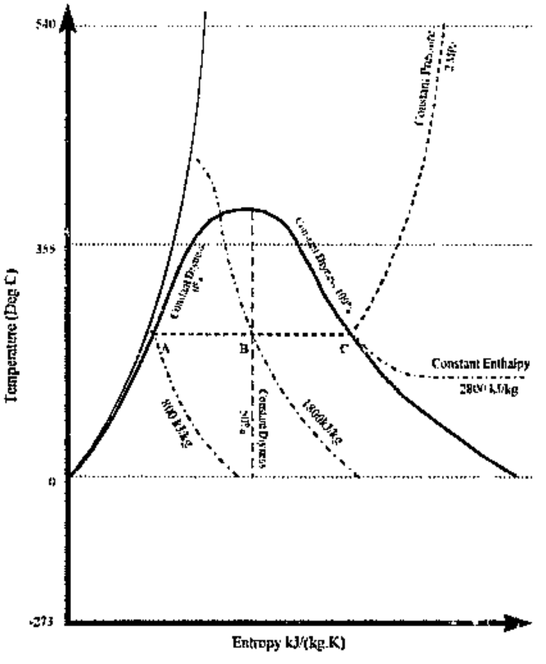


Figure 4-24: Evaporation

Assuming that the water/steam leaving the evaporator is characterized by point B on Figure 4-24, with a discharge pressure of 2 MPa, and an Enthalpy of 1900 kJ/kg. Point A represents the water which collects in the collecting vessel before being pumped back to the economizer inlet. Point C represents the dry steam that flows through the separating vessel and into superheater #1.

It is necessary to calculate the percentage of evaporator outlet flow that is pumped back to the economizer via the collecting vessel. This is done using the following formulae:

$$\%Flow_{ToSuperheaters} = \frac{Enthalpy_{PointB} - Enthalpy_{PointA}}{Enthalpy_{PointC} - Enthalpy_{PointA}} \times 100 \qquad \text{Eq 29}$$

The enthalpy of the steam entering the superheaters (during Sulzer operation) is always constant (for a constant pressure), being 2800 kJ/kg in the example above (point C). The enthalpy of the water returning to the economizer is also constant for a constant pressure, being 800 kJ/kg in the example above (point A).

Figure 4-25 illustrates the implementation of the above equation in the simulation, as well as the calculation of flows to both the superheater and economizer.

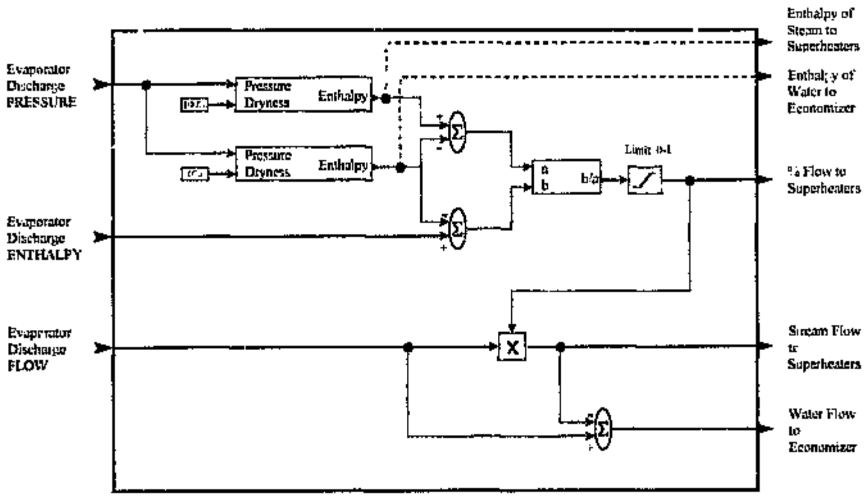


Figure 4-25: Separating Vessel Model

Once the boiler reaches approximately 50% load the steam leaving the evaporator is 100% dry, thus requiring no re-circulation. At this stage the boiler enters Benson mode.

4.5.4.1 Furnace Temperatures

The layout of the heat exchangers relative to the furnace area are shown below.

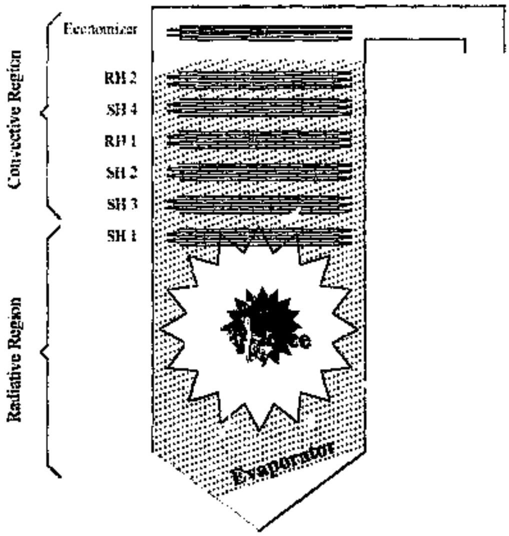


Figure 4-26: Boiler Heat Exchange Positions

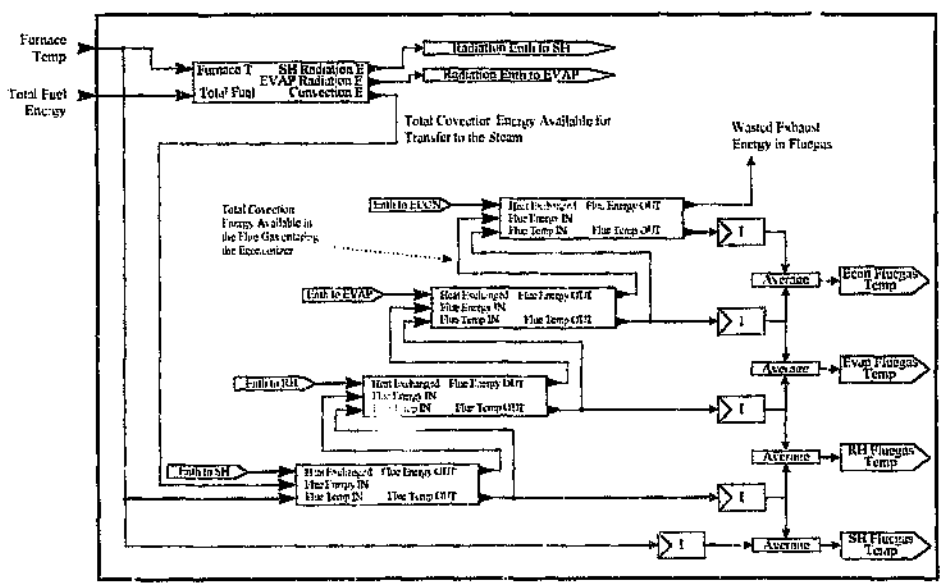


Figure 4-27: Fluegas Temperature Model

The temperature of the fluegas at each heat exchanger was modeled by assuming a constant fluegas temperature profile through the entire heat exchanger, this being equal to the average of the inlet fluegas temperature and the outlet fluegas temperature (Figure 4-29)

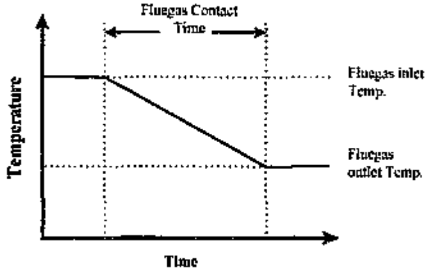


Figure 4-28: Actual Fluegas Temperature Profile Through Heat Exchanger

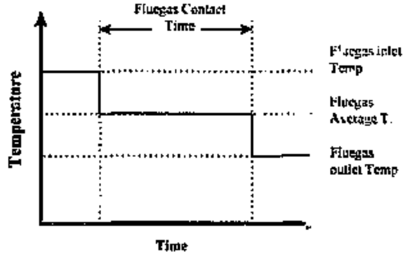


Figure 4-29: Simplified Fluegas Temperature Profile Through Heat ExcLanger

The flue gas temperature drop across each heat exchanger is modeled as follows:

$$T_{OUT} = T_{IN} \times \left(\frac{\text{TotalEnth}_{IN} - \text{TotalEnth}_{\text{Exchanged}}}{\text{TotalEnth}_{IN}} \right) \qquad \text{Eq 30}$$

This equation assumes a linear relationship between flue gas temperature and enthalpy. Total Fuel Energy is calculated as follows:

$$\text{TotalEnergy}_{\text{Coal}} = \text{CalorificValue} \times \text{FuelFactor} \times \text{MassFlow}_{\text{Coal}} \qquad \text{Eq 31}$$

In reality the furnace temperature is not a linear function of fuel burned, but has an upper limit depending on the composition of the fuel. The temperature also depends on the fuel/air ratio relative to the optimal ratio. Although increased excess air results in better combustion, this effect is canceled by the cooling effect of the additional air. A decrease in excess air below the optimum value may also cause a decline in the furnace temperature due to incomplete combustion of the coal.

The non-linear relationship between fuel quantity and furnace temperature was approximated as:

$$\text{Temperature}_{\text{Furnace}} = \left(1-\left(1-\text{MassFlow}_{\text{Fuel}}\right)^{3}\right) \times \text{MaxTemp} \qquad \text{Eq 32}$$

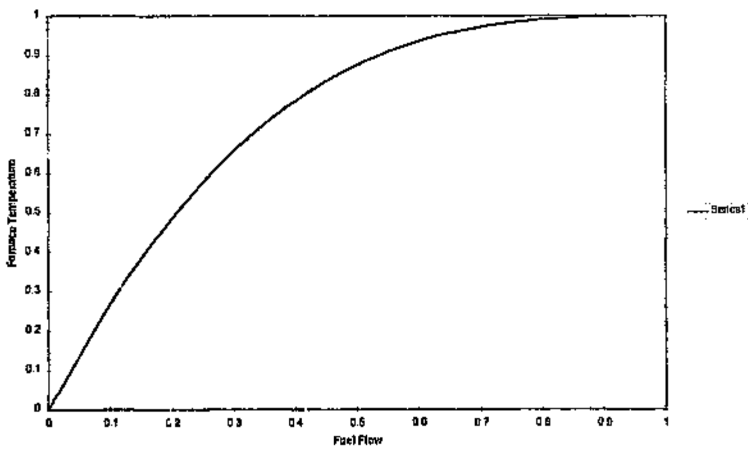


Figure 4-30: Furnace Temperature (equation 32)

In order to account for the cooling effect of the excess air, equation 32 is then divided by the total air flow.

Radiant Energy transferred to the lower superheater and the evaporator is calculated proportional to furnace temperature, with the exact ratio being dependent on the positions of the mills in service and the amount of excess air. Each mill is given a relative height weighting as follows:

- Top A: 6
- D: 5
- C: 4
- F: 3
- B: 2
- Bottom E: 1

$$\mathrm{Ratio}_{\mathrm{SH/Evap}} = \mathrm{k} \times \sum_{\mathrm{Mill\,F}}^{\mathrm{Mill\,A}} \mathrm{MillWeighting} \times \mathrm{MillMassFlow}_{\mathrm{Coal}} \qquad \text{Eq 33}$$

where k was found by experimentation to be approximately equal to 1/22. This ratio was also multiplied by the total air flow to simulate the effect of air flow on the position of the flame inside the furnace area.

4.5.5 Turbine (Acceleration and Speed)

The dynamics of the turbine have been modeled as follows[40]:

$$\alpha = \frac{\tau_{\mathrm{Steam}} - \tau_{\mathrm{BearingResistance}} - \tau_{\mathrm{GeneratorResistance}}}{I_{\mathrm{Angular}}} \qquad \text{Eq 34}$$

where

τ = Torque (Nm)
α = angular acceleration (rad/s^2)
I = moment of inertia (kg.m^2)

The following approximations were made:

$$\tau_{\mathrm{Steam}} \cong \frac{\left(P_{\mathrm{HP}} + P_{\mathrm{IP/LP}}\right) \times k}{\omega} \qquad \text{Eq 35}$$

$$\tau_{\mathrm{BearingResistance}} \cong \omega \times k_{\mathrm{Resistance}} \qquad \text{Eq 36}$$

$$\tau_{\mathrm{GeneratorResistance}} \cong \frac{k \times P_{\mathrm{GeneratorMW}}}{\omega} \qquad \text{Eq 37}$$

where

ω = Turbine Angular Velocity (rad / s) then

$$\omega = k \times \int \alpha . \mathrm{dt} \qquad \text{Eq 38}$$

giving

$$\alpha = \frac{\dfrac{\left[\left(P_{\mathrm{HP}} + P_{\mathrm{IP/LP}}\right) \times k_{\mathrm{Turb}}\right] - \left[k_{\mathrm{Gen}} \times P_{\mathrm{GeneratorMW}}\right]}{\omega} - \omega \times k_{\mathrm{Resistance}}}{I_{\mathrm{Angular}}} \qquad \text{Eq 39}$$

The implementation of this equation is as follows:

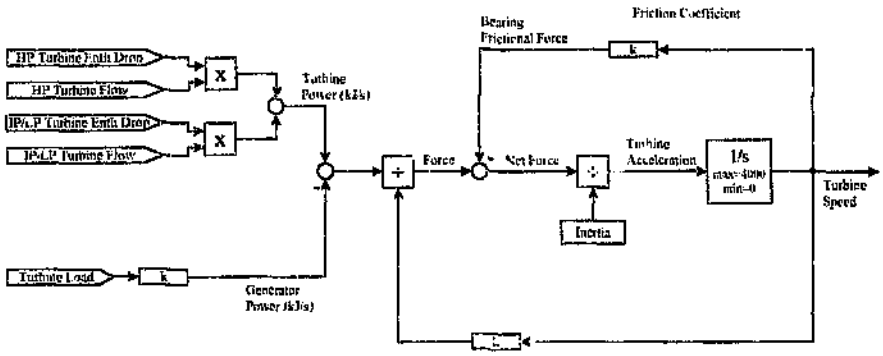


Figure 4-31: Turbine Speed and Acceleration Dynamics

4.5.6 SH and RH Attemporator Spraywater

The SH and RH spraywater attemporators play an important part in the control of the boiler temperatures. The implementation of the spraywater injection point is illustrated as follows:

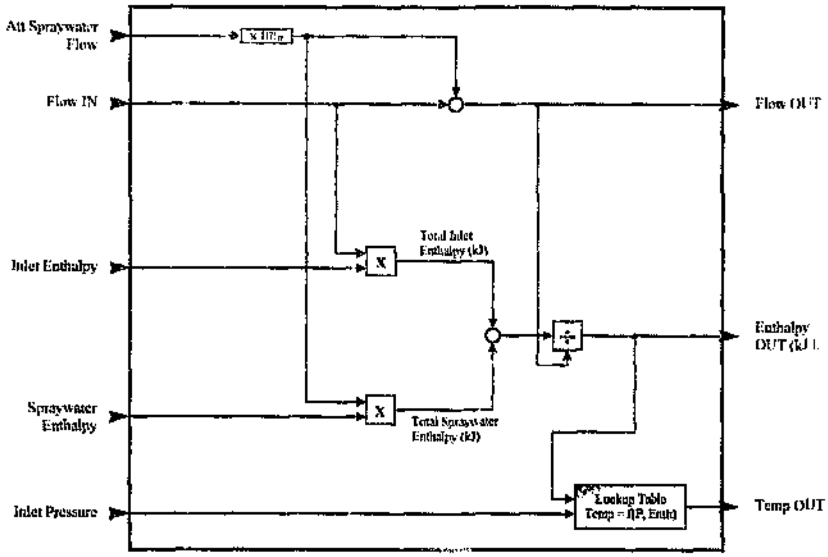


Figure 4-32: Attemporator Block

This block implements the following formula:

$$\text{Enthalpy}_{\text{OUT}} = \frac{\text{Enthalpy}_{\text{AttSW}} \times \text{Flow}_{\text{AttSW}} + \text{Enthalpy}_{\text{IN}} \times \text{Flow}_{\text{IN}}}{\text{Flow}_{\text{AttSW}} + \text{Flow}_{\text{IN}}} \qquad \text{Eq 40}$$

The pressure drop across the attemporator is negligible and has thus been ignored in the simulation.

4.5.7 *The Boiler, Turbine and Valves*

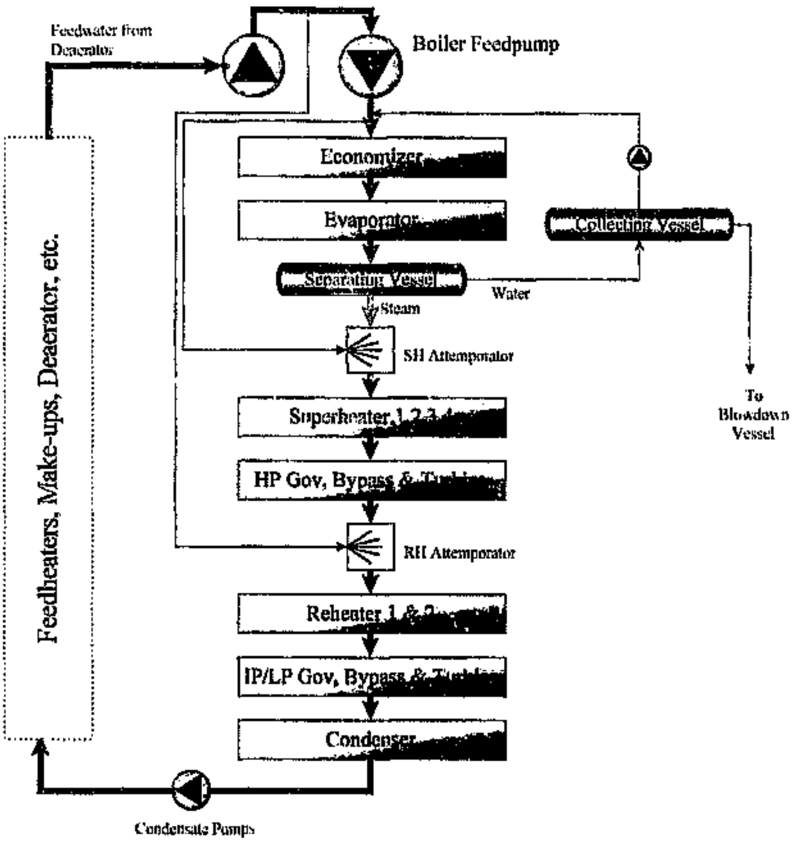


Figure 4-33: Feedwater and Steam Flow

The feedwater temperature leaving the boiler feed pump is assumed to be proportional to the furnace combustion temperature, while the delivery flow is regulated by the autocontrols. The condenser pressure is also assumed in the simulation to be constant. This allows the calculation of the boiler feed-pump discharge pressure using the lumped element building blocks for pressure and flow.

The B.F.P. discharge pressure and temperature are used to calculate the enthalpy of the water entering the economizer.

Four high pressure (HP) heaters provide extra heating of the feedwater before it enters the economizer. Due to the fact that the pressure drop across these heaters is so small, they have not been included as a separate block in the simulation, but have been included in the BFP block. The discharge temperature of the BFP has thus been increased by the appropriate amount to take into consideration the temperature effect of the HP heaters.

The implementation of the alternative boiler as simulation in VisSim is seen to be very modular in design. This is a result of the design of the pressure/flow and temperature/enthalpy lumped element blocks.

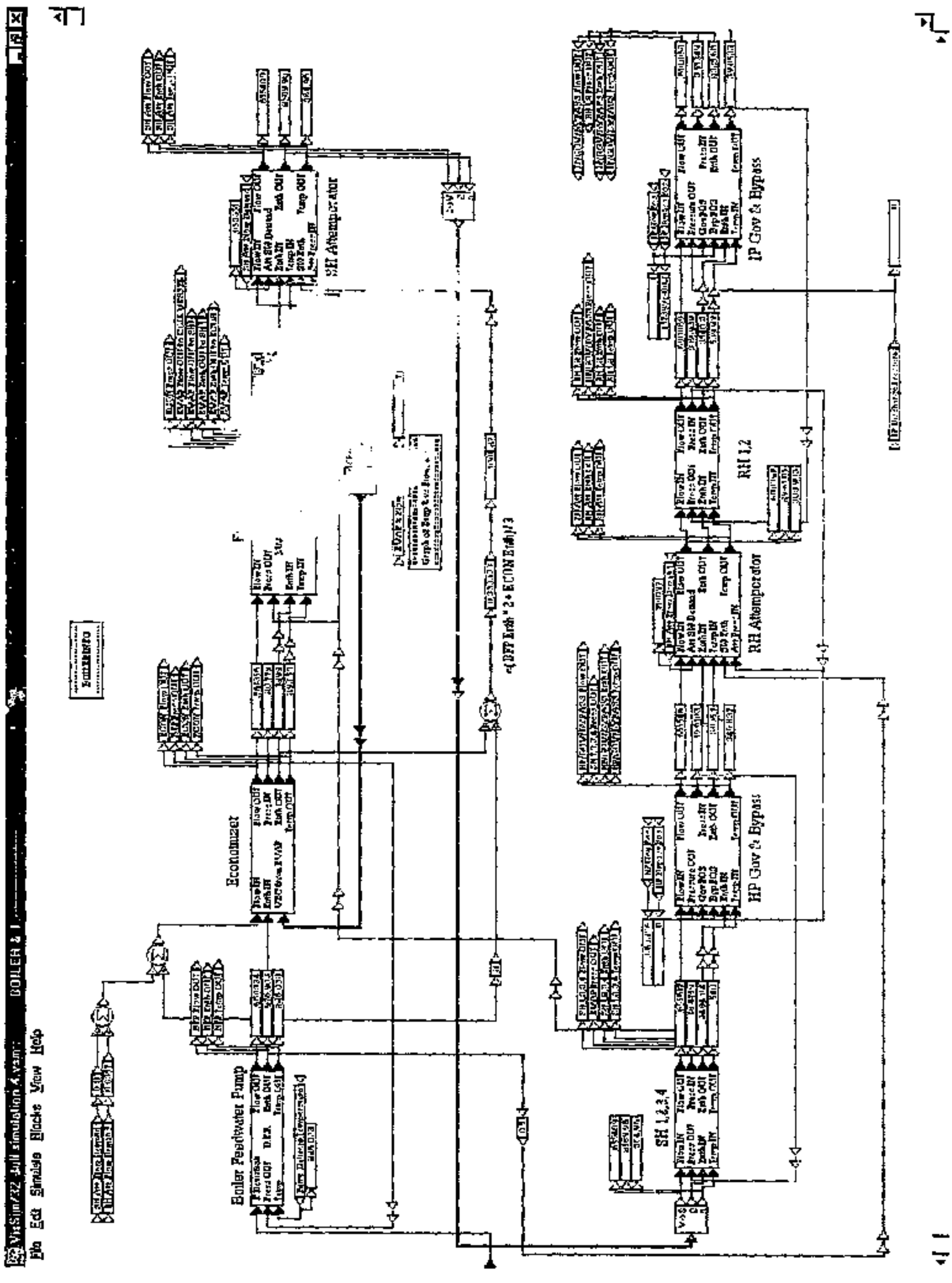


Figure 4-34: Physical Boiler Simulation in VisSim

4.6 Simulation of the Control System

4.6.1 Introduction

The unit autocontrols consist of numerous manipulated variables (outputs to plant) as well as process variables (inputs from plant). The manipulated variables are adjusted in order to control the process variables at their respective setpoints as dictated by the plant operator and protection circuitry.

Table 4-3 below summarizes the main control loops that have been implemented in the simulation. The autocontrols manipulate the primary and secondary control variables in order to control the process variable at the correct setpoint as dictated by plant conditions, operator input or other autocontrols.

Process Variable	Setpoint Source	Primary Control Variable	Secondary Control Variable
Turbine Speed	Set by Operator or 3000 rpm Constant	Governor Valve De	
Main Steam Pressure	Set by operator or 16.6MPa	Fuel Demand	HP Bypass Control Valve
Reheat Pressure	autocontrols	IP Bypass valve	
Superheater Temperature	540 °C Constant	SH Attemporator Spraywater	Boiler Feedwater Flow
Reheater Temperature	540 °C Constant	RH Attemporator Spraywater	
Evaporator Enthalpy	Load Dependent Setpoint from Autocontrols	Feedwater Flow	
Generator MW Output	Set on Control Desk or External Grid Control	no control possible as this is based on external consumption	

Table 4-3: Process Variables and Control Variables

The full simulation of the entire autocontrols is far beyond the scope of this dissertation. A complete simulation would be extremely processor intensive, possibly resulting in simulation speeds slower than real-time (using existing PC hardware). The above control loops were chosen due to their importance in the generating unit, as well as relevance to the physical boiler simulation which only includes the main turbine and boiler systems.

From Table 4-3 above, the only control sections that have not been implemented in the simple simulation are the control of evaporator enthalpy, SH and RH temperatures and turbine speed. These are, however, implemented in the advanced simulation.

4.6.2 Implementation - Initial Stages

The control system for the boiler and turbine was initially simulated in the required functionality, rather than implementing the actual controls present on the unit. This approach has the advantage that it was possible to commence testing of the physical model at an early stage, rather than having to wait for the final control simulation to be completed before being able to test the physical model.

Once the boiler/turbine physical model was at a stage where it was approximately 90% complete and operational, work began on the implementation of the control simulation based on the Duvha Power Station autocontrol diagrams.

4.6.3 Implementation - The Detailed Model

Although a great deal of work went into the implementation of the relevant autocontrols in the simulation, the majority of the work entailed converting the autocontrol design drawings into VisSim circuits. In most cases the autocontrol drawings provide adequate detail for the implementation of the simulation.

Another aspect of VisSim which makes it ideally suited for this simulation task is the fact that the basic building blocks used in the software are very similar to those found in the Teleperm-C control system autocontrol drawings.

4.6.3.1 Standard Integration Block

The standard VisSim integration block was modified to allow time-scaling of the simulation. This has been achieved by multiplying the input by a "system speed" variable. This standard block also imposes default limits of 0-1 on the integration output. These default limits can be overridden by connecting values to the "Max Output" and "Min Output" inputs.

This time scaling allows the simulation to be executed at any speed, either faster or slower than real time. The speed of the system cannot be increased indefinitely, as the system will become unstable above a certain speed. Time scaling is also dealt with in detail by Matko[41].

This block is used throughout the simulation for integration, as well as being used to derive all other blocks (e.g. the derivative block - Figure 4-36). This block has also been used in the physical plant simulations. A value of 1 for the variable "System Speed" represents real-time simulation, while a value of 2 represents simulation at twice the real-time speed. The simulation has been designed for operation up to a speed of 10 time real-time, after which it becomes unstable.

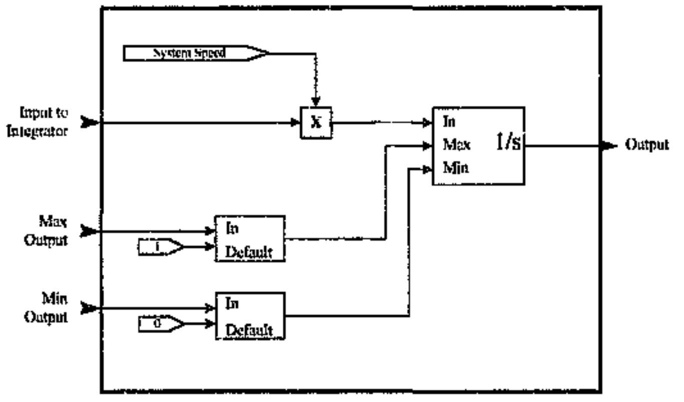


Figure 4-35: Standard Integration Block (time scaled)

4.6.3.2 Standard Derivative Model

The standard derivative block makes use of the standard integration block to enable time scaling.

The model used to approximate the derivative of a signal is based on a lag filter. The derivative is valid for frequencies up to (1/ time constant) rad/s. For higher frequencies the time constant must be decreased.

Limitations:

a) time constant > 0
b) Simulation stepsize must be less than the time constant for stability.
c) First Pass Results are inaccurate (i.e. when simulation is started at time zero)

Due to the internal time scaling inside the integration block, the simulation stepsize must be less then (time constant / system speed) seconds to ensure stability.

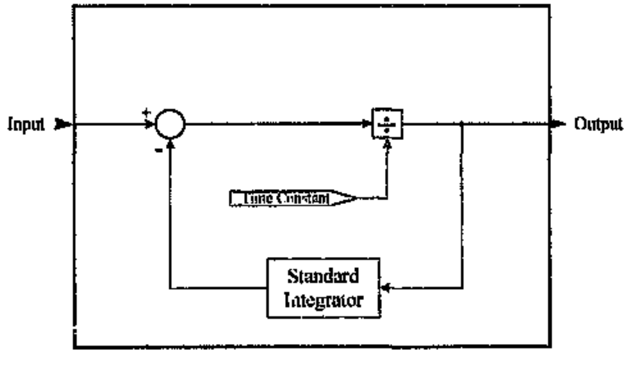


Figure 4-36: Derivative Block

The above block functions as a low-pass filter followed by a derivative block. For this reason it is important that the time constant be correct to ensure that no signals are filtered out by the low-pass filter.

An more accurate implementation of a derivative block is based on the following equation:

$$\frac{\mathrm{d}}{\mathrm{dt}}[x(t)] = \frac{x(t) - x(t - \mathrm{dt})}{\mathrm{dt}} \qquad \text{Eq 41}$$

Where, in the case of this simulation, *dt* = simulation time-step.

This implementation was found to produce extremely unstable results due to the small size of the time-step which then magnified small changes in the input signal.

4.6.3.3 Auto / Manual Override Block

The Auto/Manual block is used to manually override an input signal. The block ensures that the changeover from auto to manual and vice versa is completely bump-less (smooth). When the input signal ' Manual" is low, the output tracks the input.

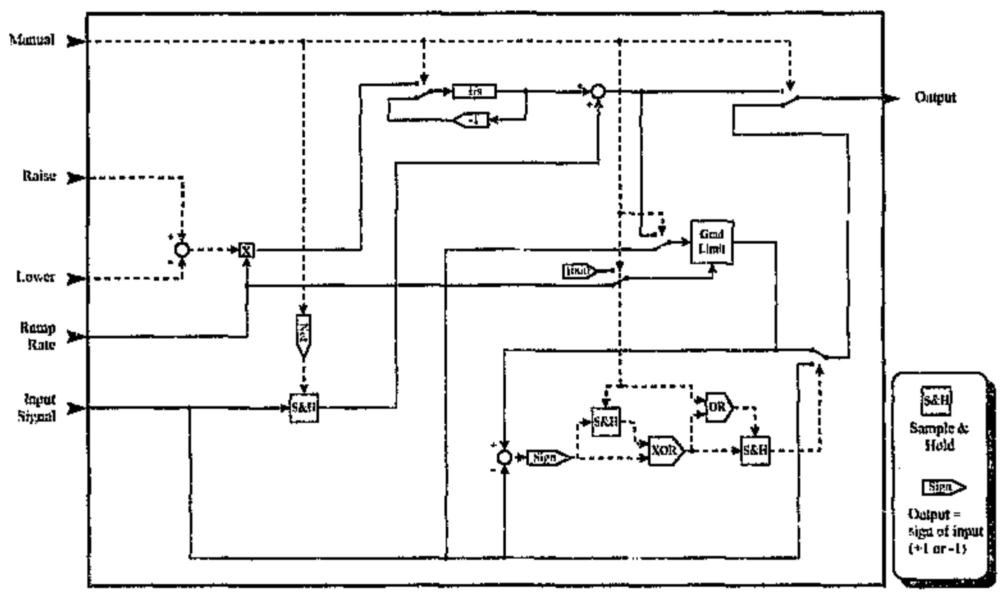


Figure 4-37: Auto / Manual Block

4.6.3.4 Unit Capability Computation

The simulation of the unit capability was derived directly from the unit autocontrols as explained in Section 3.1.

4.6.3.5 Unit Coordinator

Only the main areas of the unit coordinator have been included in the simulation. These are described in Section 3.1.

4.6.3.6 HP Bypass Control

The documentation on the HP bypass control is limited to the detailed circuit diagrams comprising the design of the control system. Due to the complexity of these drawings, it was decided to simulate the HP bypass controls in terms of their functionality, rather then creating an exact replica of the controls as was done for most of the other control loops.

4.6.3.7 LP Bypass Controls

The LP bypass controls have been implemented exactly as described in Section 3.4.

4.6.3.8 Turbine Governor Valve Controls

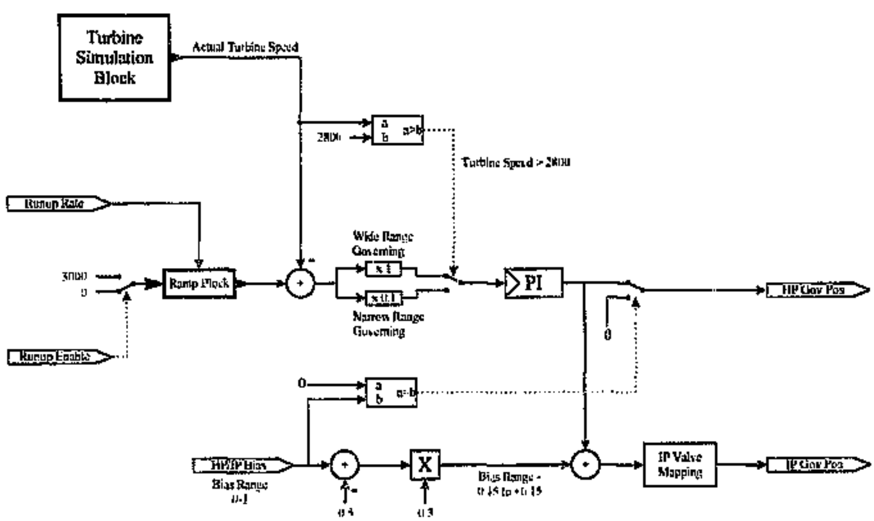


Figure 4-38: Governor Valve Controls

The time response of the generator to load changes has been neglected (due to its quick response). Load generated by the generator has been assumed to equal the load setpoint at all times. Due to this, the load error signal (see Section 3.8) in the actual governor controls has not been incorporated in the simulation.

The simulation compares the speed setpoint with the actual turbine speed, deriving the error signal which is the input to the PI controller. The output of the PI controller sets the HP governor valve position and, when summed with the HP/IP Bias signal, also sets the IP governor position. If the HP/IP Bias is set to zero then the HP governor position is also set to zero.

The ramp block simulates the implementation of the ramp rates. In order to run-up the turbine from standstill it is necessary to activate the "Run-up Enable" digital signal, which then ramps the turbine speed setpoint up to 3000rpm (via the ramp block). The controls initially start in "Wide Range" governing and switch over to "Narrow Range" governing once the turbine reaches 2800 rpm.

4.6.3.9 Superheater Temperature Control

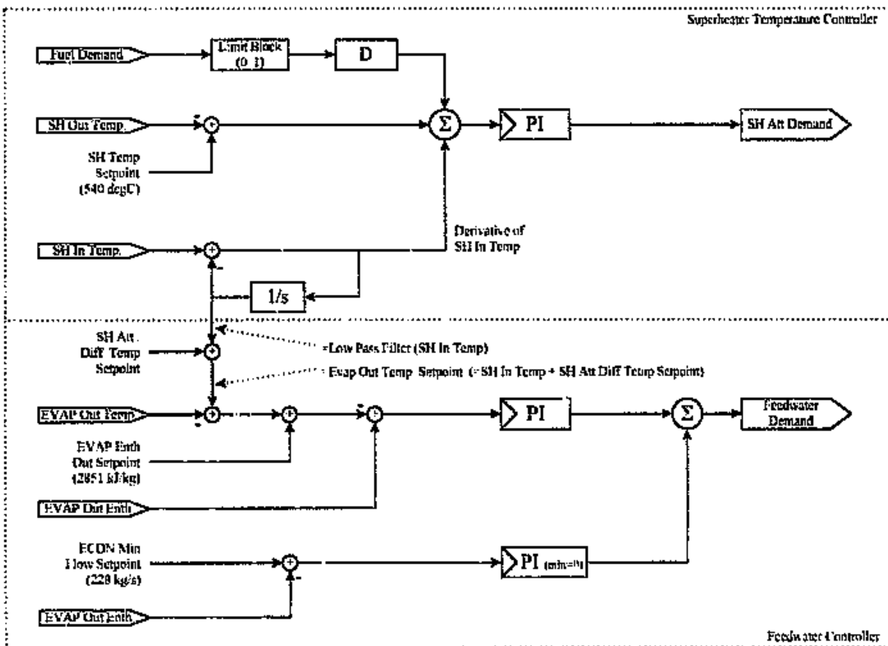


Figure 4-39: SH Attemporator Control and Feedwater Control

Refer to Section 3.6 for a description of the superheater attemporator controller. Due to the fact that the simulated boiler consists of only one superheater, it is necessary to modify the autocontrols to allow for a single superheater preceded by a single attemporator as opposed to 4 superheaters and 3 attemporators.

The difference between the superheater outlet actual temperature and setpoint is added to the derivative of the fuel demand signal and the derivative of the SH inlet temperature signal. The result of the summation forms the input to a block containing P and I elements. The output from the PI block is the SH attemporator flow demand signal.

4.6.3.10 Feedwater Controls

The feedwater controls are directly linked with the superheater attemporator controls. As can be seen in Figure 4-39, a setpoint for the evaporator outlet temperature is derived from the superheater temperature controller. The actual evaporator outlet is subtracted from this setpoint, which then forms part of the feedwater deviation signal. The other component of the feedwater deviation signal is derived from the difference between the actual evaporator outlet enthalpy and the setpoint. A second PI block has the function of ensuring a minimum flow (228kg/s) of water through the evaporator.

4.6.3.11 Collecting Vessel Level Control

The collecting vessel level controller consists of two separate PI controllers. The first attempts to control the level at a low setpoint by means of the two circulation pumps which pump the water back to the economizer inlet. Each pump has a capacity of up to 150kg/s, giving a total capacity of 300kg/s with both pumps in service.
The second controller, which operates on a higher level setpoint than the first, controls the flow to the blowdown vessel from where the steam is wasted to the ash sump.

4.7 The Operator Interface

The operator interface was designed to display as much information as possible about the simulation as well as providing access to the most important controls.

In order to optimize the execution speed of the simulation it was necessary to calculate the optimum user interface layout based on the requirements for maximum information and maximum speed. VisSim offers numerous input and output blocks, some of which execute faster than others. The execution times listed in Table 4-4 only apply to blocks when they are being displayed and not if they are hidden from view inside compound blocks.

Block	Max Number of Variables	Execution Speed
SIMULATION INPUTS		
Push-button (with text)	1	n/a
Push-button (with one bitmap icon per state)	1	n/a
Slider	1	n/a
SIMULATION OUTPUTS (DISPLAYS)		
Numeric Display Block	1	Relatively Slow
Gauge / Barchart	4	Fast
Graph	4	Fast
Digital Light	1	Fast
Scrolling Graph (stripchart)	4	Fast, but uses large amounts of memory

Table 4-4: VisSim I/O Blocks

The scrolling graph, although very useful, was found to consume approximately 1.5MB of system memory per variable, which becomes a problem when numerous scrolling graphs are used. This graph provides a scrollable window which displays any portion of the entire simulation time-range.

Numerous simulations were carried out with different types of display blocks in order to evaluate the best display configuration. The full boiler simulation was executed for 50 seconds starting from the same initial conditions for each test (1186 seconds after simulation start). The results of these tests (Table 4-5) show that the fastest practical display combination (shaded) contains only graphs as output blocks, with no numeric display blocks (shaded option).

Active Display Block(s)	Real Time	Sim. Time	Speed Rating
No Active Display Blocks	50 s	37 s	100%
1 Numeric Display Block	50 s	33 s	89%
55 Numeric Display Blocks	50 s	4 s	10.8%
4 Graphs (15 variables) + 7 Gauges	50 s	31 s	83.8%
22 Numeric Display Blocks	50 s	11 s	30%
4 Graphs (15 variables) + 7 Gauges + 1 Numeric Display	50 s	27 s	73%
4 Graphs (15 variables) + 7 Gauges + 7 Numeric Displays	50 s	18 s	48.6%

Table 4-5: Display Execution Speed

The main interface screen was designed to display only the most important variables, while sub-screens provide additional details where necessary. All controls are accessible on this main interface screen.

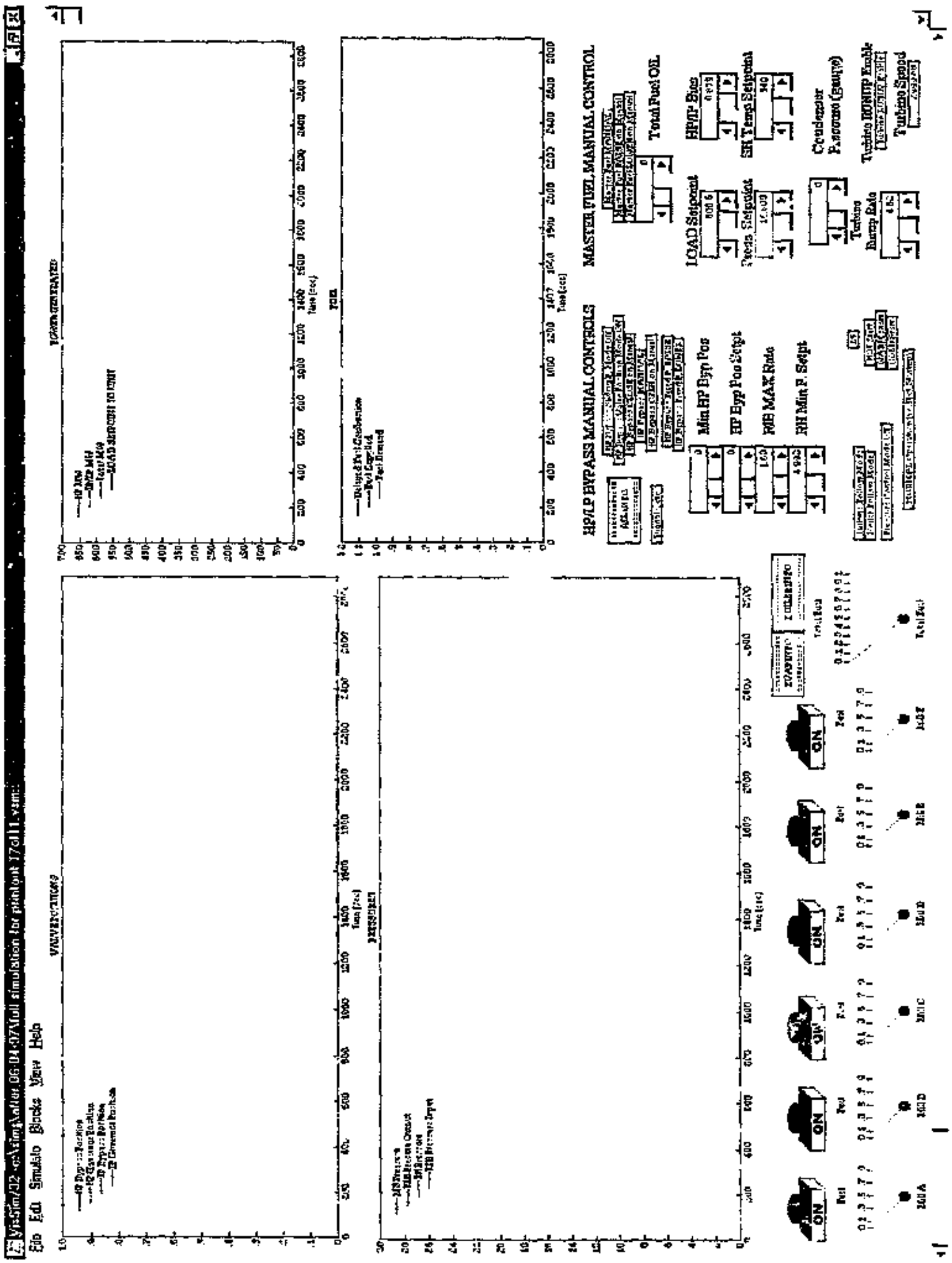


Figure 4-40: Main Simulation Interface

CHAPTER 5

5. Simulation Results

5.1 Introduction

In order to adequately asses the research with respect to the goals formulated in Section 1.2, it is important that suitable comparisons are made between the simulation and actual plant.

In assessing any simulation there are a number of different ways that comparisons may be made depending on the type and objectives of the simulation:

- Steady state comparison
- Dynamic comparison
- Full comparison (dynamic and steady-state)
- Comparison of responses to specific occurrences

In order to make a valid assessment of a simulation it is important that the simulated plant be well defined with a limited number of disturbances, each of which are either included in the simulation or may be accounted for when comparing the results.

The scope of this dissertation does not include the tuning of the simulation to match that of the real plant. For this reason any comparison of the simulation with that of the simulated plant can be very misleading unless certain restrictions are placed on the type and duration of the tests. If, for instance, a simulated startup is compared with a live startup the results will be different and could possibly confuse the engineer into thinking that the simulation is inaccurate. The reasons for these discrepancies are, among others:

a) The physical elements have not been tuned in terms of pressure drops, volumes, flows, enthalpy increases and temperature increases. This may give rise to the incorrect ratios of pressure and temperature between the different elements in the water/steam flow. Due to the fact that the feedwater flow controls react to the enthalpy at the evaporator outlet, these incorrect ratios may lead to incorrect flow rates, which in turn influence the pressures at the various stages. A slight miscalculation of the evaporator heat transfer coefficient affects the evaporator enthalpy, which affects the feedwater flow, which in turn influences the temperatures throughout the boiler. This knock-on effect is common to many of the systems, and makes tuning of the generating unit very difficult.

b) The unit control loops have not been tuned in terms of amplitude and response times. The tuning of these controls determine, along with the physical plant characteristics, frequency response, stability, and the damping of the control outputs. The correct tuning of the controls plays a major role in the dynamic response of the simulation, while having less of an influence on the steady state simulation.

In order to asses the accuracy of the simulation compared to the simulated plant certain isolated scenarios have been identified which limit the time of the simulation and the number of disturbance variables. This approach supports the primary objective of this research, which is to asses whether a reduced/simplified model of a generating unit is adequate for use as an engineering tool. The different tests were specifically chosen to evaluate the areas where the simulation would be most useful in solving engineering problems.

In order to make an accurate assessment of the results, is necessary that the boiler operate in a steady state both before and after the disturbance(s). Each test was performed at a different time, as overlapping tests would be difficult to interpret. Two tests were carried out on the simple simulation (Section 4.4) and a further three tests were performed on the alternate (more advanced) simulation (Section 4.5).

Throughout the rest of this chapter, live plant data collected from the SCADA is termed "test data" or "actual plant data" while simulation data is referred to as "simulation data".

5.1.1 Limitations

Any tests which require a change in unit load or are considered a trip risk will only be authorized to be carried out on a live unit under exceptional circumstances. The tests for the purpose of this dissertation are not of high enough priority to warrant a load loss or trip risk. For this reason the tests have been chosen carefully to allow critical comparisons to be drawn between the simulation and the actual plan, while still complying with these limitations.

5.1.2 Plant Data Collection

An extensive SCADA system has been installed on each Duvha generating unit as an aid to the unit operators. This system has the facility to record plant data for long periods of time for subsequent analysis. For the purpose of these tests, this historical plant data was exported from the SCADA at the end of each test and imported into spreadsheets for comparison with the simulated data.

5.2 Test 1: Unit Startup - Simple Simulation

5.2.1 Description of Test

This test involved the simulation of a generating unit startup from cold boiler conditions until automatic operation at full load. The primary objective of this test is to asses the simulation of the control system with respect to the different pressure control modes encountered during startup. The simple boiler model was chosen above the alternative model due to its lesser complexity and higher stability.

A secondary objective is to determine the accuracy of the simple physical boiler/turbine model in simulating the boiler main steam and hot reheat pressures during startup.

The simulated startup has not been compared with an actual startup due to the reasons outlined in Section 5.1, as well as due to the difficulty in determining, during an actual startup, which modes are active at which times.

5.2.2 Test Results

Time 00:00:00

FUEL DAMAND:	MANUAL
PRESSURE CONTROL:	OFF
HP Bypass Pressure Control Mode:	SLIDING
MIN HP Bypass Valve Position:	10%
LOAD:	0 MW

With the Fuel manually increased to 5%, the boiler main steam pressure starts to rise. During a real boiler startup, the pipework would already have been partially pressurized and heated. After approximately 3 minutes the HP bypass pressure ramp rate limiter caused the bypass valves to open in order to limit the pressure increase.
At approximately 00:25:00 the HP bypass valves were placed in Fixed Pressure Mode:

Time: 00:25:00

FUEL DAMAND:	MANUAL - 5 %
PRESSURE CONTROL:	OFF
HP Bypass Pressure Control Mode:	FIXED PRESSURE - 4MPa
MIN HP Bypass Valve Position:	10%
LOAD:	0 MW

With Fixed pressure mode enabled, the bypass valves open in order to control the main steam pressure at the fixed pressure setpoint (approx. 4MPa)

Time 01:00:00

FUEL DAMAND:	MANUAL - increased to 23%
PRESSURE CONTROL:	OFF
HP Bypass Pressure Control Mode:	FIXED PRESSURE - increased from 4MPa to 12MPa

MIN HP Bypass Valve Position:	10%
LOAD:	0 MW
Hot Reheat Max. Pressure:	3.8 MPa

The manual increase of the fuel resulted in an initial opening of the bypass valves. Manually increasing the fixed pressure setpoint from 4MPa to 12MPa then caused the bypass valves to close considerably in order to follow the increasing pressure setpoint. Once the main steam pressure reached the setpoint (01:45:00) the bypass valves once again opened, resulting in additional steam flowing to the hot reheat. This additional steam resulted in the hot reheat pressure rising above the setpoint, which caused the IP bypass valves to open.

The fixed pressure setpoint was again increased manually to 16.6MPa while also increasing the fuel demand manually. Once at this pressure the boiler was allowed to settle.

Time:03:00:00

FUEL DAMAND:	MANUAL - 40%
PRESSURE CONTROL:	OFF
HP Bypass Pressure Control Mode:	FIXED PRESSURE - 16.6MPa
MIN HP Bypass Valve Position:	10%
LOAD:	70 MW
Hot Reheat Max. Pressure:	3.8 MPa

Once the boiler is at the correct pressure, the turbine load is increased to 70 MW which results in both HP and IP governor valves opening to allow steam into the turbine. This also causes the HP bypass valves to close slightly to maintain the falling main steam pressure at 16.6MPa.

Time: 04:00:00

FUEL DAMAND:	AUTO
PRESSURE CONTROL:	ON
HP Bypass Pressure Control Mode:	VALVE POSITION - 0%
MIN HP Bypass Valve Position:	0%
LOAD:	70 MW
Hot Reheat Max. Pressure:	3.8 MPa

Once the turbine has been synchronized to 70MW and the pressure is approximately 16.6 MPa, fuel demand is placed on auto, pressure control is activated and HP bypass controls are placed in valve position mode, with a setpoint of 0%. The HP bypass valve is driven closed, while the fuel demand and pressure controller regulate the fuel to control the main steam pressure at 16.6MPa. The closure of the HP bypass results in a decrease in fuel demand due a decrease in the amount of energy wasted through the bypass valves.

Time: 04:45:00

FUEL DAMAND:	AUTO
PRESSURE CONTROL:	ON
HP Bypass Pressure Control Mode:	VALVE POSITION - 0%

MIN HP Bypass Valve Position:	0%
LOAD:	Ramped Up to 602MW
Hot Reheat Max. Pressure:	3.8 MPa

The load setpoint is ramped up gradually to 300MW, causing the HP and IP bypass valves to open. The increased flow through the IP governor valve causes the hot reheat pressure to decay, which results in the gradual closing of the IP bypass valve. This also causes a decrease in fuel demand as a result of the reduction in wasted steam.

The load is increased to 602 MW, after which the simulation is allowed to settle. At this stage all bypass valves are closed, and all controls are on AUTO except the HP bypass controls.

As the load increases from 300 to 600 MW, the hot reheat pressure decreases from 3.8 MPa to below 2 MPa. During a load increase on a real unit, the hot reheat pressure would increase slightly.

5.2.3 Analysis of Results

The results from this simulated startup have not been compared with an actual startup for the reasons outlined in Section 5.2.1 above. This makes detailed analysis of the results difficult.
It is, however, still important to analyze the results in terms of known inaccuracies and limitations. Further discussion is also required as to the suitability of the simulation as an engineering tool.

5.2.3.1 Known Inaccuracies and Limitations

a) *Recirculation*: During unit startup the separating vessels play an important role in the dynamics of the boiler. Their main function is to prevent wet steam and water from entering the superheater. This is achieved by recirculating this wet steam / water back to the economizer. This also occurs during periods of low load, typically below 300MW.

   Initially this recirculation causes a delay in the build up of boiler pressure, which was not seen in the simulation (Chart 1-2). The simulated total feedwater flow is exactly proportional to the boiler fuel (Chart 1-5). This is also incorrect as the separating vessels provide a buffering effect to disturbances.

b) *Turbine Synchronization*: The model of the turbine and generator (turbogenerator) is based on the assumption that output power is proportional to the steam mass flow (≈ volume x pressure) through the HP and IP/LP governor valves. This is an extremely simplified view of the turbogenerator, and does not take into account the energy required to overcome rotational friction.

Once the turbogenerator is at full speed and high generator output, this effect is negligible and can be ignored. During startup, however, a large amount of energy is required to bring the speed of the turbine up to synchronizing speed (3000 rpm) from barring speed (5 rpm). This would have a significant effect on the charts which would show a decrease in the bypass valve open positions, as well as changes to the boiler flows during this period of turbine runup.

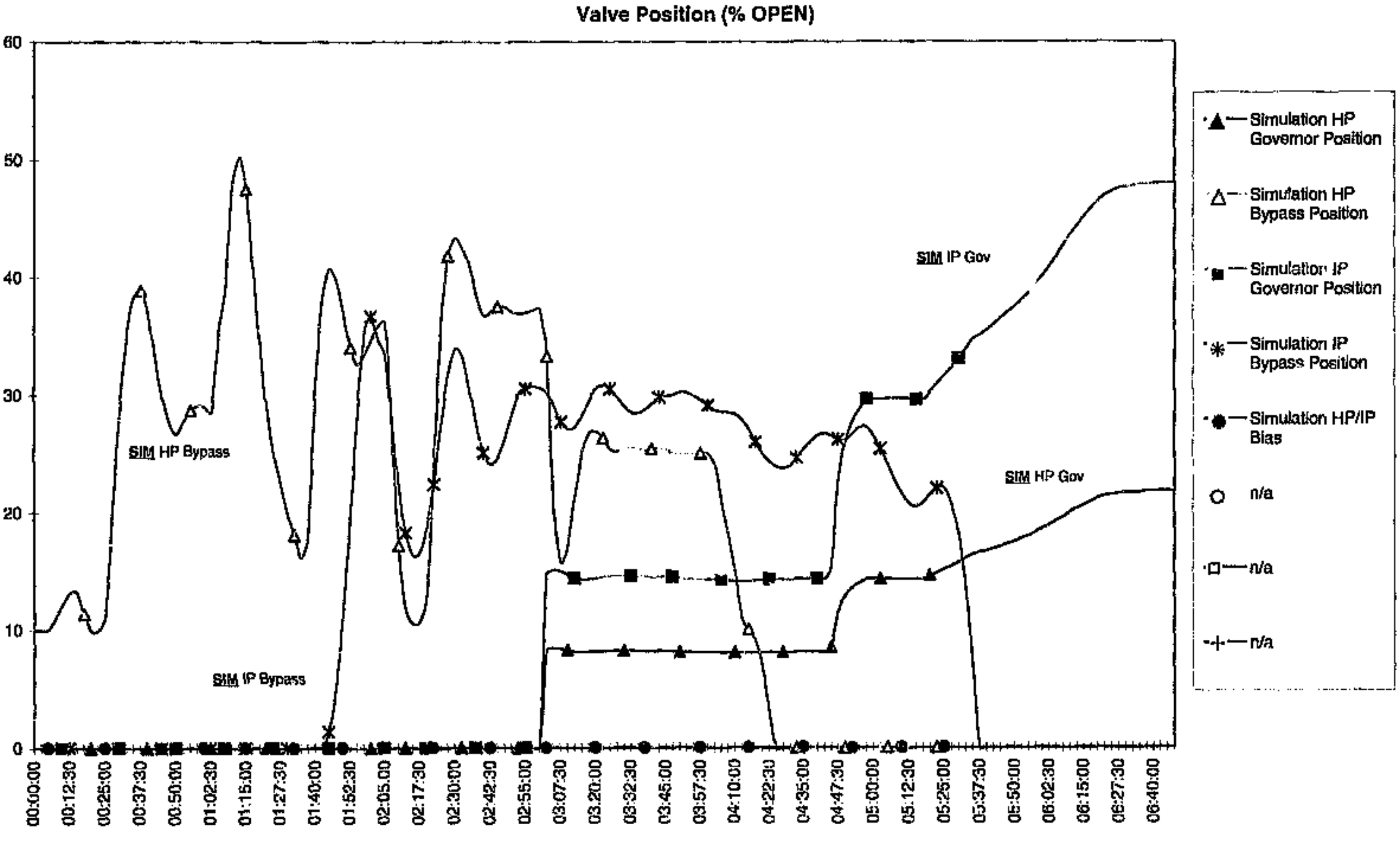
5.2.3.2 Significance of Results

At first glance the results show a reasonable resemblance to an actual boiler startup. The correct startup procedure is simulated in terms of fuel, pressure, load, and operating modes. The response to input changes is also as expected on an actual generating unit, with the exception of the reheat steam pressure which responds incorrectly to load changes.

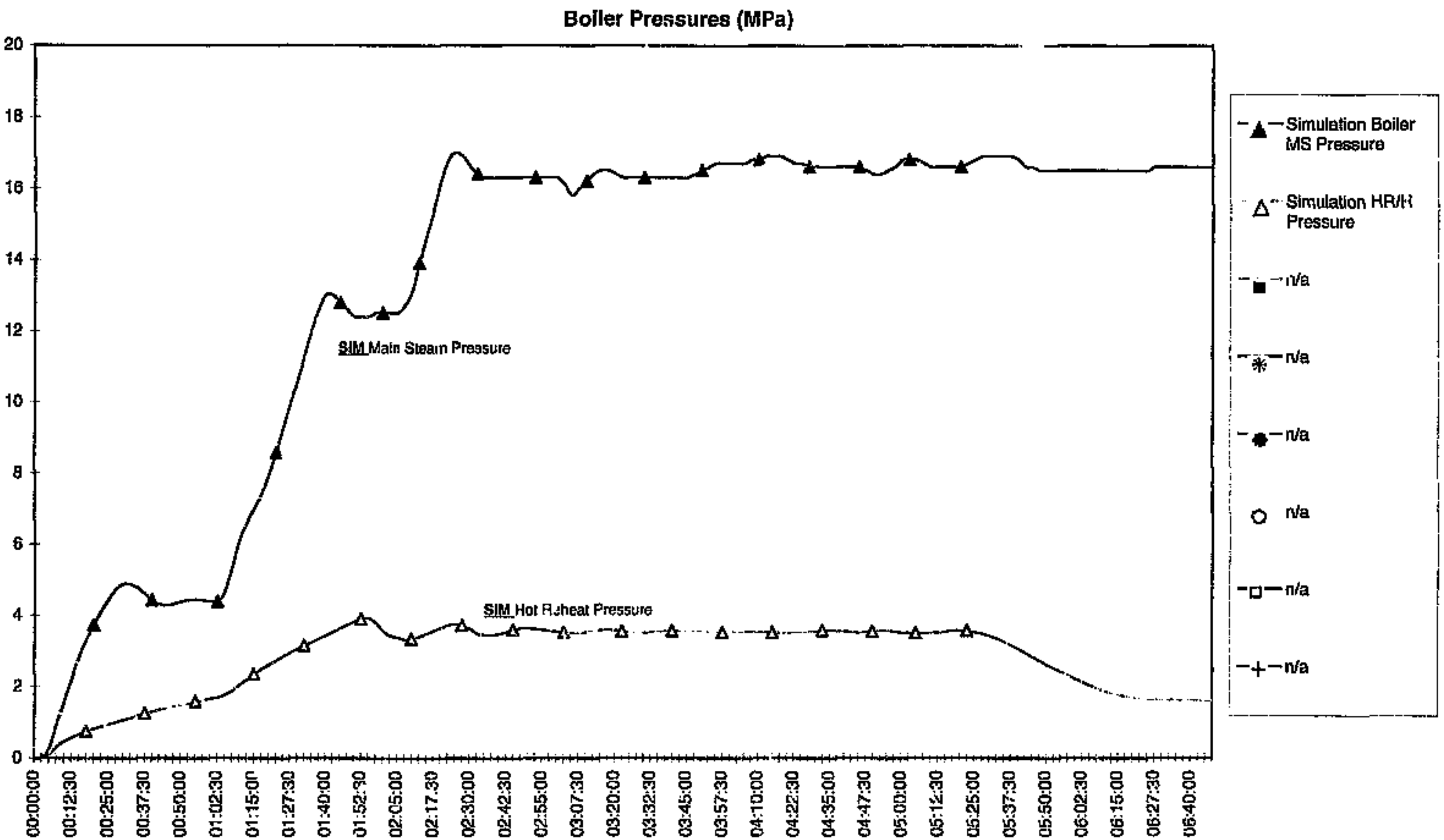
Further assessment of the model is required in terms of specific disturbances applied to the simulation. Section 5.3 compares the simulated results with actual results of a load dip.

The results of this simulation show that this simplified model is adequate as a training tool to give an understanding of the startup procedure. This includes timing of the steps, the different operational modes, and the expected response to these modes and steps.

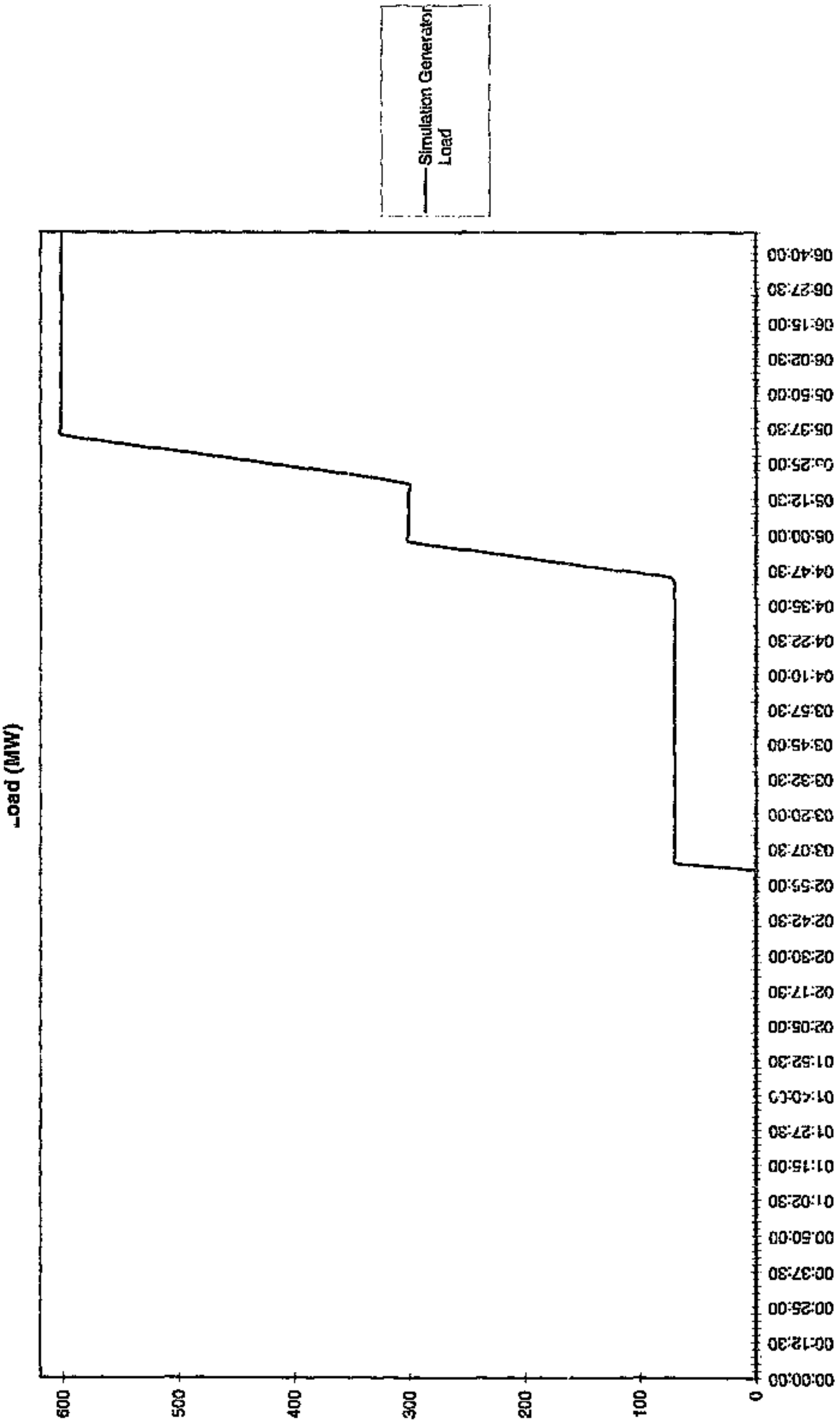
It could not, however, be used as a unit simulator for advanced training due to the many inaccuracies in the results, as well as the lack of many important simulated variables (temperature, enthalpy etc).



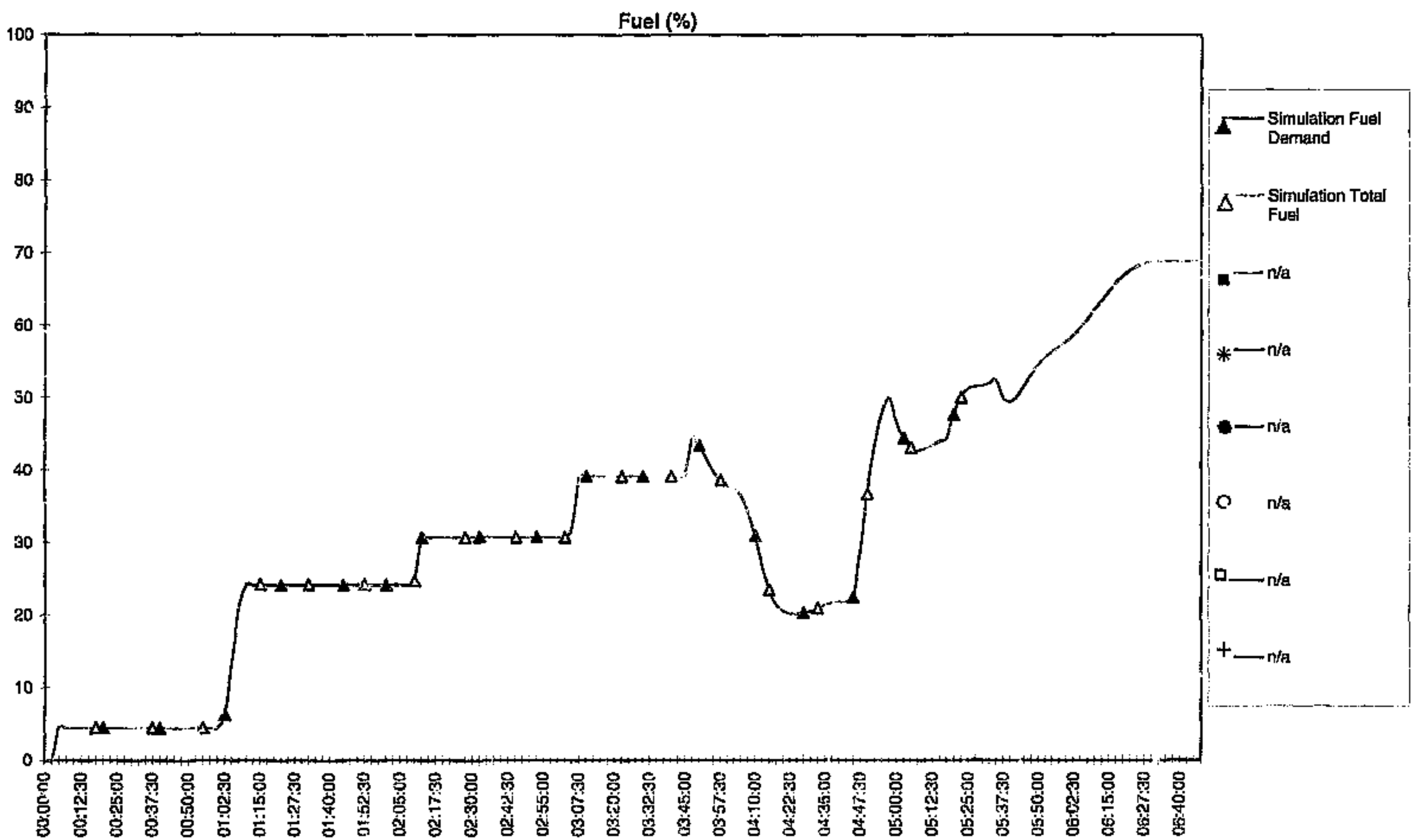
Test Chart 1-1: Unit Startup - Simple Simulation



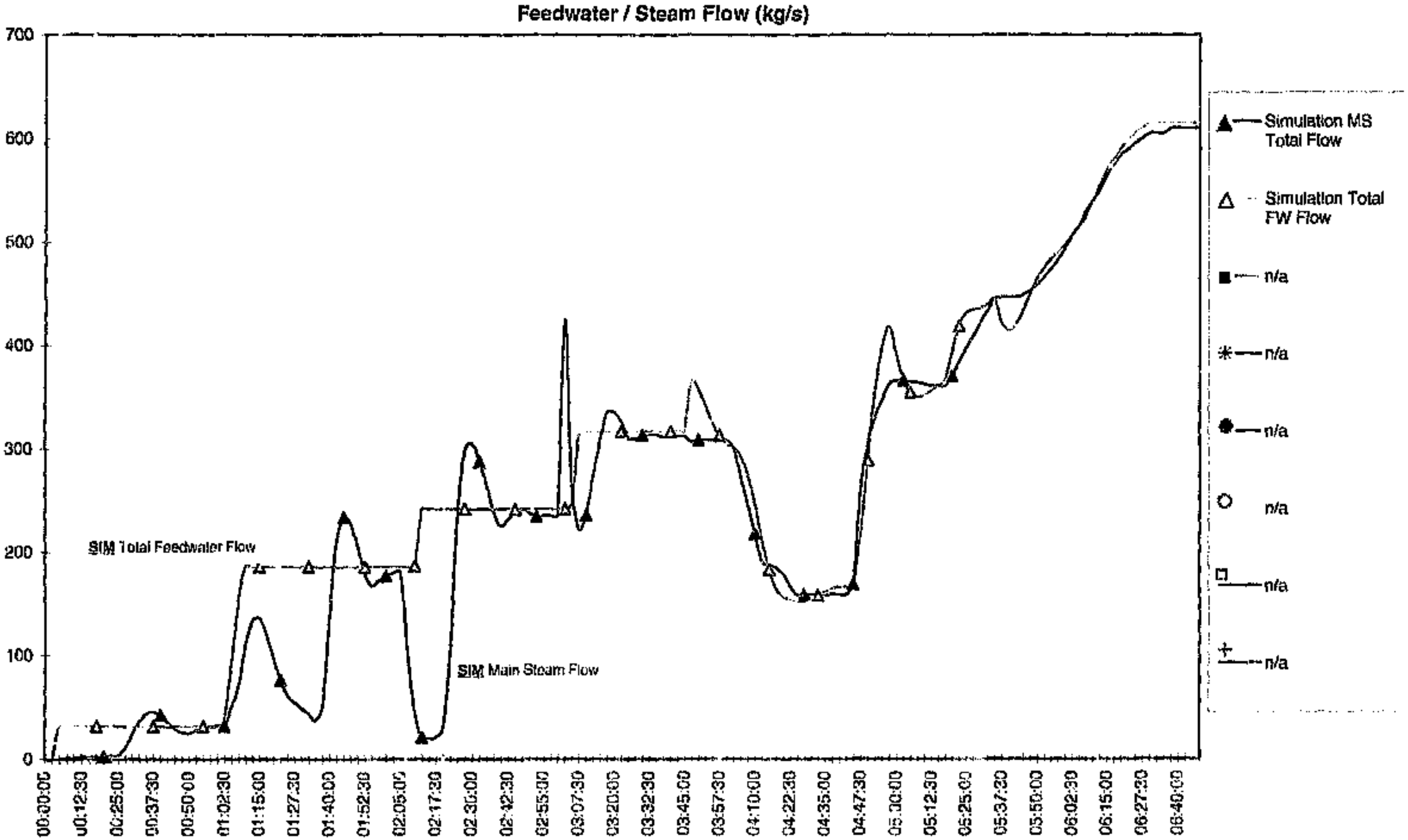
Test Chart 1-2: Unit Startup - Simple Simulation



Test Chart 1-3: Unit Startup - Simple Simulation



Test Chart 1-4: Unit Startup - Simple Simulation



Test Chart 1-5: Unit Startup - Simple Simulation

5.3 Test 2: Load Change - Simple Simulation

A common operation on a generating unit at Duvha Power Station is a change in generator load, which may happen as many as 4 times a day. It is thus important that the simulation be compared with a "real" load change. Some of the results have been compared with the advanced simulation results for the same load disturbance.

5.3.1 Description of Test

The data from a normal load change on a generating unit was captured on the SCADA and recorded. Although a number of isolated load changes were available as historical data on the SCADA, the most suitable load change was chosen based on both magnitude and duration.

The load setpoint input to the simulation was created from the live plant load signal in terms of magnitudes and ramp rates. This was done to remove the slight instabilities and oscillations in the actual load signal which would make the simulated results even more difficult to interpret.

Test:	**<u>Load Changes</u>**
Date:	11/04/97
Unit:	3
Operator:	Mr. Piet Krieger
Start Time:	01h30
Duration	04h30

<u>Time 00:00:00 - Initial Conditions</u>	
Generating unit & Autocontrols:	Stable
Turbine Load:	595 MW
<u>Time 00:38:00 - First Load Change</u>	
Load Change:	595MW down to 507 MW
Load Change Gradient:	-20 MW/min
<u>Time 01:30:00 - Second load Change</u>	
Load Change:	507 MW down to 417 MW
Load Change Gradient:	-20 MW/min
<u>Time 03:28:00 - Third Load Change</u>	
Load Change:	417 MW up to 523 MW
Load Change Gradient:	+10 MW/min
<u>Time 04:30:00 - End Conditions</u>	
Generating unit & Autocontrols:	Stable
Turbine Load:	523 MW

5.3.2 Test Results

One of the shortfalls of the initial simple simulation is the simplifications in terms of temperature and enthalpy, both of which are completely neglected. For this reason this set of results does not include any temperatures, enthalpies or attemporation flows.

5.3.2.1 Valve Positions

The test results (Chart 2-1) show a moderate correlation between the actual test done on the boiler and the simulated results from the simple boiler. The simple boiler shows a far slower response to the load changes. The speed of the response could be improved by tuning the boiler control loops.

The advanced boiler shows a far quicker response to load changes (Chart 2-2). This is mainly due to the differences in tuning parameters of the control loops. This is also shown on Chart 2-7 where the simulation exhibits spikes in the feedwater flow due to the quick response.

5.3.2.2 Boiler Pressures

This simple boiler model only simulates the main steam pressure (at superheater outlet) and the reheater steam pressure (at reheater outlet). The main steam pressure remained constant at 16.6 MPa throughout the test and has not been shown on the graphs.

One of the important inaccuracies of the simple boiler model is the response of the hot reheat pressure to load changes. As is shown in Charts 2-3 and 2-4, the response is completely opposite to the actual response exhibited by both the actual boiler and the advanced boiler model.

This is due to fundamental differences between the simple and advanced models, especially in the calculation of boiler pressures. In the advanced simulation, as well as on the real unit, decreasing load causes the valves to close slightly, which results in an increase in flow resistance. This leads to an increase in both main steam and reheat steam pressures. In order to control the main steam pressure at the setpoint, the feedwater controls decrease the feedwater flow, which intern brings the pressures down again. The result of this complex control system is that there is a strong link between the main steam and reheat pressures, leading to relatively constant reheat pressures (at constant main steam pressure) at different loads. In the case of the simple simulation, the flow from the HP turbine is scaled proportionally to represent the loss of energy in the HP turbine. The larger proportion of the flow (which is synonymous with energy) is determined by the fuel flow rate into the furnace. The main steam pressure model is completely independent of IP pressure model (i.e. hot reheat pressure and IP governor and bypass valve positions), while the IP pressure model is largely independent of the HP turbine. The result of this is that any decrease in unit demand results in a partial closing of the IP governor valves, leading to an increase in IP pressure.

5.3.2.3 Total Fuel

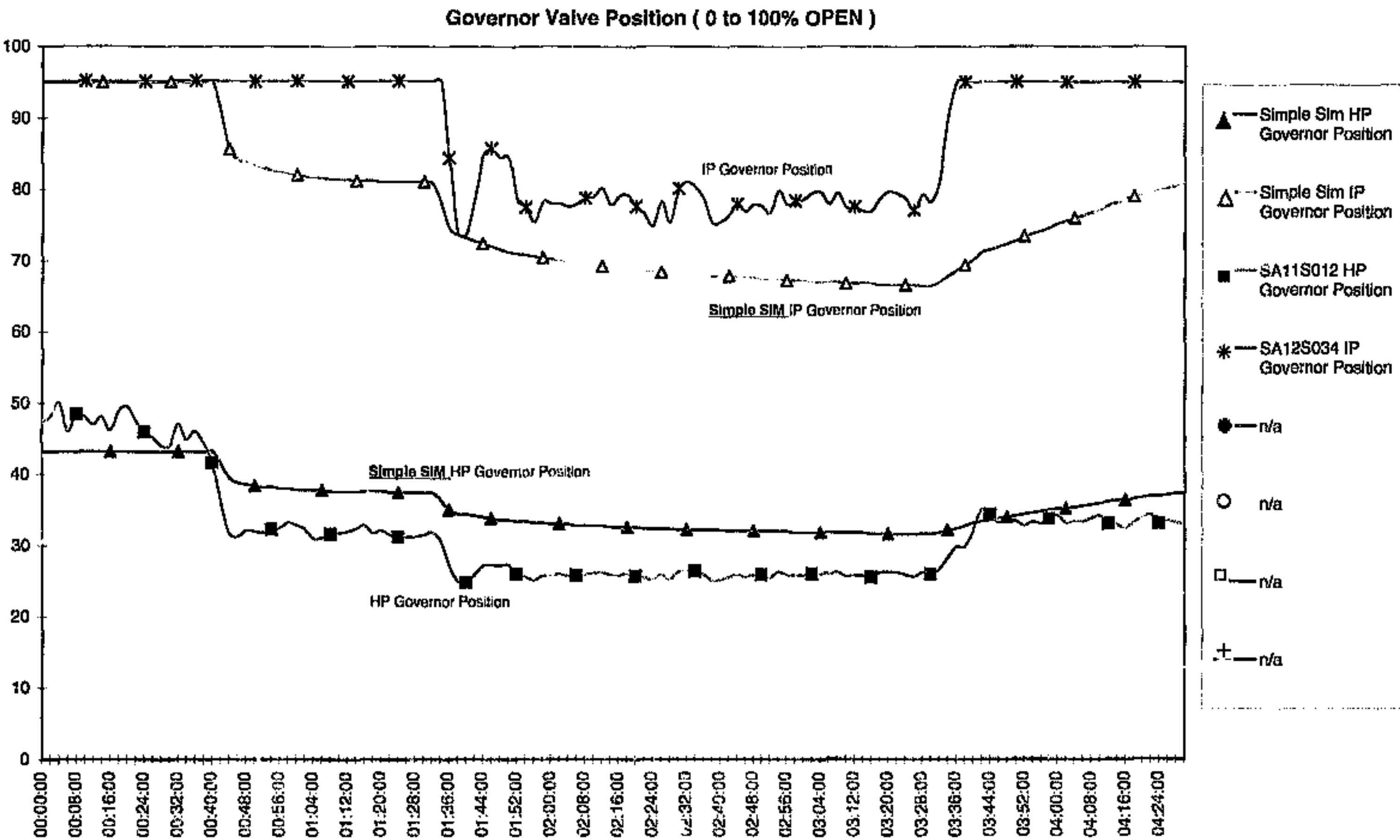
The total fuel results from the simple simulation show good correlation with the actual boiler test results (Chart 2-5). Although the response is once again slower than the actual boiler, this set of results shows a far slower response to an increase in load than to a corresponding decrease in load. Looking back at the graphs for feedwater flow and valve position confirm this phenomenon, albeit to a lesser degree. The exact reason for this discrepancy has not been investigated further. This graph also illustrates the differences in controller speed (relative to plant responses) between the simple and advanced simulations.

5.3.2.4 Feedwater Flow

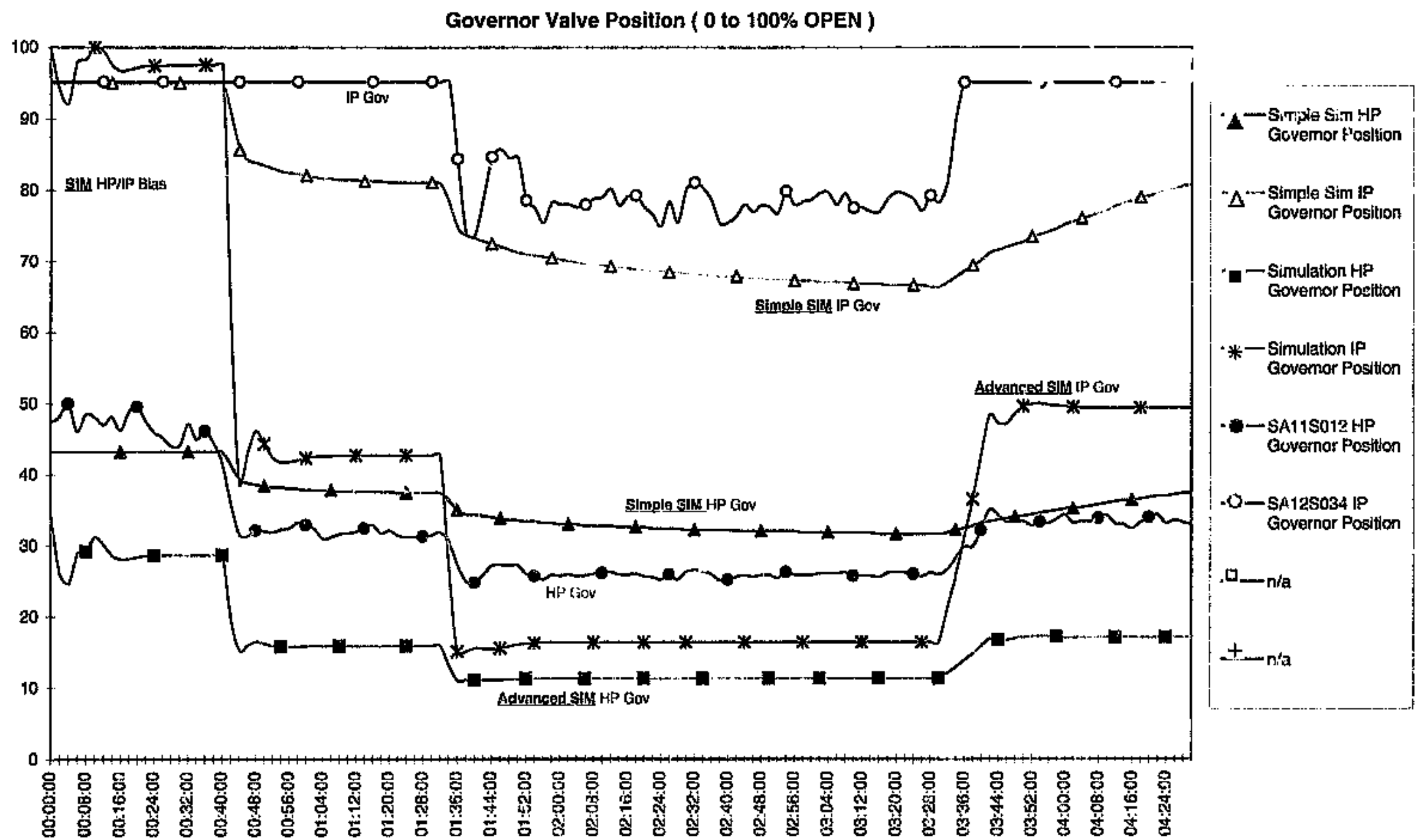
The feedwater flow results from the simple simulation show good correlation to the actual boiler results. As with the valve positions, the feedwater flow response was found to be slower than the actual boiler responses, especially following the load increase. Chart 2-6 shows the responses from both simulations, while Chart 2-7 shows only the total feedwater flows for the simple simulation, advanced simulation and actual test results.

5.3.3 Conclusions

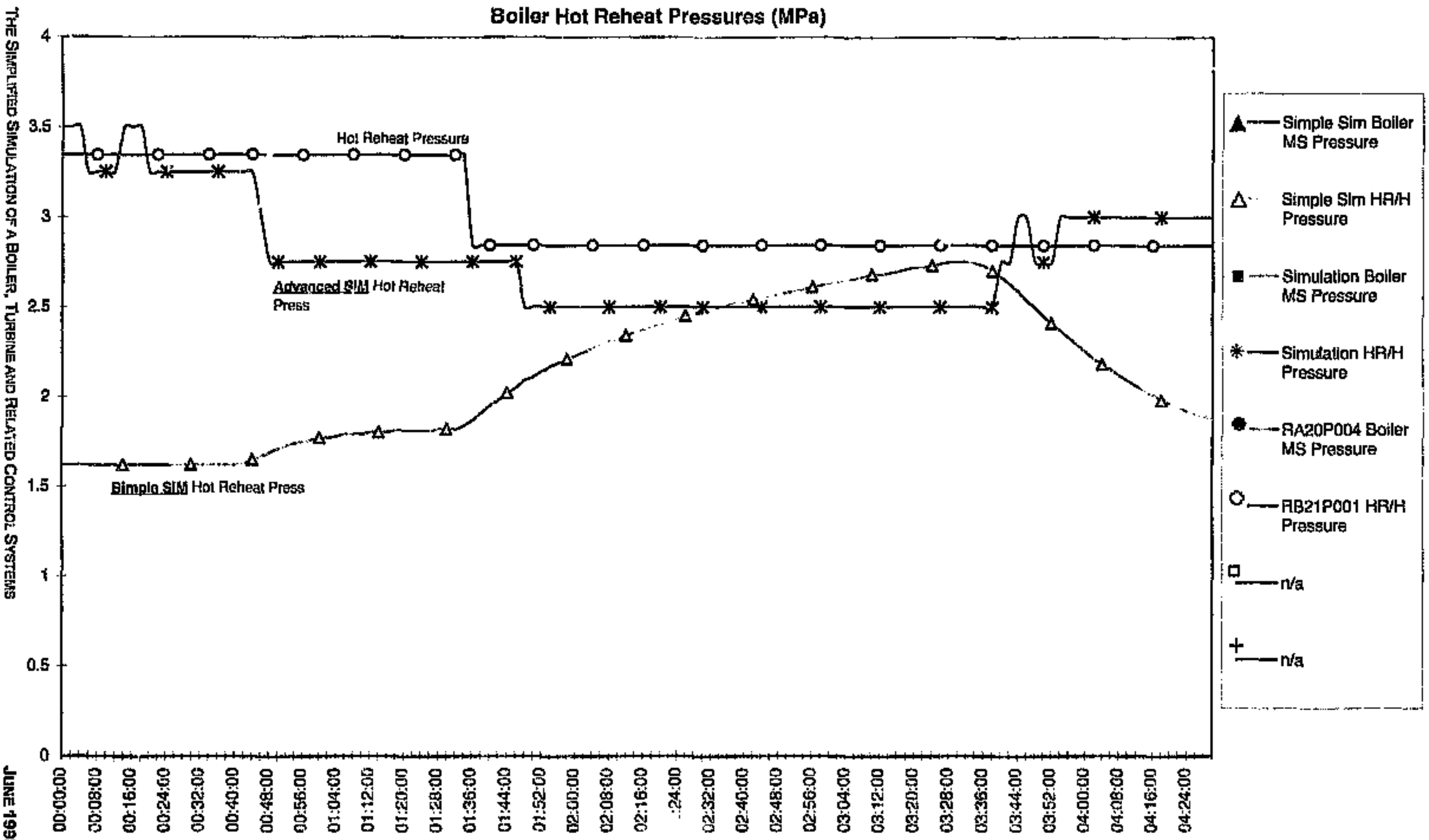
The simple simulation provides a good model of the generating unit in terms of the response of the simulated variables during a load change. The incorrect response of the IP pressure during the simulated load dip is a shortfall of the model, thus making it unsuitable for certain engineering tasks which involve the IP pressure responses.



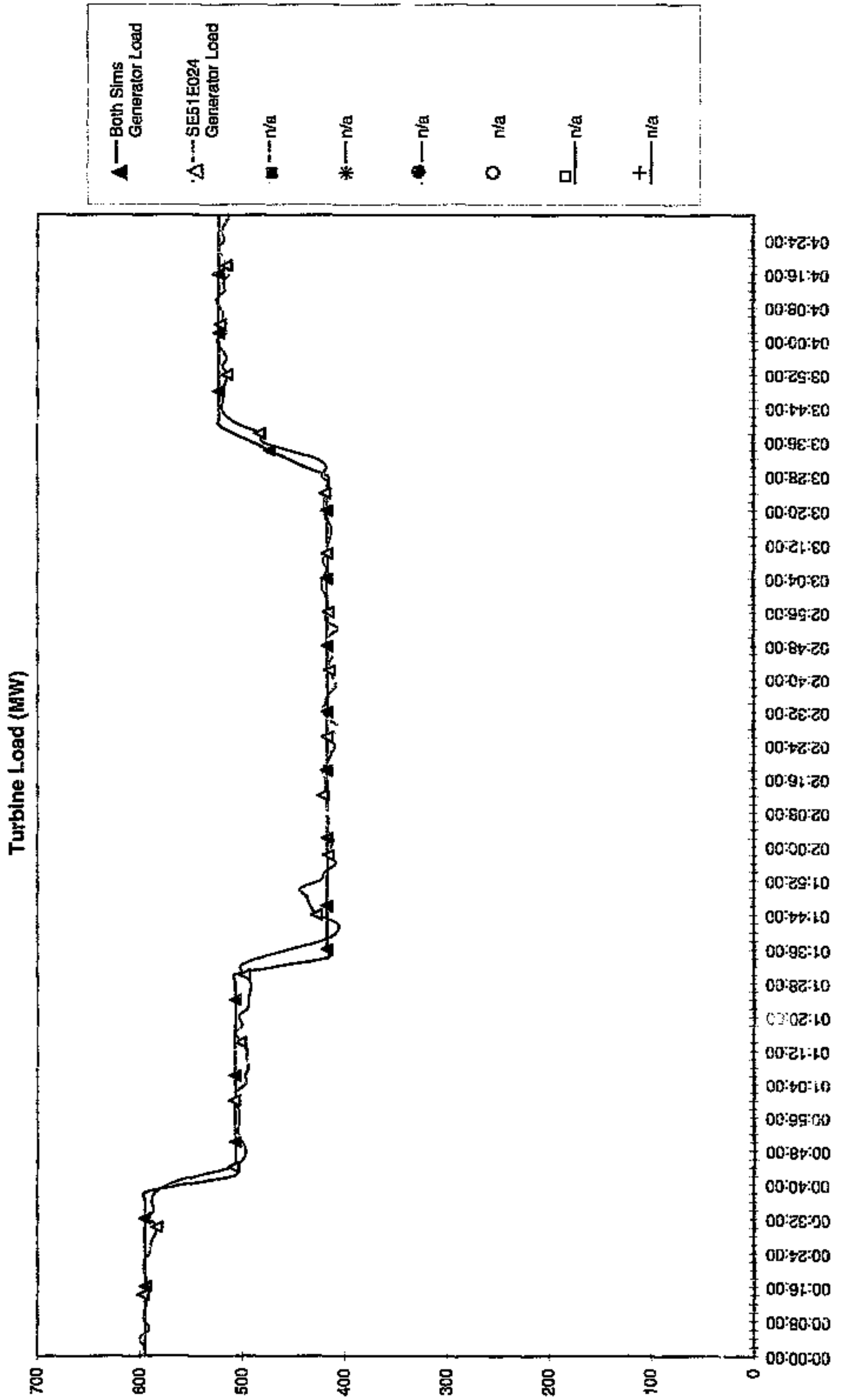
Test Chart 2-1: Load Dip - Simple Simulation



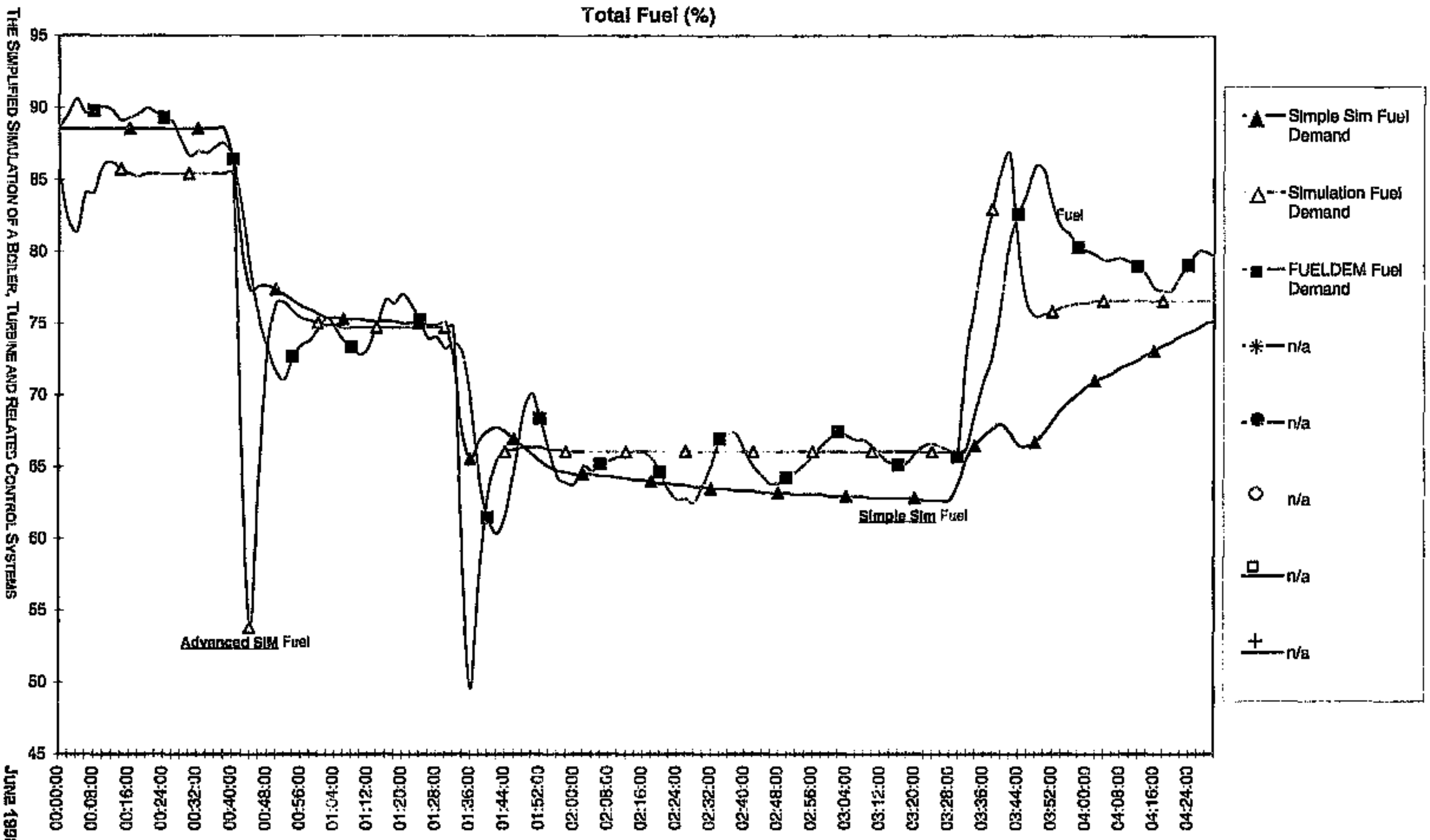
Test Chart 2-2: Load Dip - Simple Simulation



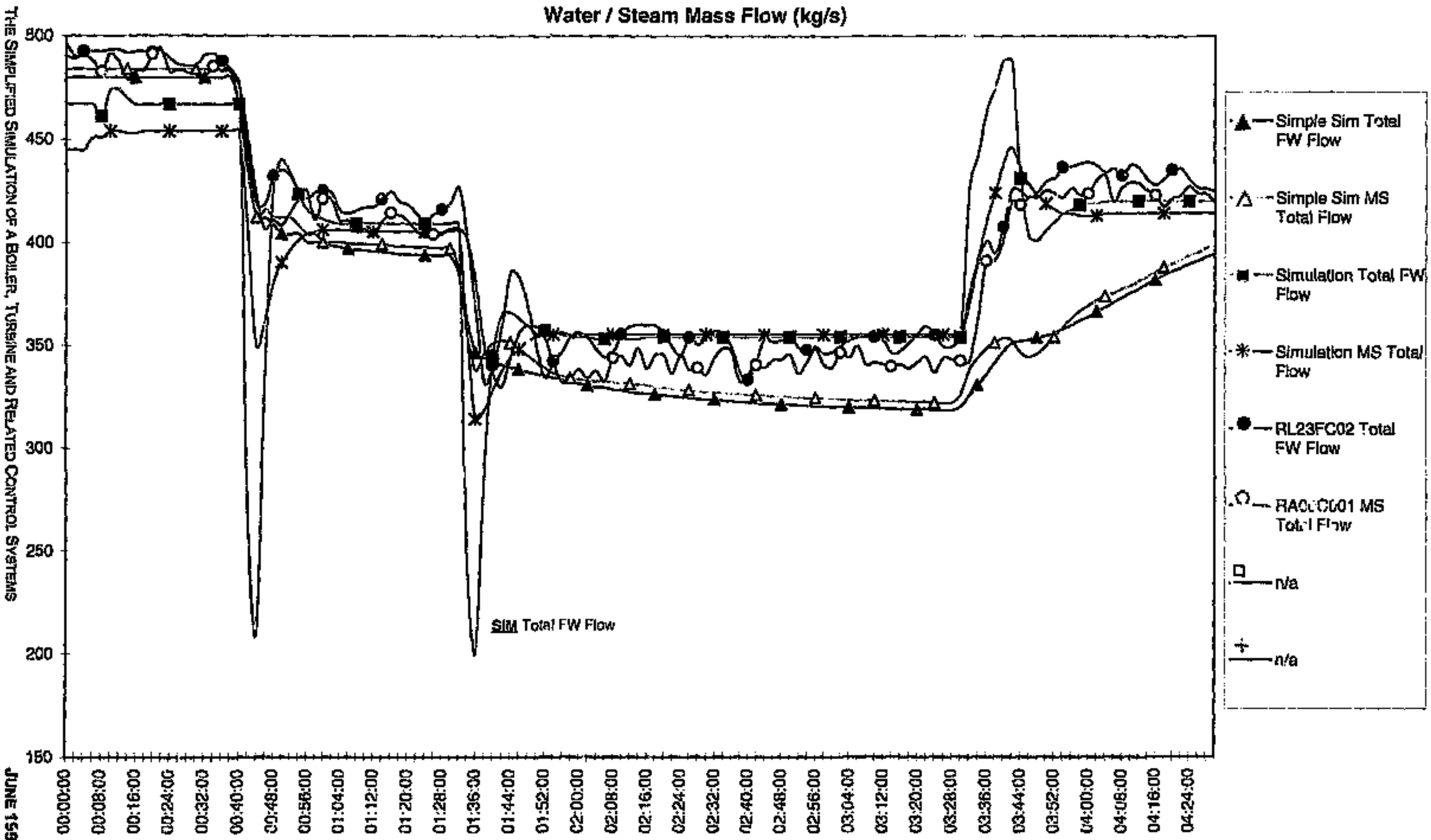
Test Chart 2-3: Load Dip - Simple Simulation



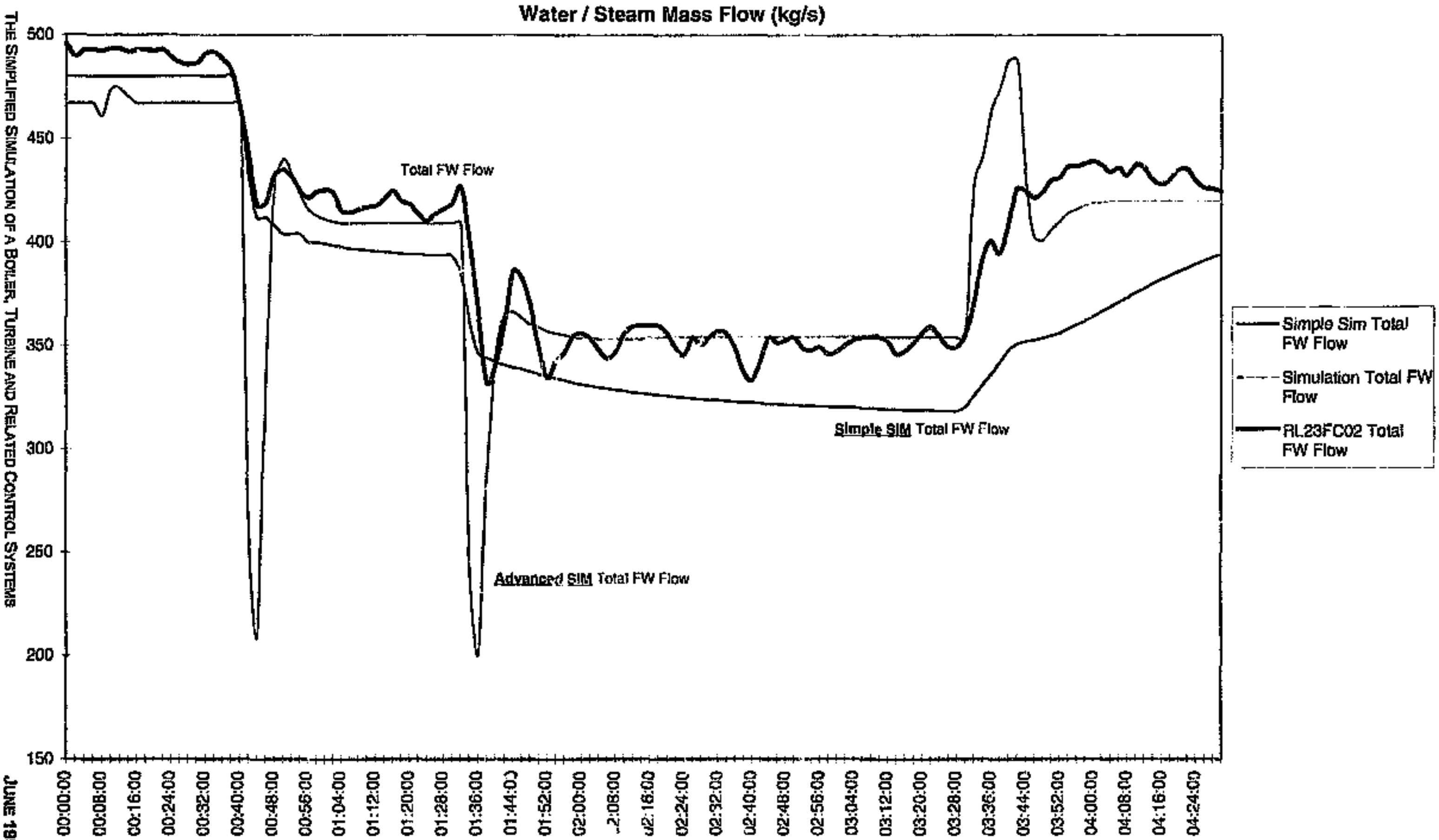
Test Chart 2-4: Load Dip - Simple Simulation



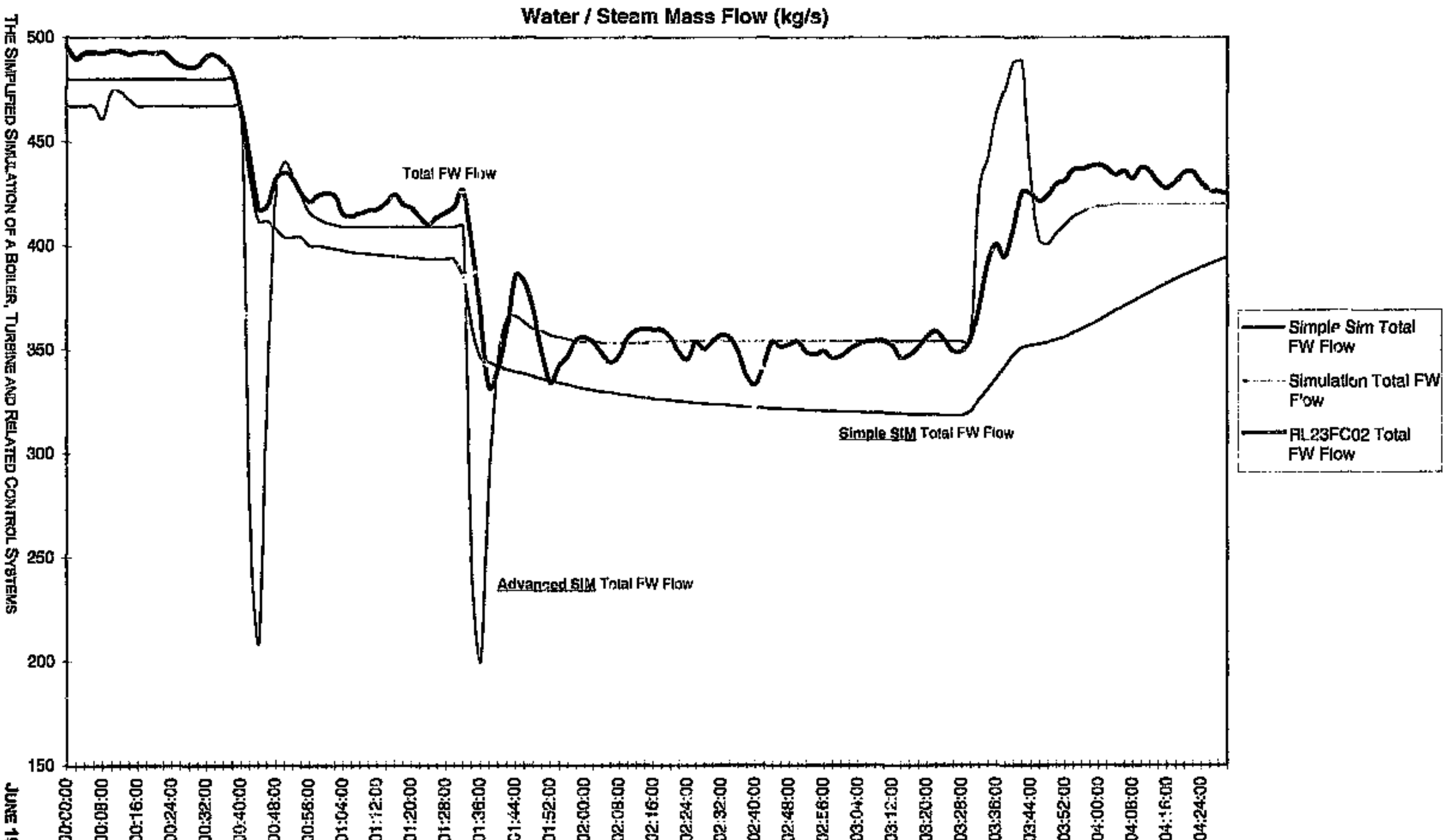
Test Chart 2-5: Load Dip - Simple Simulation



Test Chart 2-6: Load Dip - Simple Simulation



Test Chart 2-7: Load Dip - Simple Simulation



Test Chart 2-7: Load Dip - Simple Simulation

5.4 Test 3: Load Changes - Advanced Simulation

This test was performed to compare the response of the simple simulation to the same load fluctuation as used in the previous test involving the simplified power plant model.

5.4.1 Description of Test

This section compares the results of the advanced simulation with the same load dip as described in Section 5.3.1.

5.4.2 Test Results

5.4.2.1 Valve Positions

During the entire test all bypass valves remained closed. These valves would normally only operate under low load conditions, which, in the case of the Duvha generating unit, corresponds to loads below approximately 350MW.

Although the simulation of a load dip below 350MW would be very useful, it was not possible to perform such a test on the power generating unit due to the risks involved. In the absence of live data with which to compare such a simulation, no data has been included for a low load simulation.

With reference to Chart 3-1, the simulated valve positions show a strong correlation to the actual plant valve positions. Before the test, both the simulated and actual IP valves were above 95% open. The simulated and actual HP governor valves were at 30% and 45% open respectively before the load changes.

During the first load dip from 595MW to 505MW (00:30:00), the responses of the simulated and actual IP governor valves appear quite different. Both the simulated valve positions decreased considerably, especially the IP governor valve which closed from 95% to 45%. The difference in response between the simulated HP and IP valves is due to the different non-linear circuitry governing the positions of these two valves with respect to the load demand signal. Figure 5-1 shows that the IP governor only starts to close significantly after the demand drops below 25%, which corresponds to the HP governor position of 28% and an IP Governor position of approximately 78%.

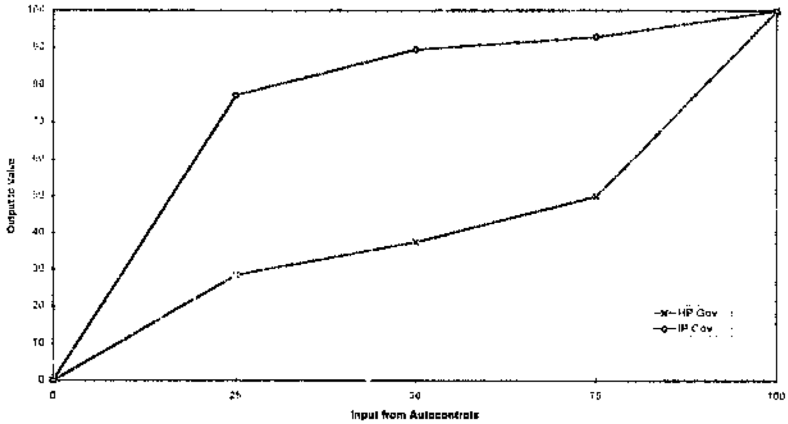


Figure 5-1: Non-Linear Drivers for HP and IP Governor Valves

Although the same non-linear responses were modeled in the simulation, it was found that during this test the simulated load demand signal dropped more than on the live plant (for the same load drop). This resulted in both HP and IP governors closing. The main cause of this is the inaccuracies in the flow models of the two governor valves. Although the flow characteristics were modeled according to the available flow data, the simulated valves reached 95% full flow at a lower % open value than the live valves.

The live plant data is not sufficient to enable a recalculation of the correct flow characteristics of these two valves.

5.4.2.2 Temperatures

Throughout the test the actual and simulated temperatures remained relatively constant (Chart 3-2). This is primarily due to the temperature control action of the superheater and reheater attemporation.

One unexpected result from this test are transients on the simulated and actual cold reheat temperatures which are approximately equal in magnitude but opposite in direction. These are only visible during load changes. These small deviations may be considered negligible when compared to other charts (e.g. flow transients, Chart 3-8) which show larger deviations.

5.4.2.3 Pressures

The pressure changes during the simulation compare very well with the test data in terms of both duration and magnitude (Charts 3-3 and 3-4). Although the boiler parameters were tuned as much as possible, the inter-relations between the physical boiler characteristics made it difficult to match the simulated pressures to the actual pressures. In order to illustrate the complexity of the problem, a partial list has been included of the main simulation parameters that contribute to the individual pressures in the boiler:

- BFPT discharge pressure

- Condenser vacuum
- Heat transfer characteristics (Econ, Evap, SH, RH)
- Flow Resistance Characteristics (Econ, Evap, SH, HP gov v/v, HP bypass, HP bypass v/v, HP cylinder, RH, RH bypass, IP gov v/v, IP bypass v/v, IP/LP cylinder)
- Flow characteristics (HP governor, IP governor, HP bypass, IP bypass)
- Control circuit gains
- Control circuit valve non-linear circuits (HP governor, IP governor)

These variables also affect all other aspects of the simulation, including the flow, attemporation, temperatures, valve positions, etc.

The volume of each of the elements (Econ, Evap, SH, HP gov v/v, HP bypass, HP bypass v/v, HP cylinder, RH, RH bypass, IP gov v/v, IP bypass v/v, IP/LP cylinder) only contribute to the dynamic response of the boiler pressures, and are thus not as important when considering the steady state response of the boiler to load changes or other tests.

The main discrepancy between the simulated results and the actual plant data is found on the IP governor differential pressure which, in the case of the simulation, is affected substantially more by the load changes than occurs on the actual plant. The reason for this is due to the differences in governor valve positions between the simulation and the actual plant. The high simulated IP governor differential pressure (relative to the test data) can be accounted for when comparing the relative valve positions between the actual data and the simulation.

Although Chart 3-3 shows that during the load increase there is no increase in actual IP pressures whereas the simulated IP pressures increased as a result of the load increase. This is due to the plant instrumentation which quantizes the pressures to a resolution of 0.5 MPa. Although no show on the graph, the actual pressures did increase during the load increase.

5.4.2.4 Total Fuel (Coal)

Chart 3-6 shows a strong correlation between the actual fuel data and the simulated data. The simulation fuel controls appear to be quicker than the plant response, resulting is control spikes immediately after the load increase/decrease. These transients may be attributed to untuned parameters in the simulation of the plant and control system and have also been observed during other tests. The steady state response of the simulation is accurate.

5.4.2.5 Feedwater / Steam Mass Flow

The comparisons between the feedwater/steam flows show very close correlation between the plant data and simulated data. As in the case of the fuel, the simulated results from Chart 3-7 and 3-8 show a good steady state response, with large control spikes during transients.

5.4.2.6 Spraywater Attemporator Flow

The simulated attemporation during load changes corresponds with the actual attemporation flow changes. In the case of the SH attemporation, the simulation matches the actual flow in terms of both proportion of change and duration. Once again the simulation produced larger than expected control spikes which have been noted on other graphs as being due to, amongst other things, untuned control parameters.

5.4.2.7 Enthalpy

Chart 3-10 is a graph showing the simulated enthalpy at each stage in the power generation cycle. No comparison has been made with actual plant data as the enthalpy values are not recorded by either the SCADA system or the autocontrols. This graph illustrates the effectiveness of the enthalpy controller at regulating the evaporator outlet enthalpy. The load independence of the boiler enthalpies (at steady state) is also show on this graph, where only the BFP and economizer enthalpies appear to be slightly affected by load changes.

5.4.3 Conclusions

The advanced generating unit model provides an accurate simulation of the responses to changes in unit load. Although not completely tuned, the simulated steady state responses correlate very well with the actual unit. The dynamic responses are too quick, but could be improved by correct tuning of the simulation parameters.

5.4.2.6 Spraywater Attemporator Flow

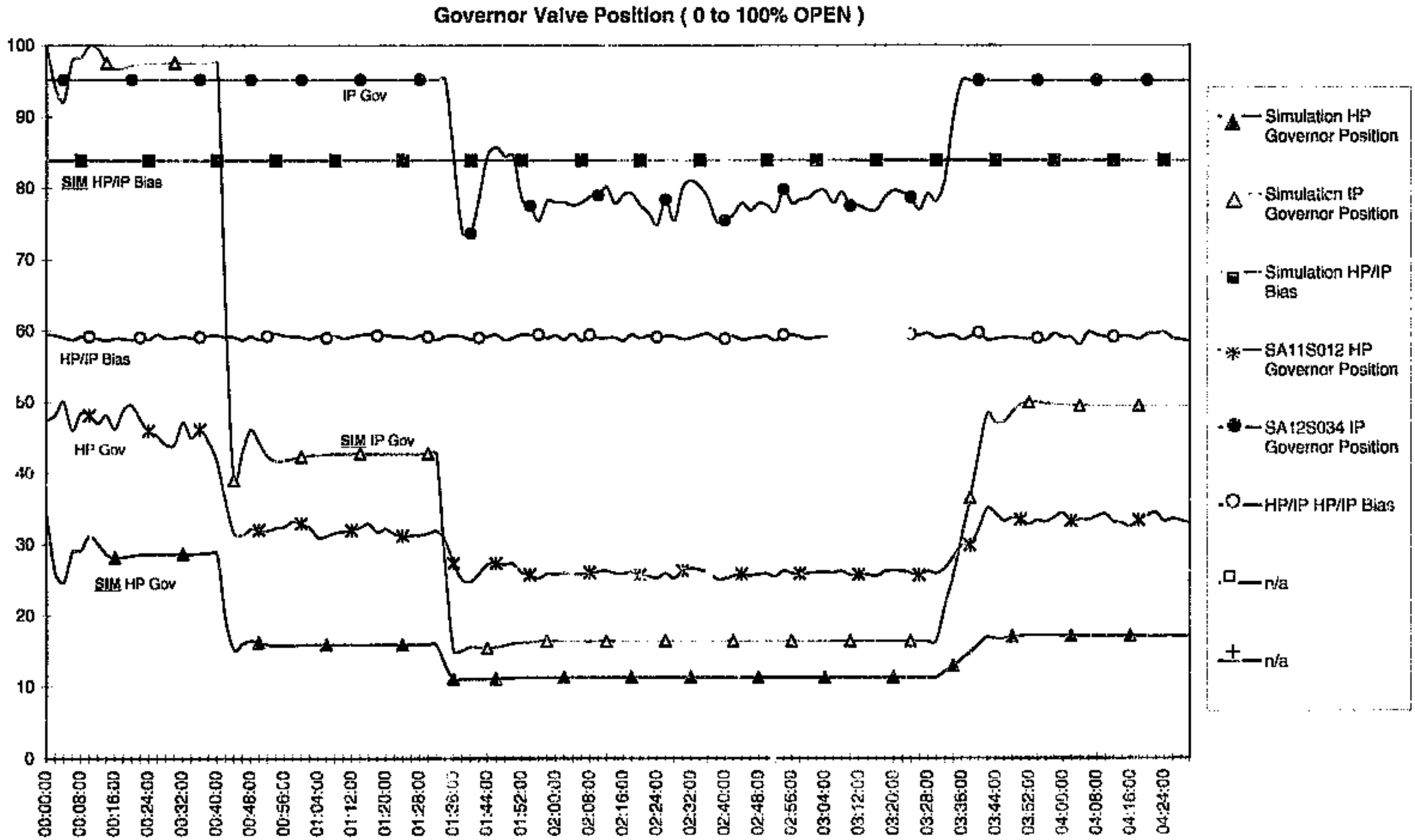
The simulated attemporation during load changes corresponds with the actual attemporation flow changes. In the case of the SH attemporation, the simulation matches the actual flow in terms of both proportion of change and duration. Once again the simulation produced larger than expected control spikes which have been noted on other graphs as being due to, amongst other things, untuned control parameters.

5.4.2.7 Enthalpy

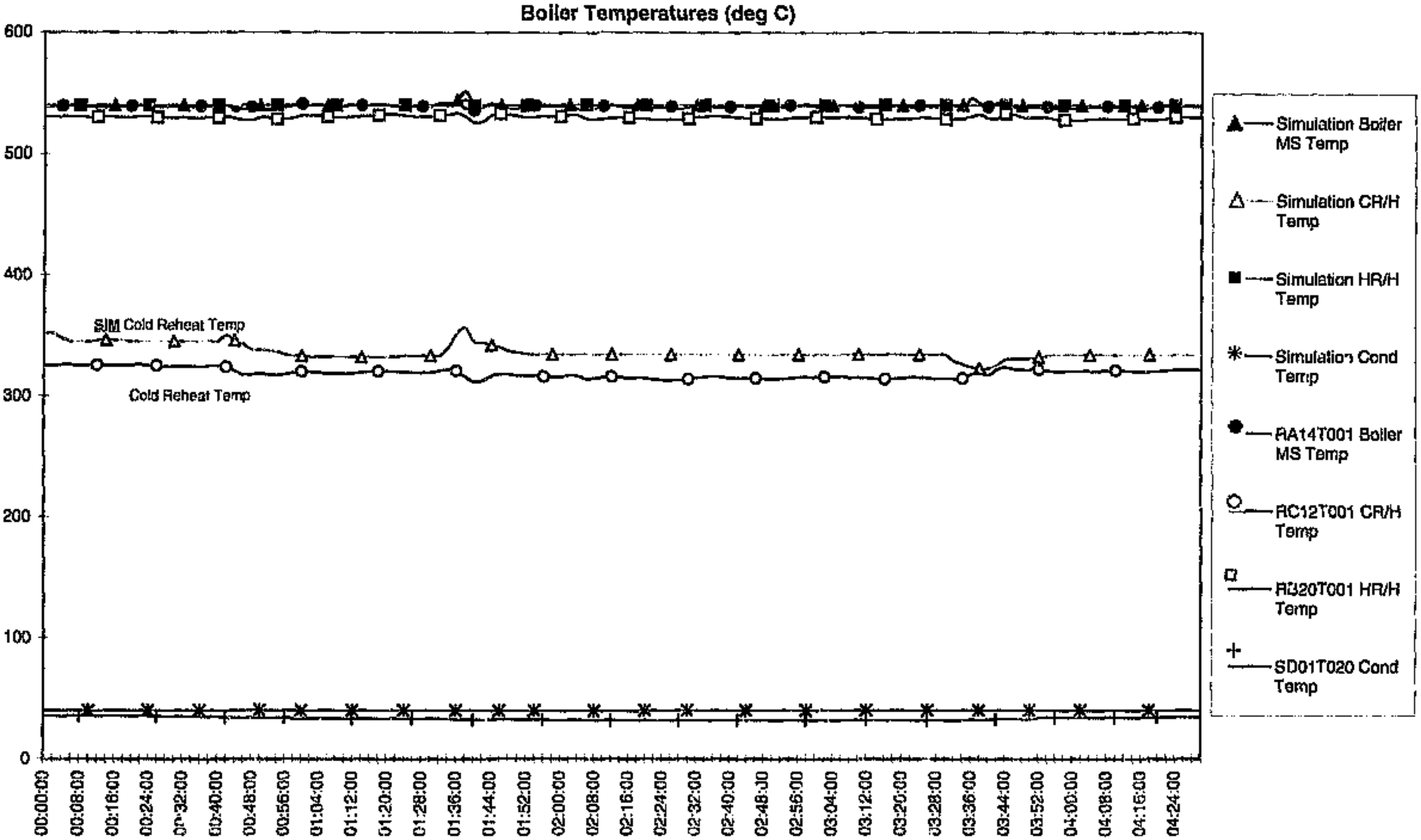
Chart 3-10 is a graph showing the simulated enthalpy at each stage in the power generation cycle. No comparison has been made with actual plant data as the enthalpy values are not recorded by either the SCADA system or the autocontrols. This graph illustrates the effectiveness of the enthalpy controller at regulating the evaporator outlet enthalpy. The load independence of the boiler enthalpies (at steady state) is also show on this graph, where only the BFP and economizer enthalpies appear to be slightly affected by load changes.

5.4.3 Conclusions

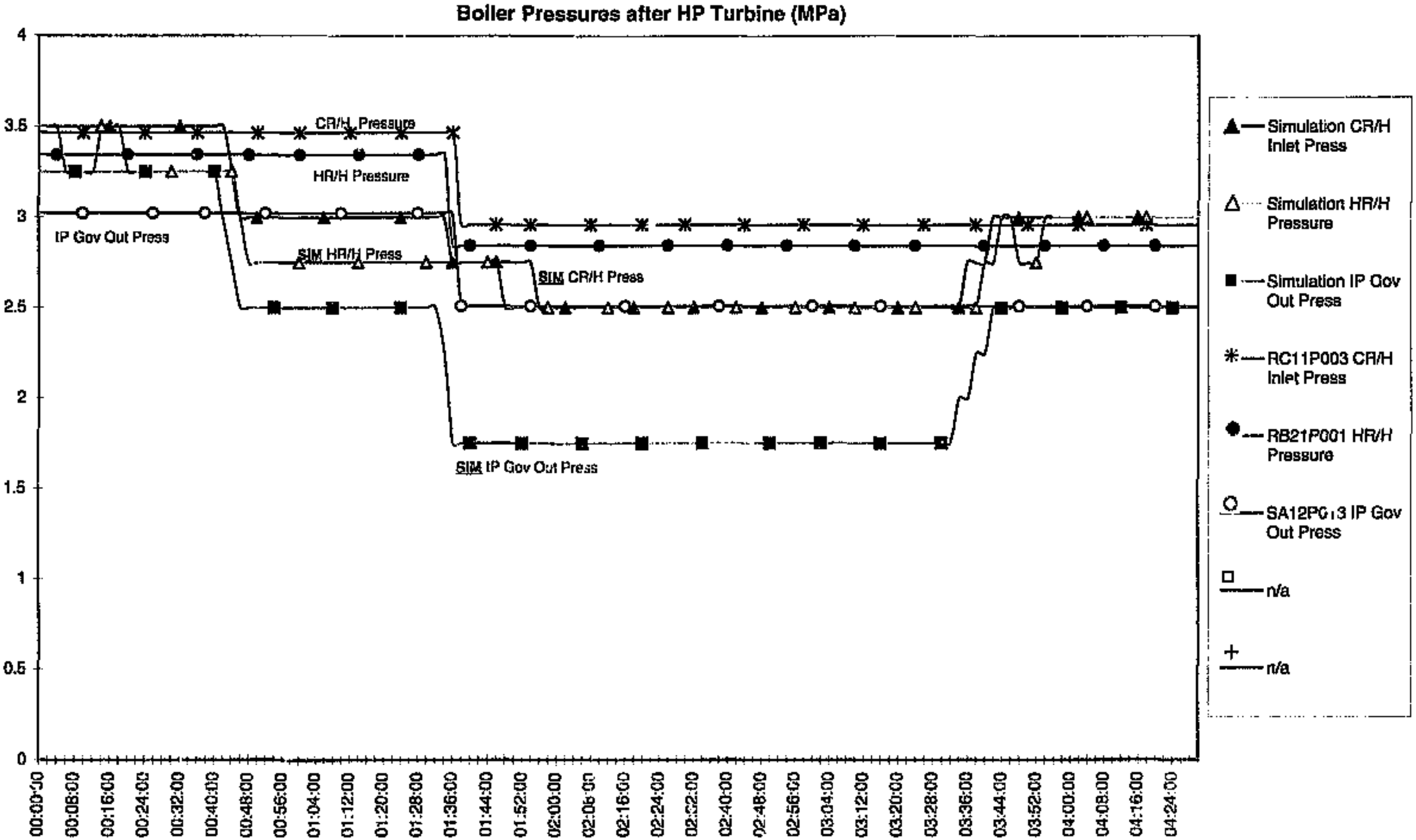
The advanced generating unit model provides an accurate simulation of the responses to changes in unit load. Although not completely tuned, the simulated steady state responses correlate very well with the actual unit. The dynamic responses are too quick, but could be improved by correct tuning of the simulation parameters.



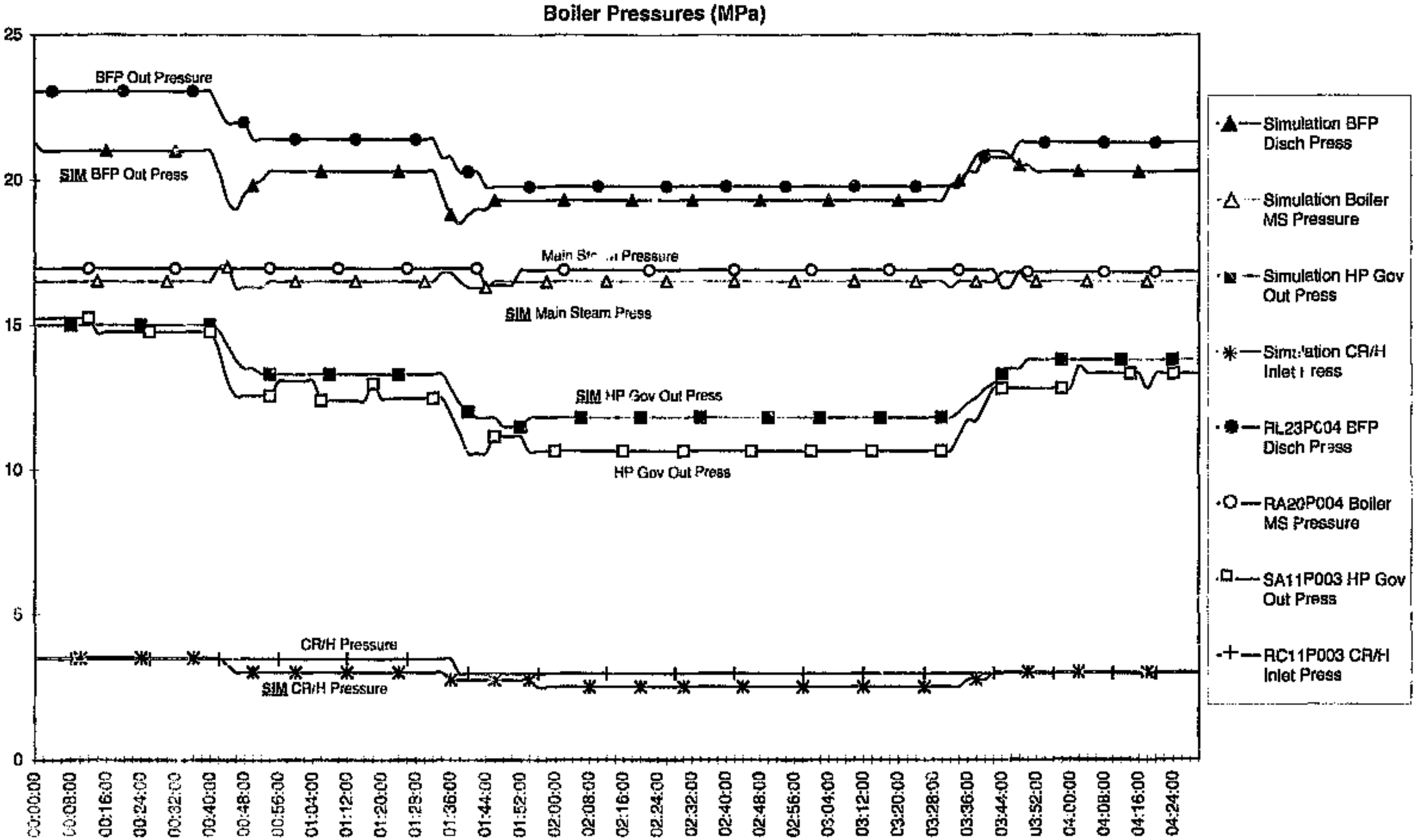
Test Chart 3-1: Load Changes - Advanced Simulation



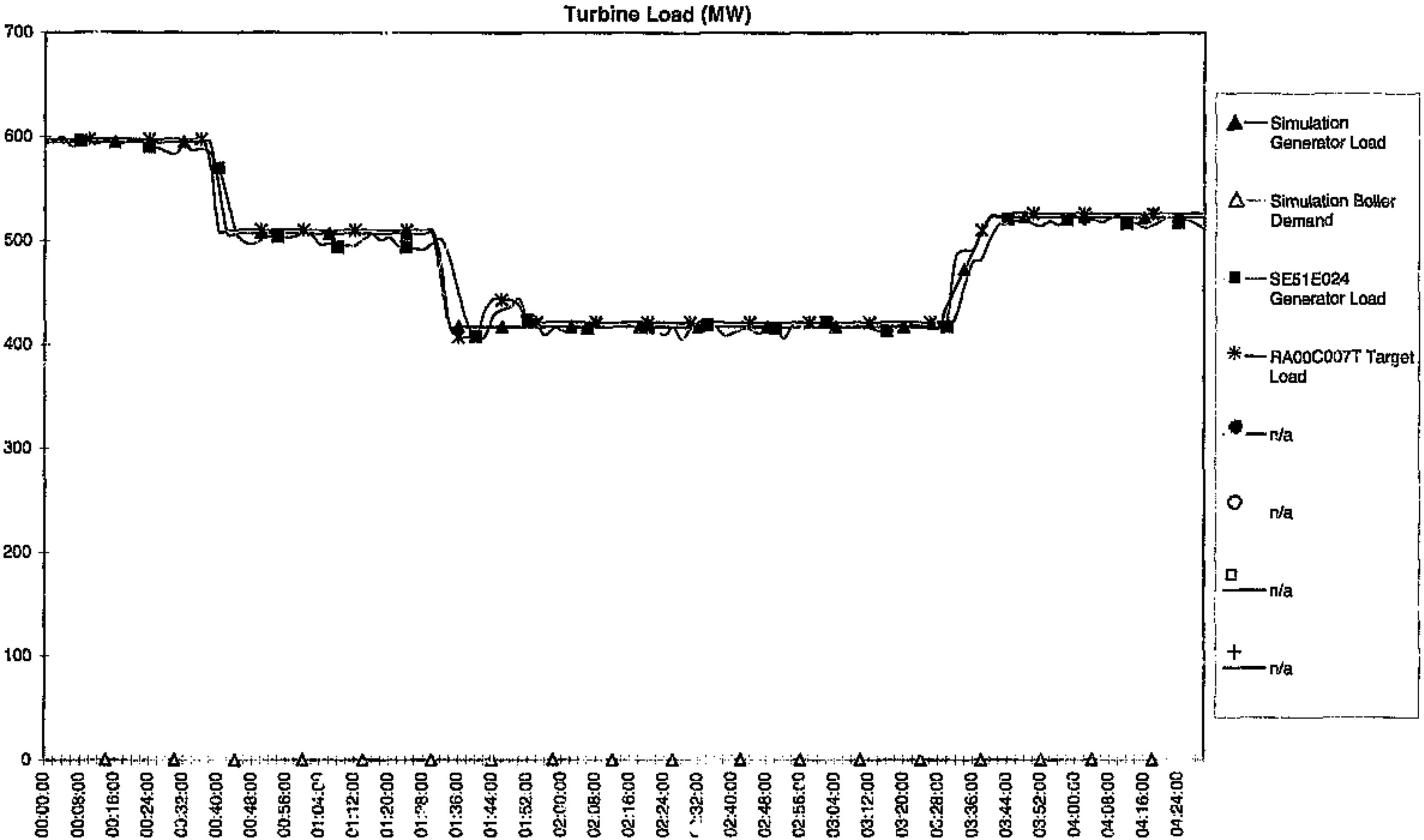
Test Chart 3-2: Load Changes - Advanced Simulation



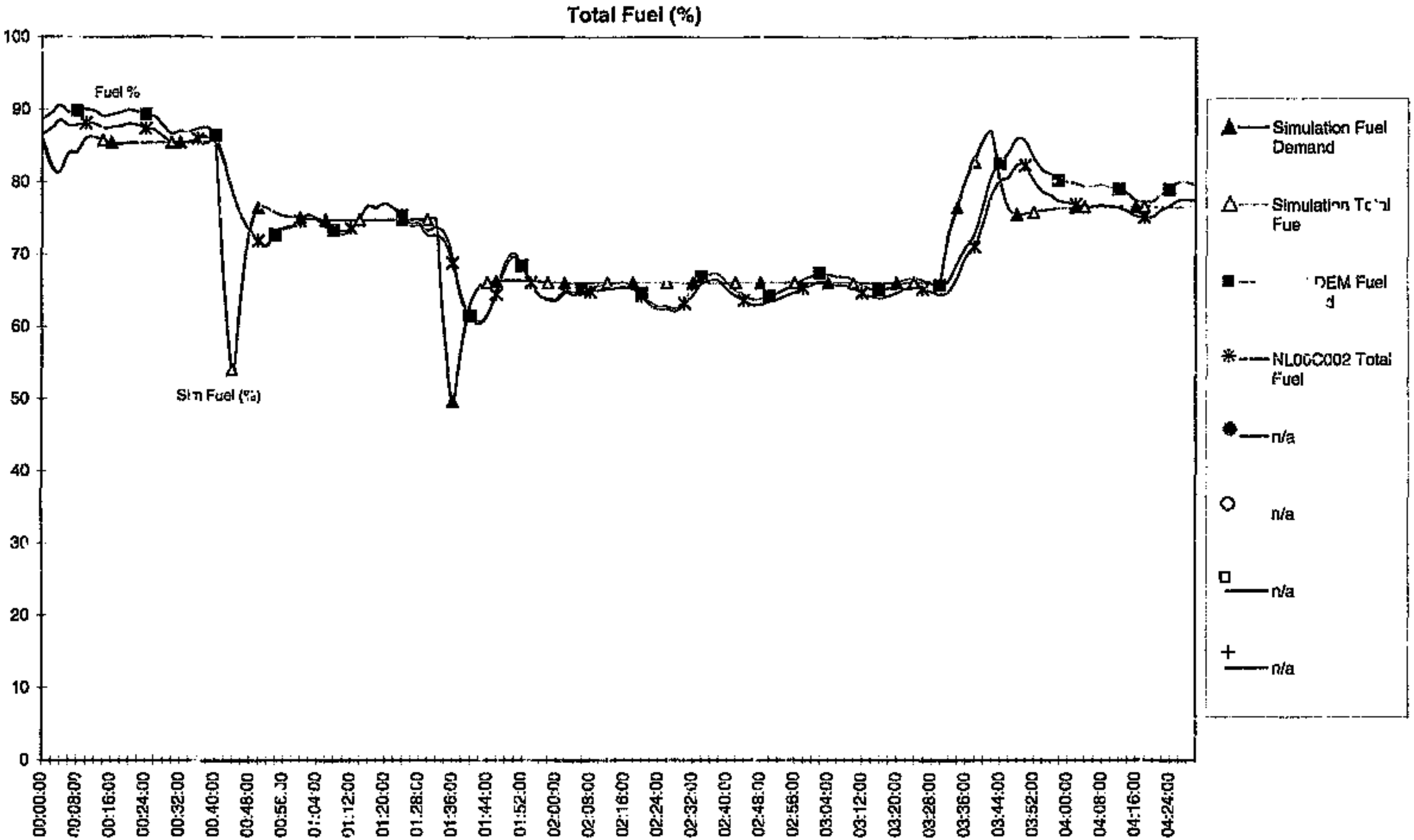
Test Chart 3-3: Load Changes - Advanced Simulation



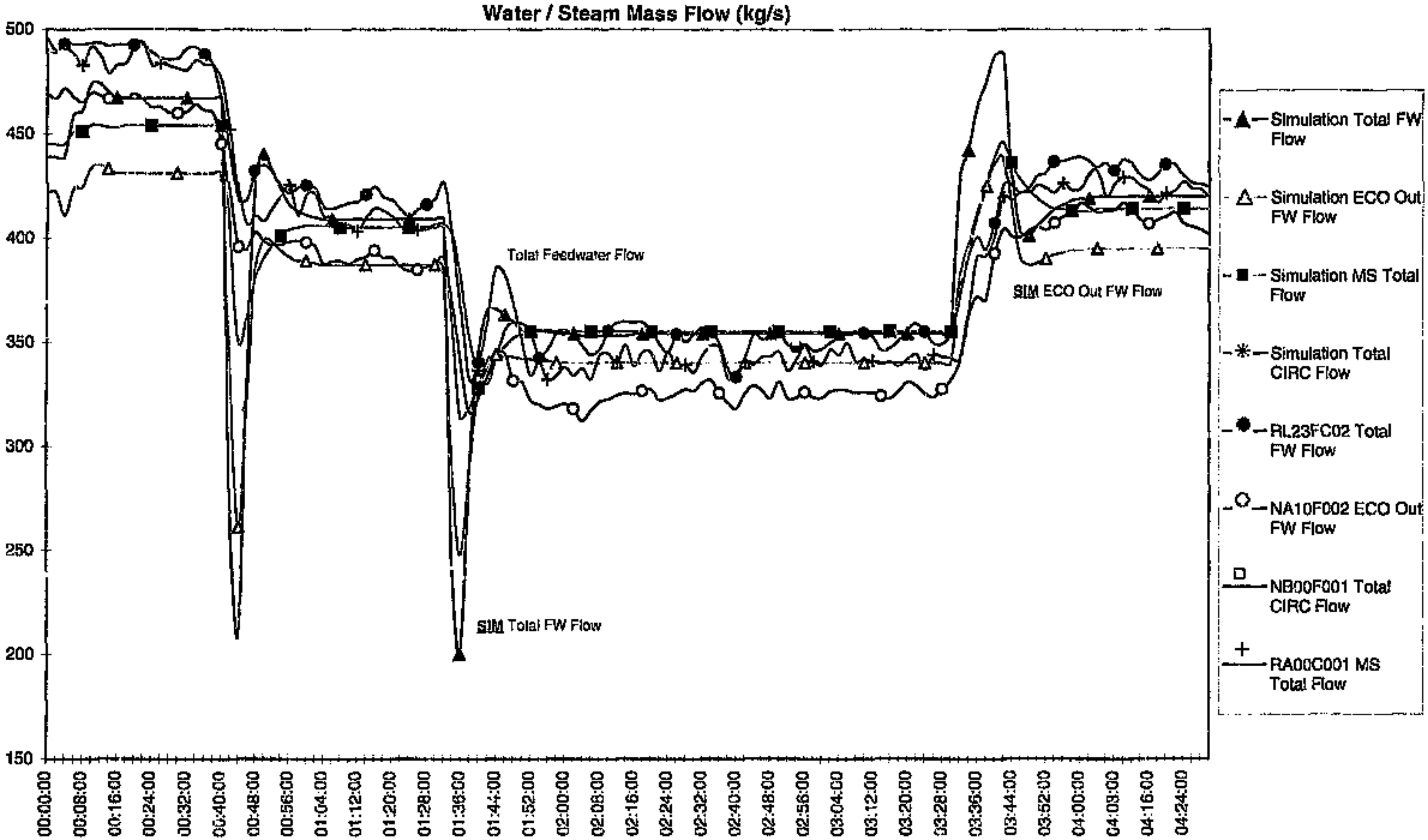
Test Chart 3-4: Load Changes - Advanced Simulation



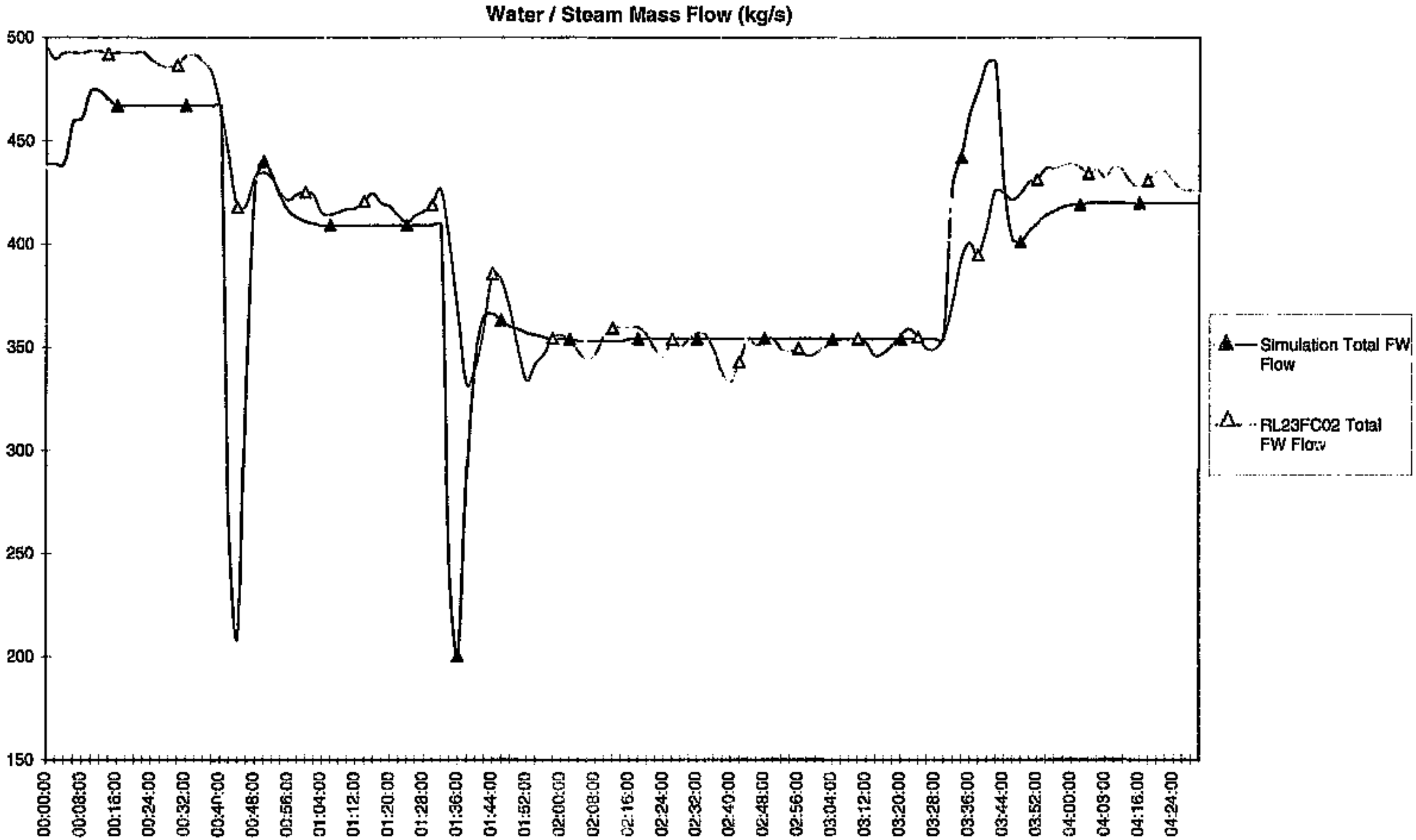
Test Chart 3-5: Load Changes - Advanced Simulation



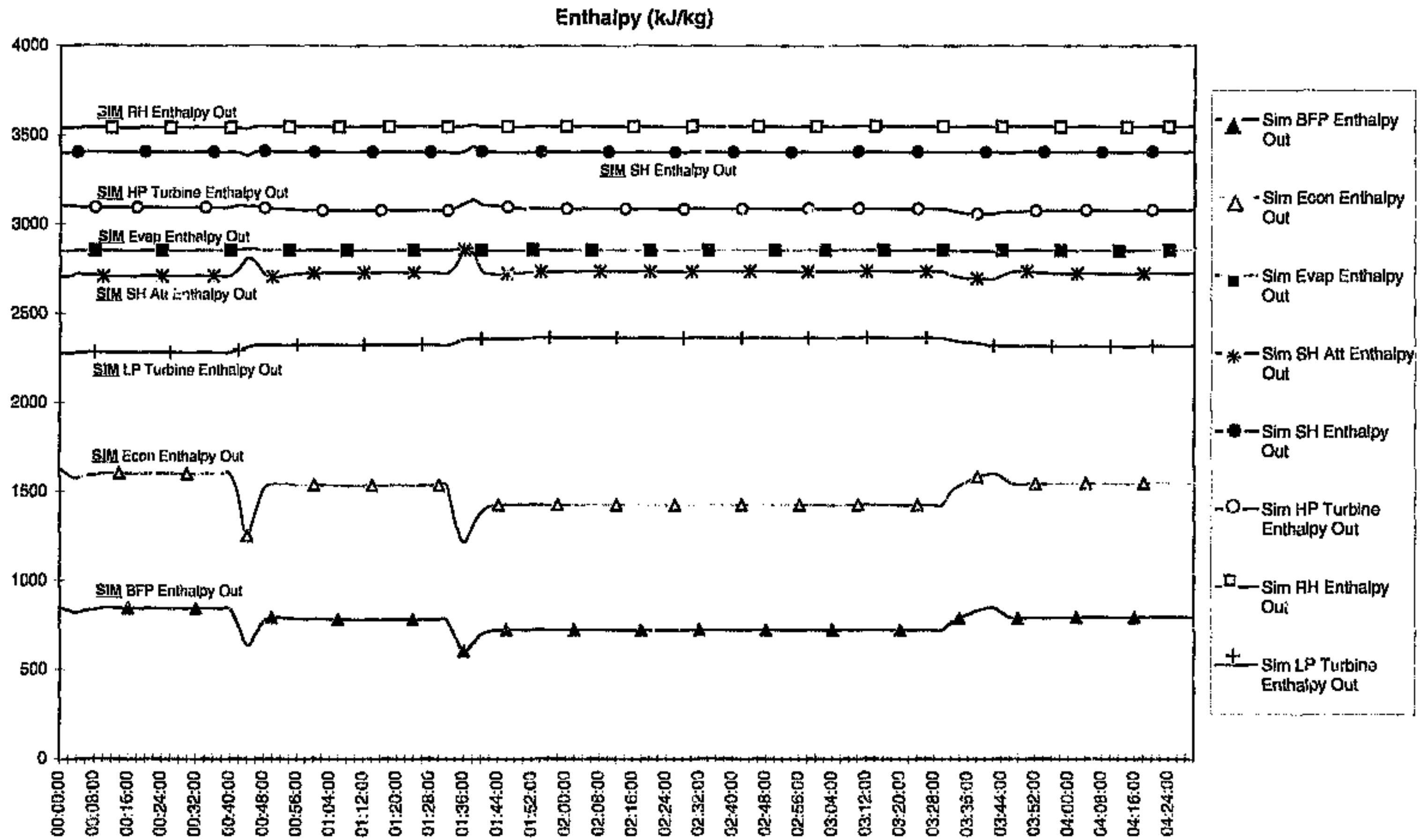
Test Chart 3-6: Load Changes - Advanced Simulation



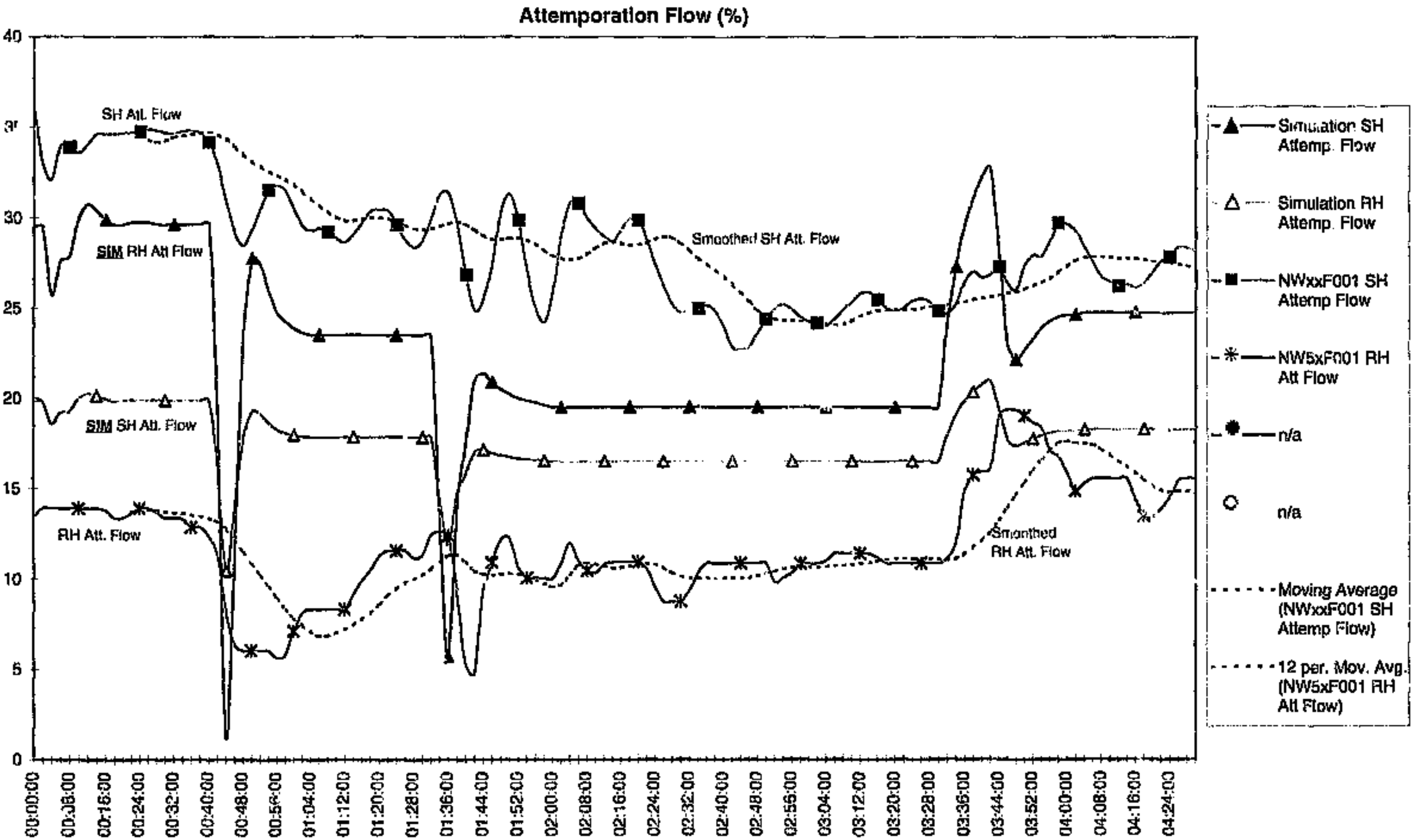
Test Chart 3-7: Load Changes - Advanced Simulation



Test Chart 3-8: Load Changes - Advanced Simulation



Test Chart 3-10: Load Changes - Advanced Simulation



Test Chart 3-9: Load Changes - Advanced Simulation

5.5 Test 4: Changing Mills - Advanced Simulation

5.5.1 Introduction

One of the variables which plays an important role in the steady state boiler temperatures, pressures, flows, and enthalpies is the positioning of the flame within the furnace area. Certain types of boilers make use of this property to control the boiler by tilting the burners upwards or downwards in order to shift the flame position. Although this is not possible at Duvha Power Station, it is possible to vary the flame position slightly by selecting certain combinations of burners at different heights in the boiler furnace area. Changes in air flow also affect the flame height.

The Duvha Power Station generating units normally run at full output capacity with five out of the six mills in operation. This allows the sixth mill to be taken out of service for maintenance or repairs, or alternatively ensures that there is a standby in the event of a mill failure or trip. When possible, the unit operators prefer not to use the uppermost row of burners (row A - refer to Figure 4-6) which are all supplied with pulverized fuel from Mill A. The use of these burners causes the flame to move further into the superheat region and away from the evaporator region, which results in a decrease in the evaporator enthalpy. The evaporator enthalpy controller corrects this by decreasing the feedwater flow, which also has a knock-on effect throughout the rest of the boiler in terms of the temperatures, pressures and enthalpies. This leads to operation outside the optimum control range which results in less stable operation of the controllers.

5.5.2 Description of Test

The objective of this test is to monitor the responses of the actual and simulated boiler to a change in the burner firing positions. During this test mill E, which feeds the bottom row of burners, was replaced by mill A (top row of burners). These two rows were chosen to maximize the effect on the flame position caused by the change in firing positions.

Test:	**Mill Change on Actual Unit**
Date:	16/04/97
Unit:	3
Operator:	Mr. Piet Krieger
Start Time:	13h20
Duration:	34 minutes
Average Unit Load:	595 MW

Time 00:00:00 - Initial Conditions	
Mills in Service:	D, C, F, B, E
Mills on Standby:	A
Generating unit & Autocontrols:	Stable
Time 00:10:45 - Stop Coal Feeder to Mill D	
Mills in Service:	D, C, F, B
Mills in Purge Cycle:**	E
Mills on Standby:	A
Time 00:14:00 - Open Primary and Secondary Air Dampers on Mill A	
Mills in Service:	D, C, F, B
Mills in Purge Cycle:**	E
Mills in Startup:***	A
Time 00:16:30 - Start Mill A, Complete Mill E Purge	
Mills in Service:	A, D, C, F, B
Mills on Standby:	E
Time 00:33:00 - Final Conditions	
Mills in Service:	A, D, C, F, B
Mills on Standby:	E
Generating unit & Autocontrols:	Stable

** : During this cycle, all pulverized fuel is purged from the mill by stopping the coal feeder while leaving the primary and secondary air dampers open.

***: During mill startup the primary air and secondary air dampers are opened prior to starting the mill feeder. This is done to heat up the mill to the correct operating temperature.

The simulation does not include any air flows and is thus not able to mimic the operation of the mills during the purge cycle and during startup. For this reason the simulation differs slightly from the actual test performed on the unit.

Test:	**Simulated Mill Change**
Duration:	34 minutes
Unit Load:	600 MW

Time 00:00:00 - Initial Conditions	
Mills in Service:	D, C, F, B, E
Mills on Standby:	A
Generating unit & Autocontrols:	Stable
Time 00:12:15 - Stop Mill E	
Mills in Service:	D, C, F, B
Mills on Standby:	A, E
Time 00:12:15 - Start Mill A	
Mills in Service:	A, D, C, F, B
Mills on Standby:	E
Time 00:00:00 - Initial Conditions	
Mills in Service:	A, D, C, F, B
Mills on Standby:	E
Generating unit & Autocontrols:	Stable

5.5.3 Test Results

5.5.3.1 Load

During the test the actual load remained relatively constant, with only a slight increase within the last 4 minutes. Due to the constraints imposed while performing the tests, it was not possible to clamp the actual boiler load at a fixed value.

A constant load of 600MW was used as the input to the simulation.

5.5.3.2 Fuel

The simulated and actual total fuel flows remained relatively constant during the entire test. The actual value was found to be slightly less stable than the simulated fuel.

5.5.3.3 Pressures

The boiler pressures remained constant throughout the test. The simulated pressures are comparable with the actual pressures recorded during the test.

5.5.3.4 Temperatures

The simulated and actual boiler temperatures remained constant throughout the test. This is due to the action of the temperature controllers (SH attemporation control, RH attemporation control, and Enthalpy controller). With these temperature controllers deactivated, the action of changing the firing position would have a direct influence on the boiler temperatures and would be visible on the graphs.

5.5.3.5 Governor and Bypass Valve Positions

The governor and bypass valve positions remained constant throughout the simulation. This is attributed to the constant load, constant pressures and constant temperatures throughout the test.

Both the simulated and actual bypass valves remained shut throughout the test. This is normal when operating above 350MW.

5.5.3.6 Attemporation

A. Actual Results

Before commencing the test the boiler was allowed to stabilize. At time 00:10:45 the Mill E coal feeder was stopped. The result of this is that the fuel demand to the other 4 in-service mills (D, C, B, F) increases to make up for the loss of Mill E Even after stopping the feeder to mill E, there is sufficient coal stored in the mill and ducting to supply fuel to the furnace for some time. This causes an increase in the total amount of fuel that is supplied to the furnace until mill E is completely purged. Although this increase is not recorded on the 'Total Fuel' graph (Chart 4-3), it manifests as an increase in SH and RH steam temperatures. Stopping the mill E feeder also causes the flame position to move upwards in the boiler furnace, leading to a decrease in the evaporator enthalpy. This causes the enthalpy controller to reduce the feedwater flow through the boiler. This reduced flow also causes an increase in the SH and RH temperatures.

The increase in SH and RH temperatures causes the attemporation controllers to increase the SH and RH attemporation flow. The increase of the SH attemporation results in an increase in the flow through the RH, which thus helps to bring down the RH temperature. This is shown on Chart 4-3 by the fact that the RH attemporation increased by a small amount (approximately 2 %) whereas the SH attemporation increased by approximately 21% (from 27% to 48%).

By the time that mill A was placed into 'Startup Mode' (at time 00:14:00) most of the fuel would have been purged from mill E. This reduction in fuel supplied to the furnace causes a decrease in the SH attemporation. This is complemented by the action of opening the primary and secondary air dampers to the top row of burners (mill A on Startup Mode). The additional air at the top of the furnace area cools the furnace in the SH and RH area as well as reducing the heat exchange due to the increased flow rate of the flue gas over the heat exchange surfaces. The graph shows a 11 % reduction in SH attemporation and a 2 % reduction in RH attemporation.

At time 00:16:30 mill E was taken out of service completely and mill A feeder started. The fuel supply from mills B,C,D and F were decreased slightly by the autocontrols to maintain the total fuel supplied to the furnace at a constant value. The result of the additional mill at the top of the furnace and the decrease in the firing of the lower mills is that the firing position moves higher up, causing a further large increase in SH attemporation (approximately 20%). This additional SH attemporation causes the RH temperatures to drop, resulting in a drop in RH attemporation.

B. Simulation Results

Although the simulation involved less steps than the actual test on the live unit, the results are similar to those recorded during the test. After stopping the lowest mill (E) the SH attemporation increased from 17% to 31% (approximately 14%) while the RH attemporation decreased very slightly. After starting the top mill (mill A), the simulation shows a further increase in SH attemporation (approximately 14%) with a corresponding decrease in RH attemporation (approximately 1%).

Although the simulated and actual RH attemporation do not follow similar responses during the test, they are comparable if only the initial and final values are compared.

The difference in SH attemporation graphs is caused by the simplified model of the mills which does not include the primary and secondary air. Ignoring the intermediate SH values, the simulated and actual attemporation responses are comparable.

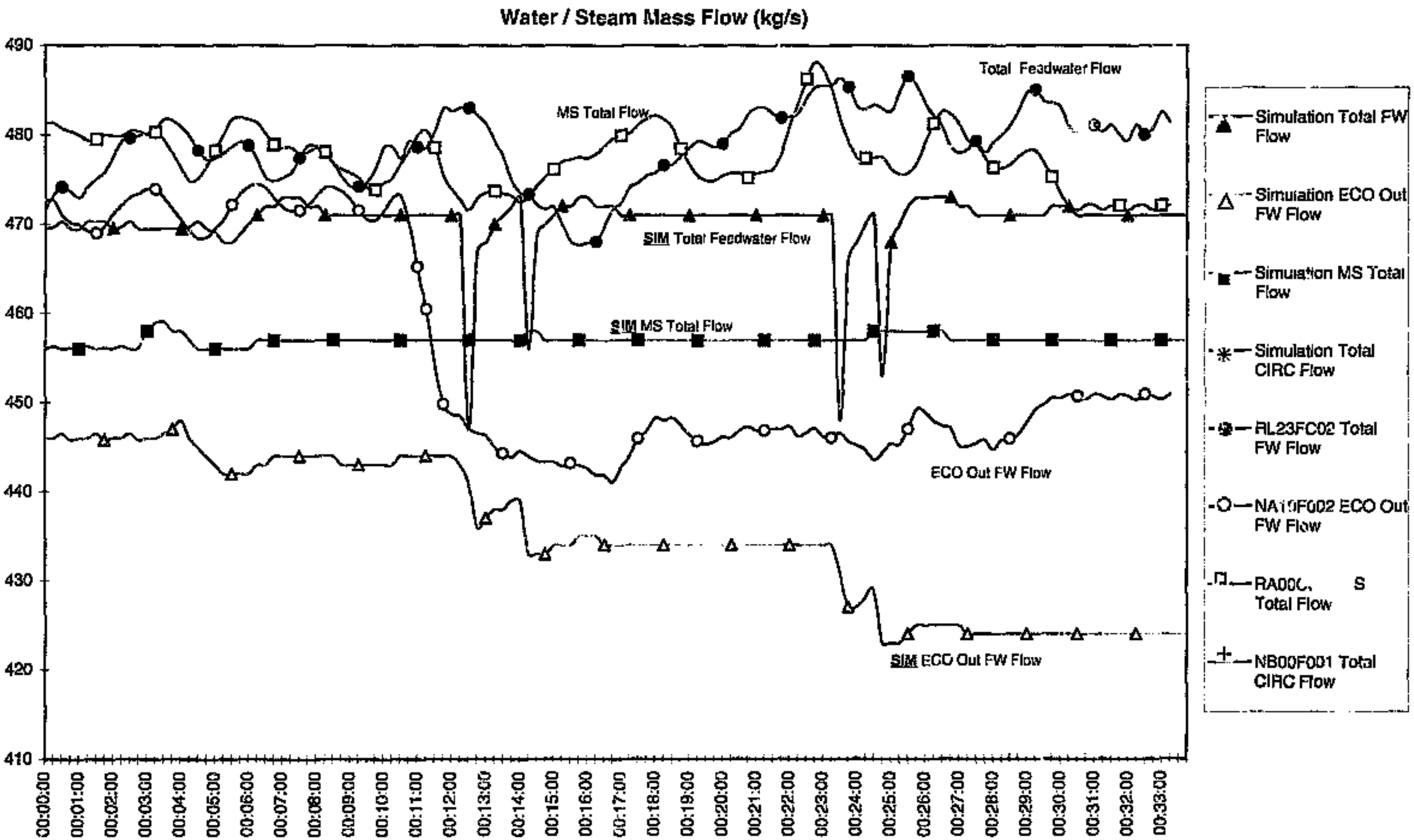
5.5.3.7 Flow

Although the flow results are more difficult to analyze, the similarity between the simulated and actual economizer outlet flow is immediately apparent. The simulated economizer outlet flow is shown to drop both when mill E is taken out of service and again when mill A is placed in service. The reasons for this have been explained in Section 5.5.3.6 above. The actual economizer outlet flow is seen to decrease only once when mill E is placed in the purge cycle.

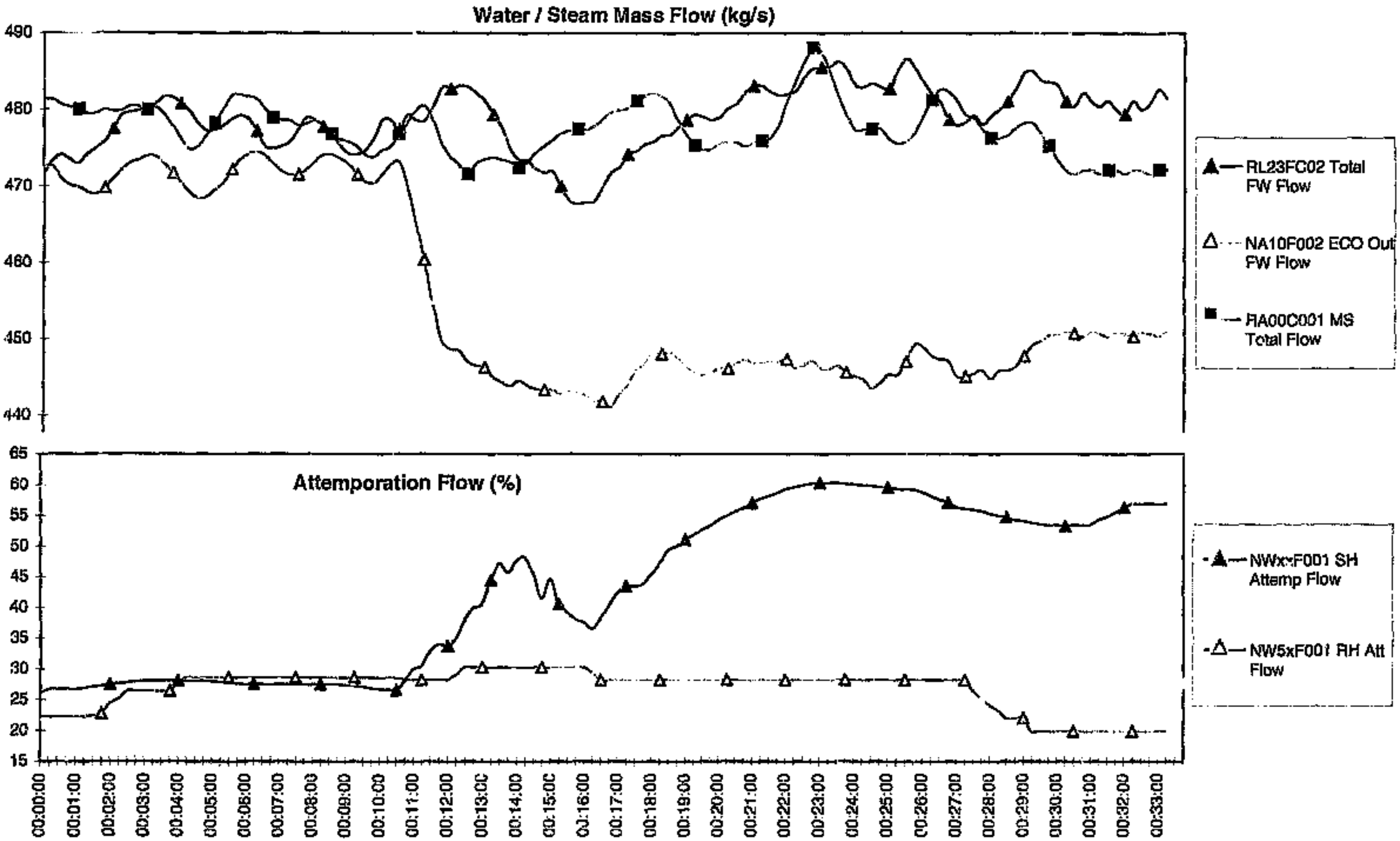
The simulated Total Feedwater Flow and Main Steam Flow remained constant throughout the simulation, except for control spikes during the mill changes. This does not correspond with the actual test results which show the Main Steam Total Flow to decrease slightly and the Total Feedwater Flow to increase slightly over the duration of the test.

5.5.4 Conclusions

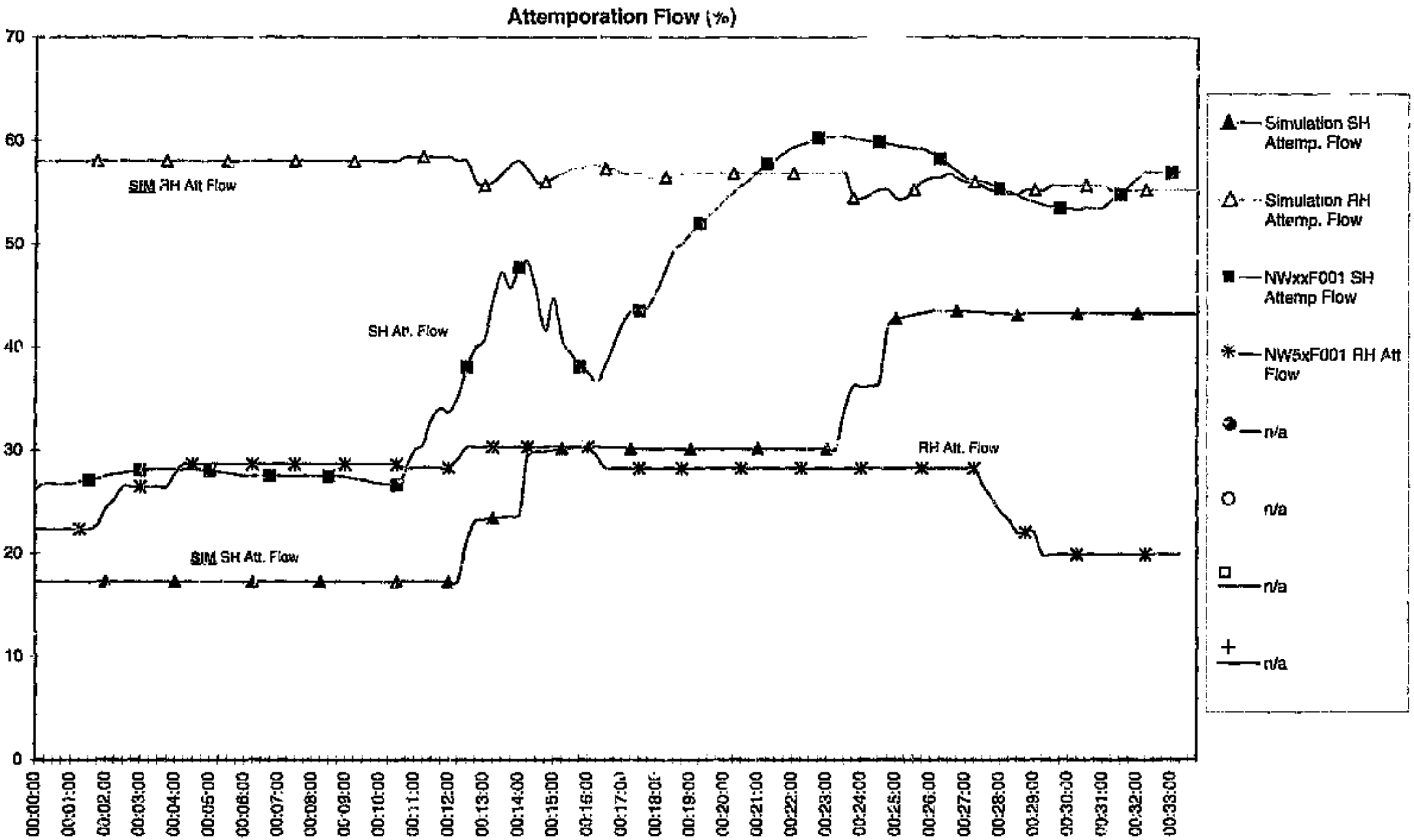
The simulation shows good correlation with the actual test results, especially the steady state response resulting from the mill change. The response of the simulated RH attemporator is far smaller in magnitude to the actual response.



Test Chart 4-1: Mill Change - Advanced Simulation



Test Chart 4-2: Mill Change - Advanced Simulation



Test Chart 4-3: Mill Change - Advanced Simulation

5.6 Test 5: Decrease in Excess Air - Advanced Simulation:

5.6.1 Introduction: Importance of Excess Air

In an ideal situation, the combustion of coal (carbon) in the presence of oxygen folk vs the following formula:

$$C + 2O \rightarrow CO_2 \qquad \text{Eq 42}$$

If too much oxygen is supplied, after all the carbon reacts with the oxygen a certain amount of oxygen remains unburned. If, on the other hand, too little oxygen is supplied for the reaction, a certain number of the carbon atoms would react with only one oxygen atom to form carbon monoxide according to the following formula:

$$C + O \rightarrow CO \qquad \text{Eq 43}$$

Apart from producing a poisonous gas, this reaction also releases less energy than the previous reaction.

In the furnace of a generating unit it is important to limit the amount of air supplied to the combustion process as this causes cooling of the furnace. It is also important to achieve complete combustion according to equation 42 to ensure that the maximum energy is released from the coal.

In an ideal furnace, the optimal conditions would be a 2:1 ratio of oxygen/carbon, which would result in complete combustion without any unnecessary excess air. This could easily be achieved by controlling the amount of air supplied to the furnace so as to ensure that the flue gas oxygen content remain slightly above zero.

In practice, however, the combustion process inside a boiler is far from ideal. This is illustrated as follows:

Property	Ideal Boiler Furnace	Real Boiler Furnace
Combustion Time	infinite	limited
Particle Size	atomic size	varies from 10um to 1mm
Mixing of Carbon and Oxygen	complete	partial
Combustion Temperature	high throughout furnace	high in center, low toward the outsides

Table 5-1: Requirements for Ideal Combustion

5.6 Test 5: Decrease in Excess Air - Advanced Simulation:

5.6.1 Introduction: Importance of Excess Air

In an ideal situation, the combustion of coal (carbon) in the presence of oxygen follows the following formula:

$$C + 2O \rightarrow CO_2 \qquad \text{Eq 42}$$

If too much oxygen is supplied, after all the carbon reacts with the oxygen a certain amount of oxygen remains unburned. If, on the other hand, too little oxygen is supplied for the reaction, a certain number of the carbon atoms would react with only one oxygen atom to form carbon monoxide according to the following formula:

$$C + O \rightarrow CO \qquad \text{Eq 43}$$

Apart from producing a poisonous gas, this reaction also releases less energy than the previous reaction.

In the furnace of a generating unit it is important to limit the amount of air supplied to the combustion process as this causes cooling of the furnace. It is also important to achieve complete combustion according to equation 42 to ensure that the maximum energy is released from the coal.

In an ideal furnace, the optimal conditions would be a 2:1 ratio of oxygen/carbon, which would result in complete combustion without any unnecessary excess air. This could easily be achieved by controlling the amount of air supplied to the furnace so as to ensure that the flue gas oxygen content remain slightly above zero.

In practice, however, the combustion process inside a boiler is far from ideal. This is illustrated as follows:

Property	Ideal Boiler Furnace	Real Boiler Furnace
Combustion Time	infinite	limited
Particle Size	atomic size	varies from 10um to 1mm
Mixing of Carbon and Oxygen	complete	partial
Combustion Temperature	high throughout furnace	high in center, low toward the outsides

Table 5-1: Requirements for Ideal Combustion

In order to minimize the effect of these properties of a real boiler on the combustion process, it is necessary to supply a certain percentage of excess air to ensure optimal combustion. This optimal amount of excess air is measured as a percentage of oxygen content in the flue gas. In the case of Duvha Power Station, the optimal amount of excess air occurs when the oxygen content of the flue gas is approximately 2 to 4%. Increasing the amount of excess air reduces the efficiency of the unit by increasing the cooling effect of the air, whereas decreasing the excess air also causes a reduction of boiler efficiency due to incomplete combustion.

5.6.2 Description of Test

This test involved decreasing the excess the air from 100% to 85% after which the unit was allowed to stabilize. A setting of 100% indicates optimal value.

Although the actual boiler load remained constant at approximately 575MW, the simulation was performed at a constant load of 600MW.

Test:	**15 % decrease in Excess Air**
Date:	16/04/97
Unit:	3
Operator:	Mr. Piet Krieger
Start Time:	14h45
Duration:	09 minutes 30 seconds
Average Actual Unit Load:	575 MW
Simulation Load:	600 MW

Time 00:00:00 - Initial Conditions	
Excess Air:	100%
Generating unit & Autocontrols:	Stable
Time 00:00:16 - Decrease Excess Air by 15%	
Excess Air:	85%
Time 00:09:30 - Final Conditions	
Excess Air:	85%
Generating unit & Autocontrols:	Stable

5.6.3 Test Results

5.6.3.1 Load

The actual boiler load remained constant within 5% of 575 MW. The simulation load was set at 600 MW for the duration of the test.

In order to minimize the effect of these properties of a real boiler on the combustion process, it is necessary to supply a certain percentage of excess air to ensure optimal combustion. This optimal amount of excess air is measured as a percentage of oxygen content in the flue gas. In the case of Duvha Power Station, the optimal amount of excess air occurs when the oxygen content of the flue gas is approximately 2 to 4%. Increasing the amount of excess air reduces the efficiency of the unit by increasing the cooling effect of the air, whereas decreasing the excess air also causes a reduction of boiler efficiency due to incomplete combustion.

5.6.2 Description of Test

This test involved decreasing the excess the air from 100% to 85% after which the unit was allowed to stabilize. A setting of 100% indicates optimal value.

Although the actual boiler load remained constant at approximately 575MW, the simulation was performed at a constant load of 600MW.

Test:	**15 % decrease in Excess Air**
Date:	16/04/97
Unit:	3
Operator:	Mr. Piet Krieger
Start Time:	14h45
Duration:	09 minutes 30 seconds
Average Actual Unit Load:	575 MW
Simulation Load:	600 MW

Time 00:00:00 - Initial Conditions	
Excess Air:	100%
Generating unit & Autocontrols:	Stable
Time 00:00:16 - Decrease Excess Air by 15%	
Excess Air:	85%
Time 00:09:30 - Final Conditions	
Excess Air:	85%
Generating unit & Autocontrols:	Stable

5.6.3 Test Results

5.6.3.1 Load

The actual boiler load remained constant within 5% of 575 MW. The simulation load was set at 600 MW for the duration of the test.

5.6.3.2 Temperature

Both the simulated and actual temperatures remained constant throughout the test.

5.6.3.3 Attemporation, Feedwater Flow and Fuel Usage

A. Actual Results

Decreasing the amount of excess air affects the real boiler in the following three ways:

a)	A decrease in excess air results in a less efficient combustion process due to insufficient oxygen availability for complete combustion of the coal. (Refer to Section 5.6.1 for additional information). This leads to increased fuel usage. The increased mass flow of flue gas also leads to a decrease in the ratio of radiative energy to convective energy, resulting in a slight decrease in the amount of radiative energy transferred to the evaporator and superheater.
b)	Decreasing the amount of excess air results in a decreased flue gas flow rate through [illegible]. This decreased flow across the heat exchange surfaces results in a longer [illegible] and thus an increase in the total heat exchange from the flue gas to the [illegible] increased heat exchange results in a higher overall efficiency of the [illegible] process, resulting in a decrease in fuel consumption.
c)	The [illegible] rate causes the coal to burn lower down in the boiler furnace, thus increasing the proportion of radiated heat that is absorbed by the evaporator relative to the superheater. This results in a higher evaporator enthalpy, which the enthalpy controller corrects by increasing the feedwater flow rate.

Table 5-2 - Primary Effects of Decreased Excess Air

The effect of each of the above on the boiler is as follows (for a decrease in excess air):

Primary Effect (Table 5-2)	Effect on Evap. Enthalpy	Effect on SH Temp	Effect on RH Temp	Effect on Fuel Usage
a)	slight decrease	negligible	negligible	increase
b)	negligible	increase	negligible	slight decrease
c)	increase	slight decrease	slight decrease	negligible

Table 5-3: Effect of Decreased Excess Air on a Real Boiler

Combined Effect of a) b) c) from Table 5-3	Effect on Evap. Enthalpy	Effect on SH Temp	Effect on RH Temp	Effect on Fuel Usage
Combined effect of a), b) & c) ignoring the attemporation and enthalpy controller	medium increase	medium increase	slight decrease	slight increase
Combined effect after taking into account the action of the attemporation and enthalpy controller	enthalpy kept constant by increasing FW flow	temp. kept constant by increasing SH attemporation	temp. kept constant by decreasing RH attemporation slightly	slight increase in fuel usage

Table 5-4: Combined Effects of a), b) and c) from Table 5-3 on Real Boiler

The table above shows that a decrease in excess air causes an increase in both feedwater flow an SH attemporation, a slight decrease in RH attemporation and a slight increase in fuel usage.

B. Simulated Boiler

Decreasing the amount of excess air affects the simulated boiler slightly differently to the real boiler. This is caused by the simplification of the combustion process which does not take into account the loss of efficiency of the combustion process as a result of insufficient excess air.

Primary Effect (Table 5-2)	Effect on Evap. Enthalpy	Effect on SH Temp	Effect on RH Temp	Effect on Fuel Usage
a)	n/a	n/a	n/a	n/a
b)	minimal	increase	minimal	slight decrease
c)	increase	slight decrease	slight decrease	minimal

Table 5-5: Effect of Decreased Excess Air on Simulated Boiler

Combined Effect of a) b) c) from Table 5-5	Effect on Evap. Enthalpy	Effect on SH Temp	Effect on RH Temp	Effect on Fuel Usage
Combined effect of a), b) & c) ignoring the attemporation and enthalpy controller	increase	slight increase	slight decrease	slight decrease
Combined effect after taking into account the action of the attemporation and enthalpy controller	enthalpy kept constant by increasing FW flow	temp. kept constant by increasing SH attemporation	temp. kept constant by decreasing RH attemporation slightly	slight decrease in fuel usage

Table 5-6: Combined Effects of a), b) and c) from Table 5-5 on Simulated Boiler

C. Comparison of Actual and Simulated

The two tables above show that the simulated boiler response should be the same as the actual boiler response in terms of feedwater flow, SH attemporation and RH attemporation.

The test results show a good correlation between the simulated and actual boiler attemporation flows. Chart 5-5 shows both SH attemporator flows increasing at roughly the same rate, while the RH attemporation flows decrease at approximately the same rate.

A comparison of the flows (Chart 5-4) show a good correlation between the actual and simulated main steam flow-rates. The response of the simulation in terms of total feedwater flow and economizer flow does not follow the prediction shown in Table 5-5 and is also opposite to the actual boiler response. Further investigation is required to determine the exact reason(s) for the differing responses. This is beyond the scope of this research.

The simulated fuel supply (Chart 5-3) decreased by 13% (from 59% to 46%) during the test. The actual total fuel reading during the test increased by 1.5 % (from 82% to 83.5%). This discrepancy is due to the simplification of the simulation of the combustion process which does not take into account the amount of oxygen available in the air, but rather assumes complete combustion of all fuel regardless of the amount of air available. This is in agreement with Table 5-3 and Table 5-4 which predict a slight decrease in fuel usage.

5.6.3.4 Valve Position

Only the HP governor valves are affected by the decreased excess air. In both the simulation and on the real unit the HP governor valves opened slightly (Chart 5-1):

- actual HP governor: opened from 43% to 48% (average)
- simulated HP governor: opened from 34% to 42%

These changes in position are attributed to the increase in main steam total feedwater flow in both the actual and simulated boiler during the test.

5.6.3.5 Pressures

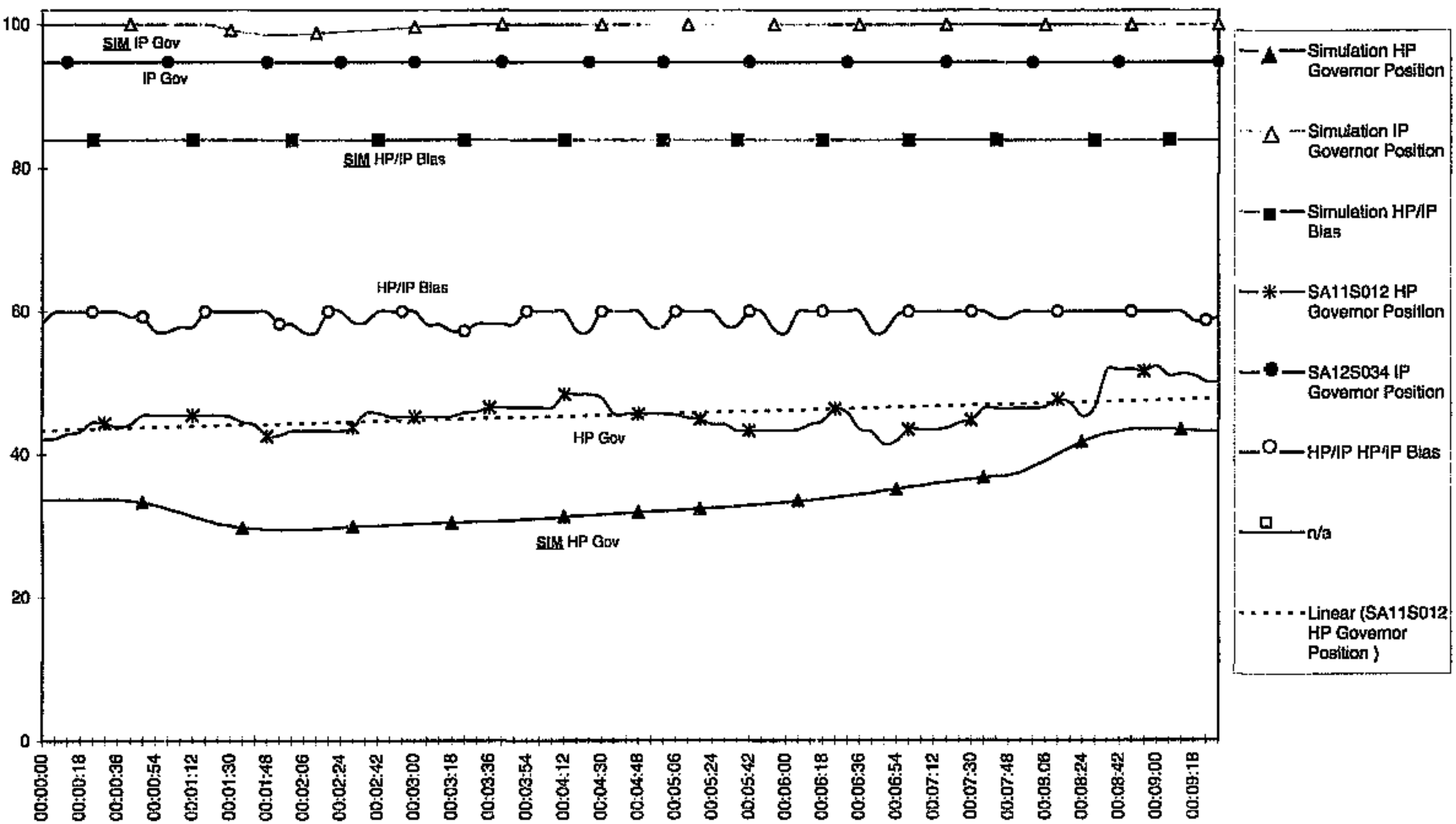
The actual HP governor outlet pressure increased by 0.6 MPa and the simulated HP governor outlet pressure increased by 0.3 MPa. All other actual and simulated pressures remained constant throughout the test.

The increases in turbine inlet pressure (HP governor outlet pressure) are due to the increased feedwater flow resulting in a higher pressure difference across the HP and IP/LP turbines.

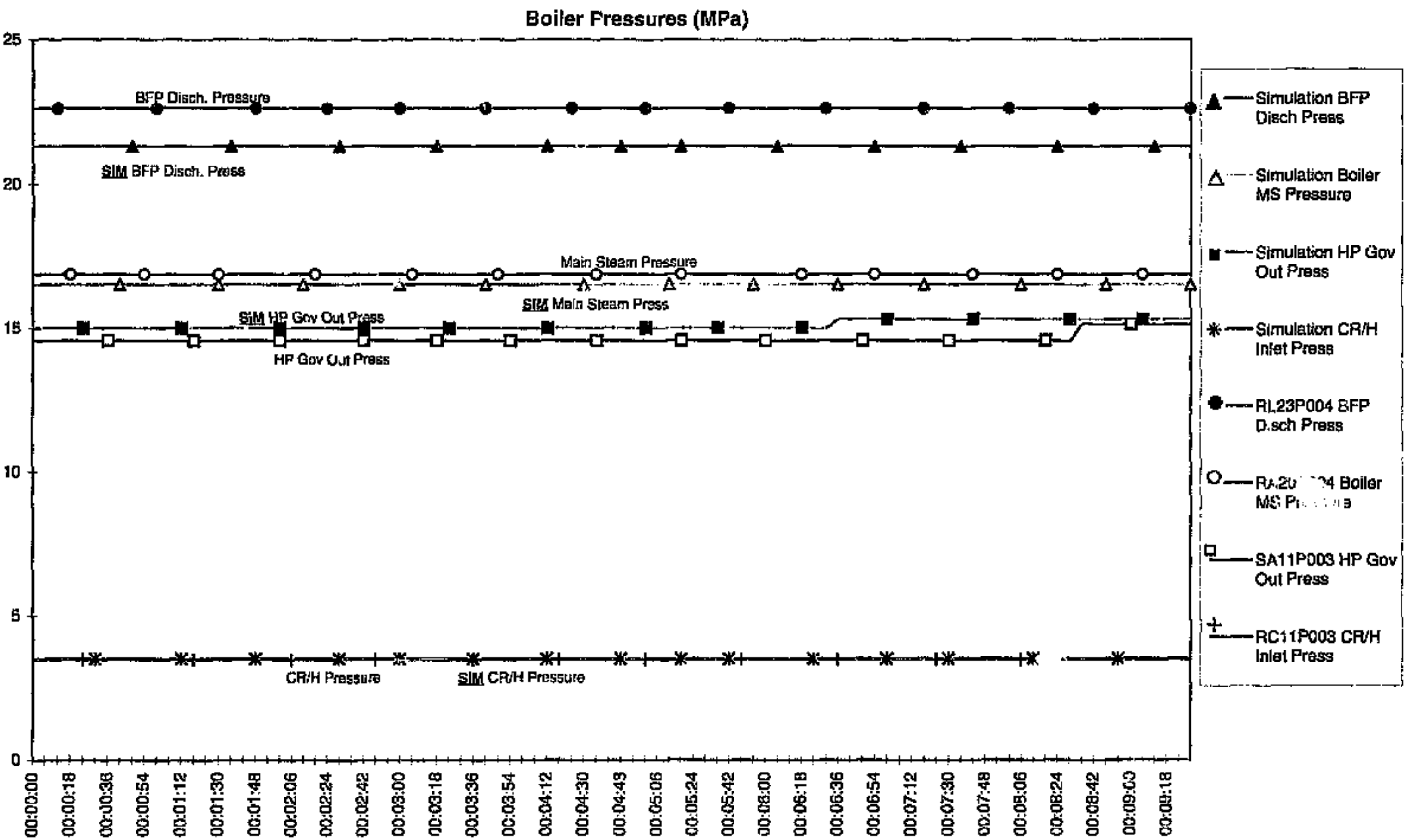
5.6.4 Conclusion

The response of a real boiler to a decrease in excess air is extremely complex, consisting of numerous conflicting effects. The overall result of these conflicting effects governs the behavior of the boiler and turbine.

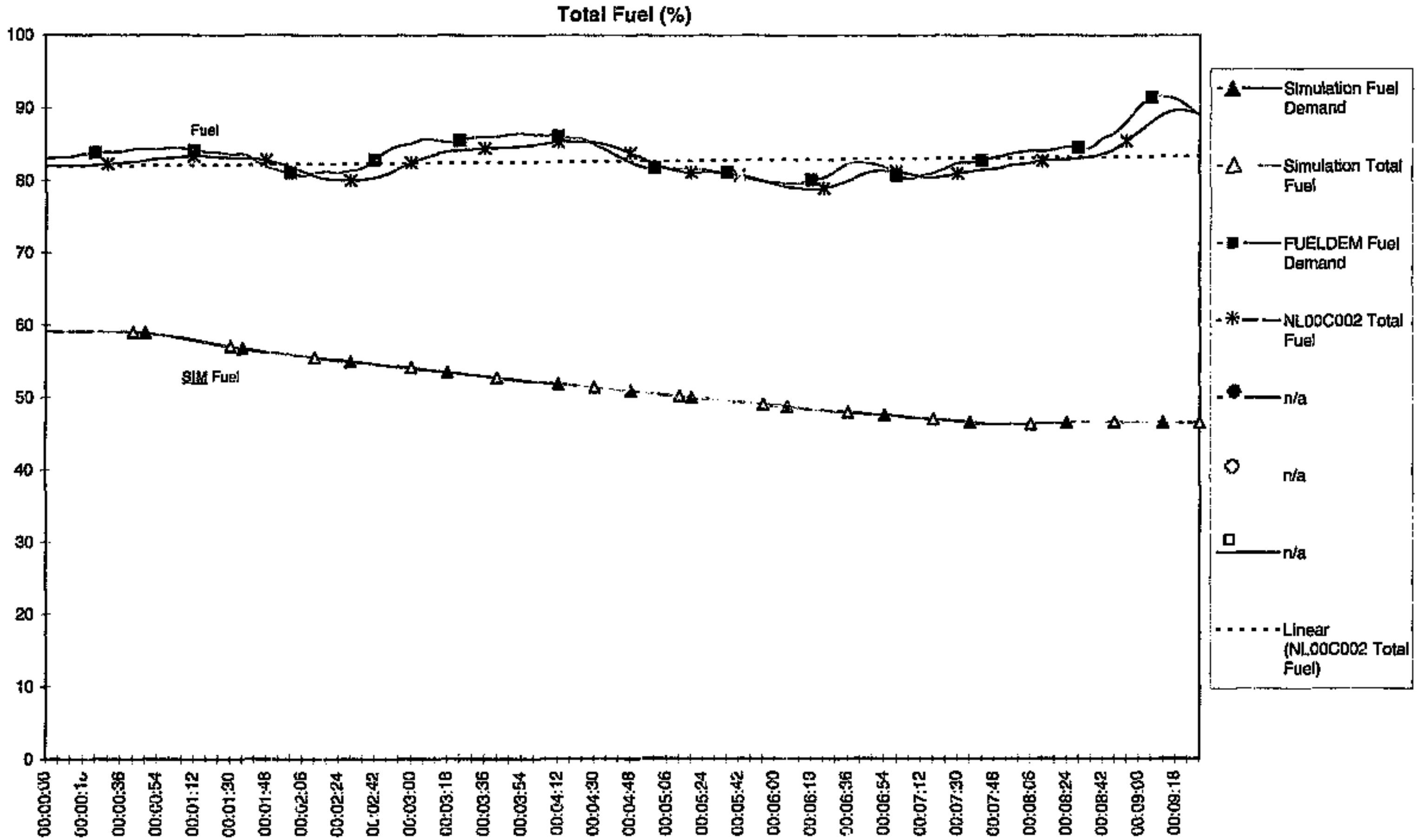
The simulation achieved very similar results to the actual generating unit, with the exception of some of the flow rates which did not correspond with the expected results. The main response to the excess air decrease is in the amount of SH and RH attemporation, which was confirmed by the simulation.



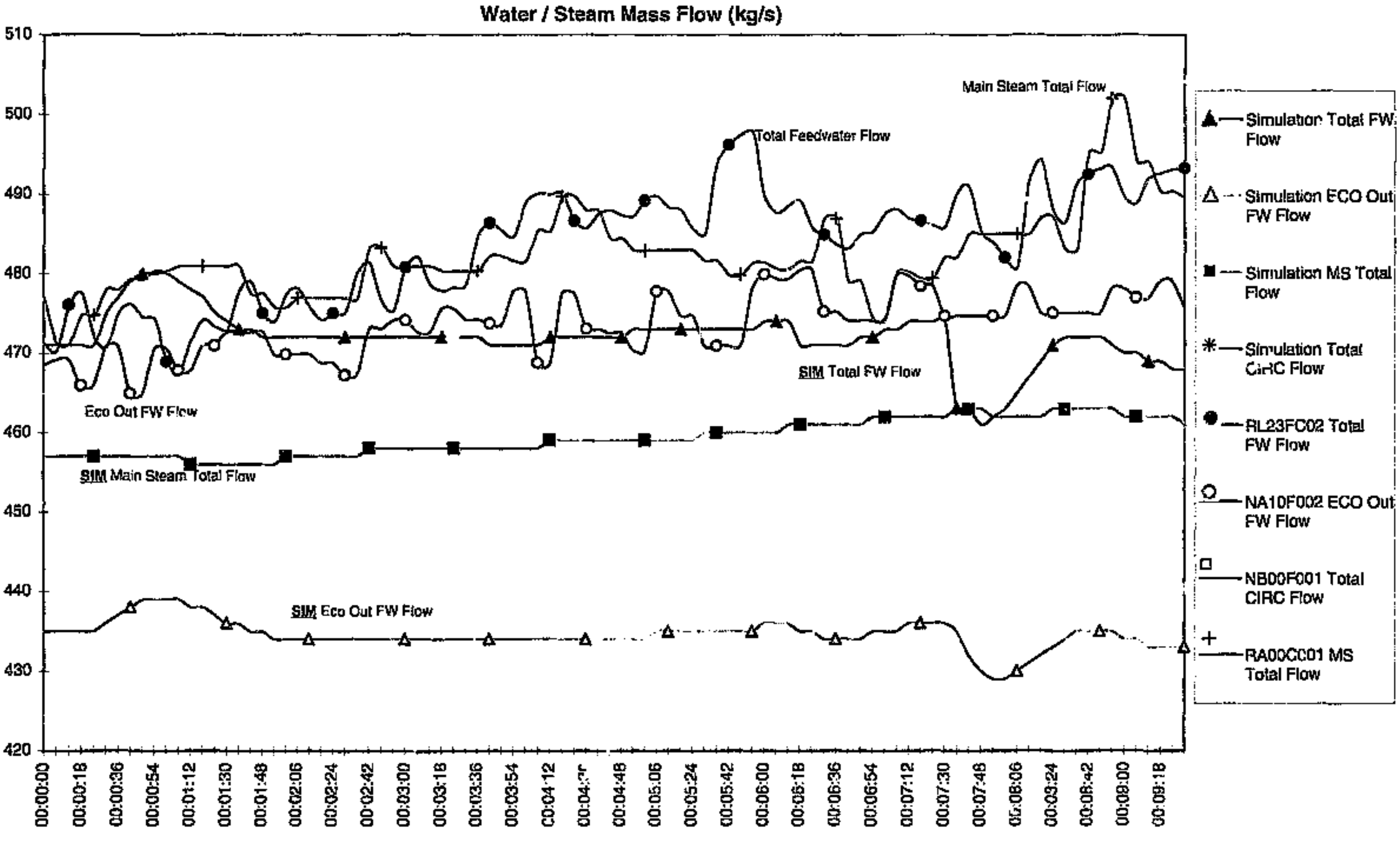
Test Chart 5-1: Decrease Excess Air - Advanced Simulation



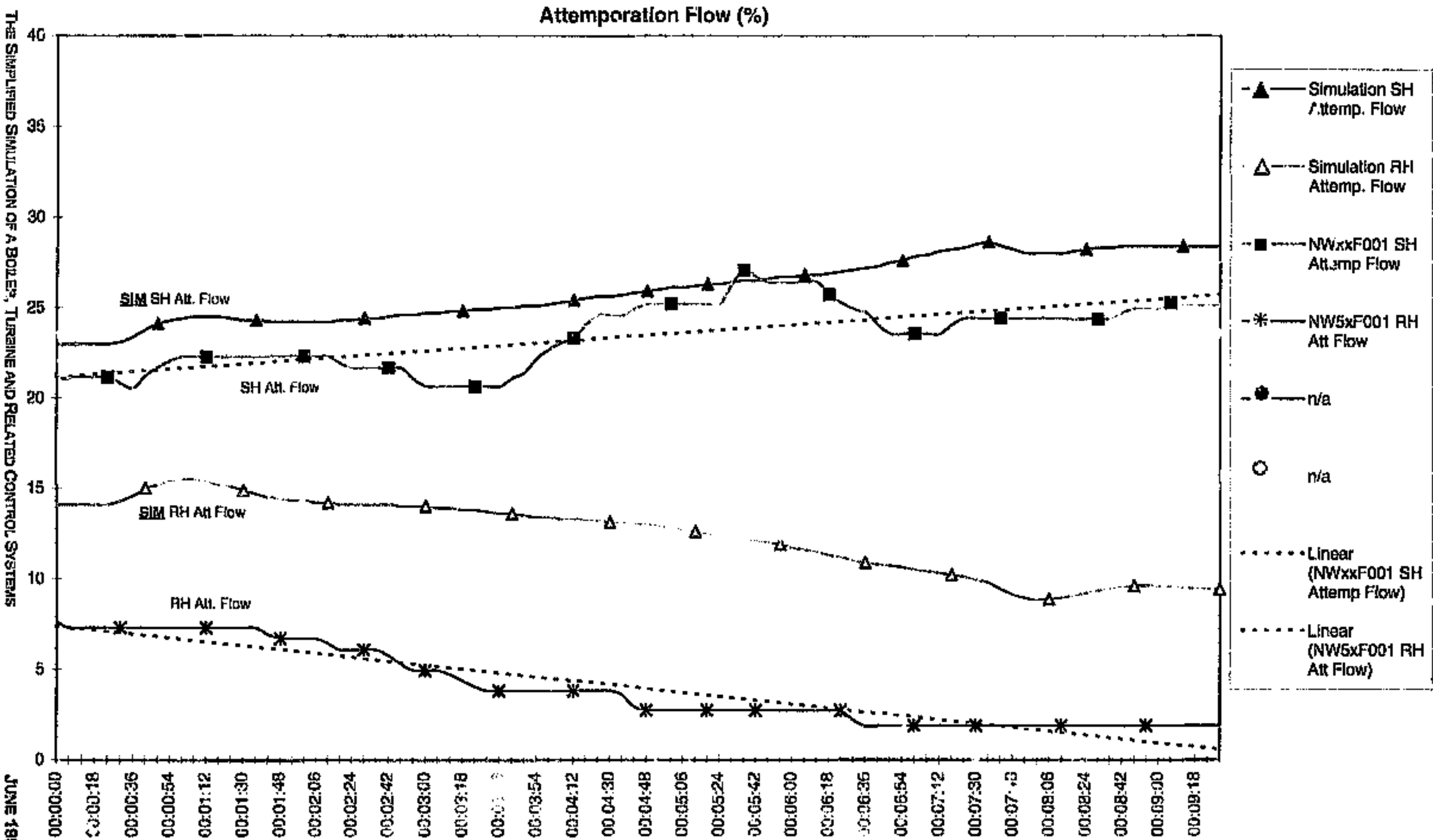
Test Chart 5-2: Decrease Excess Air - Advanced Simulation



Test Chart 5-3: Decrease Excess Air - Advanced Simulation



Test Chart 5-4: Decrease Excess Air - Advanced Simulation



Test Chart 5-5: Decrease Excess Air - Advanced Simulation

CHAPTER 6

6. Conclusions

6.1 General

6.1.1 Simple Simulation

The results of extensive comparisons between the initial simple simulation and the actual boiler responses show reasonable similarity in terms of the available simulation variables. Due to the lack of simulation of temperatures and enthalpies, as well as the simplifications and assumptions of this model, this simulation is found to be very limited in its application as an engineering tool. This model could not be used to make valid predictions for many of the normal operating disturbances, such as changes in excess air and coal quality. It would also be difficult to simulate simple mechanical plant modifications such as the addition of extra heat exchanges.

The omission of the SH and RH attemporators from the simulation is also very important when considering design changes, as most plant changes cause a change in the attemporation.

This simulation is more suited to being used as a training tool to illustrate the overall workings of the boiler, turbine and autocontrols.

6.1.2 Advanced Simulation

The advanced simulation is shown to correspond well with the actual boiler test results. In terms of the objectives set out in Section 1.2, this second simulation provides a far more accurate and complete model of the power generation processes of Duvha Power Station. The inclusion of relatively simple thermodynamics ensures that the important temperatures and enthalpies throughout the boiler and turbine are simulated, while also maintaining the simplicity of the model.

In order to assess the effectiveness of this research in achieving the objectives set out in Section 1.2, each objective is individually evaluated:

a) *To develop a* <u>*simple*</u> *dynamic simulation of a boiler and turbine in terms of fuel input, boiler pressure and megawatts (MW) output. This excludes simulation of some of the less important systems such as the internal operation of the steamfeed pumps, blowdown vessel, condenser, mills etc.*

This objective has been met by producing a model which incorporates the more important physical systems while excluding the systems which have minimal effect on the simulation as a whole. The model is accurate in simulating the dynamic as well as steady state responses of the generating unit.

b) *To simulate the physical processes through the use of simple mathematics while confining the use of complex thermodynamic equations to the absolute minimum.*

In terms of producing a simulation that is both easy to understand while avoiding the use of complex thermodynamics, this objective is achieved. In order to minimize the use of thermodynamic equations, customized lookup tables were used to convert between pressures, temperatures, enthalpies and steam dryness. It must be borne in mind that, if the simulation in its present form were converted to a set of mathematical equations, this would result in highly complex equations which would be very difficult to interpret and modify.

c) *To develop a detailed simulation of all relevant autocontrols which form part of the physical systems.*

All relevant autocontrols were simulated. In some instances appropriate simplifications were used to reduce the complexity of the simulation while maintaining functionality and accuracy. Many of the less used control modes have not been incorporated into the simulation.

d) *To develop an operator interface for the simulation.*

The operator interface was developed to allow easy operation of all controls from a single screen. This screen also includes graphs showing the main pressures, valve positions, fuel flows, and generated power. Additional simulation data has been grouped as follows: pressures, temperatures, enthalpies, flows, valve positions.

e) *To asses the question as to whether the simulation adequately fulfills the requirements of an engineering tool. This will be done by means of comparisons between live data and simulated data.*

The results of Chapter 5 show the simulation to be accurate in determining responses due to certain changes in the operating parameters. The extent to which the model may be used as an engineering tool depends on the engineering task. This issue is further discussed in Section 6.2

6.2 Significance as an Engineering Tool

Chapter 5 gives more than one example of the effectiveness of the simulation in predicting plant responses due to disturbances. Although in the tests performed on the unit all disturbances were controlled, the simulation could just as easily be applied to uncontrolled disturbances such as changes in coal quality, or sudden loss of vacuum in the condenser.

Evaluation of design changes could be carried out by making changes to the configuration parameters. For example, in order to simulate the addition of an extra row of economizer tubes, the heat exchange coefficient of the economizer would be increased and the volume of the economizer increased slightly.

It is not anticipated for this simulation be used for detailed design engineering of critical systems or major modifications. In such cases dedicated, boiler specific models would be needed to provide the accuracy and scope necessary to ensure the operation of the boiler at an optimum level of both safety and efficiency. This simulation could, in this case, be used for preliminary conceptual design. Although not providing extremely accurate results, the advanced simulation implemented during this research project is capable of indicating the approximate response expected from certain disturbances or design changes. In the case of a power generating unit, the ability to predict the approximate response is extremely useful due to the complexity of the system.

This dissertation shows that a relatively simple model of the boiler, turbine and related autocontrols is adequate for use as an engineering tool. It is also significant as it achieves its results using an IBM compatible PC running off-the-shelf simulation software. The fact that the simulation has been written in a purely graphical language makes it possible for the model to be understood by engineers without too much difficulty. This results in a powerful tool which, in the past, was only available to a limited number of specialists, and which may be used and modified by the engineers at Duvha Power Station to suite their particular engineering needs.

6.3 Limitations

As in most research projects, the scope of the dissertation was reduced due to the limitations of both time and resources.

6.3.1 Limitations on Scope

Due to the complexity of the generating unit, it was necessary to impose certain limitations on the simulation to ensure completion of the dissertation within a reasonable time period. A decision was thus made to limit the scope of the simulation to the major physical systems and control loops rather than attempting to include the entire generating unit. The systems that were omitted have negligible influence on the boiler operation during normal operation and only influence the generating unit under exceptional circumstances. A good example of this is the condenser vacuum which is normally controlled at a set pressure. The only time that the boiler operation is affected by the condenser is when the vacuum deteriorates.

The processing power of the computer was also a limiting factor on the complexity of the simulation. In order to ensure the usefulness of the simulation as an engineering tool, it is necessary that it run significantly faster than real-time. This imposed a restriction on the complexity of the simulation model which, in the case of the advanced simulation, was found to execute at 10 times real-time as opposed to the original design figure of 28 times faster than real-time.

6.3.2 Limitations on Tuning of Control Loops and Boiler Parameters

Due to the limitation of time it was not possible to tune the simulation parameters adequately. This resulted in differences between the simulation and actual tests, especially in terms of magnitude and speed of responses. A further restriction on tuning of the simulation is a result of limited access to the live generating unit for testing purposes.

The task of tuning the simulation is extremely complex and time consuming, and as such is outside the scope of this dissertation. This would, however, be extremely important as further research to improve the accuracy of the model.

6.4 Further Research Possibilities

Due to the limitations of both time and computer resources, certain aspects of the simulation have not been fully researched. Extensive scope still exists for further research to improve the simulation accuracy and user interface.

6.4.1 On-line Simulation

Although the simulation has been designed to run as a stand-alone unit, the usefulness of the model would be increased considerably if it was modified to run as an on-line simulation. This would entail modification of the simulation interface to use actual plant parameters as inputs to the simulation. This would allow the simulation to take its inputs directly from the unit autocontrols without the need for user intervention.

The present SCADA system provides some of the data that would be necessary to automate the simulation. The variables that are not available on the SCADA are the system setpoints and mode settings. Figure 6-1 illustrates the data which is currently available from the SCADA system as well as the data which is not captured by the SCADA in its present configuration.

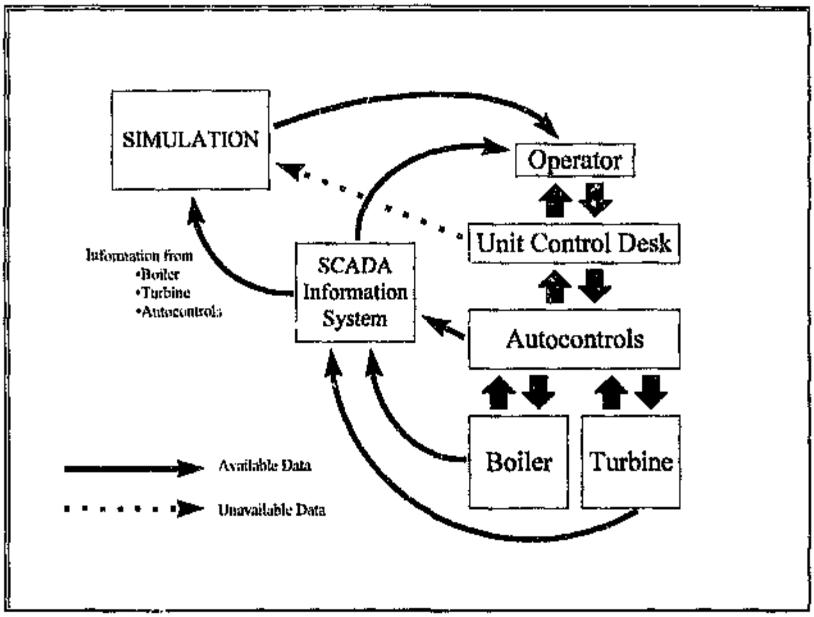


Figure 6-1: Online Simulation Data Flow

6.4.2 Expansion of Scope

As discussed in Section 6.3.1 many aspects of the generating unit are not included in this simulation. There is a possibility for further research to include these systems that have been omitted in order to increase the accuracy and significance of the simulation.

6.4.3 Tuning of Simulation Parameters

In order for the simulation to realize its full potential as an engineering tool, it is necessary that the simulation parameters be tuned to match the simulated plant. Although excluded from this research project, this may be considered to be the most important step in the improvement of this model.

Due to the size and complexity of the model, tuning of the parameters is a difficult task, and would require much testing of the live unit against certain controlled inputs. Tuning of the control loops should be less difficult as the values for each control component can (in theory) be ascertained from the controller design documents or read from the plant control equipment itself.

APPENDIX A - DUVHA POWER STATION

"Duvha coal-fired giant
Technical Information"[42]

Inleiding

Eskom staan aan die voorpunt van kragontwikkelingstegnologie. Deur reusagtige en verbeeldingryke skemas het Eskom vir hom 'n plek in die energiewêreld verwerf en internasionaal die aandag van verwante gespesialiseerde sektore getrek. Tegniese inligting is die sleutel tot 'n professionele begrip van hierdie multidissiplinêre ingenieursprojekte. Hierdie publikasie bied 'n inleidende tegniese oorsig oor Eskom se grootste ingenieursprestasies, en inligting word gereeld bygewerk om toepaslikheid te verseker.

Introduction

Eskom is in the forefront of power generation technology. Vast and imaginative schemes have ensured Eskom's pre-eminence in the energy world and attracted international attention from related specialist sectors. Technical information is the key to a professional understanding of these multidisciplinary engineering projects. This publication presents an introductory technical reference to Eskom's major engineering achievements, and this information is updated on a regular basis to ensure technical relevance.

Duvha, steenkoolgestookte reus

Duvha, een van die wêreld se grootste kragstasies, is tiperend van Eskom se nuwe geslag steenkoolgestookte installasies met ses stelle van 600 megawatt (MW) elk. Grootte is Duvha se mees besondere kenmerk en die stasie spog met 'n aantal superlatiewe. Die skoorstene is byvoorbeeld die hoogste vrystaande betonstrukture in die Suidelike Halfrond. Die steenkoolmyn, wat 'n verwagte lewensduur van 30 jaar het en slegs aan Duvha-kragstasie steenkool verskaf, is die grootste oopgroefsteenkoolmyn in die Suidelike Halfrond. Die warmtewisselingsvermoë van elke koeltoring is 'n massiese 886 MW.

Konstruksie by Duvha het in November 1975 begin en die laaste stel is in Februarie 1984 in kommersiële bedryf gestel. Die totale projekkoste vir die agt jaar het R1,6 miljard beloop.

Duvha: die basiese siklus

Steenkool word deur 'n vervoerband van die bergbakke af na die ketelbunkers gevoer, van waar dit in die meulens gevoer word. Die verpoeierde steenkool word dan van die meulens deur die ketelbranders in die oond geblaas, waar dit teen ongeveer 1 500 °C brand. Die soliede produkte van die verbranding is stof en as in 'n verhouding van 5:1. Die as val na die bodem van die ketel en word met water weggespoel, terwyl die stof in die rookgasse deur elektrostatiese ontstowwers opgevang word. Die gereinigde rookgasse gaan deur die skoorsteen, waar dit 300 m bo grondvlak in die atmosfeer vrygelaat word. Die hitte van die steenkoolverbranding word geabsorbeer deur die ketel-voedingswater in die 500 km pypwerk wat die ketelwande vorm, en die water verander teen hoë temperatuur en druk in stoom. Die stoom word dan deur superverhitters gestuur en van daar na die hoëdrukturbine. Die uitlaatstoom word na die ketelherverhitters teruggevoer en van daar na die middeldrukturbine en die twee laedrukturbines.

Die uitgewerkte stoom word dan gekondenseer en deur voedingswaterverwarmers en 'n ontlugter na die ketelvoerpomp en weer in die ketel teruggepomp om die siklus te herhaal. Die kondensator bevat meer as 22 500 pype. Die koelwater self moet verkoel word en word daarom in die laer vlakke van 'n koeltoring gesproei, waar verdamping die ongewenste hitte verwyder. Die stygstroom in die koeltoring is geheel en al aan konveksie toe te skryf.

Die generatorrotor, 'n silindriese elektromagneet wat in 'n gasdigte hulsel toegebou is, is aan die as van die vier turbines gekoppel. Die elektrisiteit gaan van die statorwikkelings na 'n transformator, waar die spanning van 22 kV verhoog word tot die spanning van die nasionale kragnet – 400 kV by Duvha. Die elektrisiteit word dan van die hoëspanningswerf via die nasionale kragnet versprei.

Die installasie

Beheer

Elkeen van Duvha se ses ketel-turbinestelle word as 'n aparte entiteit bedryf, elk met sy eie kontroles en instrumentasie in sy eie kontrolebank en -paneel. Daar is drie beheerkamers, wat elk een paar stelle bedien en van waar die bedieners elke groot taak wat verband hou met aansit, normale bedryf, afsluiting en noodbedryf, kan uitvoer. Die bedieners is deurentyd in verbinding met ander Eskom-beheersentrums, wat die geïntegreerde transmissienetwerk uitmaak.

'n Dataregistrerende rekenaar monitor voortdurend die hoofbedryf- en alarmstelsels en voorsien 'n konstante inligtingsvloei op videoskerms en drukkers.

Brandstofhantering

Vergruisde steenkool, kleiner as 25 mm, word deur middel van 'n vervoerbandstelsel van Duvha-oopgroefmyn verskaf en in twee bergbakke geberg. Lewering aan die aanleg geskied teen 'n tempo van 845 000 ton per maand en die twee bergbakke het 'n kapasiteit van 55 000 en 110 000 ton onderskeidelik. Die voorraadstapel word op 500 000 ton gehou.

Duvha, fossil-fired giant

Duvha, one of the world's largest power stations is typical of Eskom's new breed of fossil-fired plant, with six 600 megawatt (MW) sets. Size is Duvha's most striking characteristic, and the station boasts a number of superlatives. The chimneys, for example, are the tallest free-standing concrete structures in the Southern Hemisphere. The tied colliery, which has a projected life of 30 years, is the largest open-cast coal mine in the Southern Hemisphere. The heat exchange capacity of each cooling tower is a massive 886 MW.

Construction at Duvha began in November 1975, and the last set was taken into commercial operation in February 1984. Project costs totalled R1,6 billion over that eight-year period.

Duvha: the basic cycle

Coal is fed from the staithes to the boiler bunkers by conveyor belt, from where it is fed into the mills. The pulverised coal is blown from the mills through the boiler burners and into the furnace, where it burns at about 1 500 °C. The solid products of the combustion are dust and ash, in a ratio of 5:1. The ash falls to the bottom of the boiler and is sluiced away with water, whereas the dust is collected from the flue gases in the electrostatic precipitators. The cleaned flue gases are passed through the chimney to be released into the atmosphere 300 m above ground level. The heat from the combustion of the coal is absorbed by the boiler feedwater in the 500 km of tubing that forms the boiler walls, and the water turns into steam at high temperature and pressure. The steam is then passed through superheaters and thence to the high-pressure turbine. The exhaust steam is returned to the boiler reheaters and from there to the intermediate-pressure turbine and the two low pressure turbines.

The spent steam is condensed and pumped back through feedheaters and a deaerator to the boiler feed pump and thence into the boiler again to repeat the cycle. The condenser contains more than 22 500 tubes. The cooling water itself has to be cooled and is therefore sprayed into the lower levels of a cooling tower where evaporation removes the unwanted heat. The draught in the cooling tower is due entirely to convection.

Coupled to the shaft of the four turbines is the generator rotor, which is a cylindrical electromagnet enclosed in a gas-tight housing. The electricity passes from the stator windings to a transformer, where the voltage is raised from 22 kV to the voltage of the national grid – 400 kV at Duvha. The electricity is then distributed from the high-voltage yard via the national grid.

The plant

Control

Each of Duvha's six boiler-turbine sets is run as a separate entity, with its own controls and instrumentation incorporated into its own control desks and panels. There are three control rooms, each serving a pair of sets. Here the operators can execute every major task associated with start-up, normal operation, shutdown and emergency operation. The operators are in permanent contact with the other Eskom control centres which make up the integrated transmission network.

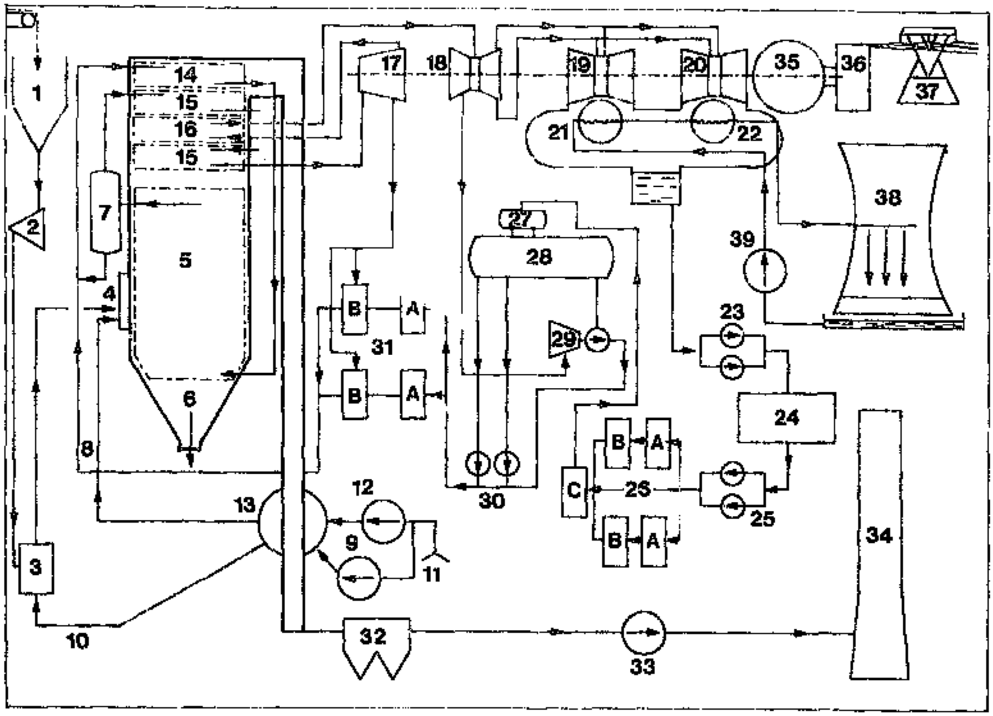
A data-logging computer continuously monitors the main operating and alarm systems, and provides a constant flow of information on video screens and printers.

Fuel handling

Crushed coal, smaller than 25 mm, is delivered from the Duvha open-cast mine by means of a conveyor system and stored in two staithes. Delivery to the plant is at the rate of 845 000 tons a month, and the two staithes have capacities of 55 000 and 110 000 tons respectively. In addition, a stockpile of 500 000 tons is maintained.

Duvha-kringdiagram

Duvha circuit diagram



1 Steenkoolbunker
2 Steenkoolvoerder
3 Steenkoolmeule
4 Ketelbranders
5 Ketelverbrandingskamer
6 As
7 Skeidingsvat
8 Sekondêre lug
9 Primêrelugwaaier
10 Primêre lug
11 Luginlaat
12 Dwangtrekwaaier
13 Lugverwarmer
14 Ketelbespaarder
15 Ketelsuperverhitter
16 Ketelherverhitter
17 Hoëdrukturbine
18 Middeldrukturbine
19 Laedrukturbine Nr 1
20 Laedrukturbine Nr 2
21 Kondensator Nr 1
22 Kondensator Nr 2
23 Kondensaatpompe
24 Kondensaatsuiweringsinstallasie
25 Kondensaatpompe
26 Laedrukverwarmers
27 Ontlugter
28 Voedingswatertenk
29 Ketelvoerpompturbine
30 Elektriese voerpomp
31 Hoëdrukverwarmers
32 a) Sakfilters (Eenheid 1-3)
32 b) Ontstowwers (Eenheid 4-6)
33 Suigtrekwaaier
34 Skoorsteen
35 Generator
36 Generatortransformator
37 Eskom se nasionale kragnetwerk: 275 en 400 kV
38 Koeltoring
39 Koelwaterpomphuis

Steenkool word van die bergbakke af gevoer na die ketelbunkers, waarvan daar ses per ketel is. Elke bunker het 'n kapasiteit van 670 ton, wat beteken dat daar voldoende bergkapasiteit by die ketels is vir 12 uur se bedryf by volle las.

Wanneer 'n oond aangesit word, word brandolie gebruik totdat die verbranding van die verpoeierde steenkool stabiel is. By Duvha word twee opgaartenks met 'n gesamentlike kapasiteit van 3 000 ton gebruik om brandolie op te gaar.

Meulens

Steenkool van die ketelbunkers word onder beheer in die meulens gevoer en die voertempo word deur die behoefte aan

1 Coal bunker
2 Feeder
3 Coal mill
4 Boiler burners
5 Boiler combustion chamber
6 Ash
7 Separator vessel
8 Secondary air
9 Primary-air fan
10 Primary xair
11 Air intake
12 Forced-draught fan
13 Air heater
14 Boiler economiser
15 Boiler superheater
16 Boiler reheater
17 High pressure turbine
18 Intermediate-pressure turbine
19 Low-pressure turbine No 1
20 Low-pressure turbine No 2
21 Condenser No 1
22 Condenser No 2
23 Extraction pumps
24 Polishing plant
25 Extraction pumps
26 Low-pressure heaters
27 Deaerator
28 Feedwater tank
29 Boiler feed pump turbine
30 Electric feed pump
31 High-pressure heaters
32 a) Bag filters (Units 1-3)
32 b) Precipitators (Units 4-6)
33 Induced-draught fan
34 Chimney
35 Generator
36 Generator tansformer
37 Eskom national grid: 275 and 400 kV
38 Cooling tower
39 Cooling-water pump house

From the staithes, coal is fed to the boiler bunkers, of which there are six per boiler. Each bunker has a capacity of 670 tons, which means that there is sufficient storage capacity at the boilers for 12 hours' operation at full load.

When a furnace is started up, fuel oil is used until the combustion of the pulverised coal is stable. At Duvha, two storage tanks are used for fuel oil, with a combined capacity of 3 000 tons.

Mills

Coal from the boiler bunkers is fed under control into the mills. The feed rate is determined by the need for steam. Inside stoom bepaal. In die meulens is daar of groot staalballe of rollers wat in ronde bane loop en die steenkool verpoeier. Die fyn, uiters brandbare poeier word van elke meule teen sowat 63 ton per uur in die ketels ingeblaas.

Daar is ses meulens vir elke ketel, wat 'n reserwekapasiteit van twee meulens per ketel voorsien wanneer steenkool van ontwerpkwaliteit verbrand word. By volle vermoë het elke ketel tussen 250 en 300 ton steenkool per uur nodig, na gelang van die hittewaarde van die steenkool.

Ketels

Die ses ketels is van die Benson-tipe, wat volgens die beginsel van geforseerde sirkulasie werk en geen stoomtrom het nie. Die voedingswater, wat hoogs gesuiwerde en gedemineraliseerde water is, voltooi een aanhoudende en ononderbroke kringloop deur die ketel, turbines en kondensator en word weer by die ketel ingevoer. Op sy baan deur die ketel verdamp die voedingswater en, namate meer hitte geabsorbeer word, word die stoom superverhit tot 'n temperatuur van 540 °C en 'n druk van 17,1 MPa.

'n Kenmerk van die stoompype is 'n hoëdrukstelsel wat toelaat dat die superverhitte stoom, wanneer nodig, verby die hoëdrukturbine en direk in die herverhitpype vloei. Van daar vloei dit verby die middeldruk- en laedrukturbines, reguit in die hoofkondensator in. Hierdie stelsel beteken dat die ketel onafhanklik van die turbines bedryf word en die korrekte stoom- en temperatuurdruk kan verkry word voordat die generator aan die gang gesit word. Die stelsel laat ook toe dat indien 'n turbine uitklink, die ketel by ongeveer 40 persent van sy maksimumlas bedryf word, gereed vir 'n snelle heraansit; en sou 'n turbine 'n groot deel van sy las afwerp, kan die turbine by 'n lae las bedryf word terwyl die ketel by 'n hoër las bedryf word. In beide gevalle verhoed die omleistelsel, binne die perke van sy vermoë, dat skielike drukstygings die stellings van die veiligheidskleppe in die herverhitter oorskry. Elke ketel is toegerus met 24 branders wat in ses rye van twee aan die voor- en agterwande van die oond gerangskik is. Elke steenkoolmeule voorsien vier branders -- twee aan die voorwand en twee aan die agterwand. 'n Oliebrander is in die middel van elke brander gemonteer en word vir aansit en die stabilisering van die verbranding van steenkool by lae las gebruik.

Druktrekwaaiers voorsien sekondêre lug aan die branderwindkaste, terwyl primêre waaiers die verpoeierde steenkool van die meulens na die branders vervoer. Die verbrandingsgasse word deur twee suigtrekwaaiers uit die oond oor die oppervlakke van die stoomsuperverhitter, die herverhitters, die brandstofbespaarder en die lugvoorverhitters, deur die elektrostatiese ontstowwers en in die skoorsteen in, getrek. Meer as 99 persent van die stof, of vliegas, word deur die ontstowwers opgevang.

Elke ketel het 'n vermoë van 507 kg/s stoom. Die voedingswater word teen 'n druk van 22 MPa deur twee elektriese pompe met vermoëns van 13 MW elk, of deur een stoomaangedrewe pomp voorsien. By maksimum deurlopende ontwerpvermoë is die rendement van die ketels 93,9 persent.

Die totale massa van een ketel en sy ondersteunende struktuur is 9 800 ton en sy kouewaterinhoud is 185 ton. Die volume van die oond is ongeveer 13 000 m³ en die oppervlakte van die warmtewisselingspype is 74 000 m².

Sakfilters (Eenhede 1, 2 en 3)

Hierdie aanleg vervang die ontstowwers vir eenhede 1, 2 en 3 en bestaan uit die ABB Flakt Optipulse-sakfilter, wat met akrieldralon- T-sakke toegerus is.

Die aanleg is verdeel in vier isoleerbare kompartemente. Elke kompartement het 6 724 sakke. Daar is altesaam 26 896 sakke per ketel. Die lengte van die sakke is agt meter.

Die aanleg is ontwerp om 1 290 m³/s ketelrookgas teen 'n snelheid van 0.015 m/s te hanteer.

Turbines

Duvha-kragstasie het ses turbines. Elke turbine-eenheid is toegerus met hoëdruk-, middeldruk- en twee dubbelvloei laedruksilinders. Die hoëdruk- en middeldruk-silinders bestaan uit 'n binne- en buite-omhulsel om vinnige aanloop en snelle laswisseling moontlik te maak.

Die fondamente en stutte van een turbine bevat 8 000 m³ of 20 000 ton beton.

the mills either large steel balls or rollers run in circular tracks to pulverise the coal to powder. This fine, highly combustible powder is blown from each mill into the boilers at some 63 tons an hour.

There are six mills to each boiler, providing a spare capacity of two mills per boiler when coal of design quality is being burnt. At full capacity each boiler requires between 250 and 300 tons of coal an hour, depending on the calorific value of the coal.

Boilers

The six boilers are of the Benson type, which depends on forced circulation and has no steam drum. The feedwater, which is highly purified and demineralised, makes one continuous and uninterrupted circuit through the boiler, the turbine, the condenser, and again through the boiler. On its path through the boiler, the feedwater evaporates and as more heat is absorbed, the steam becomes superheated to a temperature of 540 °C and a pressure of 17,1 MPa.

A feature of the steam pipework is a high-pressure system that allows the superheated steam to bypass the high-pressure turbine when necessary, to flow directly into the reheat pipework. From there it bypasses the intermediate-pressure and low-pressure turbines and flows into the main condenser. This system means that the boiler can be operated independently of the turbine, and the correct steam temperature and pressure can be obtained before the generator is started up. Also, if a turbine trips, the system allows the boiler to be operated at roughly 40 per cent of its maximum load, ready for a rapid re-start; and if a turbine sheds a large part of its load, it can be run at a low load while the boiler is maintained at a higher load. In both instances, the bypass system prevents, within the limits of its capacity, sudden pressure rises from exceeding the settings of the safety valves in the reheater.

Each boiler is fitted with 24 burners, arranged in twelve rows of two -- six rows in the front wall of the furnace and six rows in its rear wall. Each coal mill supplies four burners - two in the front wall and two in the rear wall. Mounted in the centre of each burner is an oil burner, and these are used for starting up and for stabilising the combustion of coal at low loads.

Forced-draught fans supply secondary air to the burner wind boxes while primary-air fans transport pulverised coal from the mills to the burners. Combustion gases are drawn from the furnace by two induced-draught fans, over the surfaces of the steam superheater, the reheaters, the economiser, and air preheaters, through the electrostatic precipitators and into the chimney. More than 99 per cent of the dust, or fly ash, is collected by the precipitators.

Each boiler has a capacity of 507 kg/s of steam, and the feedwater is supplied, at a pressure of 22 MPa, by two electric pumps rated at 13 MW each or by one steam-driven pump. At maximum continuous rating the efficiency of the boilers is 93,9 per cent.

The total mass of one boiler and its supporting structure is 9 800 tons, and that of its water content, when cold, is 185 tons. The volume of the furnace is approximately 13 000 m³ and the surface area of the heat-exchange tubing totals 74 000 m².

Bag filters (Units 1, 2 and 3)

This plant replaces the precipitators for Units 1, 2 and 3 and consists of an ABB Flakt Optipulse pulse-jet fabric filter, utilising acrylic dralon T-bags.

The filter is divided into four isolatable compartments. Each compartment contains a total of 6 724 bags, each of which is eight metres long, giving a total of 26 896 bags, per boiler.

The plant is designed to handle 1 290 m³/s boiler gas, which gives a filter velocity of 0.015 m/s.

Turbines

There are six turbines at Duvha Power Station. Each turbine unit has high-pressure and intermediate pressure cylinders and twin double-flow low-pressure cylinders. The high pressure and intermediate-pressure cylinders are constructed with an internal and external casing to allow for fast start-up and rapid variation in load.

The foundations and supports for one turbine include 8 000 m³ or 20 000 tons of concrete.

Voedingsverwarmerinstallasie

Die kondensaattemperatuur in die kondensaattenk is ongeveer 37 °C. Om maksimum stelseldoeltreffendheid te verkry, is dit nodig om die kondensaat te herverhit voor dit na die ketel teruggevoer word. Om dit te doen moet die kondensaat deur talle verwarmers gestuur word voordat dit in die ketel ingevoer word. Al die verwarmers is van die bogrondse tipe waarin die water deur pypbundels vloei. Stoom wat aan verskillende stadiums van die turbine onttrek word, verhit die water deur middel van geleiding deur die pypmetaal.

'n Ontlugter tussen die hoëdruk- en laedrukverwarmers verwyder die suurstof wat in die kondensaat vasgevang is en verhit die water deur direkte kontak met die stoom wat aan die turbines onttrek word. Die kondensaat word dan met behulp van die hoofkondensaat-uitsuigpompe van die kondensator af deur die laedrukverwarmers na die ontlugter gepomp.

Die kondensaat word hierna met behulp van die ketelvoerpompe van die ontlugter se voedingswatertenk af deur die hoëdrukverwarmers na die ketel gepomp. Die eindtemperatuur van die voedingswater by 600 MW is 247 °C.

Kondensators

Die kondensators is van die dubbeldruk-oppervlaktipe. Die vakuum in die kondensators word bewerkstellig deur stoomstraallugverdrywers en word in normale bedryf deur waterstraallugverdrywers in stand gehou. Die stoom wat uit die laedruksilinder gelaat word, kondenseer oor 22 552 geelkoperpype, wat 'n totale oppervlakte van 23 400 m² beslaan.

Elke ketel-turbinestel het een kondensator met 'n hittewisselingsvermoë van ongeveer 400 MW.

Generators

Elke generator het 'n ontwerpvermoë van 667 MVA en lewer 600 MW by volle las. Die uitstuurvermoë is 575 MW, aangesien die verskil van 25 MW gebruik word om hulptoerusting van die turbine en ketel te laat loop.

Generatorkoeling word deur twee stelsels bewerkstellig. Die stator en rotor word verkoel deur waterstof wat teen 'n druk van 400 kPa deur oppervlaktipe waterkoelers sirkuleer. Sirkulasie van die waterstof vind plaas deur twee waaiers wat aan die rotor-as geheg is. Die statorwikkelings en die kern word deur die sirkulasie van gedemineraliseerde water verkoel.

Die generators werk teen 22 kV en hul lewering word tot 400 kV verhoog vir distribusie via die nasionale netwerk.

Skoorstene

Duvha se twee skoorstene is met 'n hoogte van 300 m die hoogste vrystaande betonstrukture in die Suidelike Halfrond. Elke skoorsteen bevat drie rookgasgange, sodat daar een gang per ketel is. Die diameter van die skoorstene by die basis is 22 m en beide staan op 60 heipale van 1,2 m by 18 m. Nagenoeg 40 000 ton beton is in die konstruksie van elke skoorsteen gebruik.

Die hoogte van die skoorstene beteken dat die rookgasse deur die normale inversielaag van Mpumalanga se Hoëveld-atmosfeer uitgewerp word en sodoende oor baie groot afstande verstrooi kan word.

Koeltorings

Die ses koeltorings by Duvha is ontwerp vir 'n totale termiese las van nagenoeg 19 000 GJ/h. Die drie grootste torings is 149 m hoog en 114 m in diameter by grondvlak, en die minimum wanddikte is 180 mm. By volle las beloop die daaglikse verdamping uit een toring 30 Mℓ per dag.

Watervoorsiening

Daar kan uit een van twee bronne in Duvha se waterbehoeftes voorsien word. Eerstens kan water gepomp word uit die Vygeboom- en die Nooitgedachtdam, wat in die oostelike gebied van Mpumalanga deur die Komatirivier gevoed word. Hierdie damme kan ook aangevul word met water uit die Usutu-stelsel. Tweedens kan water van die Witbankdam in die bolope van die Olifantsrivier verkry en aangevul word met water uit die Grootdraaidam in die Vaalrivier naby Standerton.

Die tweede bron word so min moontlik gebruik omdat dit nie so suiwer as die Komatirivier is nie. Suiwer water is uiters wenslik, aangesien suiwerings- en demineraliseerkoste in verhouding is met die konsentrasie materiaal wat verwyder moet word. Hierdie bron word slegs gebruik indien die Komati-stelsel nie water kan voorsien nie.

Feed-heating plant

The condensate temperature in the condenser hotwell is about 37 °C. In order to obtain maximum system efficiency it is necessary to reheat the condensate before returning it to the boiler. To achieve this, the condensate is passed through numerous heaters before feeding it into the boiler. All the heaters are of the surface type in which the water flows through tube nests. Steam extracted from different stages of the turbine heats the water by conduction through the tube metal.

A deaerator situated between the low-pressure and high-pressure heaters removes entrained oxygen in the condensate and heats the water through direct contact with steam extracted from the turbines. The condensate is pumped from the condenser to the deaerator through the low-pressure heaters by means of the main condensate extraction pumps.

The condensate is then pumped from the deaerator feedwater tank to the boiler through the high-pressure heaters by the boiler feed pumps. The final feedwater temperature at 600 MW is 247 °C.

Condensers

The condensers are of the dual-pressure surface type. The vacuum inside the condensers is established by steam-jet air ejectors and maintained in normal operation by water-jet air ejectors. The steam exhausted from the low-pressure cylinder condenses over 22 552 brass tubes, which have a surface area totalling 23 400 m².

Each boiler-turbine set has a condenser with a heat exchange capacity of approximately 400 MW.

Generators

Each generator is rated at 667 MVA and produces 600 MW at full load. The sent-out capacity is 575 MW, the difference of 25 MW being used to run the auxiliary equipment of the turbine and boiler plant.

Generator cooling is achieved through two systems. The stator and rotor are cooled by hydrogen at a pressure of 400 kPa, which is circulated through water coolers of the surface type. Circulation of the hydrogen is done by means of two fans that are locked to the rotor shaft. The stator windings and core are cooled by the circulation of demineralised water.

The generators operate at 22 kV and their output is transformed to 400 kV for distribution via the national grid.

Chimneys

At 300 m each, Duvha's two chimneys are the tallest free-standing concrete structures in the Southern Hemisphere. Each chimney contains three flues, so that there is one flue per boiler. The diameter of the chimneys at the base is 22 m, and both are resting on 60 piles of 1,2 m by 18 m. Approximately 40 000 tons of concrete was used in the construction of each stack.

The height of the chimneys means that the flue gases are ejected through the normal inversion layers of the Mpumalanga Highveld atmosphere, and can thus be dispersed over very great distances.

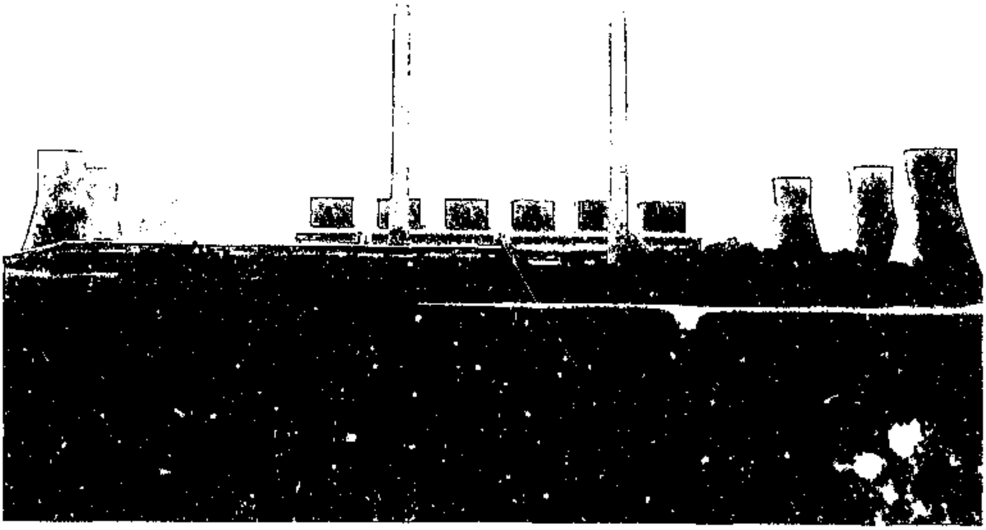
Cooling towers

The six cooling towers at Duvha are designed for a total thermal load of nearly 19 000 GJ/h. The three larger towers are 149 m high and 114 m in diameter at the base, and the minimum shell thickness is 180 mm. At full load the evaporation from one tower amounts to 30 Mℓ a day.

Water supply

Duvha's water requirements can be met from either of two sources. First, water can be pumped from the Vygeboom and Nooitgedacht dams in the eastern part of Mpumalanga, which are fed by the Komati River. These dams can also be supplied with water from the Usutu system. Secondly, water can be obtained from the Witbank Dam in the upper reaches of the Olifants River and supplemented with water from the Grootdraai Dam in the Vaal River near Standerton.

The second source is used as little as possible because it is not as clean as the Komati River. Clean water is highly desirable, as purification and demineralisation costs are proportional to the concentration of material that has to be removed. This second source is therefore only used if the Komati system cannot supply the water.



Duvha-steenkoolmyn

Duvha se oopgroefsteenkoolmyn is in 1976 begin en is nou in staat om oor die 10 miljoen ton steenkool per jaar te voorsien. Reserwes sal teen hierdie produksietempo meer as 30 jaar hou. Die projek, wat nagenoeg R190 miljoen gekos het om daar te stel, is die grootste oopgroefsteenkoolmyn in die Suidelike Halfrond.

[illegible]

Benuttingsprojek vir steenkoolreserwes

[illegible]

Asversameling en -verwydering

[illegible]

Duvha coal mine

Duvha's open-cast coal mine was established in 1976 and it is now capable of supplying just over 10 million tons of coal a year. At that rate of production reserves will last for more than 30 years. The project, which cost some R190 million, is the largest open cast coal mine in the Southern Hemisphere.

Large diameter holes are drilled into the overburden and charged with slurry type explosives. The blasted overburden is stripped by three walking draglines and the exposed coal is drilled, blasted, and loaded into coal haulers for transport to the crushing plant. The capacity of a walking dragline shovel is 53 m³, while that of a coal hauler is 136 tons.

Coal from the crushing plant is transported by a double conveyor belt system, four kilometres in length.

Following on behind the strip mining, the spoil piles are flattened, covered in topsoil and fertilised, and then replanted with various types of grass, trees and shrubs. A nursery has been established for the reclamation programme.

Coal reserves utilisation project

A project has been established whereby the life of the coal reserves for use by Duvha Power Station has been extended to 40 years. The project involves the washing and blending of coal obtained from three different mines, namely Douglas, Middelburg and Wolwekrans.

The long term benefits of this project include the optimal utilisation of coal reserves from these mines, as well as a consistent supply of a better quality blended coal to the power station.

A new coal washing plant was commissioned before the new coal supply agreement between Eskom and Ingwe took effect in [illegible].

Ash collection and disposal

At full load each boiler produces [illegible] tons of coarse ash and [illegible] tons of fly ash every day. The coarse ash and the clinker is removed from the boilers, crushed and fed into sumps, from where it is pumped to the ash dam. The fly ash extracted from the precipitator hoppers is pumped separately to the ash dam [illegible]. Water from the dam [illegible] in a closed loop.

Eventually the dams will be covered with topsoil and grassed to make them an integral part of the landscape.

Tegniese data

Ontwikkelvermoë	3 600 MW
Werknemers	1 110
Werk	3 skofte van 70 elk per 24 uur
Brandstof	
Mynboumaatskappy	Ingwe Groep
Hittewaarde	23,32 MJ/kg (droë basis)
Asinhoud	28,85 persent (droë basis)
Totale jaarlikse verbruik	10 135 000 ton
Steenkoolbergbakkapasiteit	
Nr 1	55 000 ton
Nr 2	110 000 ton
Ketelbunkerkapasiteit	24 120 ton
Steenkoolverbruik by volle las	1 500–1 800 t/h
Meule-installasie	
Stelle 1–4	
Vervaardiger	Babcock & Wilcox (SA)
Tipe	Vertikale spil (ring-en-koeël)
Getal	6 per ketel
Spoed	26 r/min
Ontwerplewering	66 t/h
Stelle 5 en 6	
Vervaardiger	Loesche
Tipe	Vertikale spil (roltipe)
Getal	6 per ketel
Spoed	36 r/min
Ontwerplewering	66 t/h
Ketels	
Vervaardiger	Steinmüller (Afrika) (Edms) Bpk
Tipe	Benson
Getal	6
Maksimum deurlopende ontwerpvermoë	507 kg/s
Finale stoomdruk	17,1 MPa
Finale stoomtemperatuur	540 °C
Herverhitterstoomdruk	3,91 MPa
Herverhitterstoomtemperatuur	540 °C
Aantal branders	24
Hoogte (dak tot vultregter)	94,0 m
Wydte by brandervlak	20,2 m
Diepte by brandervlak	17,2 m
Keteluitsetting (afwaarts)	340 mm
Benaderde volume	13 080 m³
Sakfilters (Eenhede 1, 2 en 3)	
Vervaardiger	ABB Power Tech
Inlaatbeheertemperatuur	126,5 °C maks en 120 °C min
Differensiale druk	2,3 kPa
Filtersnelheid	0,015 m/s
Stelselbeheer	Siemens S135 PLC

Technical data

Generating capacity	3 600 MW
Employees	1 110
Operating	3 shifts of 70 in 24 hours
Fuel	
Mining company	Ingwe Group
Calorific value	23,32 MJ/kg (dry basis)
Ash content	28,85 per cent (dry basis)
Total annual consumption	10 135 000 tons
Coal stathe capacity	
No 1	55 000 tons
No 2	110 000 tons
Boiler bunker capacity	24 120 tons
Coal consumed at full load	1 500–1 800 t/h
Milling plant	
Sets 1–4	
Manufacturer	Babcock & Wilcox (SA)
Type	Vertical spindle (ball and ring)
Number	6 per boiler
Speed	26 r/min
Rated output	66 t/h
Sets 5 and 6	
Manufacturer	Loesche
Type	Vertical spindle (roller type)
Number	6 per boiler
Speed	36 r/min
Rated output	66 t/h
Boilers	
Manufacturer	Steinmuller (Africa) (Pty) Ltd
Type	Benson
Number	6
Maximum continuous rating	507 kg/s
Final steam pressure	17,1 MPa
Final steam temperature	540 °C
Reheater steam pressure	3,91 MPa
Reheater steam temperature	540 °C
Number of burners	24
Height (roof to hopper)	94,0 m
Width at burner level	20,2 m
Depth at burner level	17,2 m
Boiler expansion (downwards)	340 mm
Approximate volume	13 080 m³
Bag filters (Units 1, 2 and 3)	
Manufacturer	ABB Power Tech
Control temperature inlet	126,5 °C max and 120 °C min
Differential pressure	2,3 kPa
Filter velocity	0,015 m/s
System control	Siemens S135 PLC

Effektiewe materiaaloppervlakte	86 170 m²
Lugbesoedeling	< 50 mg/Sm³
Elektrostatiese ontstowwers	
Vervaardiger	Lurgi SA
Beheergastemperatuur	150 °C
Bedryfsvolume	2 000 000 m³/h
CO_2-inhoud	13,76%
Gasdruk (staties)	41 mbar
Gasstoflading (inlaat)	30 g/m³
Suiwergasstoflading	120 mg/m³
Turbines	
Vervaardiger	GEC Turbine Generators (Edms) Bpk
Tipe	Veelsilinder-impulsreaksie
Ontwerpvermoë (generatorlewering)	600 MW
Spoed	3 000 r/min
Superverhittestoom-druk, HD-afsluitklepinlaat	16,1 MPa (abs)
Superverhittestoom-temperatuur, HD-afsluitklepinlaat	535 °C
Stoomdruk, HD-uitlaat	4 MPa (abs)
Stoomtemperatuur, HD-uitlaat	332 °C
Herverhittestoom-druk, MD-inlaat	3,75 MPa (abs)
Herverhittestoom-temperatuur, MD-inlaat	535 °C
Stoomdruk, MD-uitlaat	0,42 MPa (abs)
Stoomtemperatuur, MD-uitlaat	252 °C
Stoomdruk in kondensator	8 kPa (abs)
Gemiddelde temperatuur in kondensator	37 °C
Hitteverbruik (MDO)	8 220 kJ/kWh
Stoomvloei	520 kg/s
Generators	
Vervaardiger	GEC Turbine Generators (Edms) Bpk
Ontwerpvermoë	600 MW (maksimum deurlopende ontwerpvermoë)
Aansluiterspanning	22 kV (50 Hz)
Arbeidsfaktor	0,9 (naloop)
Koelmedium	Waterstof teen 400 kPa
Generatortransformators	
Vervaardiger	ASEA Electric (SA) Bpk
Opwerpvermoë	700 MVA
Aansluiterspanning: Primêre	22 kV
Aansluiterspanning: Sekondêre	420 kV
Koeltorings	
Getal	6
Tipe	Hiperbolies met natuurlike trek

Totale afmetings	Stelle 1–3	Stelle 4–6
Hoogte	133,7 m	147 m
Damdiameter	99,7 m	113,5 m

Effective cloth area	86 170 m²
Stack emission	< 50 mg/Sm³
Electrostatic precipitators	
Manufacturer	Lurgi SA
Control gas temperature	150 °C
Operating volume	2 000 000 m³/h
CO_2 content	13,76%
Gas pressure (static)	41 mbar
Gas dust load (inlet)	30 g/m³
Clean gas dust load	120 mg/m³
Turbines	
Manufacturer	GEC Turbine Generators (Pty) Ltd
Type	Multicylinder impulse reaction
Rating (generator output)	600 MW
Speed	3 000 r/min
Superheated steam pressure, HP stop valve inlet	16,1 MPa (abs)
Superheated steam temperature, HP stop valve inlet	535 °C
Steam pressure, HP outlet	4 MPa (abs)
Steam temperature, HP outlet	[illegible] °C
Reheated steam pressure, IP inlet	3,75 MPa (abs)
Reheated steam temperature, IP inlet	535 °C
Steam pressure, IP outlet	0,42 MPa (abs)
Steam temperature, IP outlet	252 °C
Steam pressure inside condenser	8 kPa (abs)
Average temperature inside condenser	37 °C
Heat consumption (MCR)	8 220 kJ/kWh
Steam flow	520 kg/s
Generators	
Manufacturer	GEC Turbine Generators (Pty) Ltd
Rated capacity	600 MW (maximum continuous output)
Terminal voltage	22 kV (50 Hz)
Power factor	0,9 (lagging)
Cooling medium	Hydrogen at 400 kPa
Generator transformers	
Manufacturer	ASEA Electric (SA) Ltd
Rated capacity	700 MVA
Terminal voltage: Primary	22 kV
Terminal voltage: Secondary	420 kV
Cooling towers	
Number	6
Type	Hyperbolic natural draught

Overall dimensions	Sets 1–3	Sets 4–6
Height	133,7 m	147 m
Pond diameter	99,7 m	113,5 m

Totale afmetings	Stelle 1-3	Stelle 4-6
Keëldiameter	51,4 m	56,5 m
Boonste diameter	55,25 m	60,85 m
Minimum rompdikte	180 mm	180 mm
Nominale vloeitempo koeltoring	45 000 m^3/h	48 800 m^3/h
Temperatuur in	39,44 °C	34,7 °C
Temperatuur uit	22,5 °C	19,1 °C
Verdamping by dienspunt	1 270 m^3/h	1 270 m^3/h
Nominale warmteverlies	886 MW	886 MW
Maksimum gewaarborgde warmteverlies	1 275 MW	1 275 MW
Sirkuleerwaterpompe		
Getal	12	
Totale vermoë	78 m^3/s	
Skoorstene		
Getal (3 rookgasgange elk)	2	
Hoogte	300 m	
Windskermdiameter (voetstuk)	22 m	

Hoofkontrakters

Grondwerke	Grinaker Construction (Edms) Bpk
Siviele ingenieurswerk	LTA Construction
Staalwerk (subkontrakteur)	Dorbyl Ltd
Ketels	Steinmüller (Afrika) (Edms) Bpk
Turbines	GEC Turbine Generators (Edms) Bpk
Generatortransformators	ASEA Electric (SA) Bpk
Koeltorings	Hamon Sobelco (met LTA Construction - siviele subkontrakteur)
Skoorsteen 1	Monahan en Frost (Edms) Bpk
Skoorsteen 2	Murray & Roberts (Transvaal) (Edms) Bpk
Kabelwerk	Hubert Davies (Edms) Bpk
Instrumentasie	Siemens Bpk
Prosesrekenaars	Honeywell
Brandbeheerstelsel	Mather & Platt (Edms) Bpk
Steenkoolhantering (bergbakke)	High Structures (Edms) Bpk
Vervoerbande	Spencer Melksham (Edms) Bpk
Asinstallasies 1-3	DB Thermal
Asinstallasies 4-6	Steinmuller (Afrika) (Edms) Bpk
Konstruksie van waterring	Hall Longmore & Kie
Koelwaterpompe 1-3	Amalgamated Power Engineering (SA) (Edms) Bpk
Koelwaterpompe 4-6	Sulzer Bros (SA) Bpk
Rioolaanleg	Satec Hudamec
Materiaalontstowwersakke 1-3	ABB
Elektrostatiese ontstowwers 4-6	Lurgi (SA) (Edms) Bpk
Waterbehandelingsaanleg	Foster Wheeler

Overall dimensions	Sets 1-3	Sets 4-6
Throat diameter	51,4 m	56,5 m
Diameter at top	55,25 m	60,85 m
Minimum shell thickness	180 mm	180 mm
Nominal flow rate CT	45 000 m^3/h	48 800 m^3/h
Temperature in	39,44 °C	34,7 °C
Temperature out	22,5 °C	19,1 °C
Evaporation at duty point	1 270 m^3/h	1 270 m^3/h
Nominal heat rejection	886 MW	886 MW
Maximum guaranteed heat rejection	1 275 MW	1 275 MW
Circulating-water pumps		
Number	12	
Total capacity	78 m^3/s	
Chimneys		
Number (3 flues each)	2	
Height	300 m	
Windshield diameter (base)	22 m	

Main contractors

Earthworks	Grinaker Construction (Pty) Ltd
Civil engineering	LTA Construction
Steelwork (subcontractor)	Dorbyl Ltd
Boilers	Steinmüller (Africa) (Pty) Ltd
Turbines	GEC Turbine Generators (Pty) Ltd
Generator transformers	ASEA Electric (SA) Ltd
Cooling towers	Hamon Sobelco (with LTA Construction - civil subcontractor)
Chimney 1	Monahan and Frost (Pty) Ltd
Chimney 2	Murray & Roberts (Transvaal) (Pty) Ltd
Cabling	Hubert Davies (Pty) Ltd
Instrumentation	Siemens Ltd
Process computers	Honeywell
Fire control system	Mather & Platt (Pty) Ltd
Coal handling (staithes)	High Structures (Pty) Ltd
Conveyors	Spencer Melksham (Pty) Ltd
Ash plants 1-3	DB Thermal
Ash plants 4-6	Steinmüller (Africa) (Pty) Ltd
Construction water ring	Hall Longmore & Company
Cooling-water pumps 1-3	Amalgamated Power Engineering (SA) (Pty) Ltd
Cooling-water pumps 4-6	Sulzer Bros (SA) Ltd
Sewage plant	Satec Hudamec
Fabric filter plants 1-3	ABB
Electrostatic precipitators 4-6	Lurgi (SA) (Pty) Ltd
Water treatment plant	Foster Wheeler

Gepubliseer deur
Kommunikasiedienste, Eskom
Mei 1985 (hersien Desember 1996)
Navrae moet gerig word aan:
Kommunikasiedienste
Eskom
Posbus 1091
Johannesburg 2000
Republiek van Suid-Afrika
Faks (011) 80)-

Ontwerp deur Design Unit
Reproduksie deur Typesetting & Repro Services bk
Gedruk deur Hortors Print

Published by
Communication Services, Eskom
May 1985 (revised December 1996)
Address enquiries to:
Communication Services
Eskom
PO Box 1091
Johannesburg 2000
Republic of South Africa
Fax (011) 800-4390

Designed by Design Unit
Reproduction by Typesetting & Repro Services cc
Printed by Hortors Print

APPENDIX B - PC SIMULATION SOFTWARE

Choice of the most suitable software is c[illegible] the success of this simulation in meeting the objectives of the dissertation. The [illegible] considerations when evaluating the software were:

a) Is it able to perform the required task?
b) How Fast is the software?
c) How user friendly is the software?
d) Is the simulation "program" self documenting?
e) Is the language 100% graphical, or does it require "behind the scenes" programming?
f) How stable/reliable is the software (i.e. does it ever crash)?
g) Is it possible to interface live external data to the simulation?
h) How accurate are the integration algorithms?
i) Would the simulation be relatively easy to figure out and understand by a 3rd person?
j) Is it possible to develop a sophisticated user interface for the simulation?

The following hardware was used for the development of the simulation as well as for the testing of the different software packages

- Intel Pentium 133MHz CPU
- PCI Motherboard with 256Kb Pipeline Burst Cache
- 24MB RAM
- 64 bit High Speed graphics card, capable of 1152 x 864 resolution (or higher)
- 20" Phillips Monitor (1152 x 864 capable)
- 1 GB SCSI Hard Drive

The test PC was configured with Windows NT version 4.0 which, although being slightly slower than the previous version 3.51 (due to additional RAM requirements), appeared to be more stable.

Comparision	VisSim	Simulink (Matlab)	Labview
Ease Of Use			
Getting Started	Very Easy - 10 minutes for first simulation	Fairly Simple	Difficult
For the Experienced User	Very Easy	Fairly Simple	Easy, but complex
Functionality			
Integration	Euler, Runge Kutta 2nd, Runge Kutta 4th, Adaptive Runge Kutta 5th, Adaptive Bulirsh-Stoer, Backward Euler	Linsim, Runge Kutta 3rd, Runge Kutta 5th, Gear's Predictor-Corrector, Adams Predictor-Corrector, Euler	No simulation methods are bundled with the program, although some of the complex methods may be implemented by the programmer
Equations	It is not possible to write out equations in the english format. All functions must be visually constructed using the fundamental blocks (adder, mul, div power etc)	Functions are implemented with the Fcn block using C-language type syntax	Functions are possible
Signal Analysis	Available as add-on packages for Bode, Nyquist etc	Only Spectral and related analysis	Very powerful signal analysis
Fourier Transforms	Add-on	Limited	Very Comprehensive
Filter Implementation	Add-on	Many	Many
Fuzzy Logic	Add-on	??	??
Neural Networks	Add-on	Add-on	Yes
Optimization	Limited version included in main package. More powerful version available as an add-on	??	??
Feedback Control	Very Easy	Very Easy	Very Awkward. This language is not suited for feedback
Virtual Instruments	No	No	Yes - Very Powerful
Grouping	Yes	Yes	This is achieved by means of virtual instruments which are far more powerful than grouping of blocks
Performance			
Timing	Running in Real time very easy as well as faster or slower than real time	Running in Real time very easy as well as faster or slower than real time	Real time operation does not appear to function correctly. It is difficult to determine or regulate the simulation speed
Speed	Very fast. Add-on module compiles the code into C for extra speed	Very fast. Add-on module compiles the code into C for extra speed.	not tested
Accuracy	Depends on integration method	Depends on integration method	Depends on integration method implemented by the programmer
Stability	Depends on Step size and integration method	Depends on Step size and integration method	not tested
Stability / Robustness	Crashes sometimes during specific editing operations. These operations can be avoided. Does not crash when running simulation.	not tested	not tested
User Interface	Very simple. All signal inputs, control blocks and signal outputs are displayed on the same screen. In display mode it is possible to hide blocks and lines. The advantage of this is that the simulation is self documenting.	Very similar to Vissim	A simulation consists of two "planes", the first being the user interface and the second being the simulation itself. The advantage of this is the ability to construct very functional and pleasing user interfaces
Documentation	The simulation is self documenting as each module's use is self explanatory	The simulation is self documenting as each module's use is self explanatory	The blocks are less self explanatory than those emplyed by Vissim and Labview. The separation of the simulation from the user interface further complicates the understanding of the simulation
Cost	At the time of writing, Labview and MatLab were both approximately twice as expensive as VisSim		
Overall Rating	Vissim is a very powerful, yet simple package designed specifically for the simulation environment	This package, along with Matlab, forms an extremely powerful and extensive environent for almost any mathematical or engineering task	This package is seen more as a powerful visual programming language than a simulation or engineering tool. Specific items/features are lacking which make use of this package for any modest simulation project very difficult, if not impossible

These results should only be used as a guideline as to the suitability of each of the software packages to simulation. In order to do a comprehensive evaluation of these and other packages it would be necessary to spend many hours getting to know the packages before running similar tests on each package. Many other simulation packages exist, most of which are prohibitively expensive and complicated to use.

APPENDIX C - SIMULATION ACCURACY

C.1. Turbine Model

In order to reduce the complexity of the initial simulation, certain simplifications were made when constructing the model. One area which was simplified was the model predicting power generated by the HP, IP and LP turbines.

In order to quantify the errors due to this simplification, it is necessary to compare the simulated power generated with the actual power generated while expanding steam through a single turbine. For this comparison it was decided to use the IP/LP turbines as they produce approximately 60% of the total power. They were also chosen due to the relatively constant discharge pressure (condenser pressure).

The simplified formula that was used to calculate the turbine power is:

$$\mathrm{TurbinePower}_{MW} = \mathrm{Constant} \times \mathrm{SteamFlow}_{kg} \qquad \text{Eq 44}$$

The more advanced boiler simulation used the following more accurate formula for calculating the power generated by the turbine:

$$\mathrm{TurbinePower}_{MW} = \mathrm{Constant} \times \mathrm{SteamFlow}_{kg} \times \mathrm{SteamEnthalpyDrop}_{kJ/kg} \qquad \text{Eq 45}$$

This formula assumes that all energy extracted from the steam is converted into electrical energy, thus neglecting the inefficiencies of the turbine and generator as well as the effects of friction and heat loss.

Steam Initial Conditions Before Turbine Governor Valves:

- Hot Reheat Pressure: 4.5 MPa
- Hot Reheat Temperature: 540 °C
- Hot Reheat Enthalpy 3540 kJ/kg

Governor Valve:

- Pressure Drop: from 0 MPa to 4.45 MPa (depending on governor valve position)

IP/LP Turbine Steam Inlet Conditions:

- Pressure: from 4.5 MPa to 0.05 MPa (depending on governor valve position)

IP/LP Turbine Steam Discharge Conditions:

- Pressure: 200 Pa

- Temperature: 20 °C

In order to reduce power generation, the turbine governor valve is checked in, resulting in an increase in the pressure drop across the valve. The flow of steam through the partially open governor valve results in an increase in the steam entropy. With all other influences ignored (friction etc.) the enthalpy remains constant throughout inlet throttling.

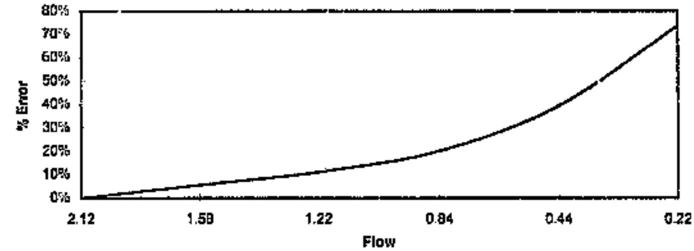
A set of 6 arbitrary pressures were chosen to represent the turbine inlet pressures for various positions of the governor valve, from 0% to almost 100% closed. Steam tables were used to tabulate the corresponding discharge enthalpies, assuming no increase in entropy during the expansion through the turbine.

Equations 44 and 45 were then evaluated at each pressure, normalized to give equal power at high pressure, and then plotted in two formats; percentage error and absolute magnitudes. These graphs show that equation 44 results in a high percentage error at low flow (74% error). The absolute magnitude of this error is negligible (204 kJ/kg) when compared to the governor valve inlet enthalpy of 3540kJ/kg.

Turb. Inlet Press.	4.50	2.50	1.50	0.70	0.20	0.05
Inlet Enthalpy	3540.00	3540.00	3540.00	3540.00	3540.00	3540.00
Disch Enth	2078.20	2154.50	2224.20	2323.00	2491.00	2698.00
Enthalpy Drop	1461.80	1385.50	1315.80	1217.00	1049.00	842.00

Flow = Constant * SQRT (InletPressure - OutletPressure) where Constant = 1
Power (equation 44) = Constant * Flow where Constant = 1461.8
Power (equation 45) = Constant * Enthalpy Drop * Flow where Constant = 1

Flow	2.12	1.58	1.22	0.84	0.44	0.22
Power (eq. 43)	3100.26	2310.38	1789.14	1221.28	650.46	320.26
Power (eq. 44)	3100.26	2189.79	1610.44	1016.76	466.78	184.47
Percent Error	0%	6%	11%	20%	39%	74%



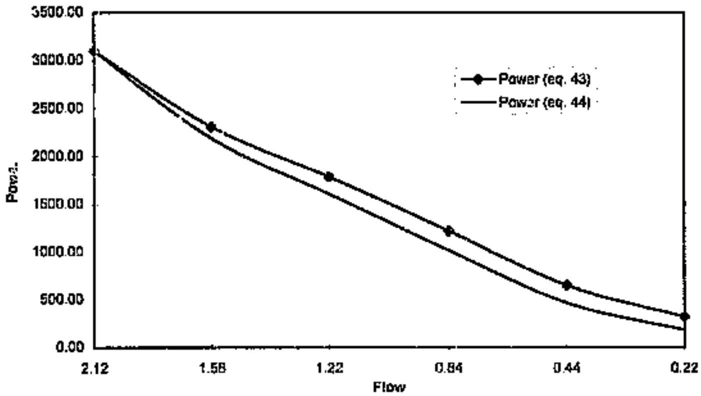


Figure 6-2: Power Generation: Comparison of Two Methods

C.2. Accuracy of Alternative Simulation

C.2.1. Finite Element Methods

Breaking the boiler and turbine down into a number of separate "cells" (which are then treated in terms of mass and energy balance) is a finite approximation to the infinite number of "cells" which obey the physical laws momentum and energy conservation along any length of boiler tube pipe.

The main limitation to the number of "cells" used in the simulation is the computation time which increases linearly with an increase in the number of "cells". Figure 6-3 below shows the relative computation times for a simulation consisting of 1, 2, 4 and lastly 8 cells. The x-axis is the number of "cells" in the simulation, while the y-axis shows the time taken to complete a certain task. The y-intercept of 10 time units may be attributed to fixed overheads such as plotting the graphs. The graph shows that computation time is roughly proportional to the number of "cells".

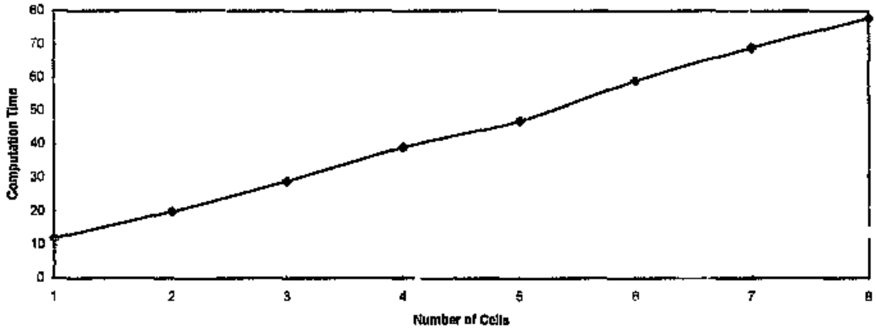


Figure 6-3: Computation Time vs Number of Cells

Figure 6-4 and Figure 6-5 were calculated according to the same formulas used in the simulation to model the pressure drop and flow through a pipe. The gain in accuracy appears to decrease exponentially with the increase in number of cells.

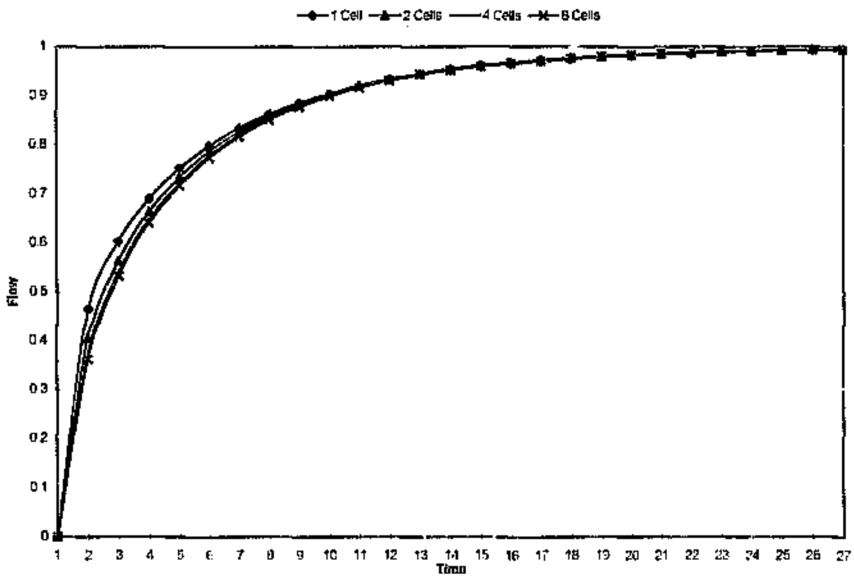


Figure 6-4: Superheater Flow Simulation (1,2,4 and 8 Cells)

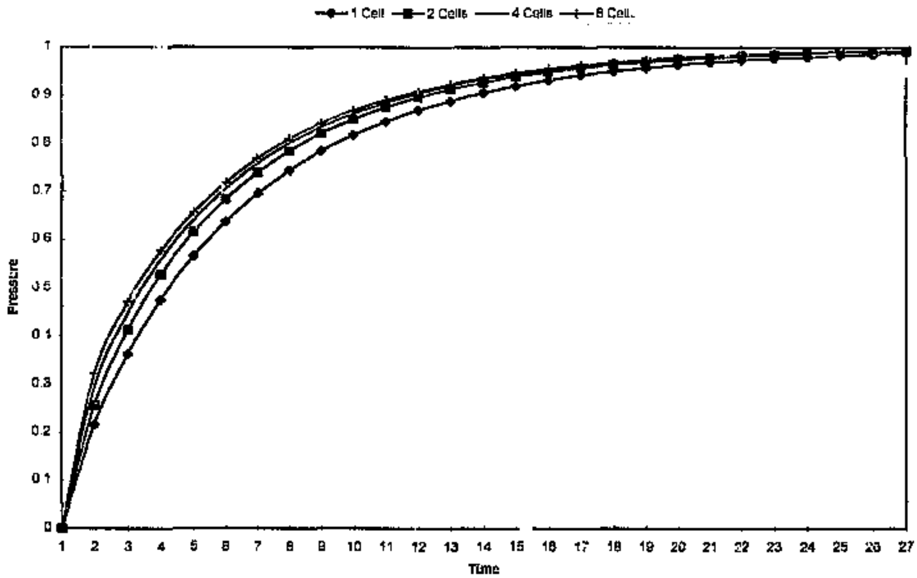
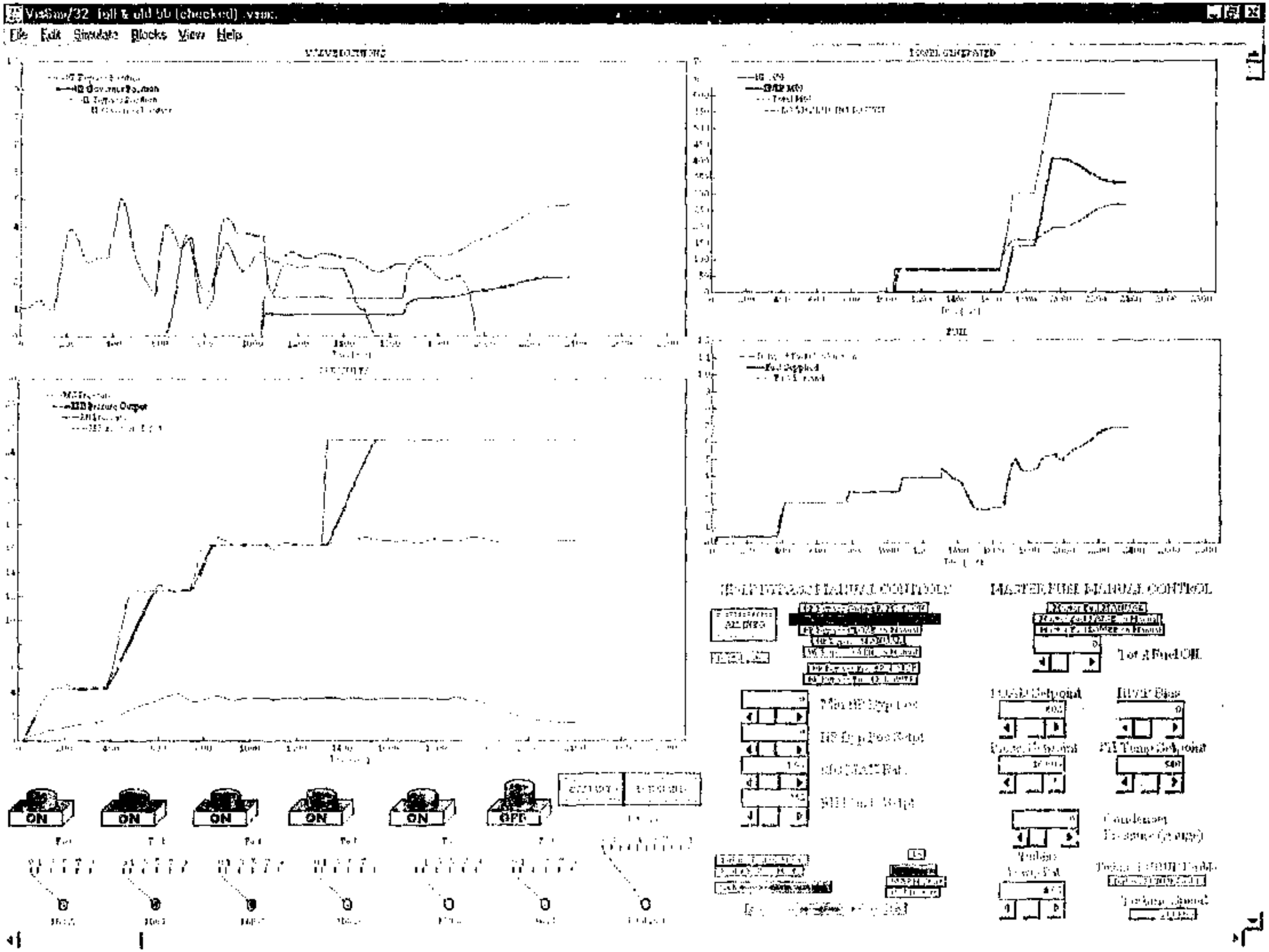
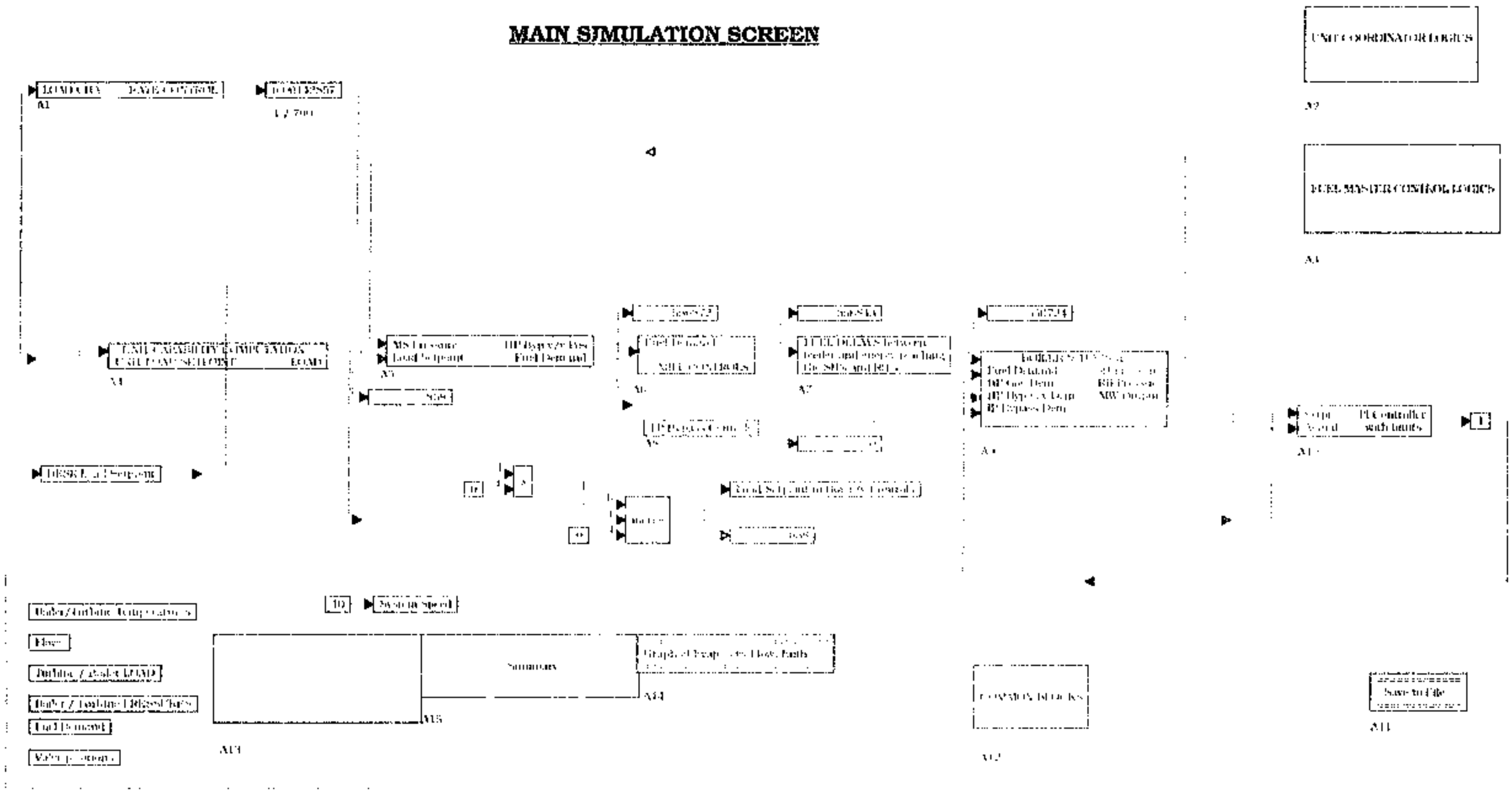


Figure 6-5: Superheater Pressure Simulation (1,2,4 and 8 Cells)

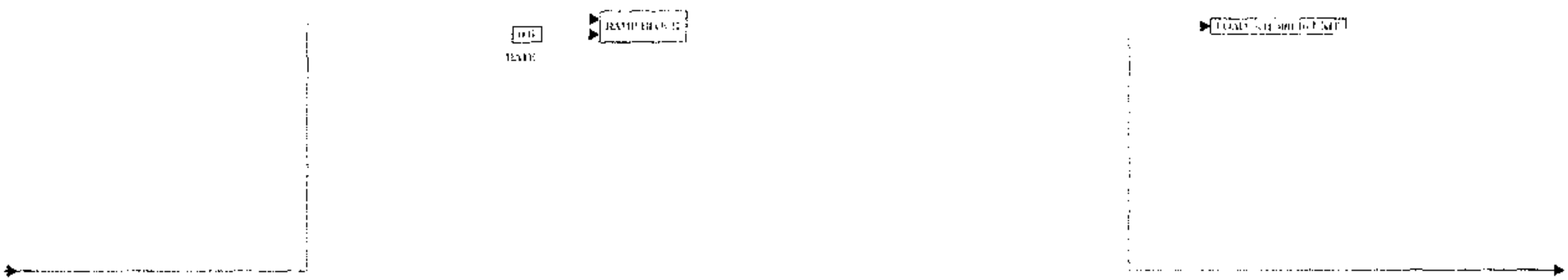
APPENDIX D - SIMULATION SOURCE CODE


VisSim/32 - full & old bb (checked).vsm
File Edit Simulate Blocks View Help
POWER GENERATED
FUEL
ON
ON
ON
ON
ON
OFF

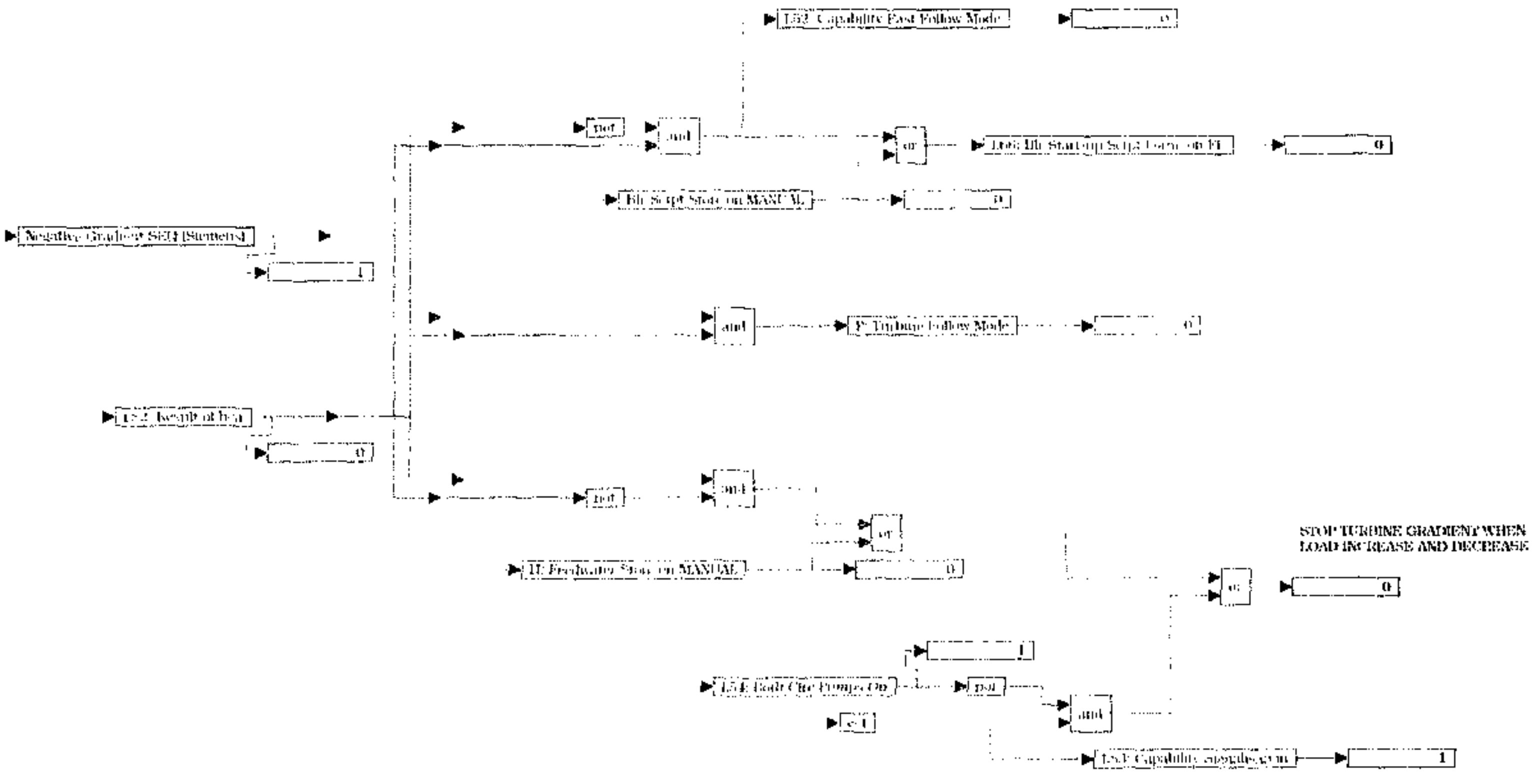
MAIN SIMULATION SCREEN



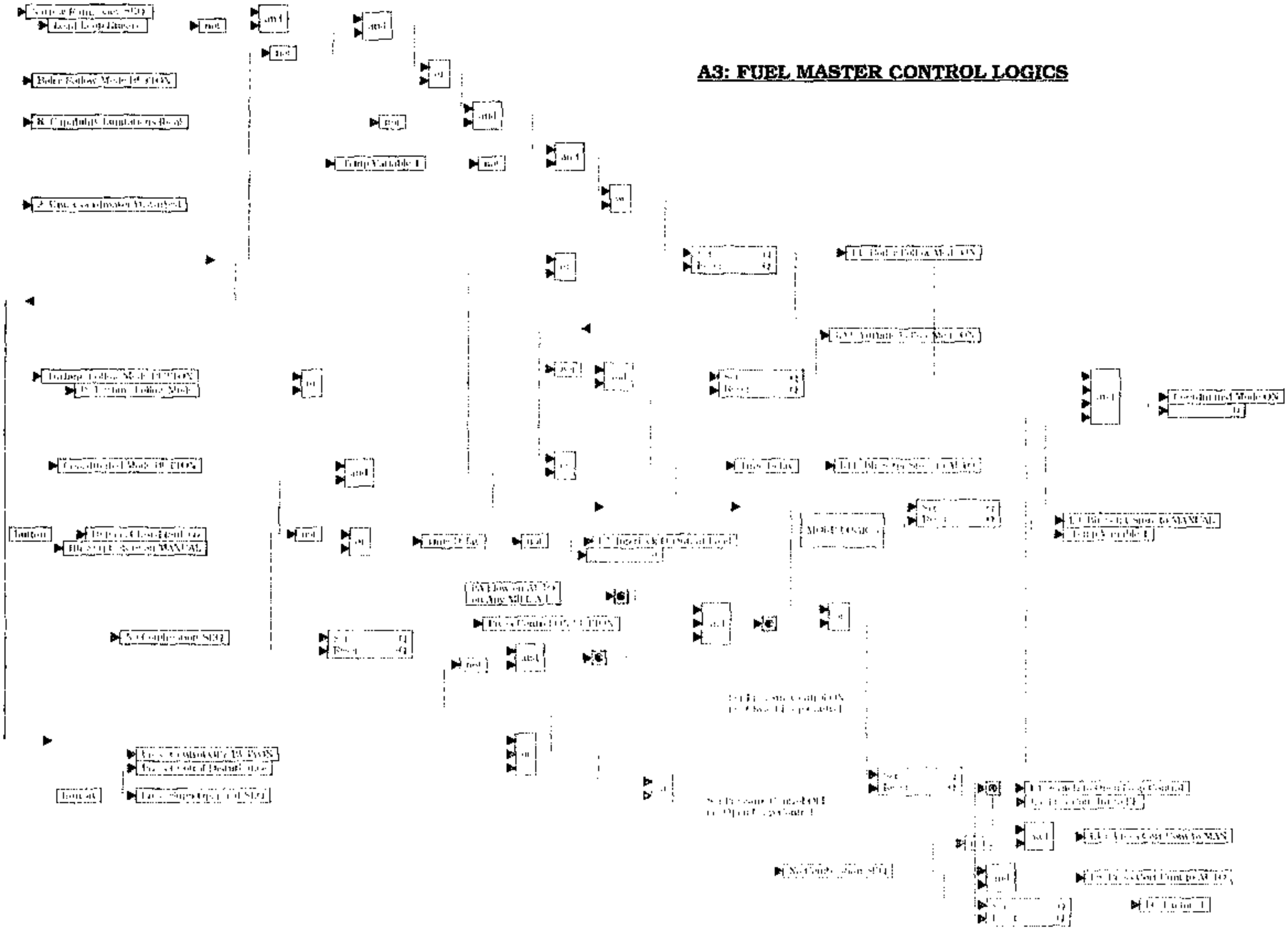
A1; LOAD CHANGE RATE LIMITER



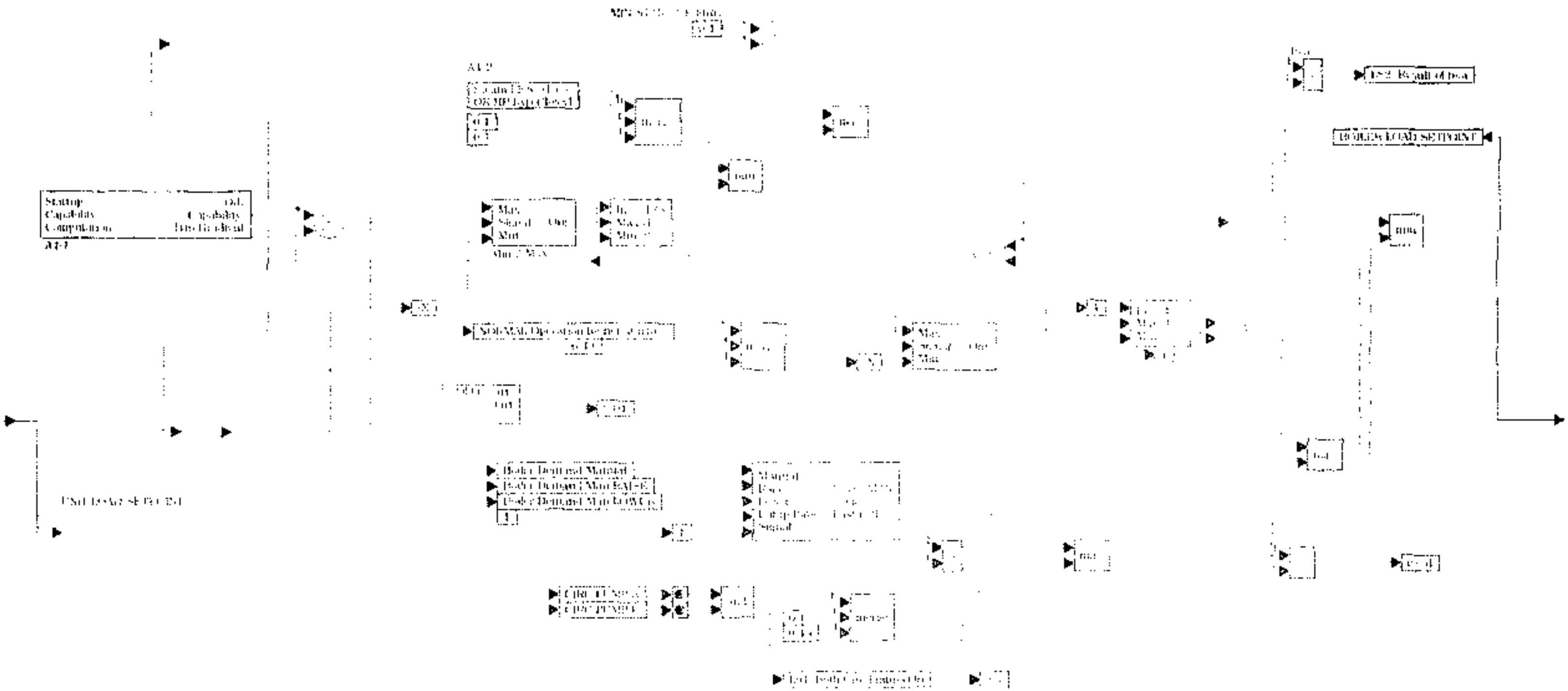
A2: UNIT COORDINATOR LOGICS



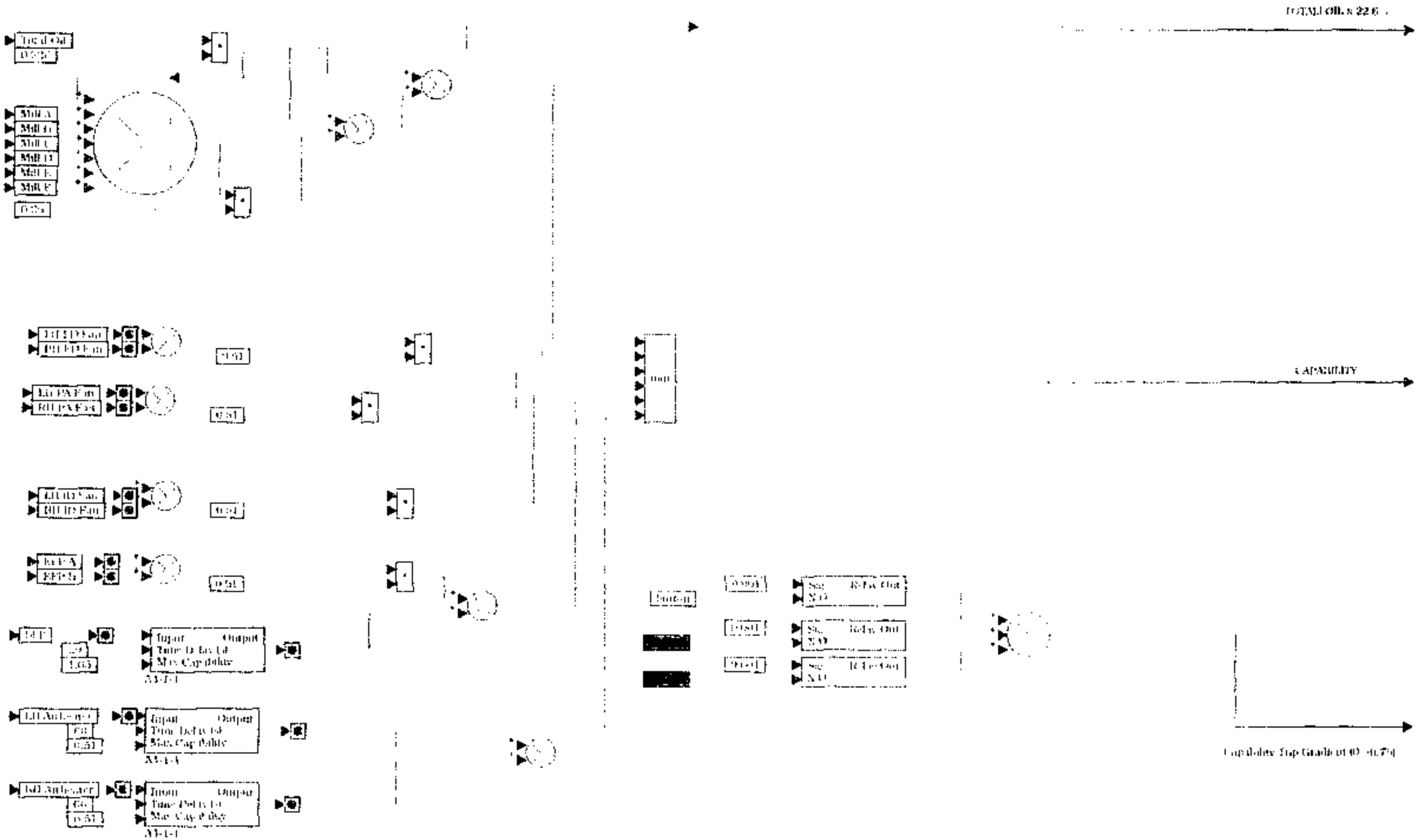
A3: FUEL MASTER CONTROL LOGICS



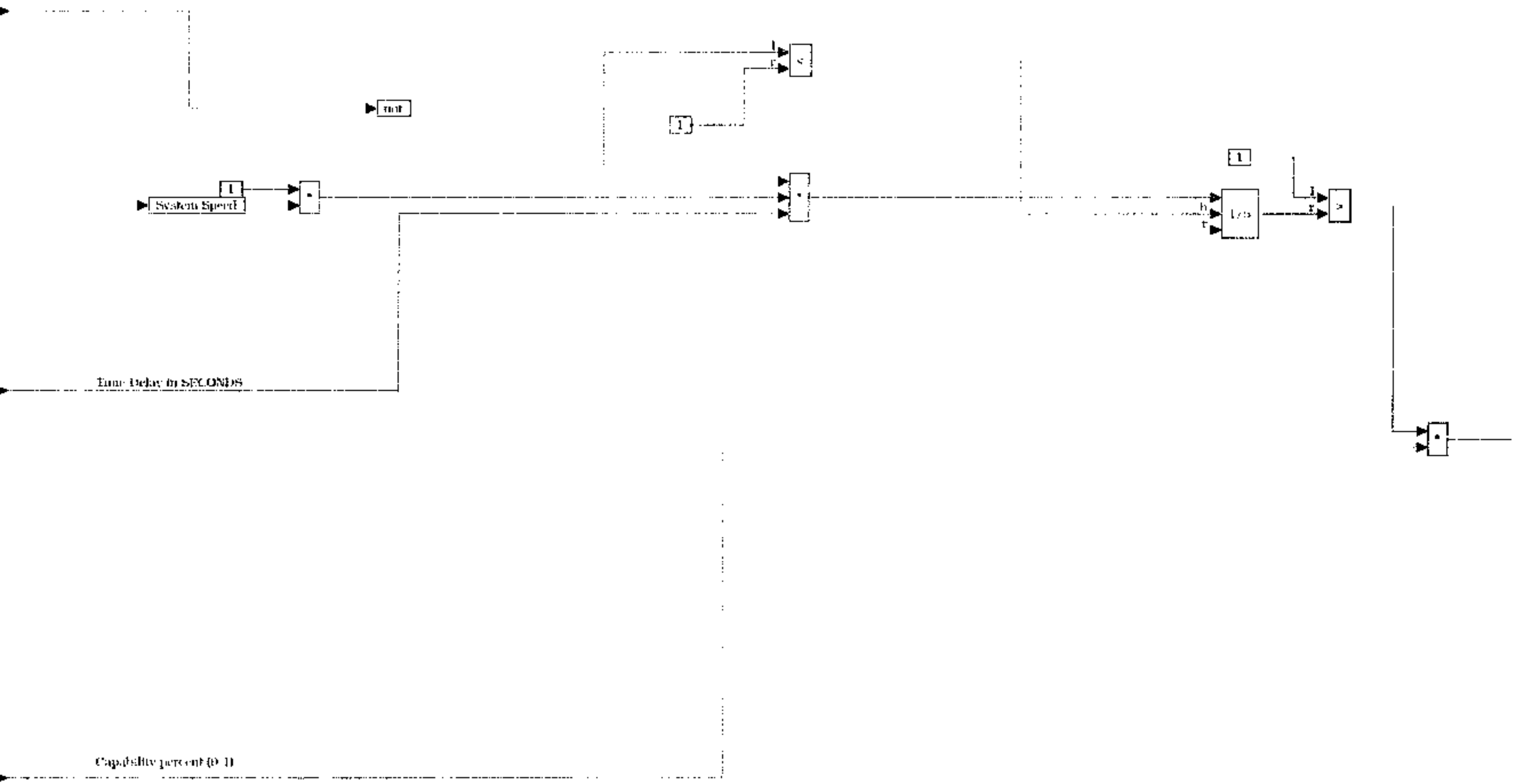
A4: UNIT CAPABILITY COMPUTATION AND UNIT LOAD SETPOINT



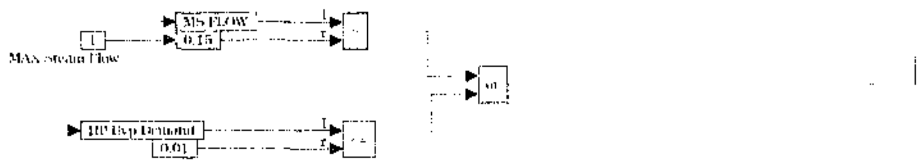
A4-1: UNIT CAPABILITY COMPUTATION



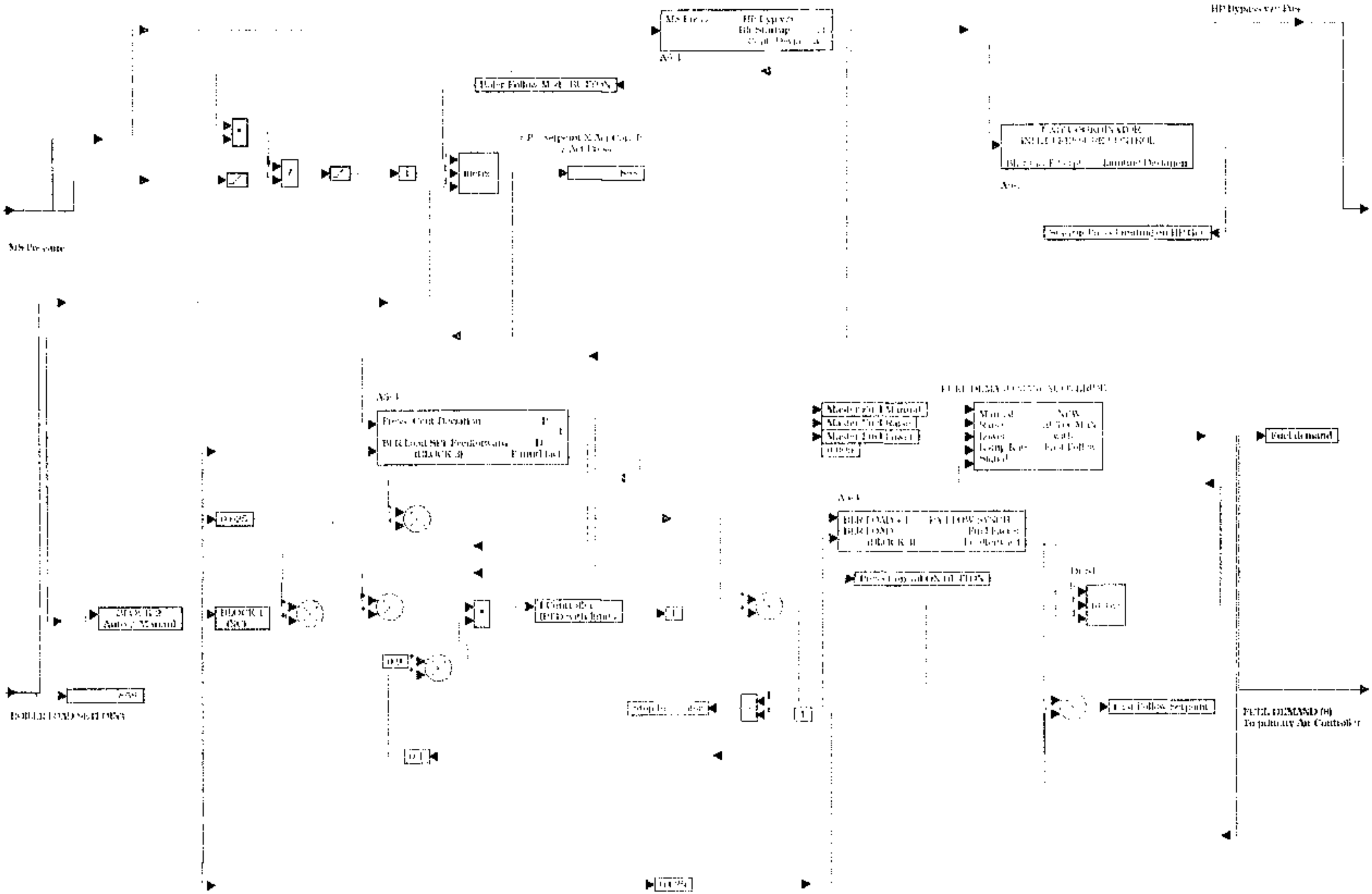
A4-1-1: TIME DELAYED RELAY



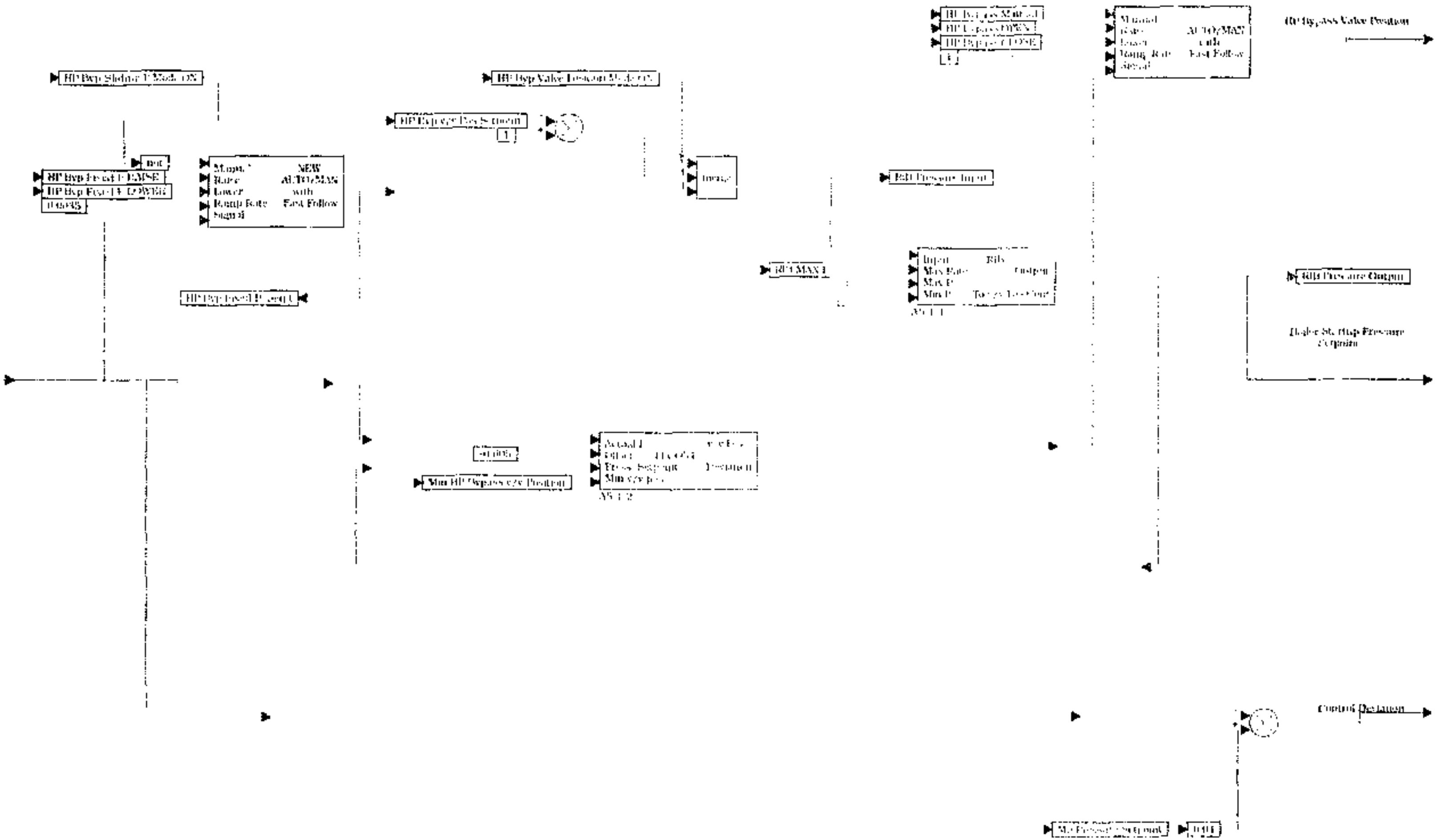
A4-2: STEAM FLOW > 15% OR HP BYPASS CLOSED



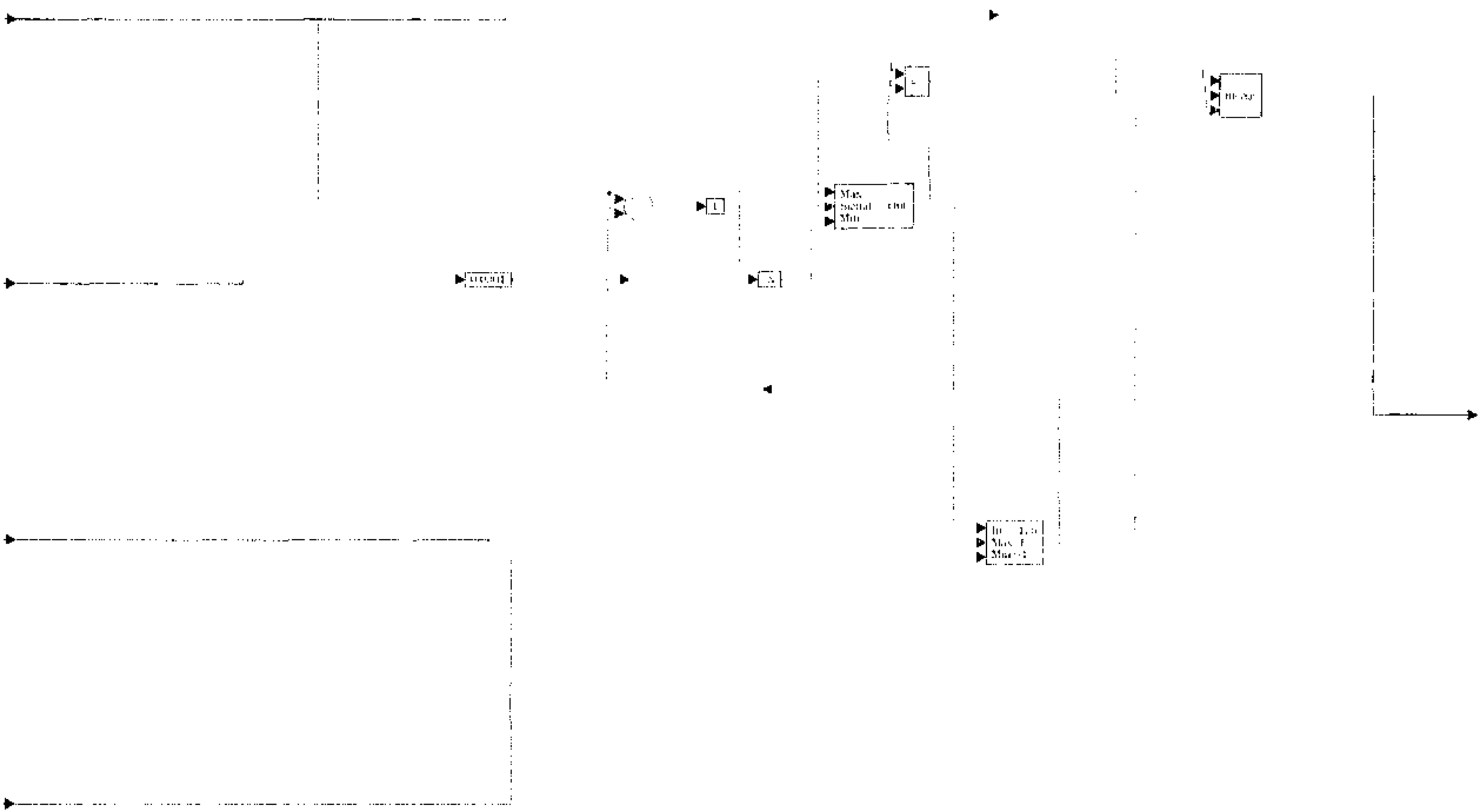
A5: HP BYPASS AND FUEL DEMAND CONTROLLER



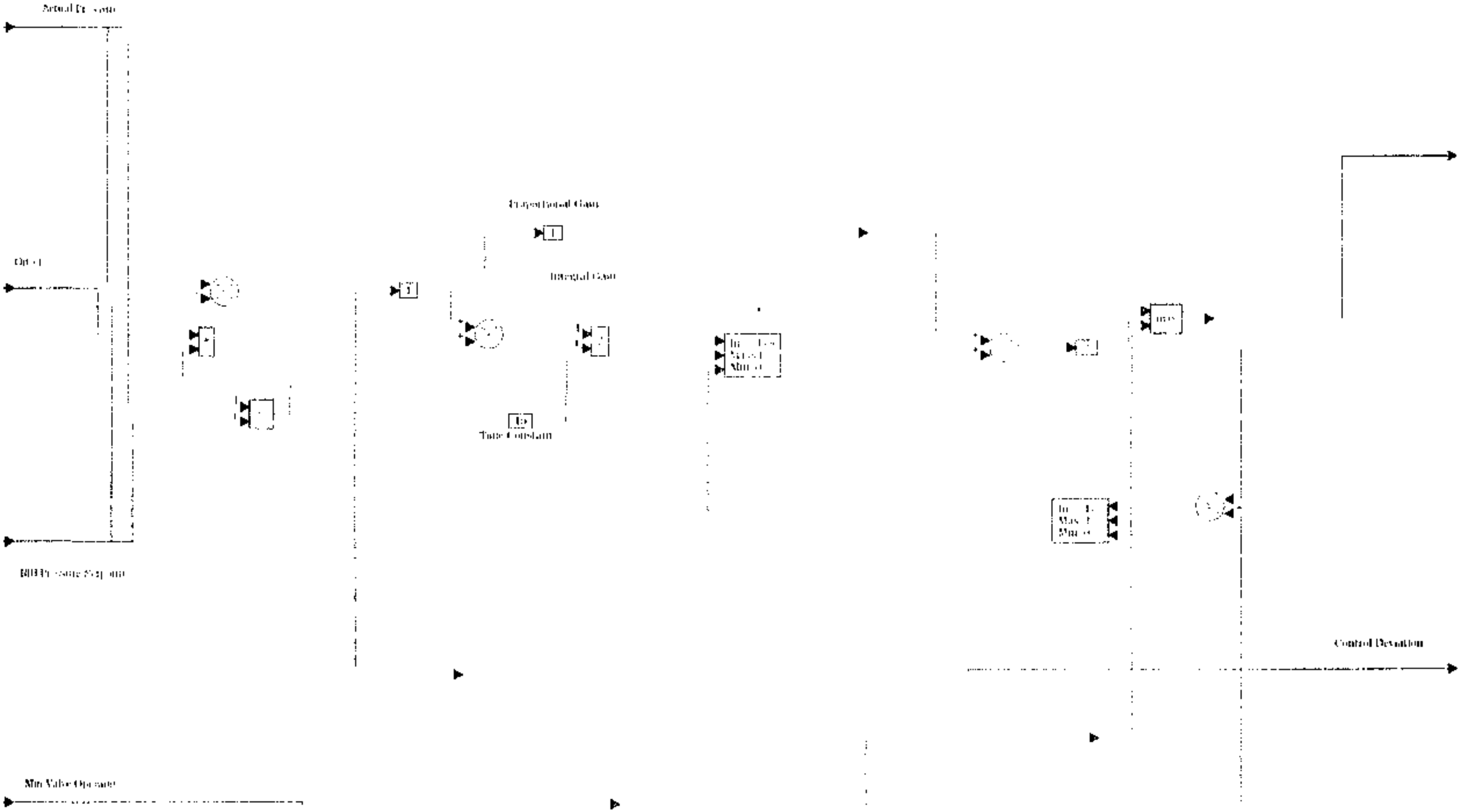
A5-1: HP BYPASS CONTROLLER



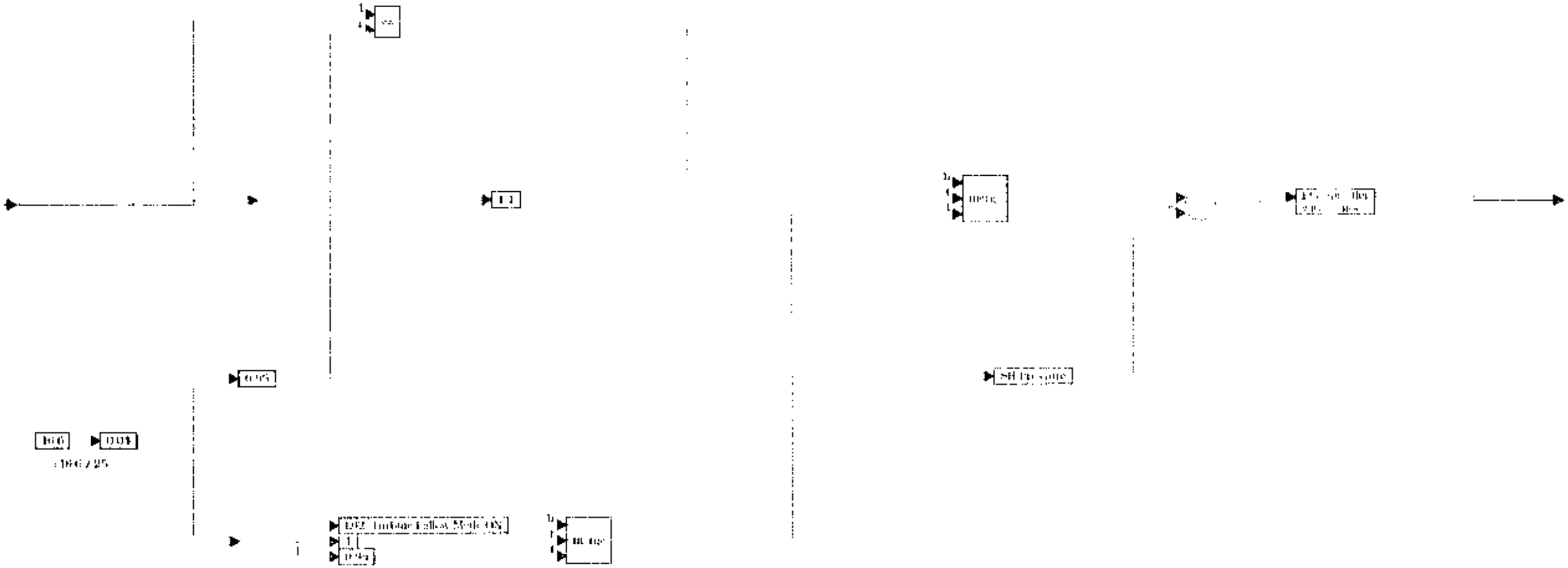
A5-1-1: LOAD SETPOINT RATE LIMITER



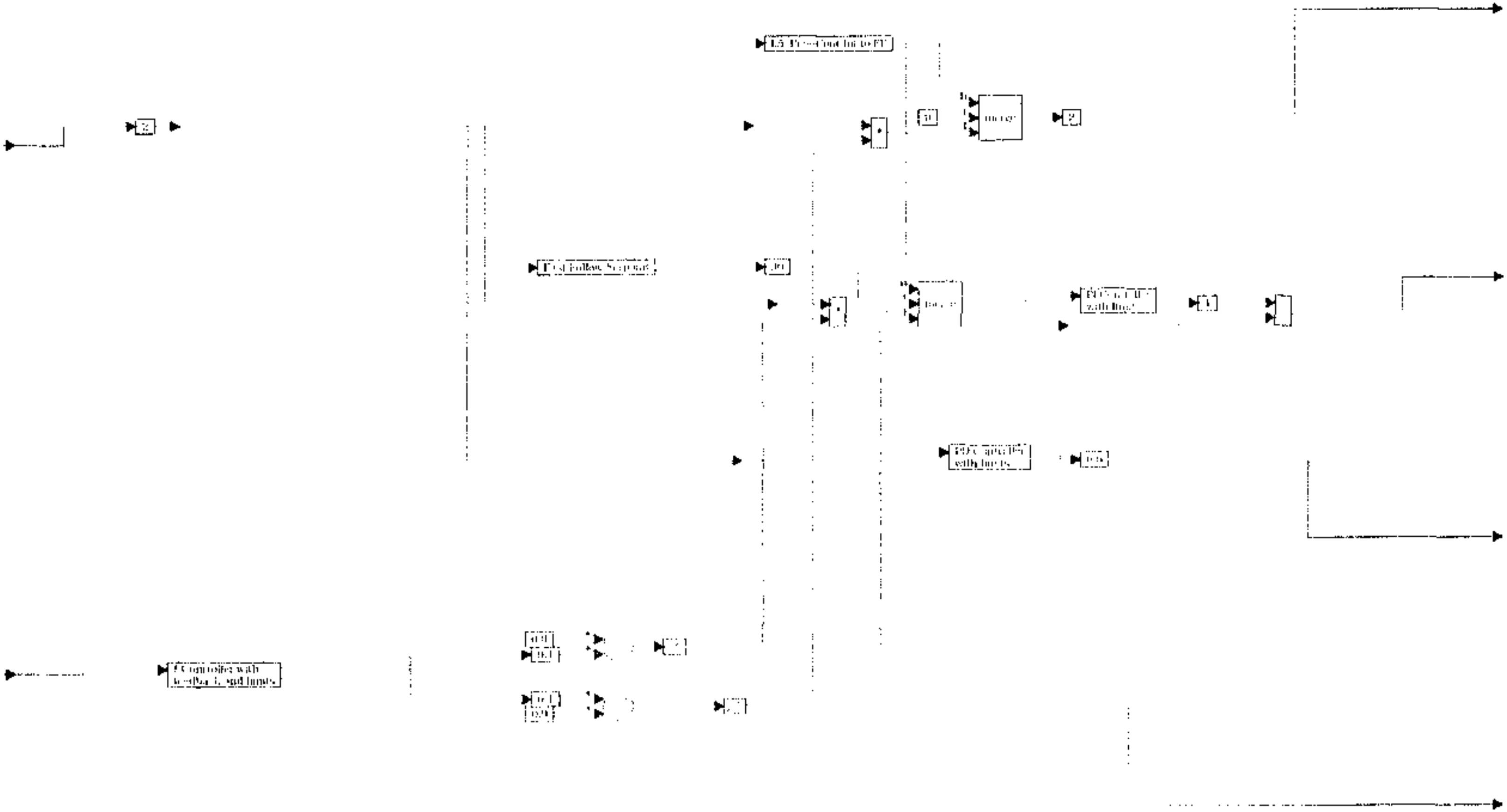
A5-1-2: MAIN HP BYPASS PRESSURE CONTROL PID



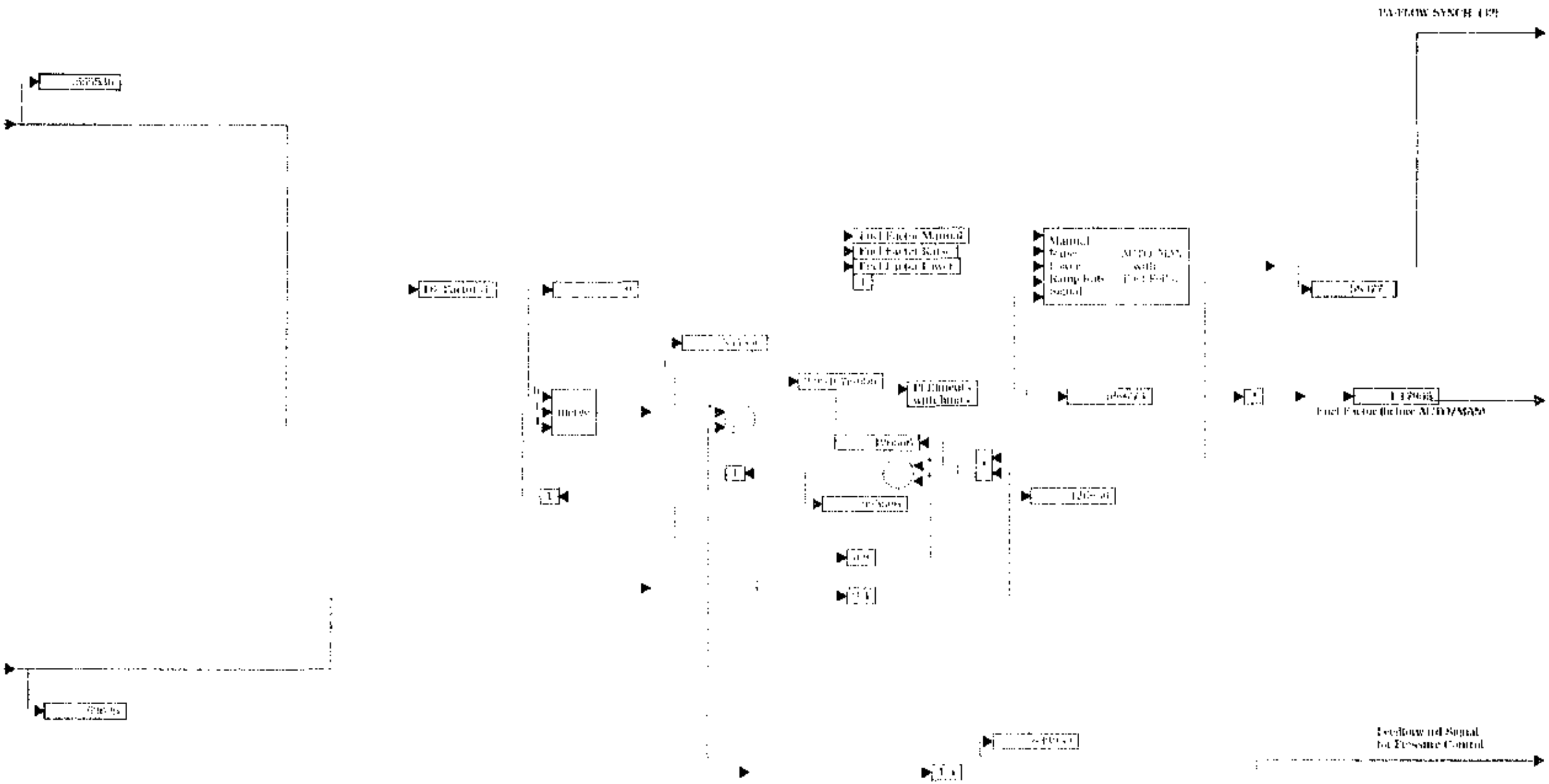
A5-2: UNIT CO-ORDINATOR



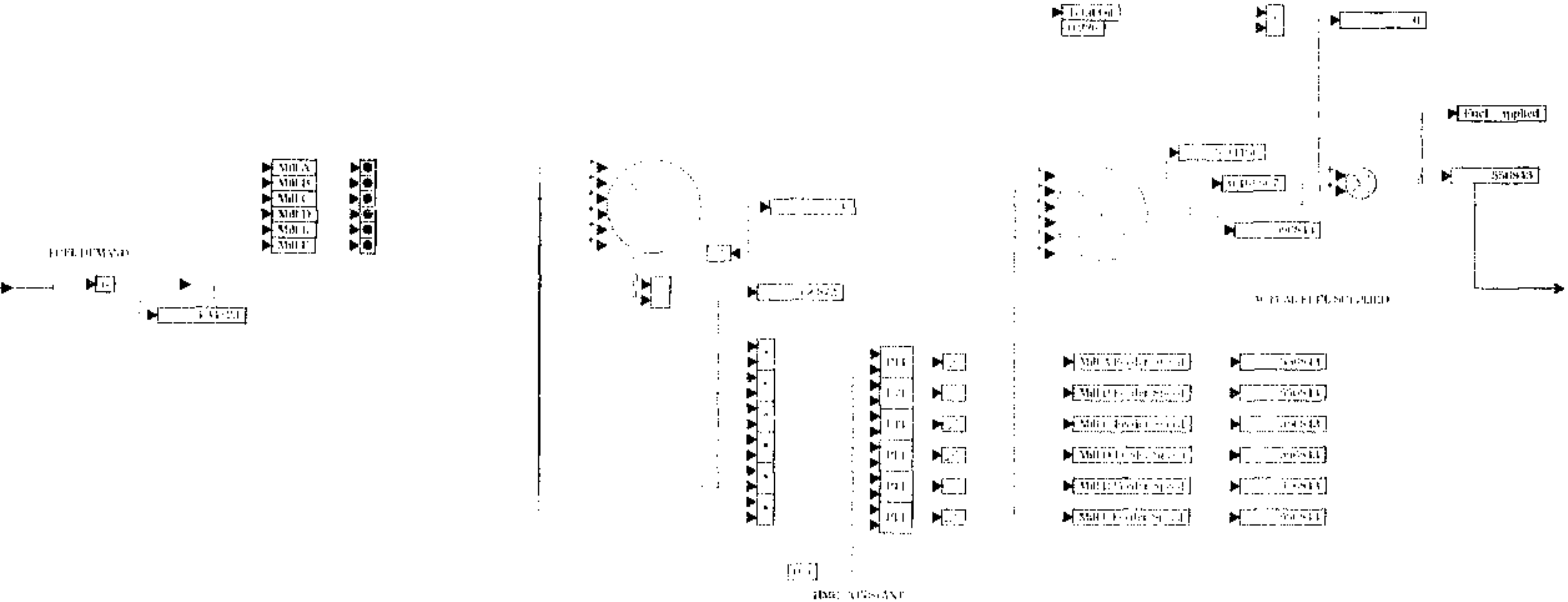
A5-3: MAIN PID PRESSURE CONTROLLER



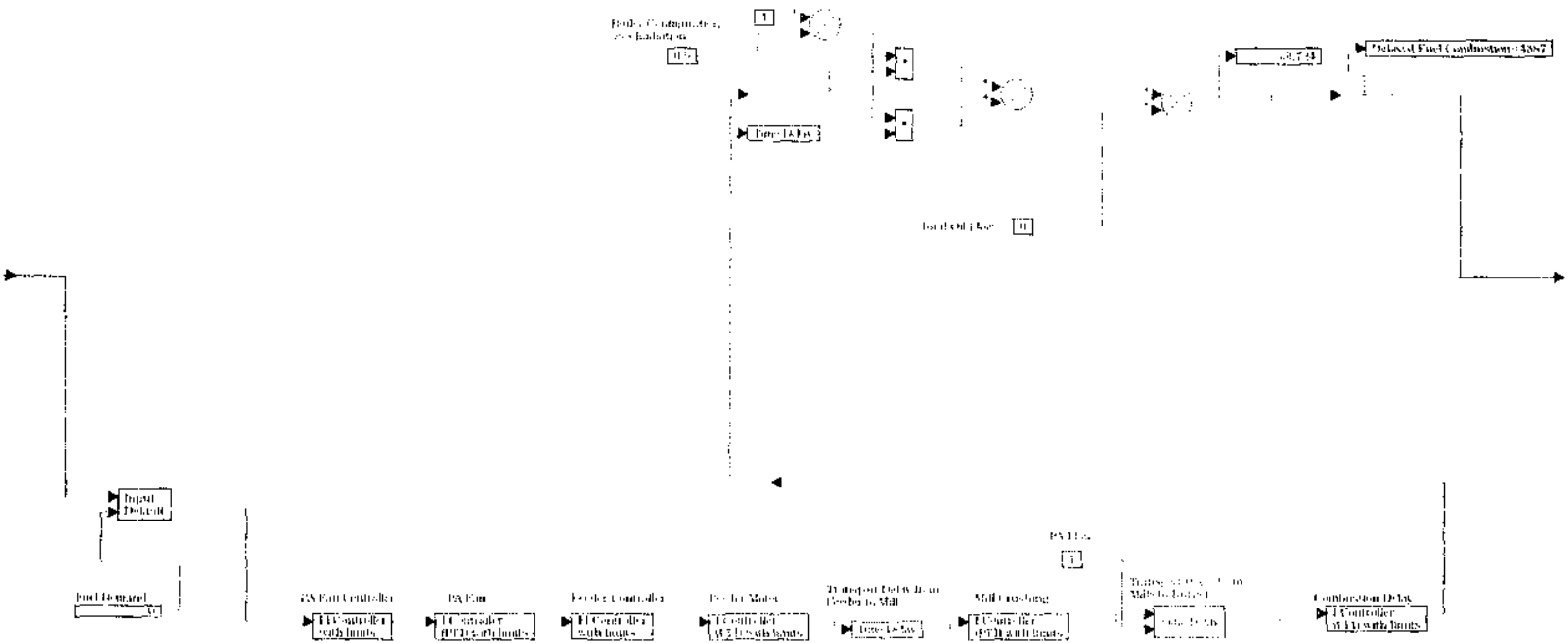
A5-4: FUEL FACTOR CONTROLLER



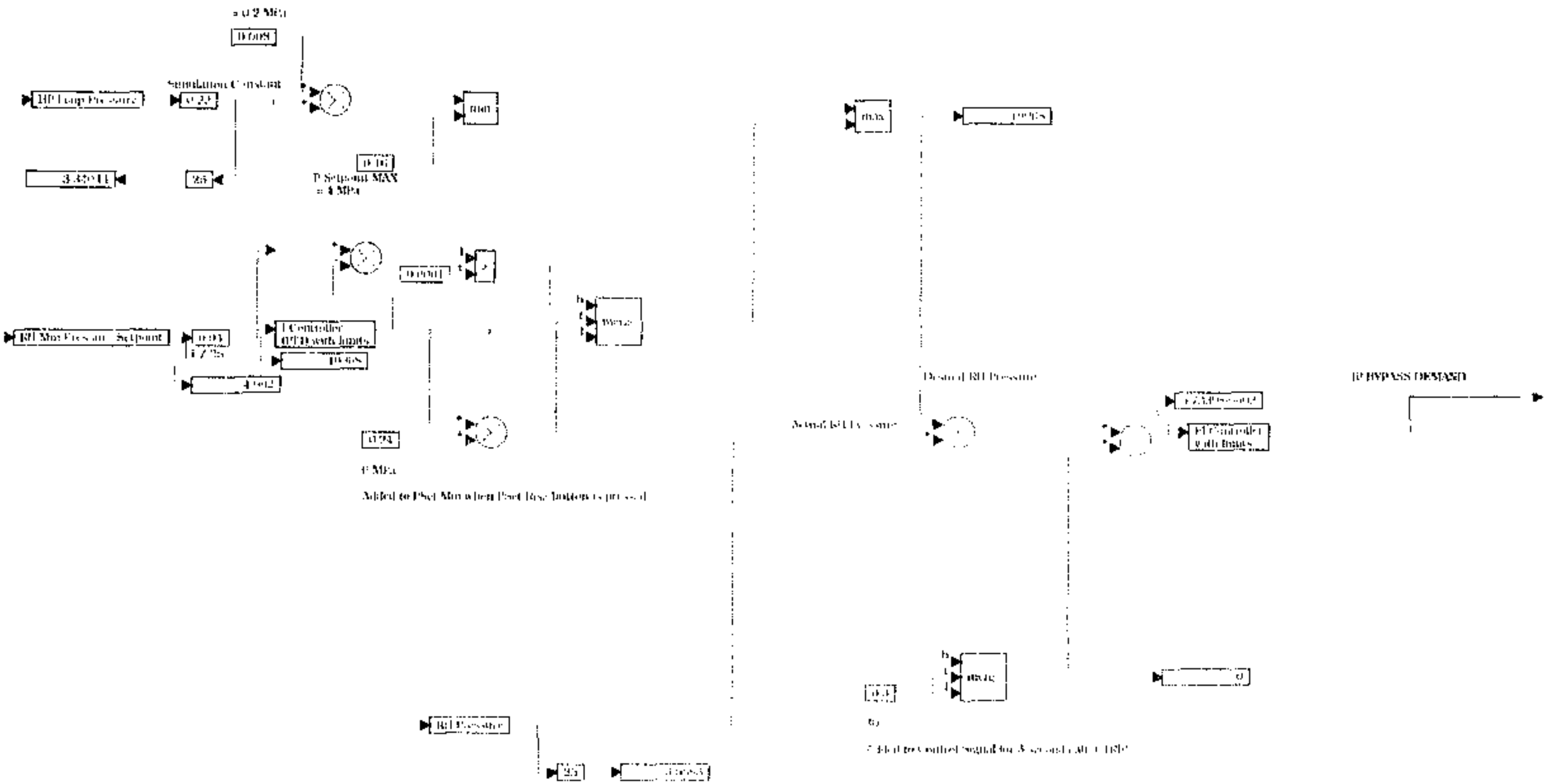
A6: FUEL DEMAND

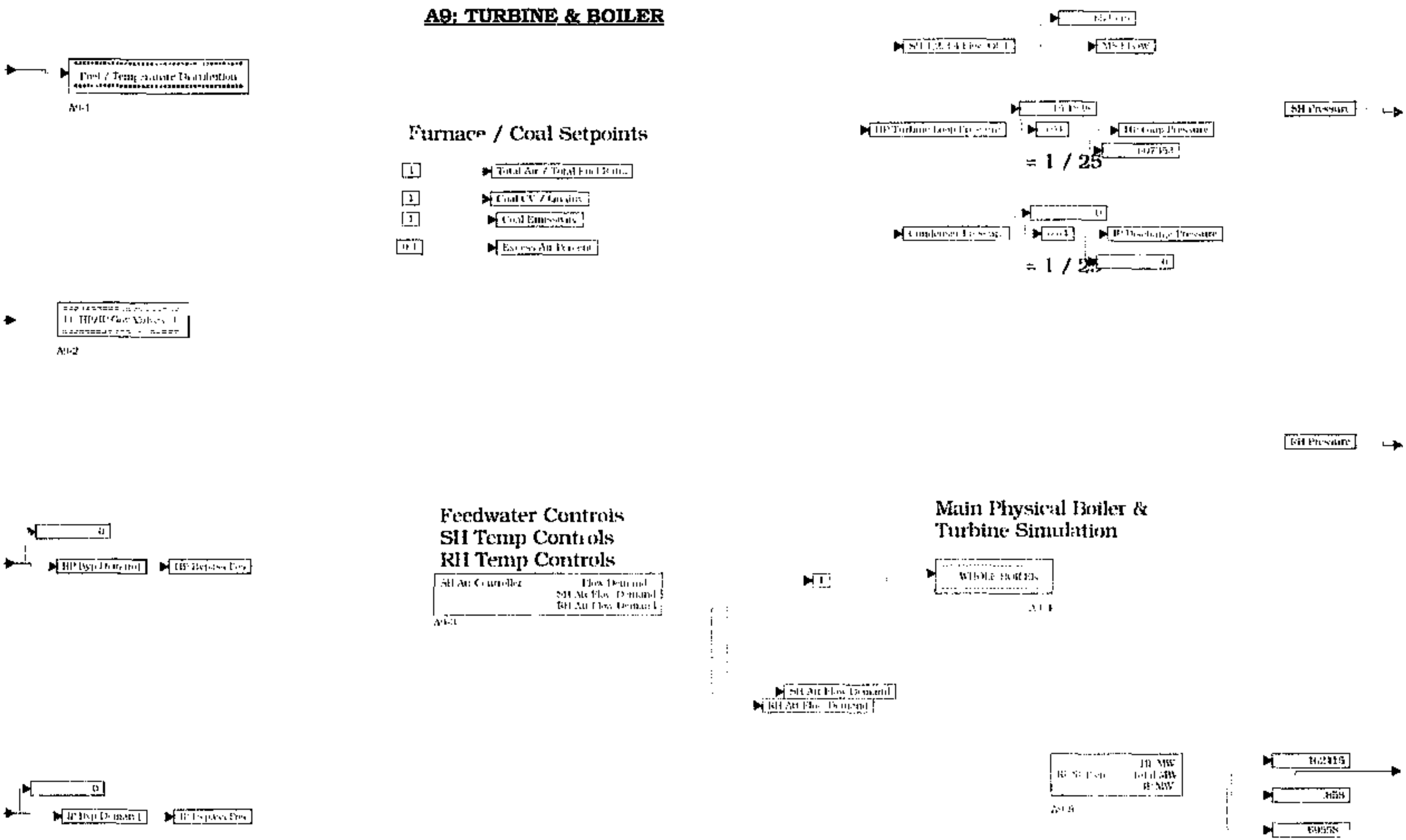


A7: FUEL DELAYS

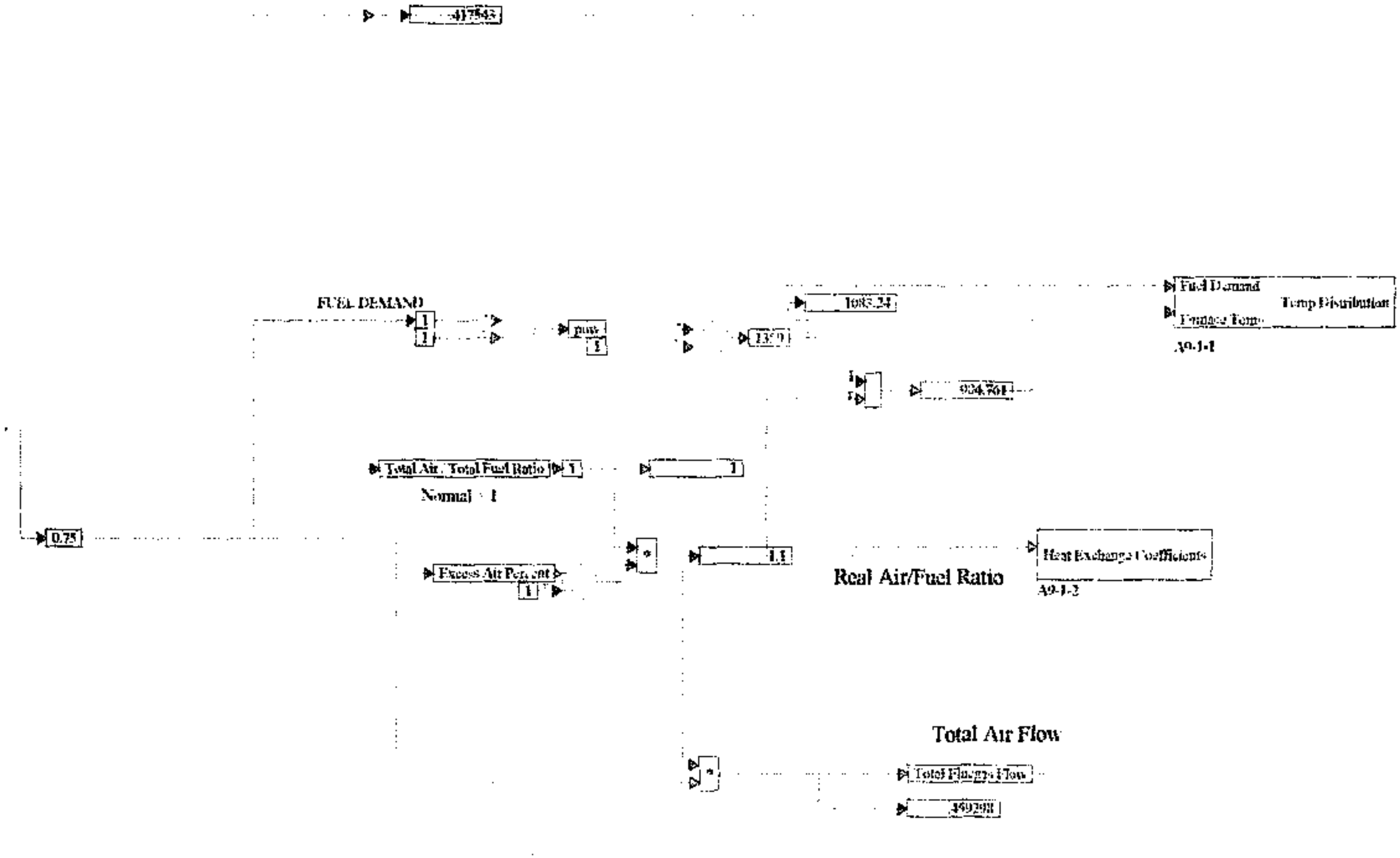


A8; LP BYPASS CONTROLLER

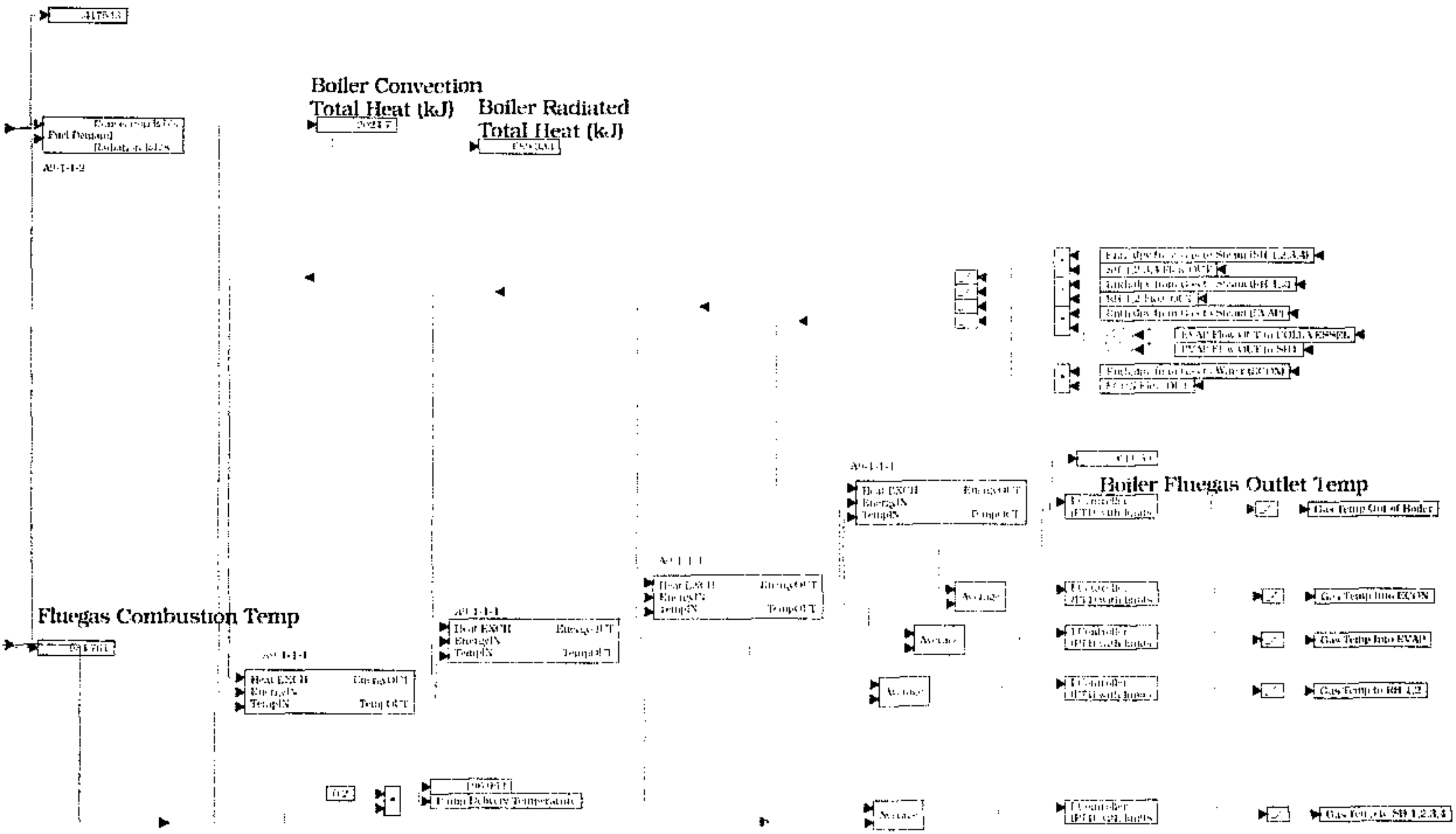



A9: TURBINE & BOILER
Fuel / Temperature Distribution
A9-1
Furnace / Coal Setpoints
Total Air / Total Fuel Ratio
Coal CV / Gravity
Coal Emissivity
Excess Air Percent
HP/IP Gov Valves
A9-2
SH Pressure
= 1 / 25
= 1 / 25
RH Pressure
HP Byp Demand
HP Bypass Pos
Feedwater Controls
SH Temp Controls
RH Temp Controls
SH Att Controller
Flow Demand
SH Att Flow Demand
RH Att Flow Demand
A9-3
Main Physical Boiler &
Turbine Simulation
WHOLE BOILER
SH Att Flow Demand
RH Att Flow Demand
HP MW
Total MW
IP MW
A9-5

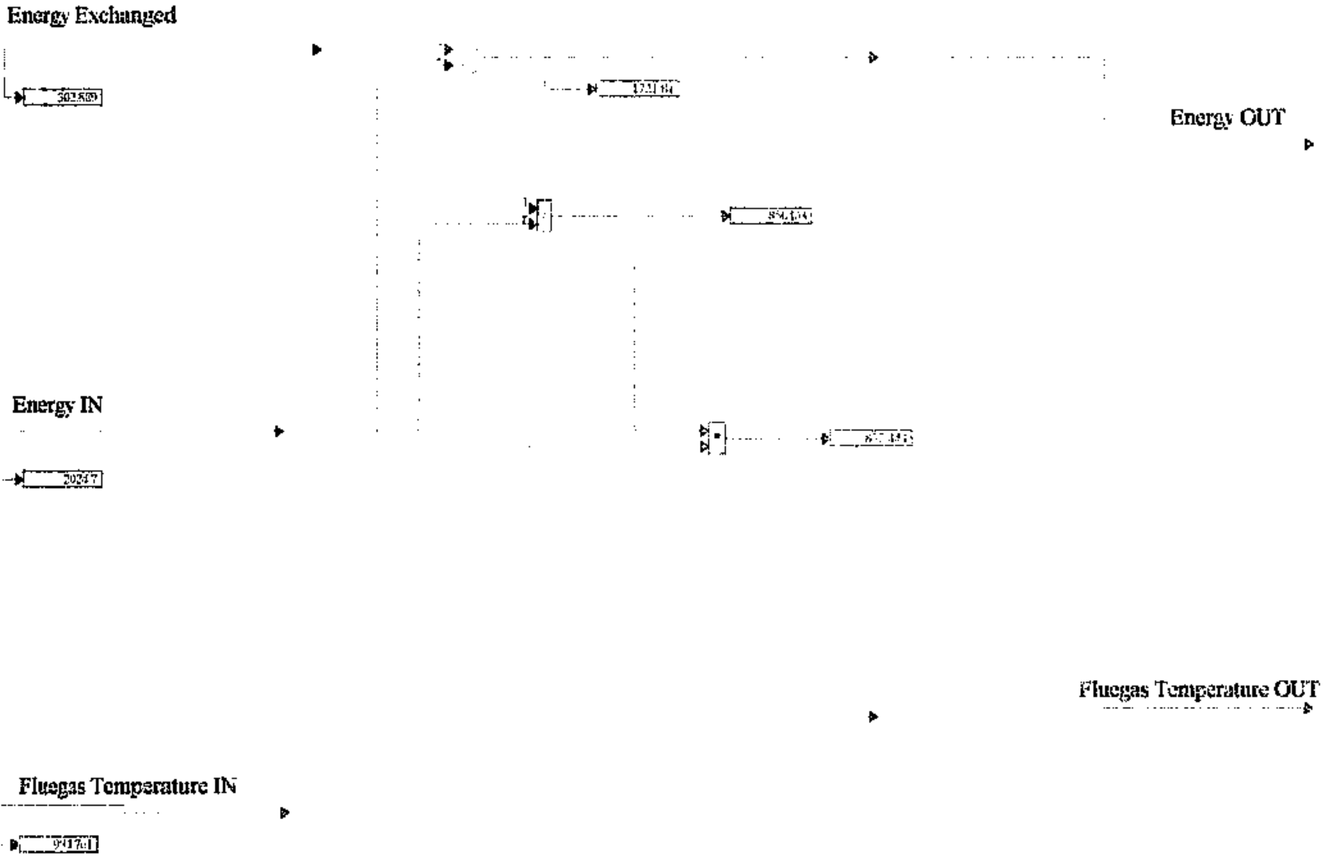
A9-1: FUEL & TEMPERATURE DISTRIBUTION



A9-1-1: TEMPERATURE DISTRIBUTION

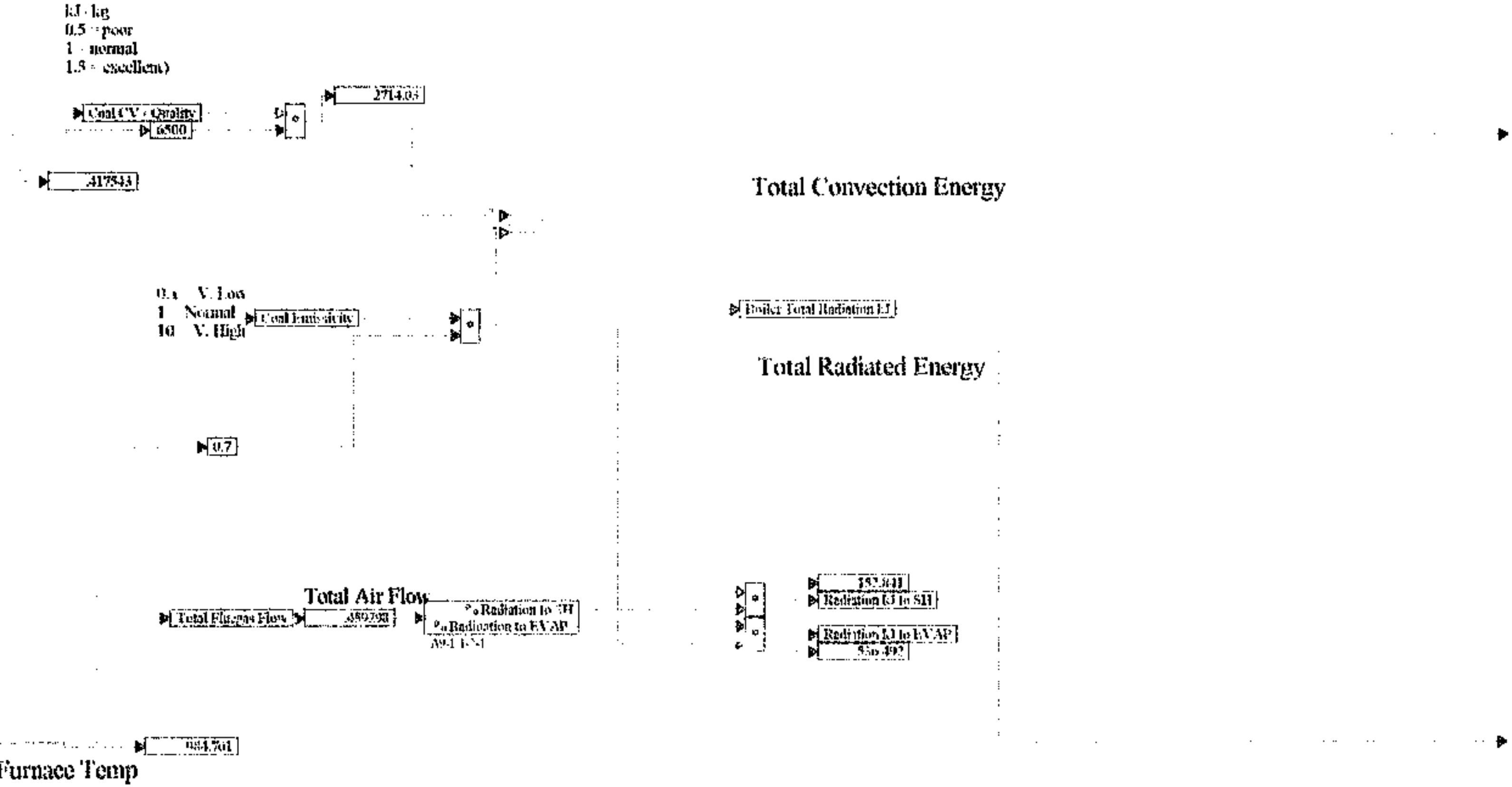


A9-1-1-1: HEAT EXCHANGER

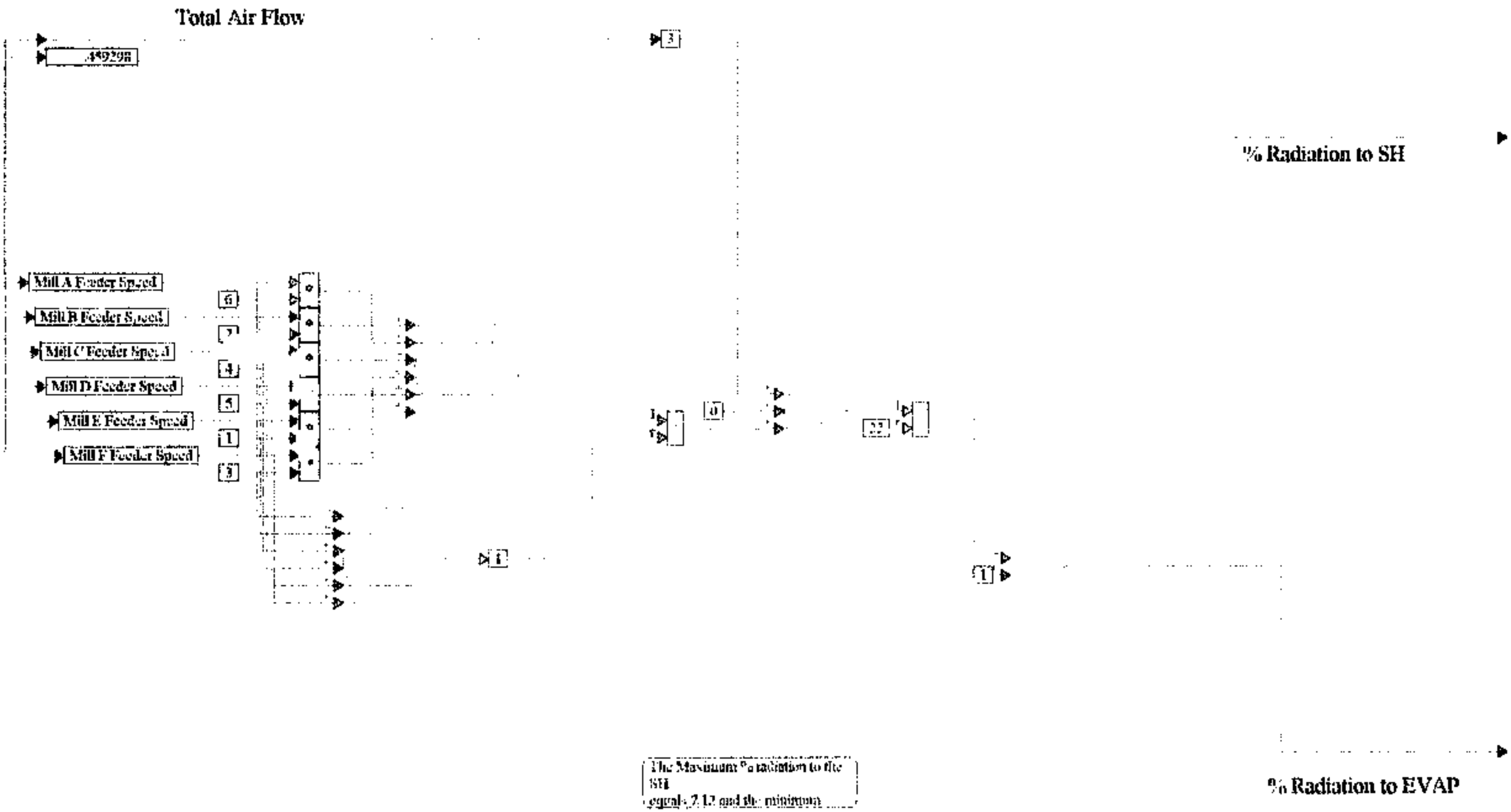


TempOUT = [(EnergyIN-EnergyEXCH) / EnergyIN] x TempIN

A9-1-1-2: CONVECTIVE / RADIATIVE ENERGY

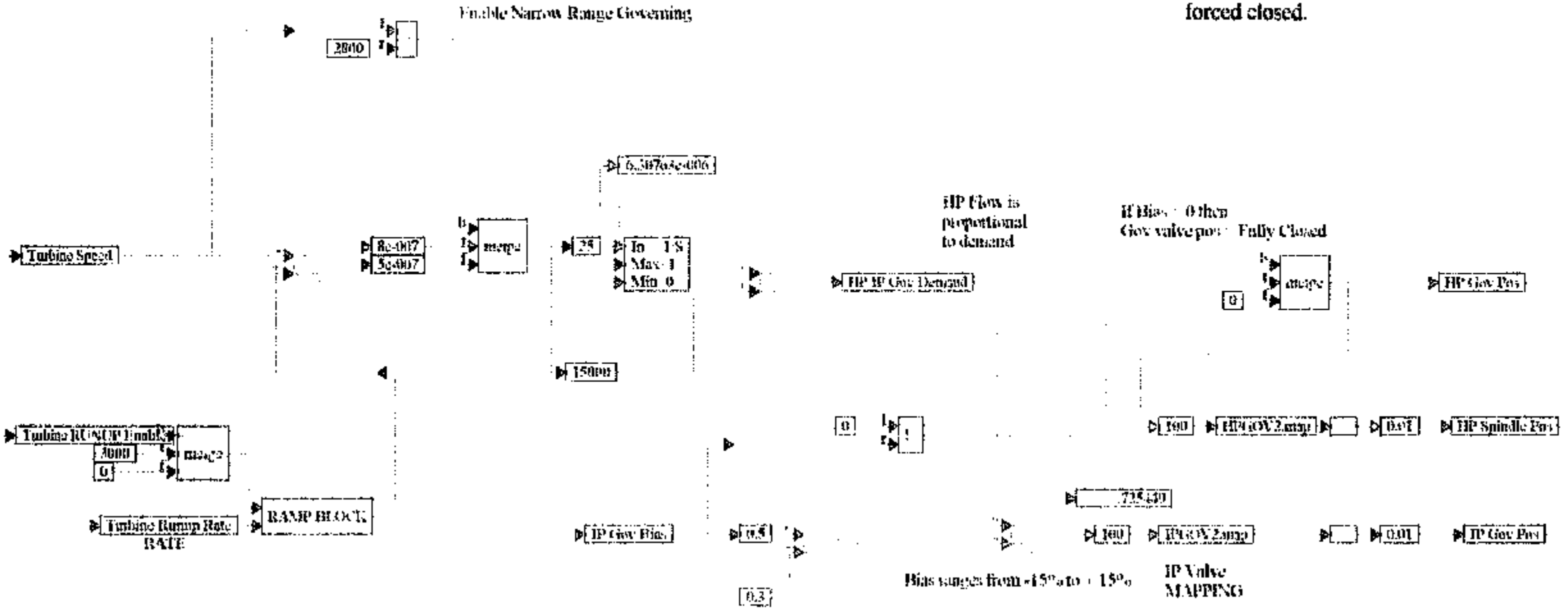


A9-1-1-2-1: ENERGY DISTRIBUTION IN BOILER

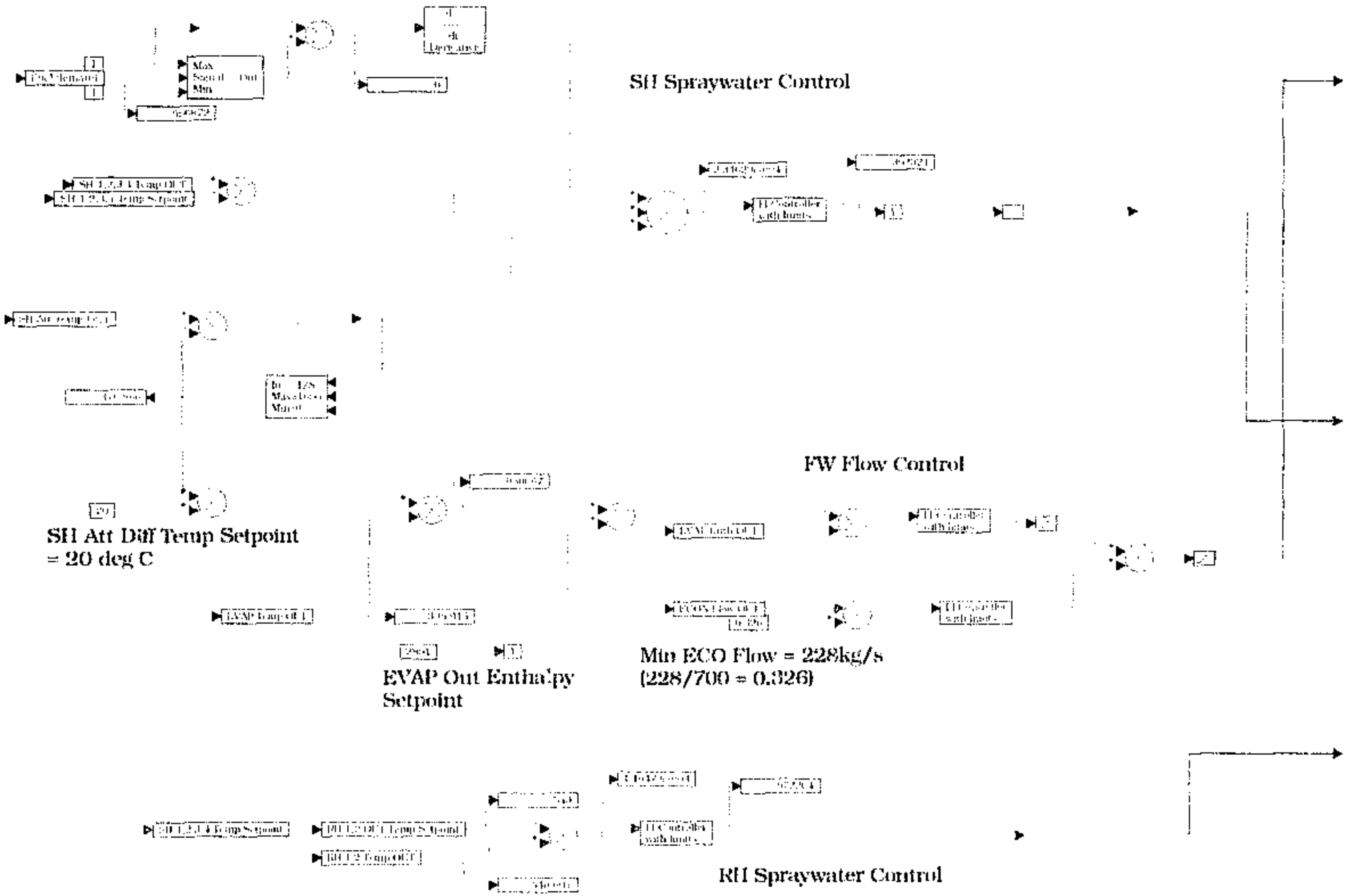


A9-2: HP & IP GOVERNOR NON LINEAR CIRCUIT AND BIAS CONTROLLER

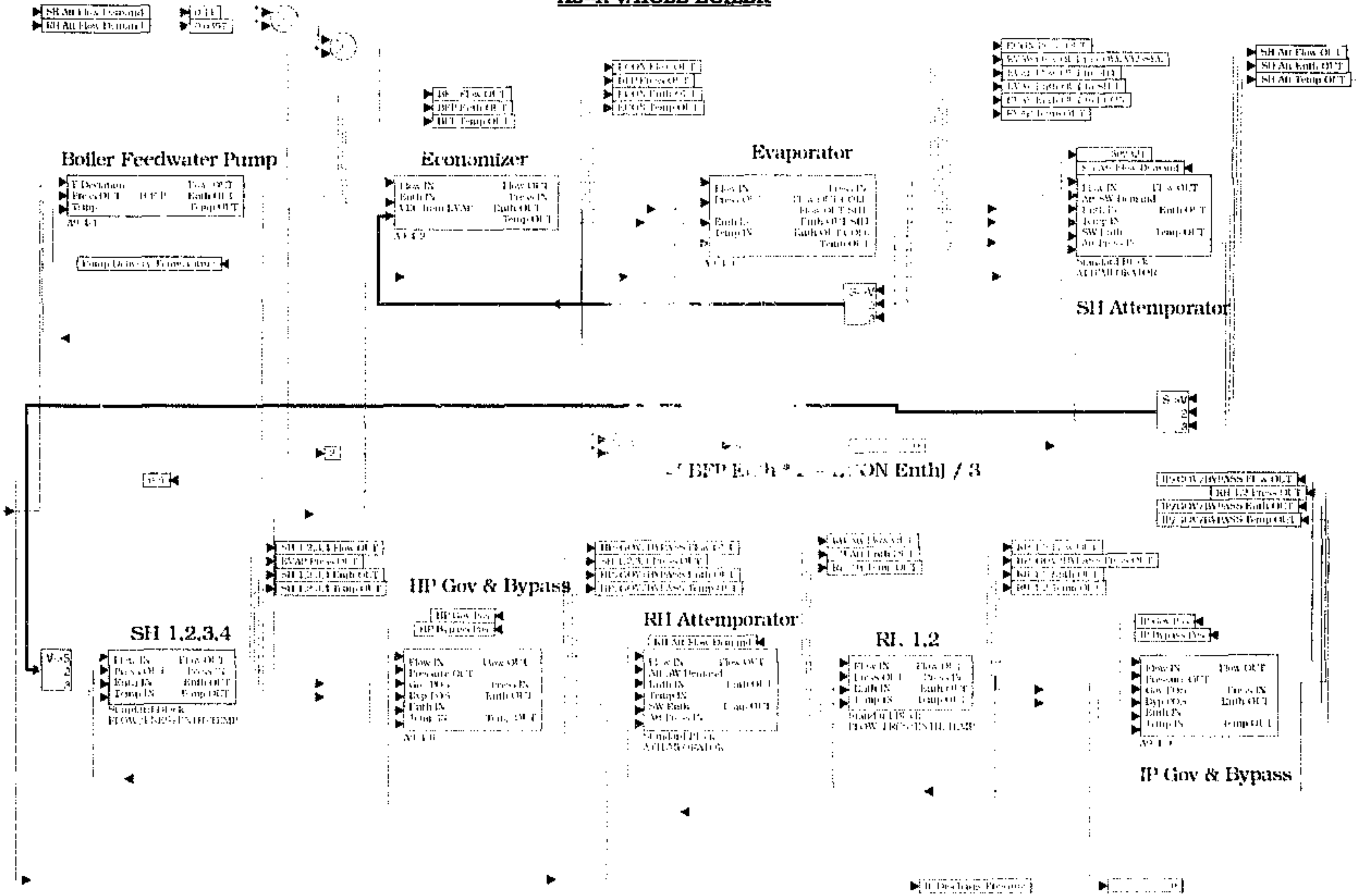
When the HP/IP Bias setting is at 0% on the control Desk (+15% actual) the HP Gov is forced closed.



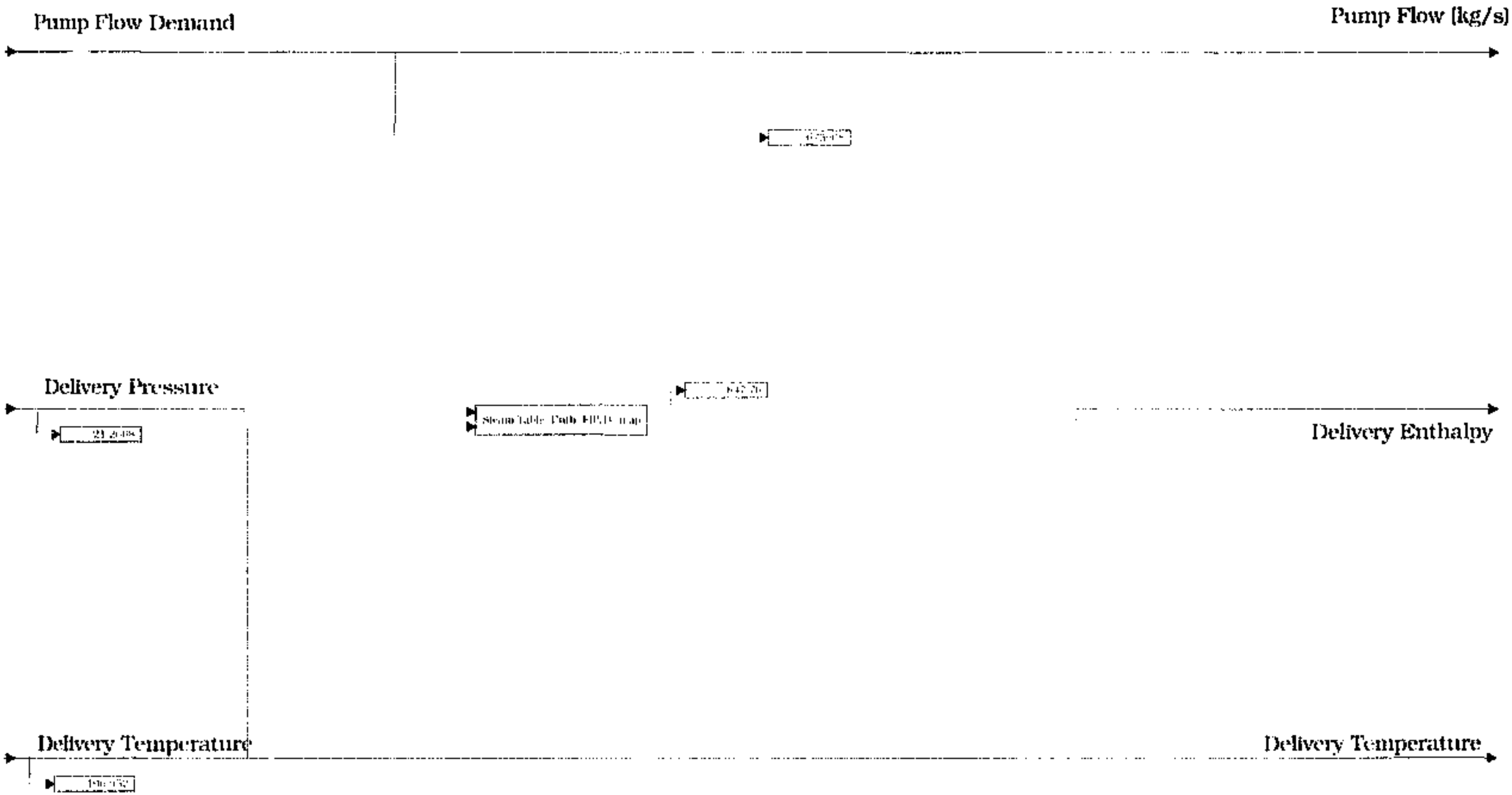
A9-3: FEEDWATER & SPRAYWATER DEMAND CONTROLLER



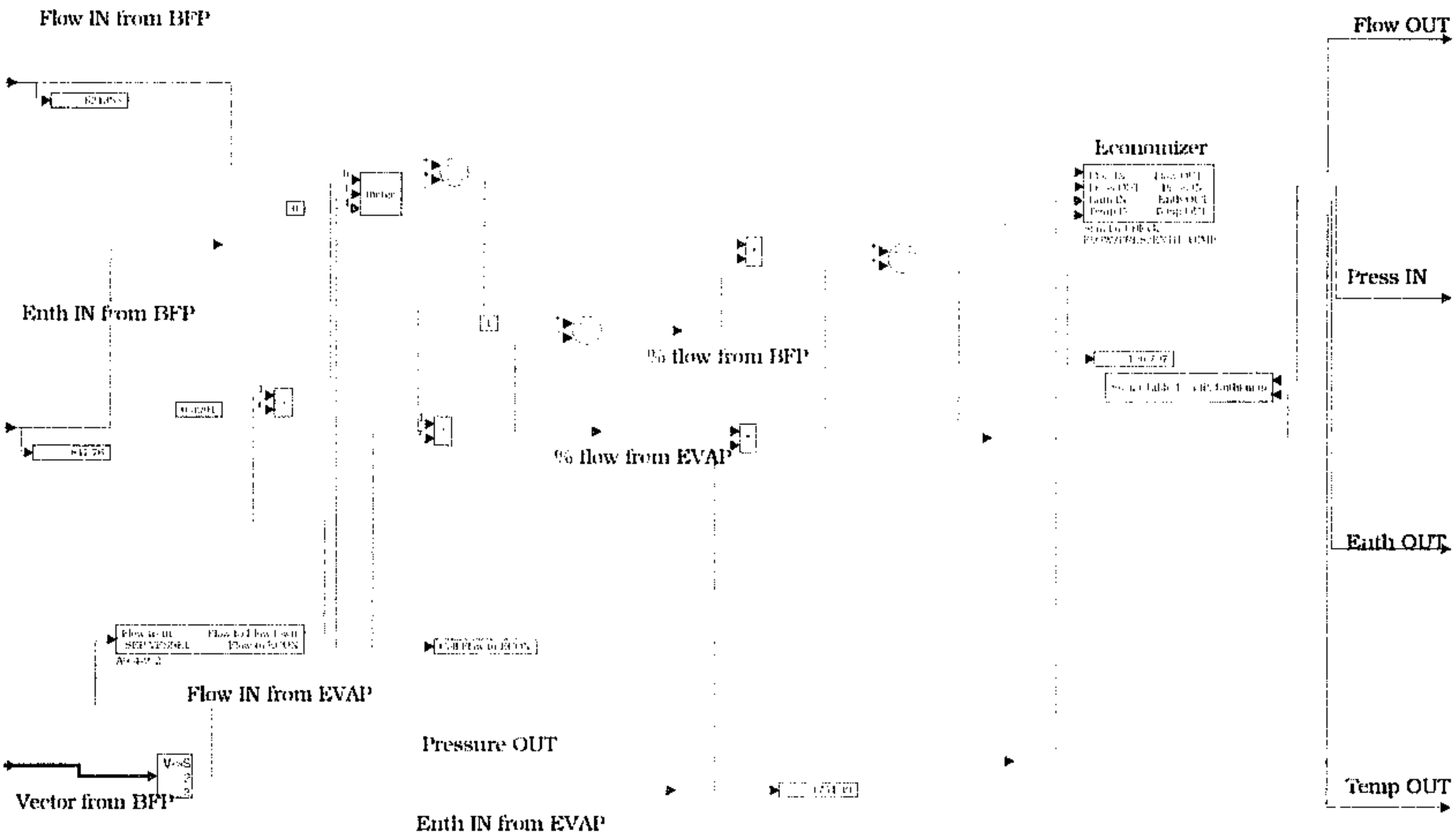
A9-4: WHOLE BOILER



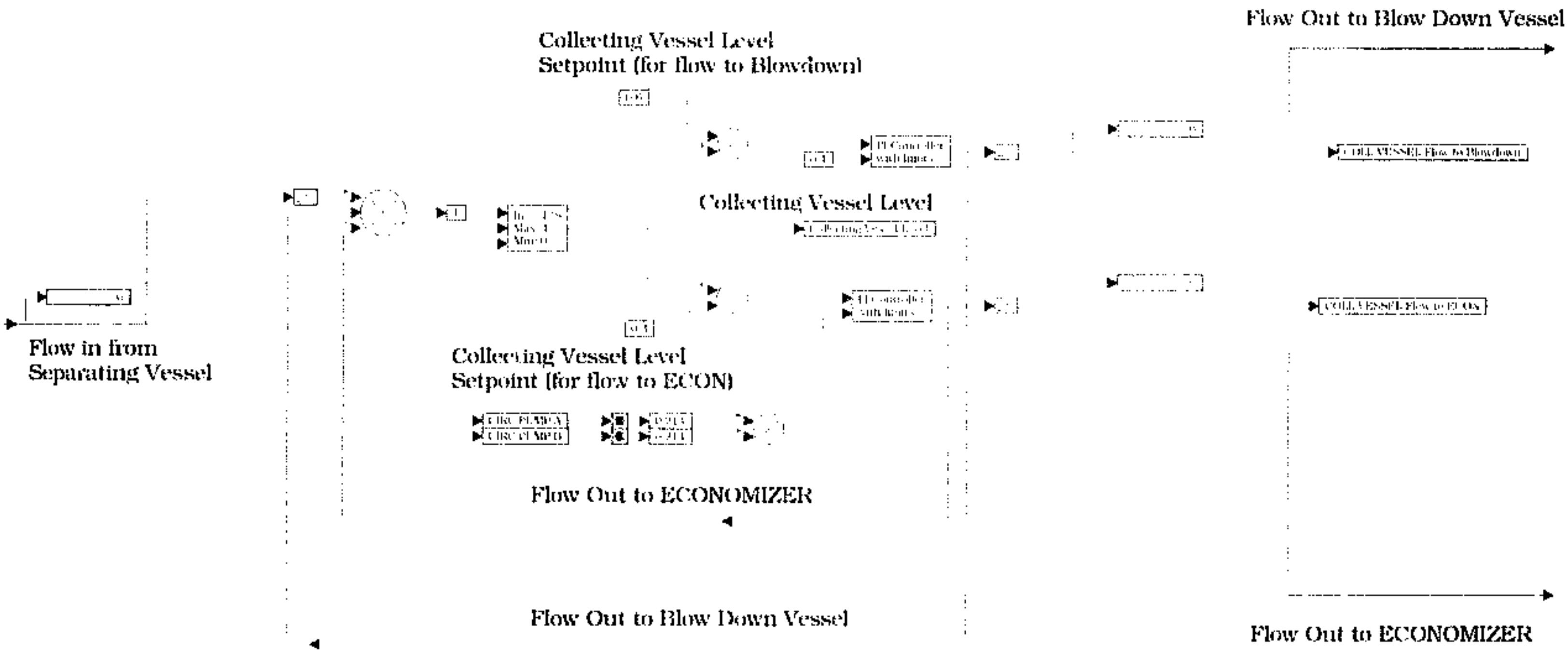
A9-4-1:BOILER FEED PUMP



A9-4-2: ECONOMIZER

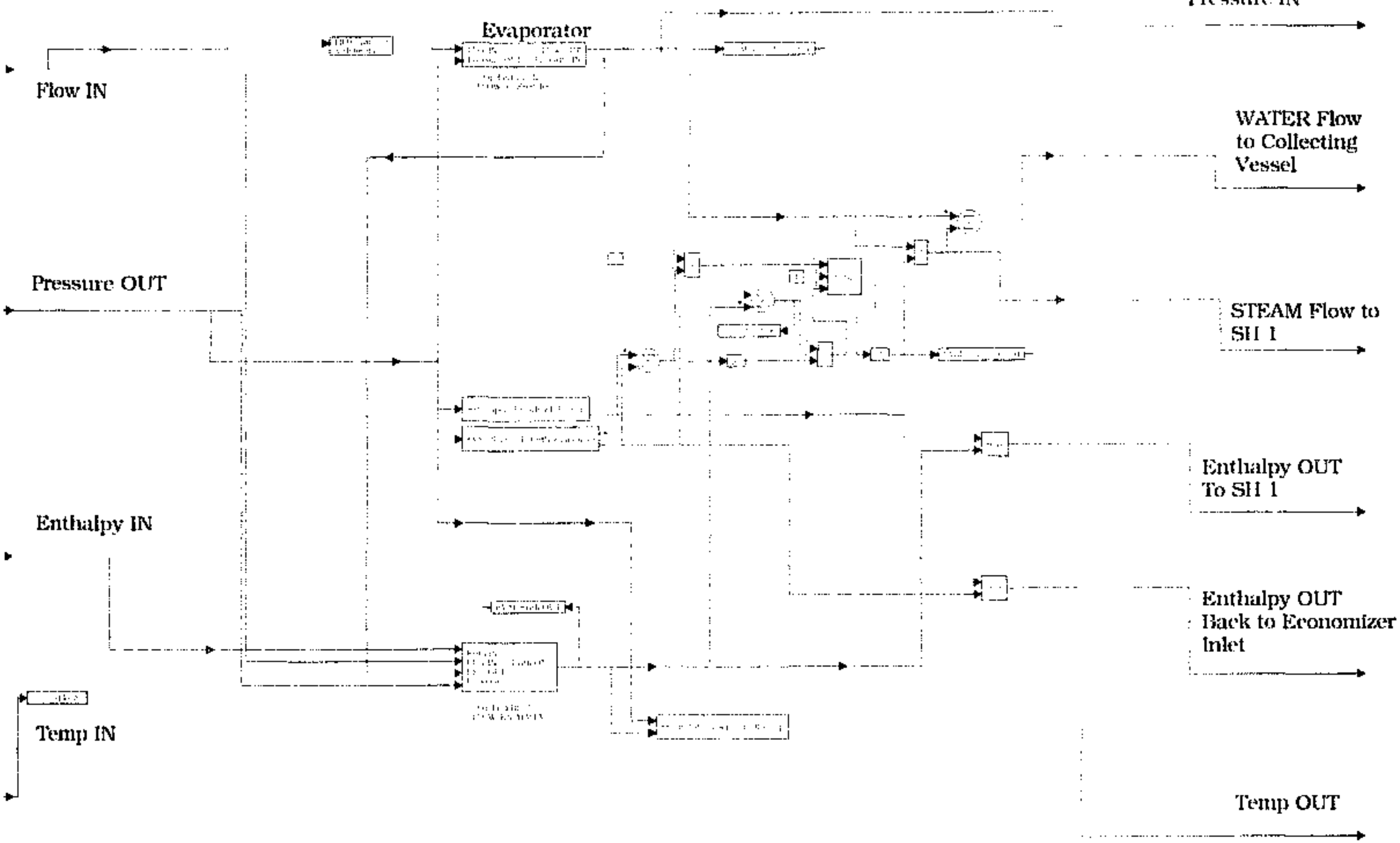


A9-4-2-2: COLLECTING VESSEL

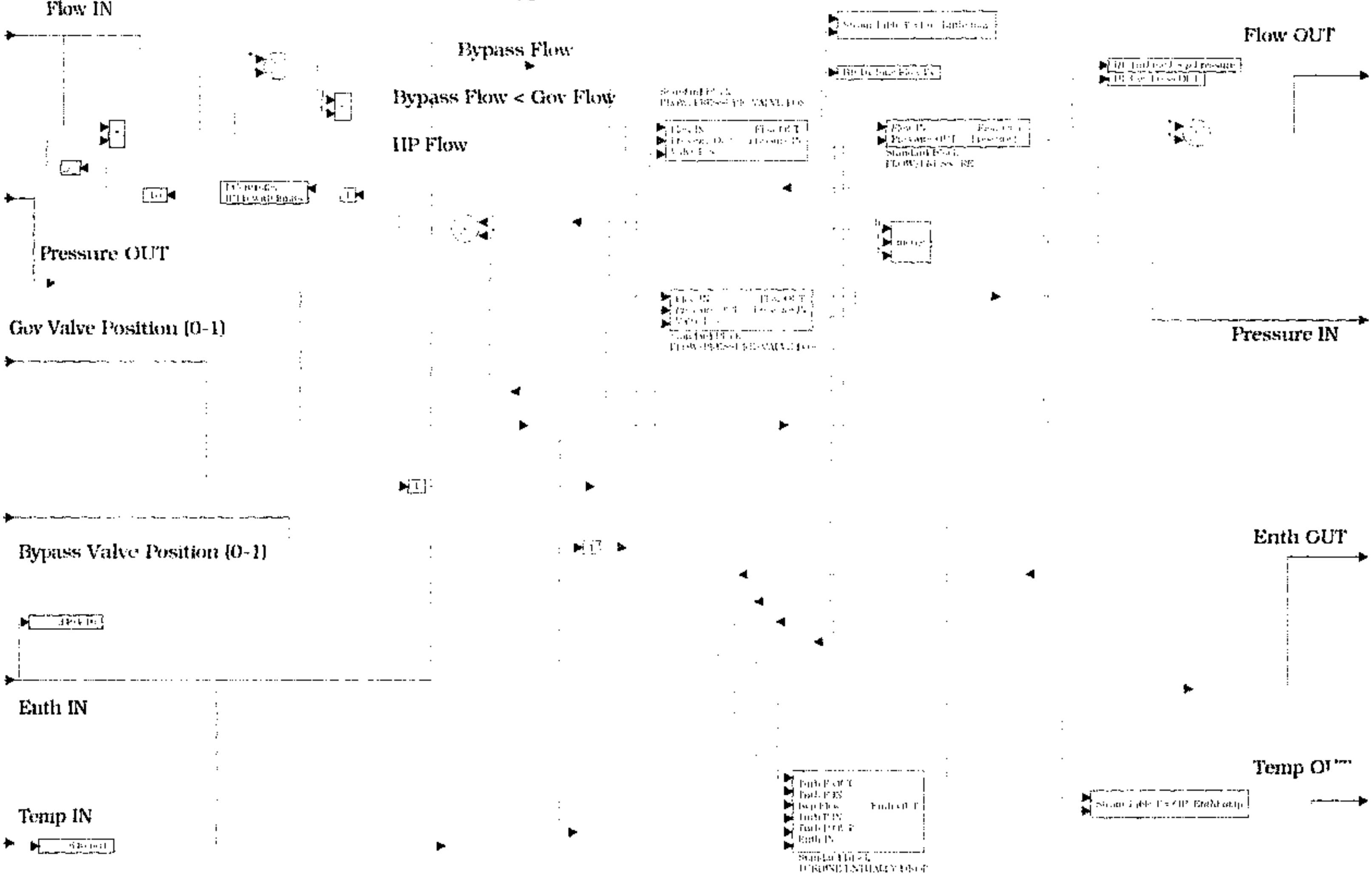


Max flow per circulation pump
= 150kg/s (=150/700=0.214)

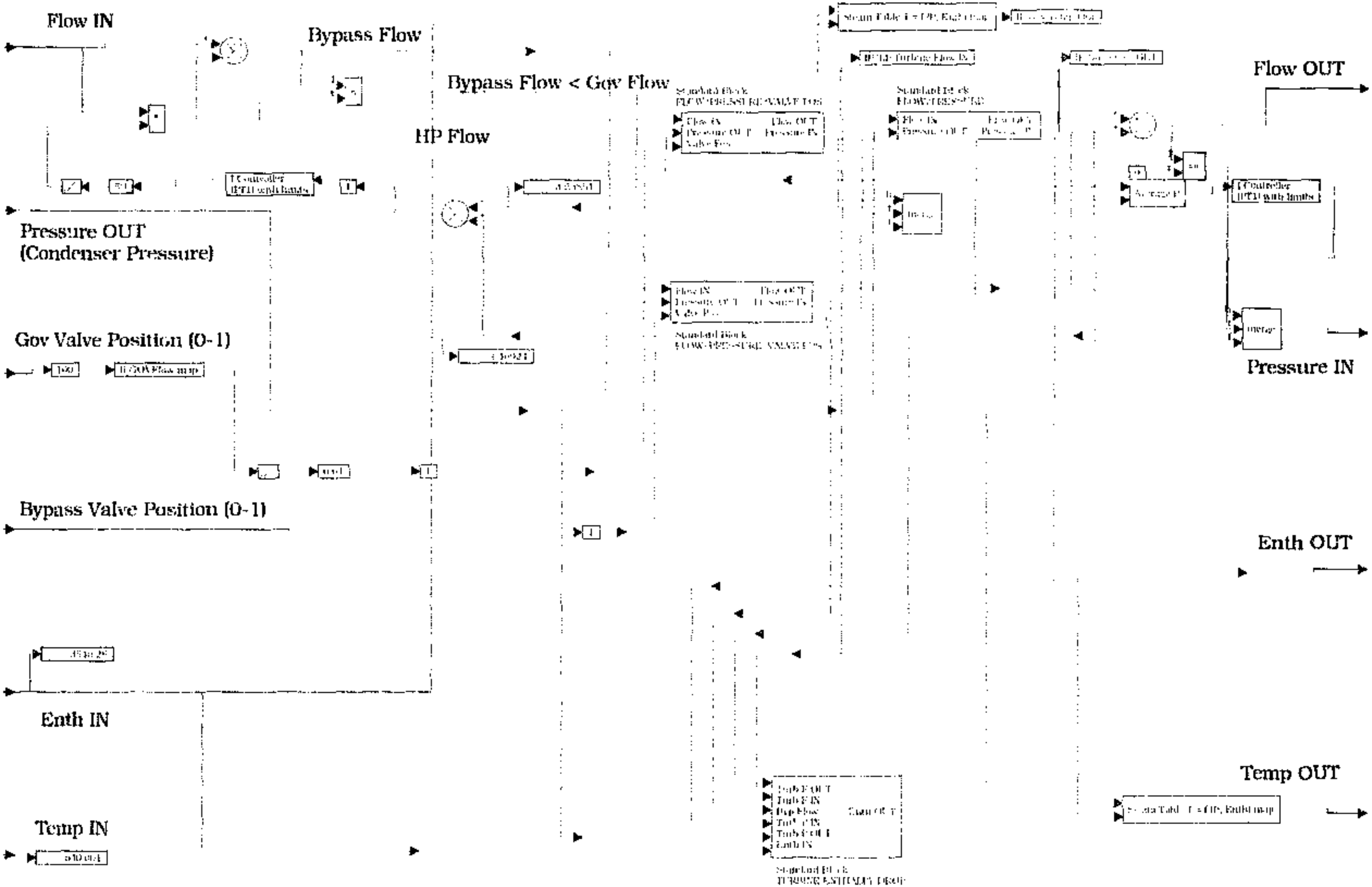
A9-4-3: EVAPORATOR



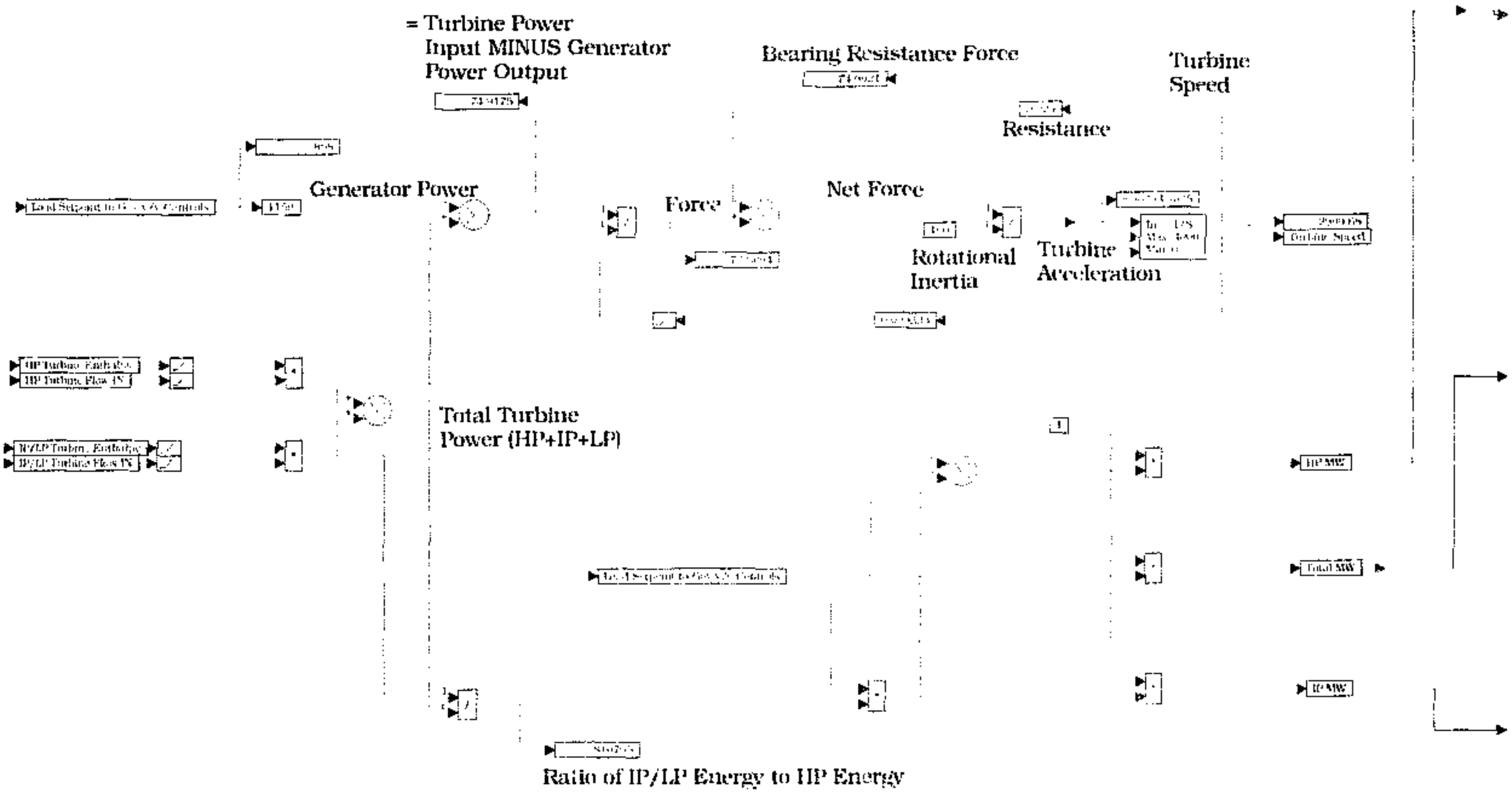
A9-4-6: HP Governor, Bypass & Turbine



A9-4-9: IP/LP Governor, Bypass & Turbine



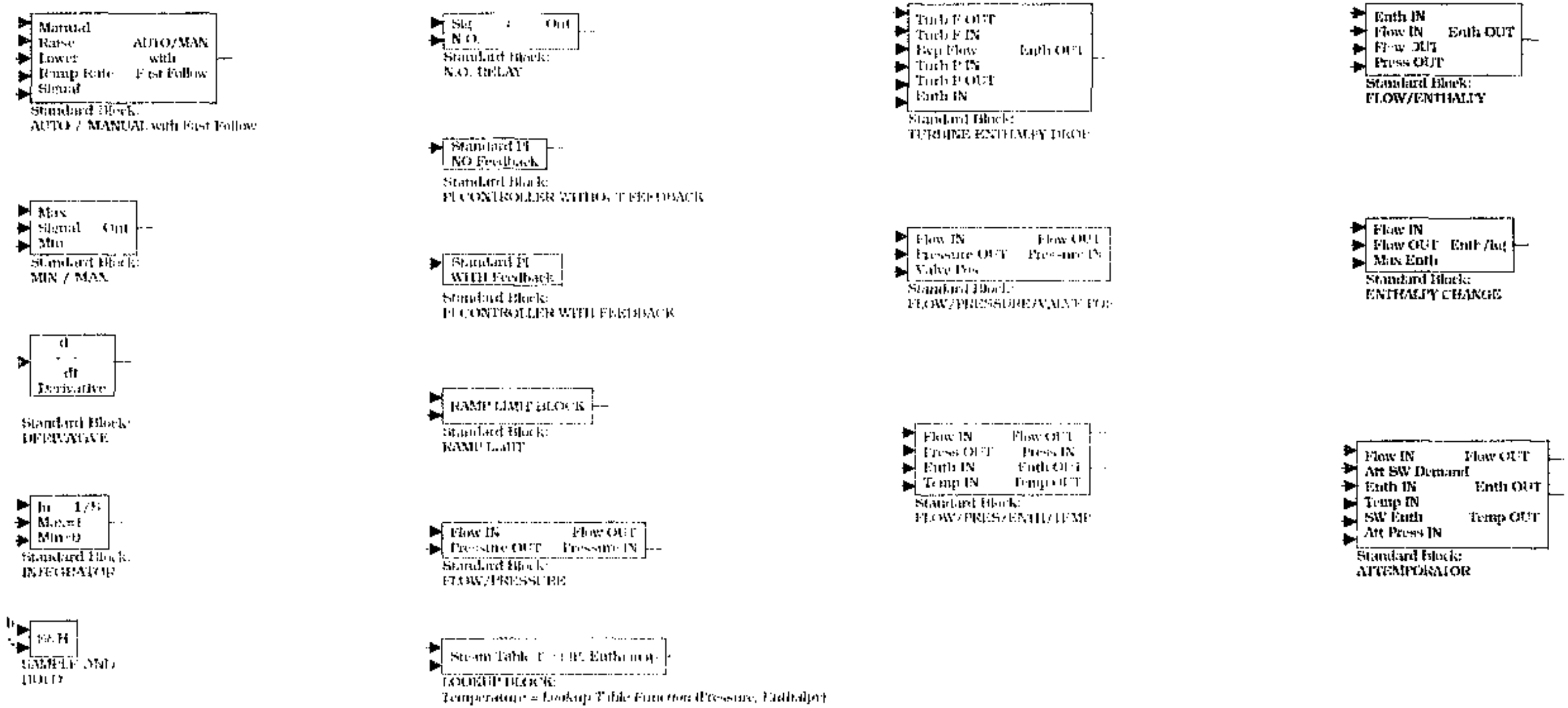
A9-5: TURBINE RUNUP CONTROLLER



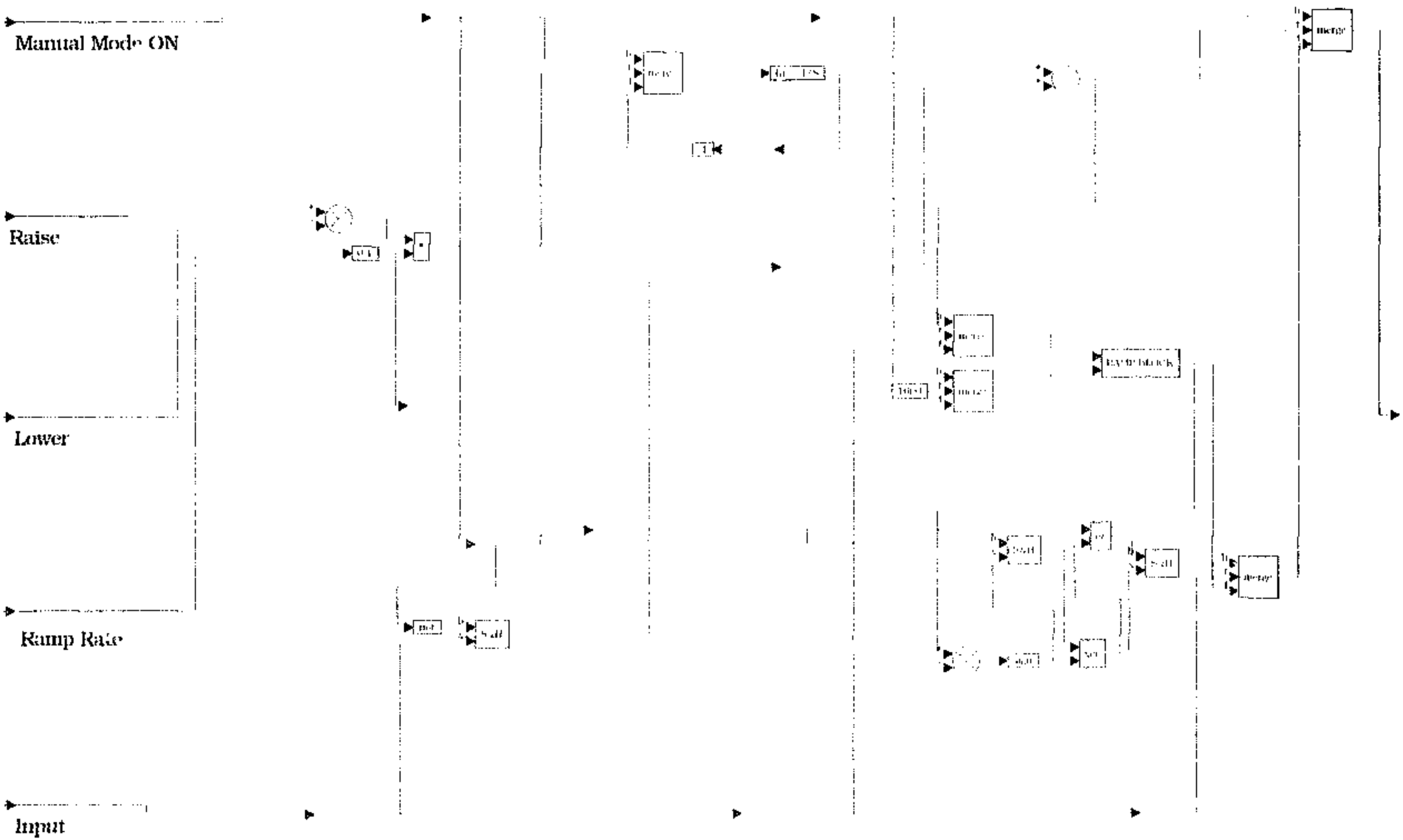
A10: LOAD DEMAND SETPOINT PI CONTROLLER

STANDARD BLOCKS

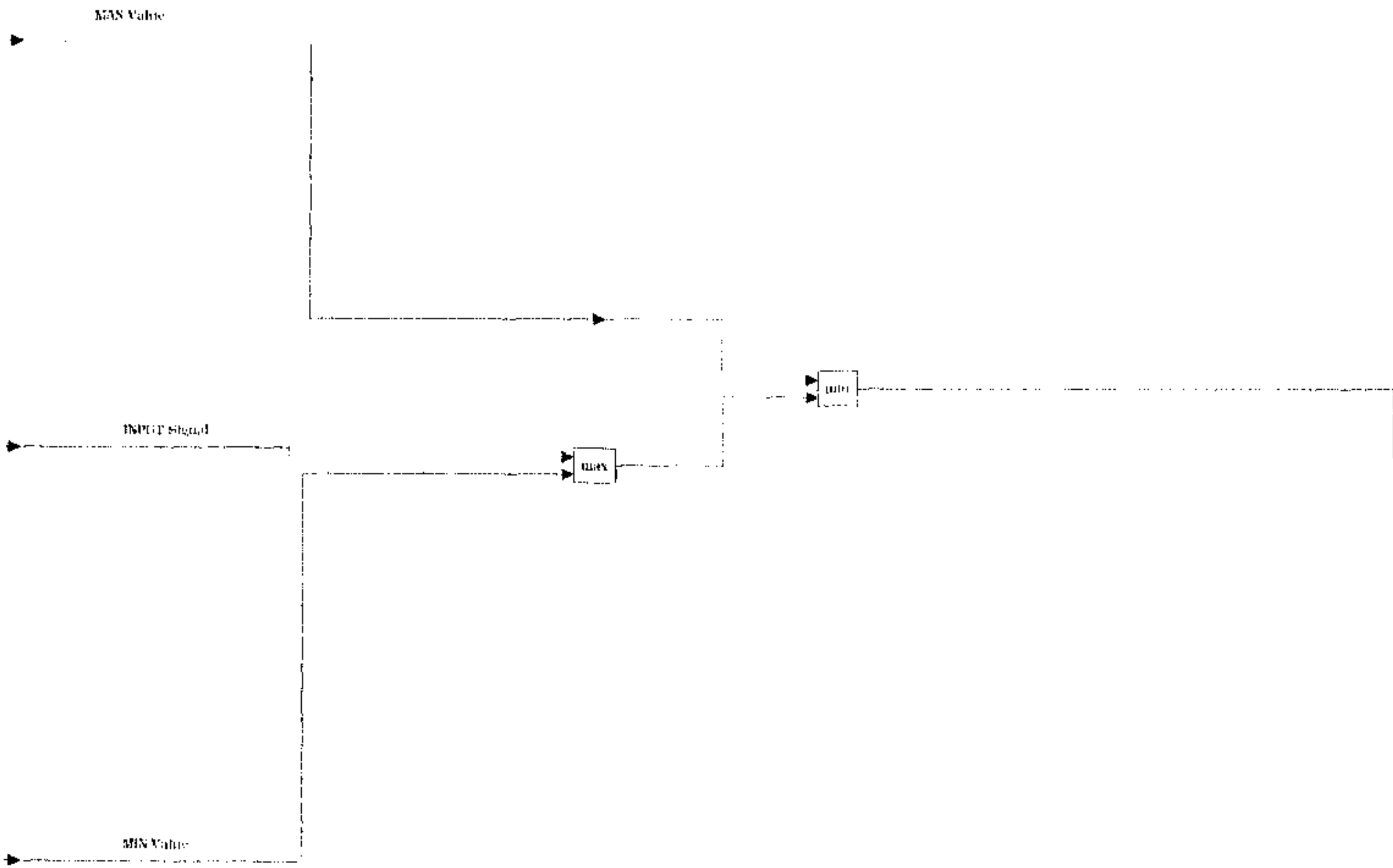
Used Throughout the Simulation



Standard Block:
AUTO/MANUAL OVERIDE

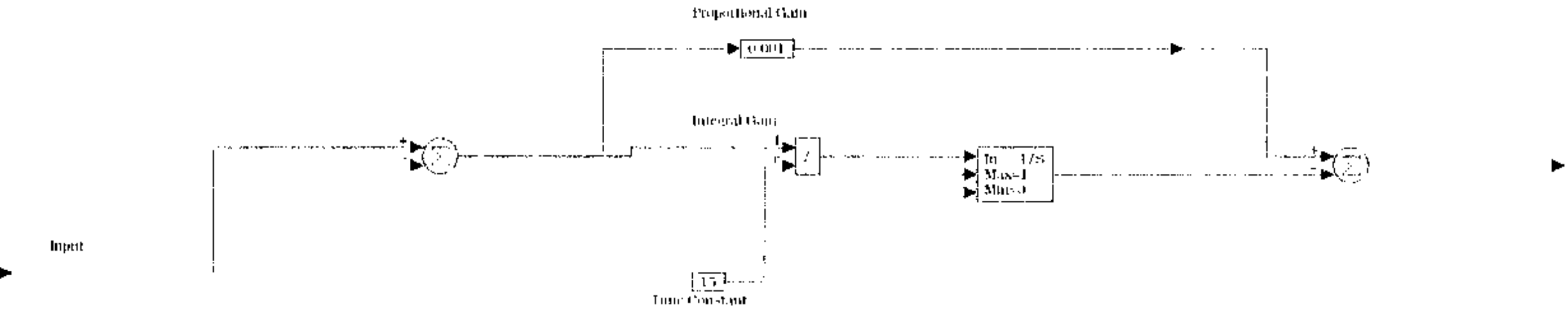


Min / Max



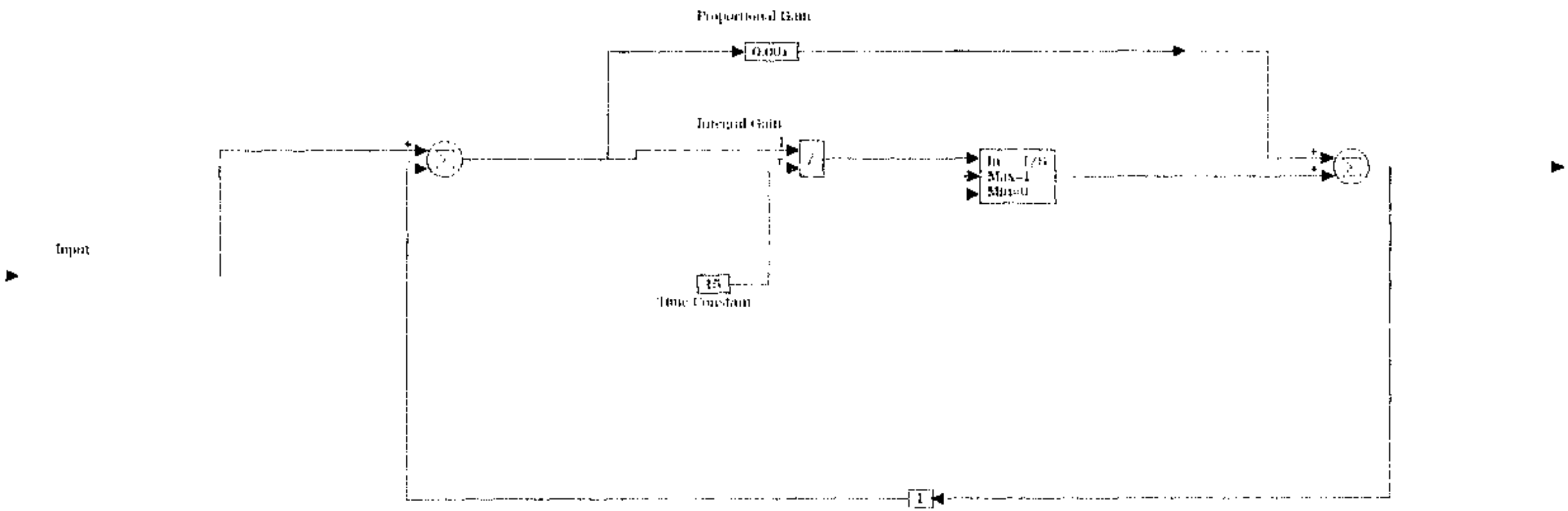
Standard Block: PI CONTROLLER

(Without Feedback)

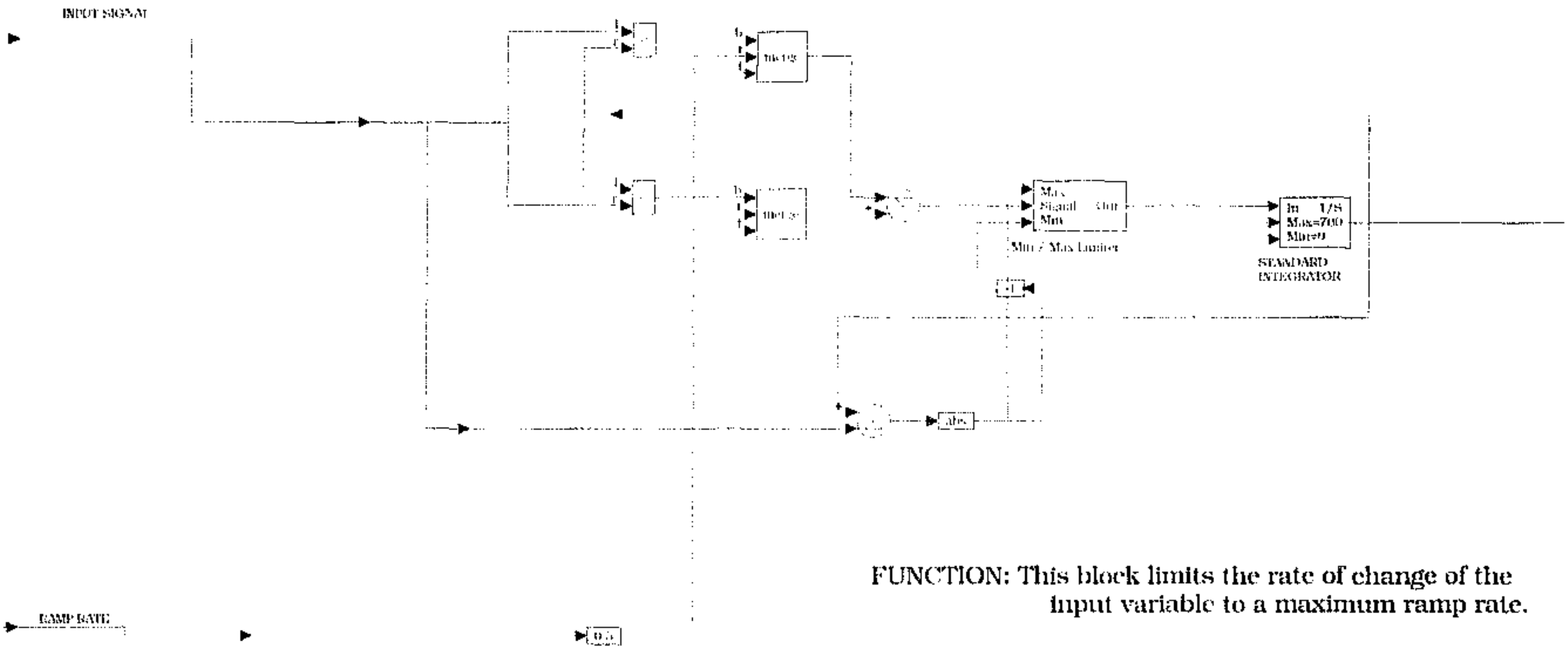


Standard Block: PI CONTROLLER

(With Feedback)

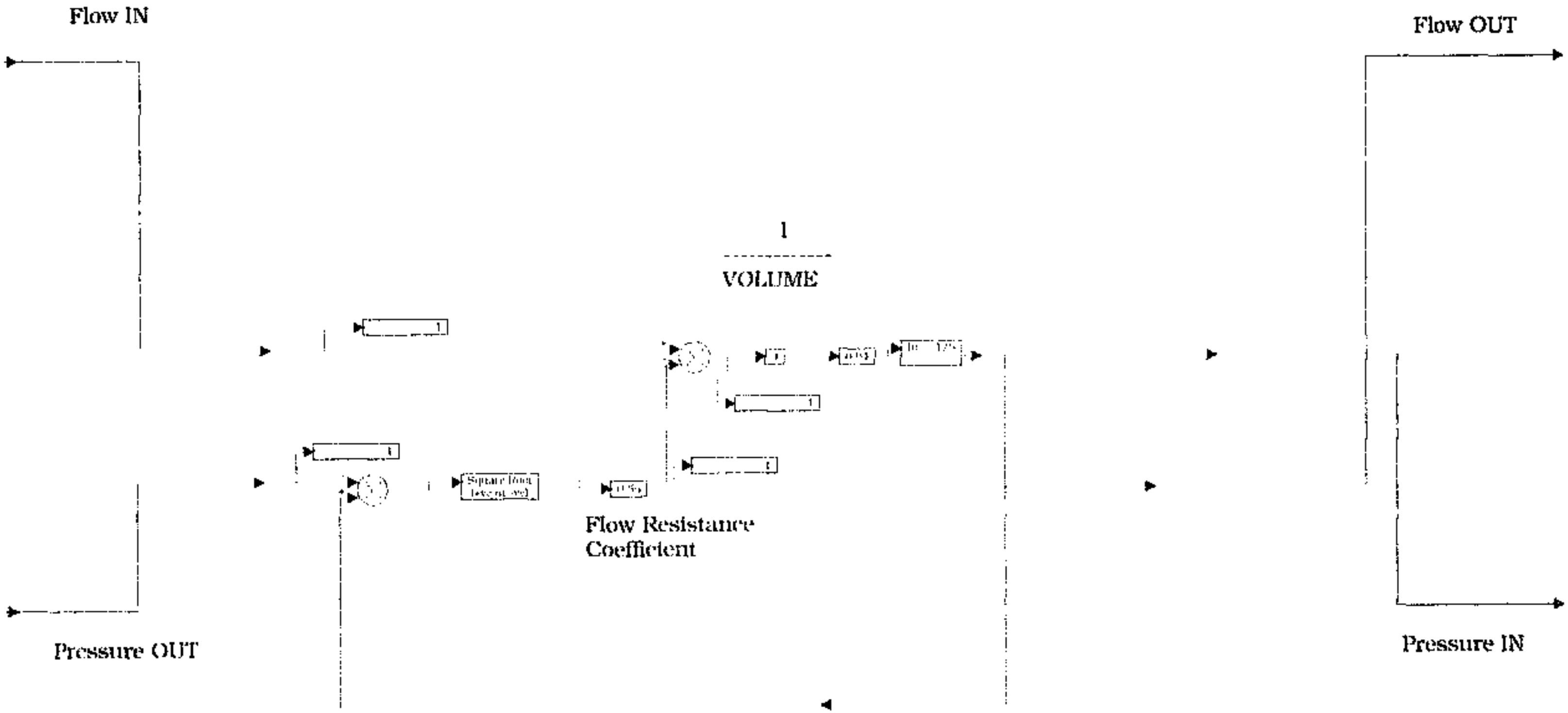


Standard Block:
RAMP LIMIT BLOCK



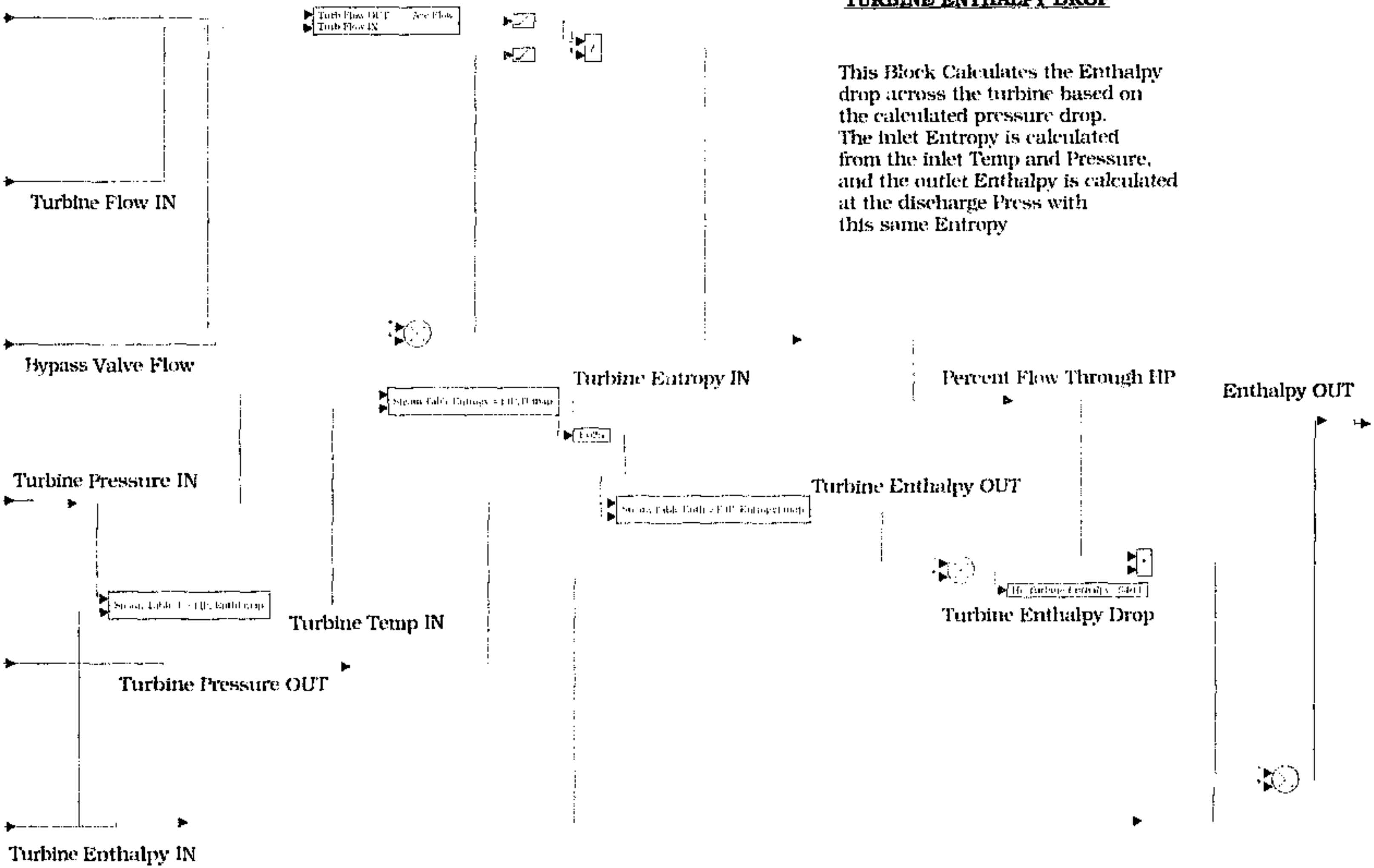
FUNCTION: This block limits the rate of change of the input variable to a maximum ramp rate.

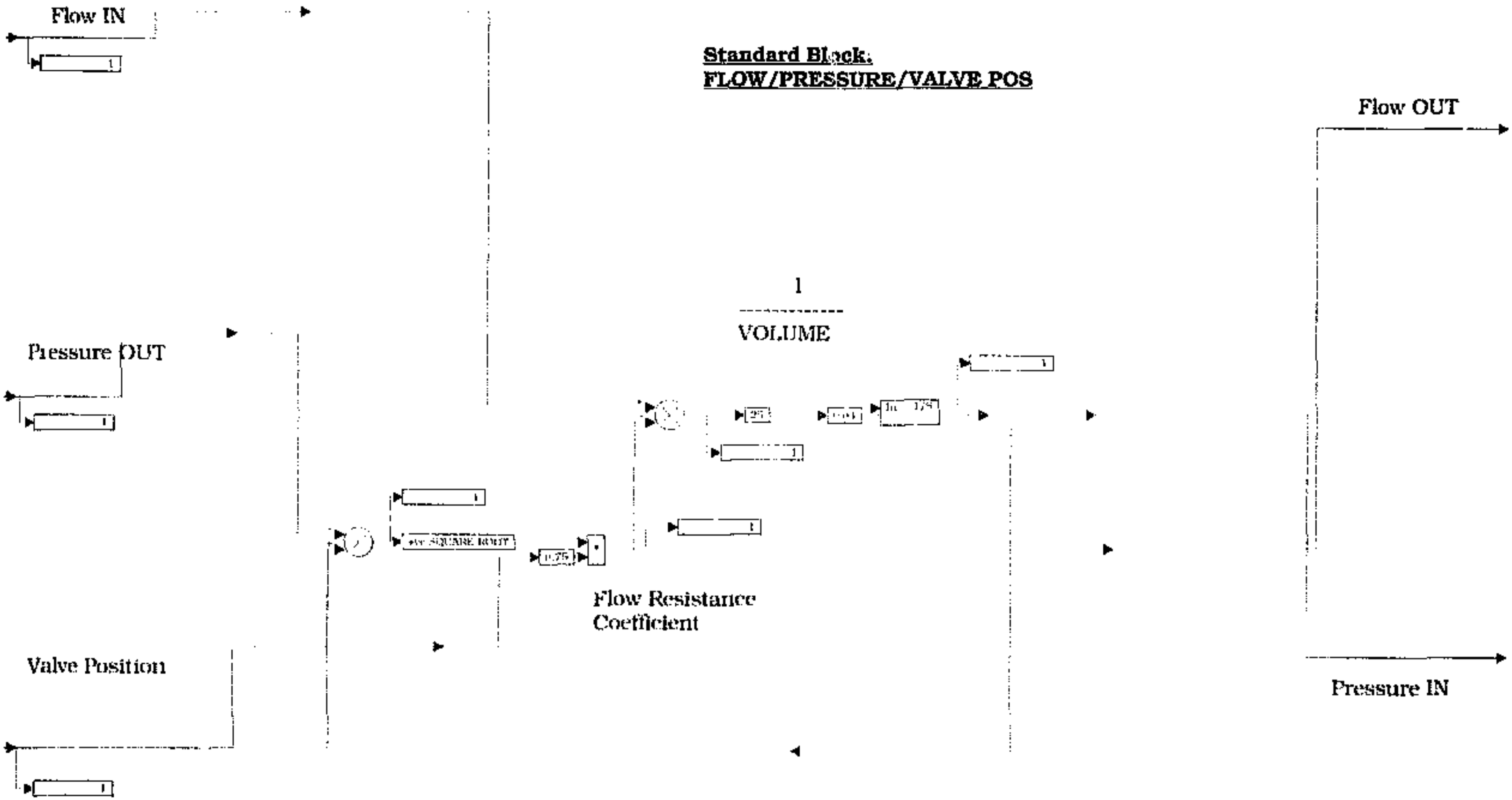
Standard Block:
FLOW/PRESSURE

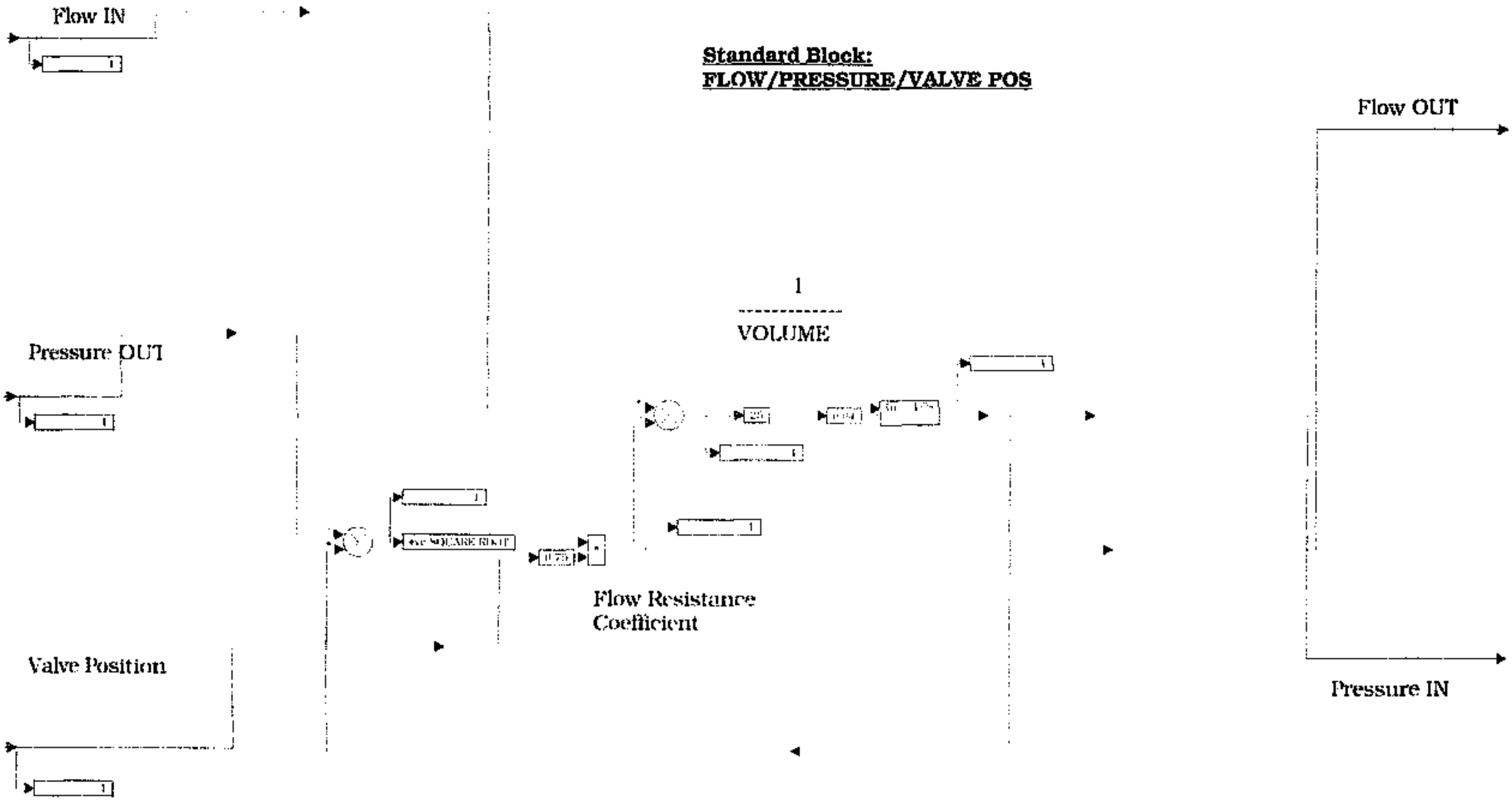


Standard Block:
TURBINE ENTHALPY DROP

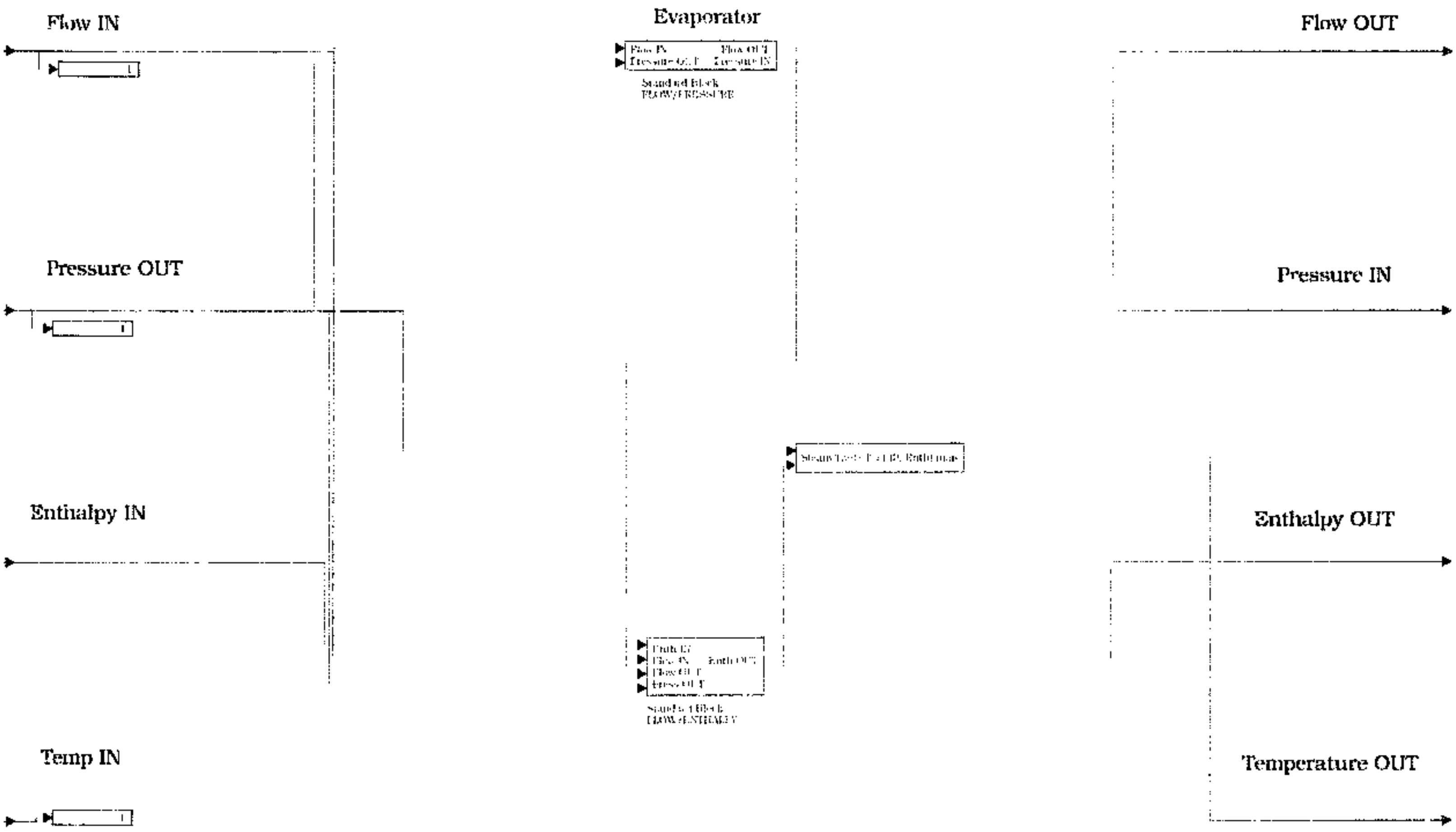
This Block Calculates the Enthalpy drop across the turbine based on the calculated pressure drop.
The inlet Entropy is calculated from the inlet Temp and Pressure, and the outlet Enthalpy is calculated at the discharge Press with this same Entropy

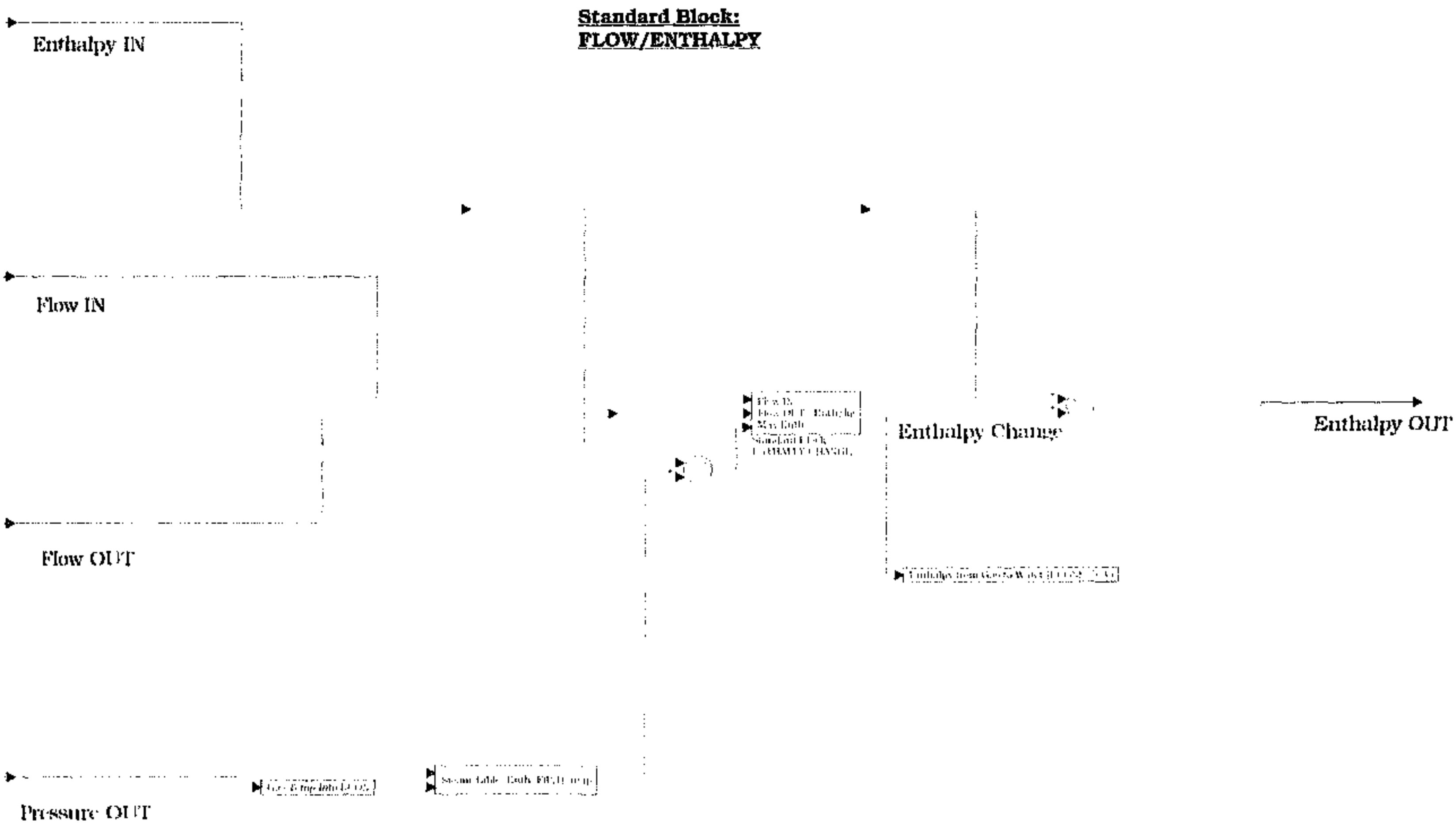


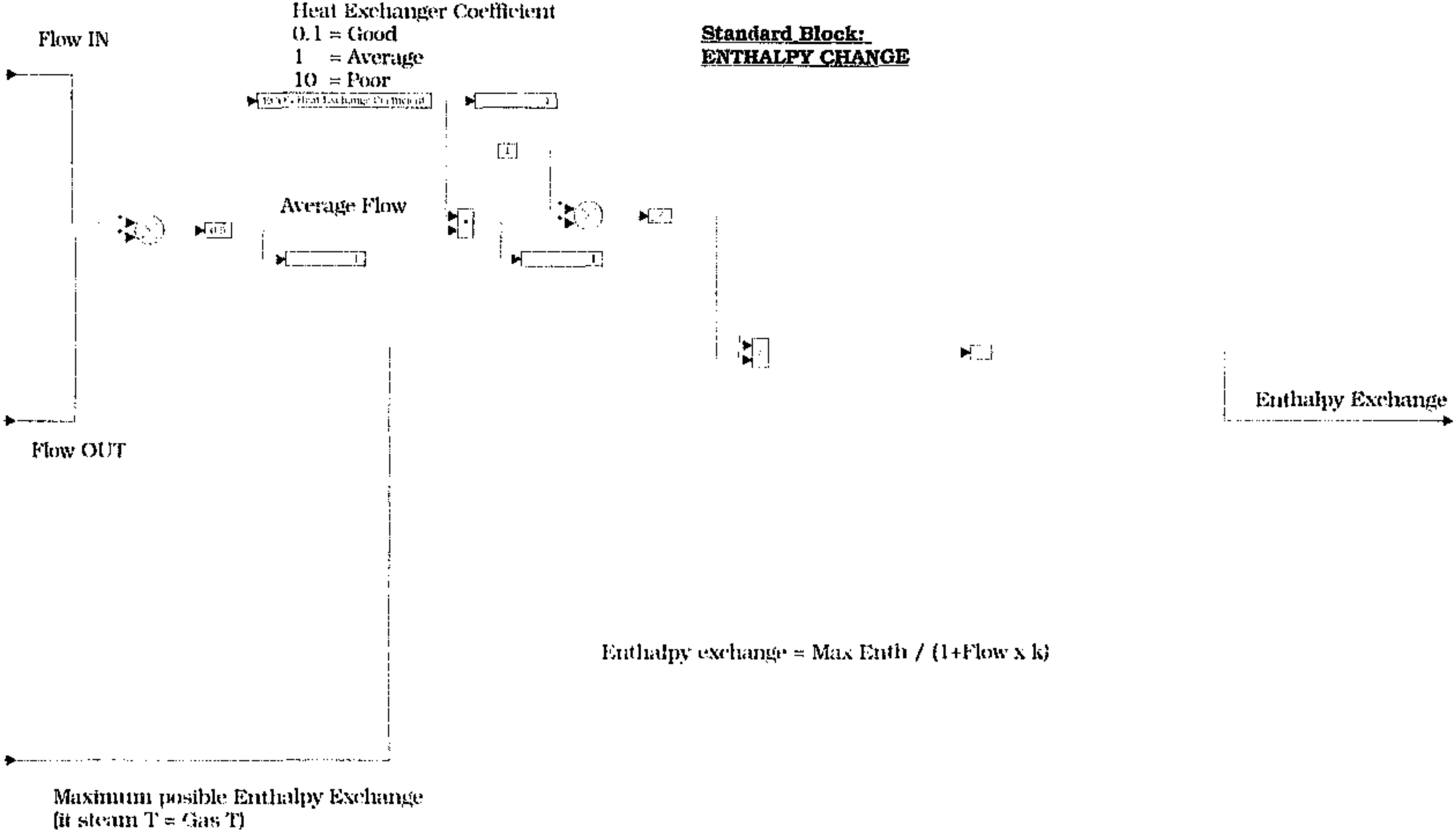

Flow IN
Standard Block:
FLOW/PRESSURE/VALVE POS
Flow OUT
1
VOLUME
Pressure OUT
SQUARE ROOT
0.75
Flow Resistance
Coefficient
Valve Position
Pressure IN

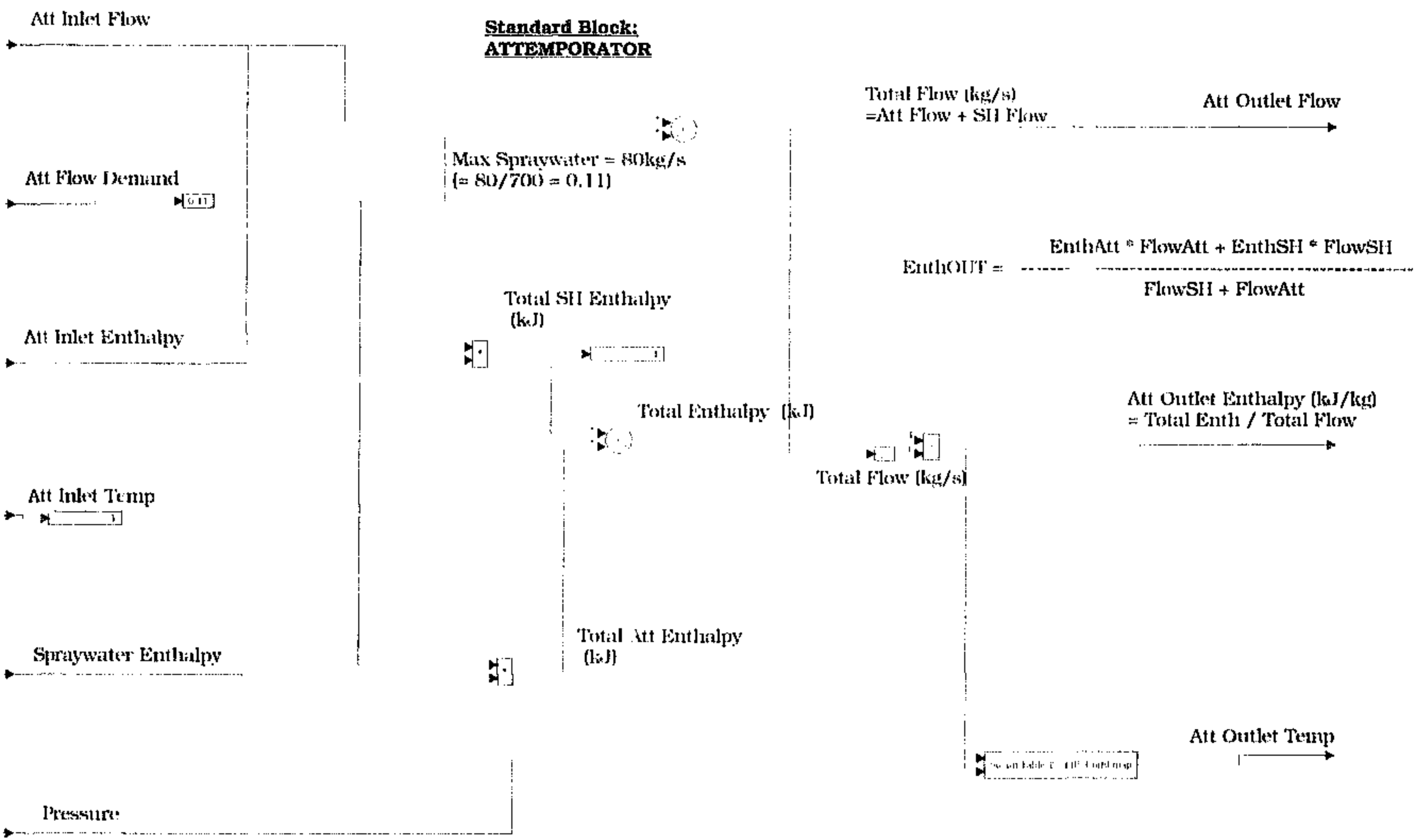

Flow IN
Standard Block:
FLOW/PRESSURE/VALVE POS
Flow OUT
1
VOLUME
Pressure OUT
Flow Resistance
Coefficient
Valve Position
Pressure IN

Standard Block: FLOW/PRES/ENTH/TEMP




Standard Block:
FLOW/ENTHALPY
Enthalpy IN
Flow IN
Flow OUT
Pressure OUT
Enthalpy Change
Enthalpy OUT


Heat Exchanger Coefficient
0.1 = Good
1 = Average
10 = Poor
Flow IN
Standard Block:
ENTHALPY CHANGE
Average Flow
Enthalpy Exchange
Flow OUT
Enthalpy exchange = Max Enth / (1+Flow x k)
Maximum posible Enthalpy Exchange
(if steam T = Gas T)


Standard Block:
ATTEMPORATOR
Att Inlet Flow
Att Flow Demand
Att Inlet Enthalpy
Att Inlet Temp
Spraywater Enthalpy
Pressure
Max Spraywater = 80kg/s
(= 80/700 = 0.11)
Total SH Enthalpy
(kJ)
Total Enthalpy (kJ)
Total Att Enthalpy
(kJ)
Total Flow (kg/s)
=Att Flow + SH Flow
Att Outlet Flow
EnthOUT = EnthAtt * FlowAtt + EnthSH * FlowSH / FlowSH + FlowAtt
Att Outlet Enthalpy (kJ/kg)
= Total Enth / Total Flow
Total Flow (kg/s)
Att Outlet Temp

REFERENCES

[1] Coulman, G.A. and I. Gotlieb, A Fundamental Study of Once Through Steam Generator Dynamics, *SCS*, Vol. 3, No. 1, 1979, pg. 67

[2] Maffezzoni, C., *ISSUES IN MODELLING AND SIMULATION OF POWER PLANTS*, IFAC International Symposium on Control of Power Plants and Power Stations, March 9-11, 1992, pg. 19

[3] Debelle, J., *SOME TRENDS IN AUTOMATIC POWER PLANT CONTROL*, IFAC Power Systems and Control Applications, Brussels, 1988, pg. 10

[4] Chien, K.L., E.I. Ergin, C. Ling and A. Lee, *DYNAMIC ANALYSIS OF A BOILER*, Trans. ASME 80, 1958

[5] Profos, P., *THE CONTROL OF STEAM PLANTS*, Berlin, Springer Verlag, 1962, (book in german, with the title "Die Regelung von Dampfanlagen")

[6] Dymek, T. , MATHEMATICAL MODELING AND COMPUTER SIMULATION OF THERMAL POWER PLANTS, ICHMT 2[nd] International Forum on EXPERT SYSTEMS AND COMPUTER SIMULATION IN ENERGY ENGINEERING, University of Erlangen, 17 - 20 March 1992

[7] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*. 1[st] ed. England: Research Studies Press Ltd., 1990, vol. 1

[8] Dukelow S.G. ,*THE CONTROL OF BOILERS*, 2[ND] Ed., USA: Instrument Society of America, 1991, vol 1

[9] Goeminnie, P., and F. Van De Meulebroeke, Real Time Simulator for Turbogenerators, *IFAC JOURNAL POWER SYSTEMS MODELLING AND CONTROL APPLICATIONS*, Brussels, Belgium 1988, pg. 517

[10] Schuck,, S.M., SIMULATION AND CONTROL OF A 500MW BENSON BOILER-TURBINE UNIT, 1977, PhD Thesis, University of the Witwatersrand, South Africa

[11] Cheres, E., Small and Medium Size Drum Boiler Models Suitable for Long Term Dynamic Response, *IEEE TRANSACTIONS ON ENERGY CONVERSION*, Dec 1990, Vol. 5, No. 4, pg. 686

[12] De Mello, F.P., Boiler Models for System Dynamic Performance Studies, *IEEE TRANSACTIONS ON POWER SYSTEMS*, February 1991, Vol. 6, No. 1, pg. 66

[13] Coulman, G.A. and I. Gotlieb, A Fundamental Study of Once Through Steam Generator Dynamics, *SCS*, Vol. 3, No. 1, 1979, pg. 67

[14] Kwan, H.W. and J.H. Anderson, A Mathematical Model of a 200MW Boiler, *INTERNATIONAL JOURNAL OF CONTROL*, 1970, Vol.12, No. 6, pg. 977-998

[15] Beato, A. and F. Marziali, Study of the Stat-up Control of a Drum Boiler By Means of a Mathematical Model, *IFAC INTERNATIONAL SYMPOSIUM ON THE CONTROL OF POWER PLANTS AND POWER SYSTEMS*, March 11-12, 1972, pg. 259

[16] Klefenz, G., *AUTOMATIC CONTROL OF STEAM POWER PLANTS*3rd ed., Bibliographisches Institut Mannheim, 1986, Vol. 1

[17] Hartley, T.T., G.O. Beale and S.P. Chicatelli, *DIGITAL SIMULATION OF DYNAMIC SYSTEMS, A CONTROL THEORY APPROACH*, 1st ed USA: Prentice-Hall, 1994, pg. 5

[18] Hartley, T.T., G.O. Beale and S.P. Chicatelli, *DIGITAL SIMULATION OF DYNAMIC SYSTEMS, A CONTROL THEORY APPROACH*, 1st ed USA: Prentice-Hall, 1994, pg. 4

[19] Hartley, T.T., G.O. Beale and S.P. Chicatelli, *DIGITAL SIMULATION OF DYNAMIC SYSTEMS, A CONTROL THEORY APPROACH*, 1st ed USA: Prentice-Hall, 1994, pg. 5

[20] Hartley, T.T., G.O. Beale and S.P. Chicatelli, *DIGITAL SIMULATION OF DYNAMIC SYSTEMS, A CONTROL THEORY APPROACH*, 1st ed USA: Prentice-Hall, 1994, Chapter 6 and 7

[21] Schwarzenbach and K.F. Gill, *SYSTEM MODELLING AND CONTROL*, 2nd ed., Edward Arnold Publishers, pg. 49

[22] Matko, D., B. Zupancic and R. Karba, *SIMULATION AND MODELLING OF CONTINUOUS SYSTEMS: A CASE STUDY APPROACH*, UK: Prentice Hall 1992, Chapter 5

[23] Rossiter, J.A., B. Kouvaritakis and R.M. Dunnett, Application of Generalised Predictive Control to a Boiler-Turbine Unit for Electricity Generation, *IEE PROCEEDINGS-D*, January 1991, Vol. 138, No. 1, pg. 59

[24] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*, 1st ed. England: Research Studies Press Ltd., 1990, vol. 1, pg. vii

[25] Dukelow S.G. ,*THE CONTROL OF BOILERS*, 2ND Ed., USA: Instrument Society of America, 1991, vol 1, pg 123

[26] Dolezal, R. , *PROCESS DYNAMICS*, 1st ed. London: Elsevier Publishing Company Ltd., 1970, vol. 1, chapter 7

[27] Dolezal, R. , *PROCESS DYNAMICS*, 1st ed. London: Elsevier Publishing Company Ltd., 1970, vol. 1, chapter 38

[28] Dymek, T. , MATHEMATICAL MODELING AND COMPUTER SIMULATION OF THERMAL POWER PLANTS, ICHMT 2nd International Forum on EXPERT SYSTEMS AND COMPUTER SIMULATION IN ENERGY ENGINEERING, University of Erlangen, 17 - 20 March 1992

[29] Dolezal, R. , *PROCESS DYNAMICS*, 1st ed. London: Elsevier Publishing Company Ltd., 1970, vol. 1, pg 11

[30] Renze, H., A New Concept for Unit Regulation using Frequency Stabilization for the 'Natural Sliding Pressure' Mode of Operation in Once-Through Forced Flow Boilers, *VGB KRAFTWERKSTECHNIK* 62, August 1982, Number 8, pg. 532

[31] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*, 1st ed. England: Research Studies Press Ltd., 1990, vol. 1, pg 187

[32] Kearton, W.J., *STEAM TURBINE THEORY AND PRACTICE*, 7th Ed., 1958, Pitman Publishers

[33] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*, 1st ed. England: Research Studies Press Ltd., 1990, vol. 1, pg 180

[34] Suzuki, Y., Sik Pak, P., *SIMULATION OF A SUPERCRITICAL ONCE-THROUGH BOILER*, Proceedings Summer Computer Simulation Conference, 1979, pg 184

[35] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*, 1st ed. England: Research Studies Press Ltd., 1990, vol. 1, pg 179

[36] Suzuki, Y., Sik Pak, P., *SIMULATION OF A SUPERCRITICAL ONCE-THROUGH BOILER*, Proceedings Summer Computer Simulation Conference, 1979, pg 184

[37] De Mello, F.P., Boiler Models for System Dynamic Performance Studies, *TRANSACTIONS ON POWER SYSTEMS*, February 1991, Vol. 6, No 1, pg. 68

[38] Schuck,, S.M., SIMULATION AND CONTROL OF A 500MW BENSON BOILER-TURBINE UNIT, 1977, PhD Thesis, University of the Witwatersrand, South Africa

[39] Knowles, J.B. , *SIMULATION AND CONTROL OF ELECTRICAL POWER STATIONS*, 1st ed. England: Research Studies Press Ltd., 1990, vol. 1, pg 179 and Appendix 14.1

[40] Bird, J.O., Chivers, P.J., *ENGINEERING AND PHYSICAL SCIENCE*, 1993, Butterworth-Heinemann Ltd., pg 294

[41] Matko, D., Karba, R., Zupancic, B., *SIMULATION AND MODELLING OF CONTINUOUS SYSTEMS - A Case Study Approach*, 1992, Prentice Hall, U.K.

[42] *DUVHA COAL-FIRED GIANT, Technical Information*, Eskom Communication Services, May 1996