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IGCC power plant simulation: Steady state, dynamic and control

Fan, Zhen, Ph.D.

City University of New York, 1993

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IGCC Power Plant Simulation
- Steady state, Dynamic and Control -

by
Zhen Fan

A Dissertation Submitted to the graduate Faculty in Engineering
in Partial Fulfillment of the Requirements for the Degree of
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1993

This manuscript has been read and accepted for the Graduate faculty in Engineering in satisfaction of the dissertation requirement for the degree of Doctor of Philosophy.

28 Apr 93
Date

Reuel Shinnar
Professor Reuel Shinnar, Chair of Examining committee

4/28/93
Date

Gerard J. Lowen
Professor Gerard Lowen, Executive Officer

28 APR 93
Date

Irven Rinard
Professor Irven Rinard, Co-Mentor

Dr. Madhav Ghatge

Professor Robert Graff, Chairman of Chem.Eng. Dept

Professor Roberto Mauri

Supervisory committee
The City University of New York

ABSTRACT**IGCC POWER PLANT SIMULATION****- Steady State, Dynamic and Control -**

by

Zhen Fan**Advisors: Professor Reuel Shinnar and Professor Irven H. Rinard**

According to today's know-how, the integrated coal gasification combined cycle (IGCC) technology is capable of meeting the requirements which will be placed on future power generation facilities with respect to low emissions and costs combined with high efficiency and availability. The fluidized bed coal gasification process has been demonstrated at the pilot plant level.

The object of this thesis was to evaluate the steady state and the dynamic performance, the operability, and the controllability of a modified IGCC power system. This system studied includes an air-blown fluidized bed gasifier, a fluidized bed combustor, and a combined cycle turbine power system. There are two related research projects - one for the overall system, and the other for the combined cycle system alone.

The primary tool for the work was dynamic simulation. First principles, non-linear dynamic models were developed for each of the major components in the system. The control system analysis utilized linearized version of the models.

It has been found that for a gasifier there are three response modes whose time constants are related to the solid, gas, and thermal capacitances. The major questions of how rapidly a gasifier can move to a new steady state from any stand-by condition and how the turndown ratio can be enhanced by appropriate integration of the total plant

control and design have been answered. An important conclusion is that a low carbon conversion operation in the gasifier is preferred both for high thermal efficiency and for good control performance in a load following mode. By which, a gain sign change is eliminated and the interactions are decoupled.

For the combined cycle system itself, a more detailed set of models were required compared to those used in the overall IGCC system studies. Control configurations and controller tunings were determined using the multivariable BLT (Biggest Log-Modules Tuning) procedure. The performance of the combined cycle system in the load following mode was evaluated via dynamic simulations. The impacts of plant design and control have been demonstrated. It is shown that good control can be obtained by appropriate plant design.

The present IGCC models allow one to simulate real processes and make judgements before going to the pilot plant or to the commercial level. This has been done for the gasifier at the Foster Wheeler pilot plant for solid mixing, steady state operation, and transient behavior. Good agreement was obtained between experimental data and simulations. For the combined cycle system demonstration for the system at Gilbert Station, NJ, shows that the model represents the system behavior and operational data at the steady state. The simulation matches the start up data for the literature.

To my wife Jiandong and my son Diandian, whose love, patience and encouragement contributed more than anything else to the completion of this work.

Zhen Fan

Spring, 1993

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NOTATION

A	surface area	m	mass flow rate
A	constant	m	number for parallel option
A _m	ash & moisture free	Mg	angular momentum
a	reactivity	N	number
A _r	Archimeter number	n	mole number
b	constant	P	pressure
B	constant	P	power
B	controller matrix	Q	heat transferred
C	constant	q	heat transfer rate
C	solid concentration	q	solid transfer constant
C _x	valve character constant	R	gas constant
C _v	heat capacity of constant volume	R	total reaction rate
C _p	heat capacity of constant pressure	Re	Renolds number
C _v	valve character constant	r	reaction rate
d, D	diameter	S	entropy
Dulong	heat combustion of coal	S	cross area
E	density distribution function	S	Laplace transfer
E _k	equilibrium constant	T	temperature
f	frequency	U	heat transfer coefficient
f	parameter	U	internal energy
F	detuning factor	U, u	linear velocity
F	flow rate	V	volume
G	transfer function	W	width
G	transfer function matrix	W	weight
H	height	W	work
H	enthalpy	X	valve position
h	heat transfer coefficient	X	steam fraction
J	Jacobin holdup	X	solid concentration in wt
K	gain	x	conversion
K	equilibrium constant	Y	output variable
k	heat conduct	Y	fraction in coal or in gas
k	solid transfer	Z	liquid level
k	reaction kinetics constant	Z	parameter
L	length		
L	bed height	α	1-C _v /C _p
M	mixing index	α	constant
M	molecular weight	β	autodamping constant
M	momentum	β	constant

β	CO/CO ₂ in carbon combustion	in	inlet
ρ	density	GC	air compressor
Δ	difference	GE	electrical generator
μ	viscosity	GT	gas turbine
η	efficiency	HP	high pressure ST
ε	void fraction	i	irreversible
ϕ	IGV adjustable range	I	integration action
ϕ	shape factor	L	unit length base
∞	angular velocity	L	liquid
ζ	decay coefficient	L	long term
τ	time constant	l	liquid
Γ	steam condensing rate	LP	low pressure ST
γ	Cp/Cv	LMTD	log mean temperature difference
γ	ratio	m	average
ε	voidage	mf	minimum fluidization velocity
δ	parameter	out	outlet
λ	heat of vaporization	P	process
θ	dimensionless time	p	particles
subscripts and superscripts		r	reversible
*	equilibrium	r	refiring
0	initial condition	S	superheater
bar	average	S	steam
0	rated condition	S	short term
1	inlet	s	set point
2	outlet	s	SURFACE REACTION
a	root	ST	steam turbine
a	first order lead	T	total
B	boiler or boiling	tr	transferred
B	bubble	U	manipulated valuable
C	condenser	V,v	vapor
c	controller	VL	vapor-liquid
C	circulated	W	wall
d	diffusion	W	water
E	economizer	w	wake
e	electric	X	valve
f	fuel	x	conversion
g	gas		
G	gas		

CHAPTER 1. INTRODUCTION

Process modeling and computer simulation have proved to be successful engineering tools for the design and optimization of physical and chemical processes. The use of simulation has expanded rapidly during the past two decades because of the availability of large hybrid computers. In the chemical process industry, large, realistic nonlinear problems are now routinely being solved via computer simulation.

Modeling and simulation are disciplines that are increasingly needed in almost all advance engineering design and development work. Modeling draws on mathematical and computational skills and the ability to relate these techniques to physical reality through physical and chemical laws and approximations on the base of elusive and highly valuable gift of physical insight. These insights and a broad of understanding to dynamical behavior are particularly essential if, to in control system, a model is to be used as a basis of design to system behavior.

In any given problem, modeling has an objective, and the effective model will be created so that the model is capable of providing reliable results in the context of engineering objective. A given piece of plant will hence frequently have many models, each tailored to provide information concerning a particular objective. A model has a little value unless the precise objective of that model are clearly stated and understood.

Control system design and control performance assessments are two areas where system dynamics are part of the properties that are essential to the engineering objective. Indeed, practitioners have claimed that modelling takes the majority of the non-hardware based effort, leaving control algorithms development in a vitally important but less time-consuming position. The two aspects are intimately entwined, however, as a control algorithms development and assessment are the objectives of the model, and control algorithm development itself is a model-based discipline.

It can be said that adequately authentic simulators will be suitable for the real applications (Ramirez,1989).

1 PURPOSES OF RESEARCH PROJECTS

The present work involves two research projects related to the IGCC (Integrated coal Gasification Combined Cycle power plant) technology.

The aims of these projects are to simulate an IGCC power plant based on an air blown fluidized bed gasifier, in order to investigate static and dynamic behavior, control and operability problems of the IGCC system where an air blown fluidized bed gasifier, gas/steam combined cycle are involved (Fig.1). The tool for these investigations will be a coal gasification model, and a combined cycle power plant model, which are suitable for the studies of both the steady state case and the dynamic responses either in open loop or in closed loop by control. A new tool is the application of DYSIM software to combined cycle system.

Since an IGCC system includes both a coal gasification subsystem and a combined gas/steam cycle power generation subsystem, the research effort has been divided into two parts (Fig.1 & Fig.2).

1.1 Gasifier Simulation

The first part is the study of overall IGCC system using a simplified model of the combined cycle subsystem. The emphasis here is the dynamic behavior and the control of the coal gasification subsystem where the gasifier is an air-blown fluidized bed. The basic system is shown in Fig.1, where it also includes a fluidized bed combustor. A question is whether or not such an IGCC system can be made to operate together under the constraints imposed by end-use and environmental considerations for a load following in an electric power distribution grid.

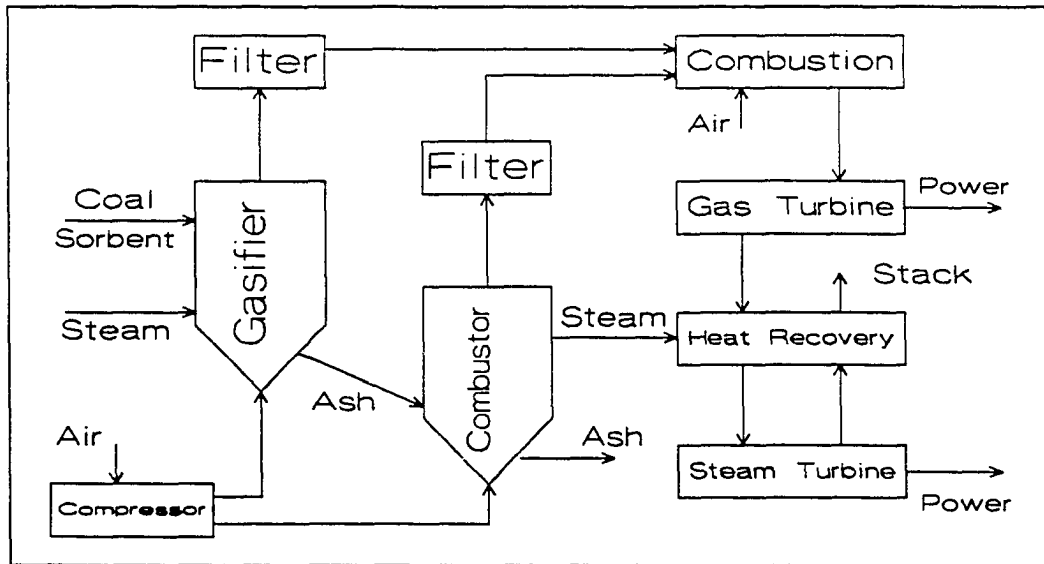


Figure 1 IGCC System

1.2 Combined Cycle Power Plant Simulation

The second part will be a more detailed study of the dynamics and control of the combined cycle subsystem. This is a problem of interest as the combined cycle part of our IGCC is identical to that widely used in the power generation using either natural gas or distilled oil as a fuel. A typical combined-cycle system is shown in Figure 2.

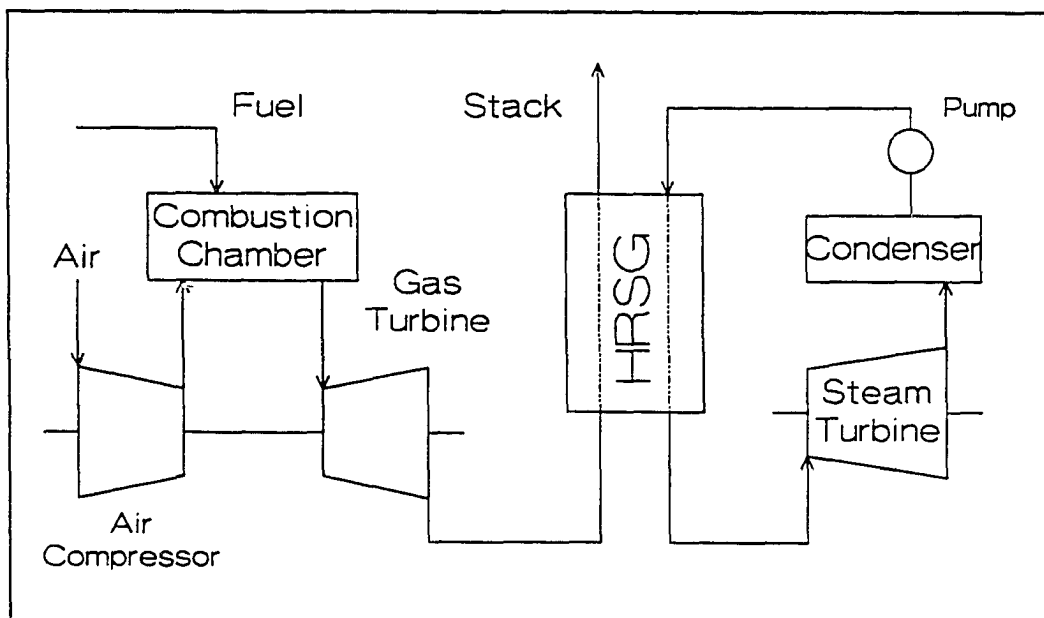


Figure 2 Scheme of Combined Cycle Power Plant

This research project includes a development of a dynamic model to simulate the

combined-cycle power plant. The model is then used to evaluate control strategies, to select controlled and manipulated variables, and configurations, to ensure a desirable steady state operation, to suppress the external disturbance, and to optimize the operation economically.

2. METHODOLOGY

Study of both of the problems outlined above requires the following steps:

(1) Development of a mathematical model

A mathematical model is needed which contains sufficient detail to expose any relevant control problems but does not contain such an overwhelming amount of detail as to obscure the underlying structure of the problem. The dynamic model of the gasifier is an extension of a steady-state model developed previously by Shinnar and coworkers (1991). A detailed model for the combined cycle plant was also developed.

(2) Incorporation of developed models in a simulator

The mathematical model must be programmed for use on a digital computer. The gasifier part was developed in a self-contained structure dynamic and steady state simulator. The combined cycle model was integrated into the DYSIM simulator developed by Professor Rinard and coworkers.

(3) Extraction of a simplified dynamic model

Computer simulations are a very useful tool for verifying that one control strategy is superior to another. However, they do not reveal much of the underlying structure of the process dynamics.

This is best done by extracting from the complete model a simplified one which, while less accurate, reveals the dynamic structure of the system. It should account for the principal poles (or time constants) of the system and reveal the presence of any non-minimum phase elements such as deadtimes and right-half-plane (RHP) zeros. It is these

latter which place strong limitations on the controllability of the system.

(4) Development of control strategies

The models developed in Items 1 through 4 allow one to develop various control strategies. There are many issues here which are discussed in more detail later in this work. Suffice it to say that between modern process control techniques and a thorough familiarity with the process and its dynamic behavior will lead to a number of alternate control strategies which can then be evaluated in a way of semi-quantitatively by dynamic simulation.

(5) Trade-offs between the process design and its controllability

Quite often the control of a process can be improved more readily by altering its design rather than by the control algorithm itself. Part of our work is to identify and evaluate trade-offs design and controllability.

3 BACKGROUND

Ten years ago, economic studies conducted by EPRI and others have identified Integrated Coal Gasification Combined Cycle (IGCC) power plants as major alternatives for generation of power from coal in an environmentally acceptable manner (Holt,1992). As mentioned, these units may be able to respond to system load changes and/or participate in system regulation to any great extent.

In order for coal conversion system to be successfully integrated into utility or industrial use, it must be demonstrated that they can meet environmental requirements and also produce a fuel that meets the specifications of the downstream equipment. A coal conversion power generation consists of two different unit process - a chemical plant as a fuel supply subsystem connected to a power generation subsystem. It is critical that these processes be able to function together and perform to the load demands of a utility grid. The individual subsystems have been operated stand alone. However, these two

subsystems have not been operated together in a load configuration. In a previous study of Li Weng, it was shown that the IGCC systems based on air blown gasifier have a potential for a higher thermal efficient power plant with low emissions. However those study dealt with steady state performance. For a power plant, controllability over a wide range of power output is essential (Weber,1983). Holt (1992) has shown that IGCC plant based on the oxygen blown Texaco gasifier should have acceptable response to load changes. Here our research concentrates on IGCC system based on an air blown fluidized bed gasifier.

3.1 Coal Gasification

Oxygen blown gasification systems based on entrained gasifiers are well demonstrated (Dow Plaquemine, Shell Houston and Texaco Coolwater). Air-blown fluidized bed gasifiers have not been operated on the same scale, but some commercial applications are in the planing stage. It is expected respond readily to changes in the electric power demand, making it suitable for daily load following service. Such a plant should be designed with the prospect for relative fast rate of response of possible load following service (Holt,1992; Quentin,1985).

Professor Shinnar and his students have done some work before on the steady state simulation and a total power plant evaluation of novel system configurations that improve the performance and the cost of the total system. It leads to better overall system results (Shinnar,1990). There are some steady state models for the gasifier in literatures (Weng, 1990), where a reactivity of solid is considered. Now a more detailed model with consideration of the reactivity, reactivity distribution and the coal property change, is needed for dynamic and control simulations, especial for a sorbent hold hot gas cleanup system. But a few dynamic models available at present time for the fluidized bed gasifier, that would allow one to evaluate its dynamic performance and to predict what

control and operability problems might be encountered from incorporating such a gasifier and a fluidized bed combustor into an IGCC system with an in situ hot gas clean up by using the sorbent (dolomite and/or limestone) in a gasifier (Shinnar,1990).

Control analysis by computer simulation to verify the dynamic performance and controllability of future process technology of an IGCC power plant is rather unique and much useful. By which, the impacts of design on control may be evaluated. Control of a plant is often hampered or prevented by multi-variable coupling, unable to measure key state variables. These might be overcome by the dynamic model base simulation, where the simulation model predict effects of changes in operating conditions, provided that model is capable of predicting the dynamic responses accurately. As a result, inferential control, using secondary process variable measurement, and model based control will provide adequate control design on the base of developed model.

An IGCC system with hot sulfur capture in the gasifier and following a combustor together to get a high coal conversion is the model base at present work. The control study of it has not been seen in the literatures, especially for a more detailed model such as coal reactivity decay and the coal properties change, those are important for the model prediction and application. We prefer to find a way to take account of them and to make better control for an desired operation. For this object a reliable dynamic model will be as a base requirement and a powerful tool.

There is an problem which need model evaluation. In fluidized bed gasifiers, the gas velocity can only be varied within narrow limits. One way to overcome this is to use a method by adjusting pressure to achieve a load change. In a cool water plant dynamic experiments showed that the gasifier itself does not restrict the turndown ratio more than that imposed by the combined cycle power plant. We need to answer what will be of the turndown ratio if a combustor is linked to the gasifier in a dynamic case. Impacts of the design on the operability and control will be very important on the turndown ratio among

many constraints for a load following case.

3.2 Combined Cycle process

In the competition for higher thermal efficiency, the combined cycle power plants have taken a leading role and for many years applied successfully worldwide. They are presently popular also with utilities all over the world because of their flexibility and relatively low cost installation, where the steam is discharged into a condenser instead of being used for a chemical process. The great advantage of combined cycle is in that they can be used as the units for a load following service in power network by its fast response and high thermal efficiency.

There are a growing number of combined cycle systems in service which use either natural gas or fuel oil as energy source. Most of these units are used to provide intermediate power, i.e., they are brought on line each weekday morning and are shut down the following evening. Gas turbines have the advantage of fast response and easier start up and shut down. They also have a higher thermal efficiency compared to conventional plants and will increasingly be used for base load service. In that case the need to use combined cycle systems for load following will increase. This is the major control study which addressed in this work.

The control systems of a power station must cope with the normal changes in demand for power as a load following, and with fault transients that arise locally or in the grid network. To achieve design-life and to restrict damage progression, the individual control loops must be co-ordinated so as to prevent the violation of operational constraints during these induced manoeuvre.

There are many reports on the gas turbine for high temperature handling, life time improvement and so on. Some of works are related to thermal efficiency by different configurations and operation strategies, which depend on the heat utility synthesis (Holt,

1992; Shinnar,1990).

Prospects are bright by the DOE that development of new gas turbine based systems able to offer high efficiency (60% or higher) will get under way for the need of 21st century. A new system (Holt,1992), called HAT (humid air turbine) cycle, is ideally made out, and received arguably the most R&D attention. Although still in the evaluation stage, recent studies suggest that the HAT can offer impressive efficiency and operating cost saving compared to combined cycles, it is particularly in conjunction with coal gasification technology. The significant changes includes a raise in firing temperatures of over 2600°F and a oust of steam turbine.

3.3 DYSIM

There are many programs developed for simulation of a specified process. As they are specified, high complexity, the structures or configurations are sometime fixed, and hard to be modified by the user.

Now in the market of computer software, more and more programs tend to have a flexible in configuration by the user, such as ASPEN, and MCS etc. (ASPEN is a software for a steady state chemical process simulation, MCS for a dynamic electrical engineering simulation), where the user determines a scheme, sets equipments and their parameters, and connects them by signal flow, material or electrical.

The DYSIM is a Dynamic Chemical Process Simulator designed by Professor Irven H. Rinard (1992) at Department of Chemical Engineering, The City College of City University of New York. It consists of basic run director and various equipment models that are written by himself and co-workers.

The DYSIM is a suite of programs for performing dynamic simulation of process systems. It is intended to be both a working simulator for the process system engineer and a research tool for dynamic simulator development. The processes are simulated by

interconnecting the appropriate equipment model. A variety of these models are well developed. Because of the master modular structure of the DYSIM, new model can be added to model library in a straight forward manner, and any different configuration can be made by the user.

4. OUTLINE OF PRESENT WORK

The major work includes two final reports in four chapters;

- 1) Coal gasification process simulation (chapter 2, 3)
- 2) Combined cycle power plant simulation (chapter 4, 5)

Data from the KRW and the FW are used to evaluate the gasification model, and data from the Gilbert Station at Jersey Central is applied to test combined cycle model.

As an additional, impacts of above control to the IGCC power plant design will be discussed. The engineering and operating considerations for integrating process technologies and providing reliability, load response, and turndown requirements will be addressed.

5. REFERENCE

Holt, N.A. (editor), Proceedings: 9th EPRI Conference on Gasification Power Plants, July, 1992

Kutten, M., "Modeling and control of a fluidized bed gasifier", Doctor dissertation, City College of CCNY, 1978

Quentin, G.H., "Control analysis of gasification/combined cycle (GCC) power plants", Chemical Process Control II, 1985

Ramirez, W.F., "Computational Methods for Process Simulation", Butterworths, Boston, 1989

Rhinehart, R.R., R.M. Felder & J.K. Ferrell, "Dynamic modeling of a pilot-scale fluidized-bed coal gasification reactor" Ind. Eng. Chem. Res., 26(1987), 738

Rinard,R.H., "DYSIM user's manual", The City College of New York, 1992

Shinnar,R., "System assessment of novel concept in coal gasification" , final report, DE-AC21-86Mc23280, 1990

Weber,David S.,G.H.Quentin & J.E.McDaniel, "Simulation objective & requirements for advanced gasification-based power plants", 3rd Annual EPRI Contractors' Conference on coal gasification, Nov.1983

Weng,L., "Modeling and analysis of the design constraints for IGCC system", Doctor dissertation, City College of CCNY, 1991

PART I COAL GASIFICATION PROCESS

The object of Part I includes steady state and dynamic modeling, and control of the coal gasification process for the IGCC system. It has three parts. The first part deals with the coal gasifier models in the steady state and dynamic cases. The second part evaluates the dynamics and control of the coal gasification process for the IGCC system in a load following simulation. In the third part, a linear model is developed for the coal gasifier with the aim of finding the response times for the gasifier in order to get a direct understanding of the long and the short terms responses and use it further in controller design.

The comprehensive process models developed in the project were made available for detailed control studies. These models reflect both dynamic and static effects and are based on physical principles and unit dimensions. The simulation runs were made using those non-linear process models to evaluate the dynamic performance and to provide guidelines for design modifications and control configurations.

After finalizing the gasifier model, we connect it to a combined cycle power plant, where we include a fluidized bed combustor that uses the unconverted carbon from the gasifier. The flue gas and the fuel gas are then sent to a gas turbine along with compressed air to drive a generator. This configuration is further used in chapter 3, to study fuel plant control for the IGCC process. By combining the detailed knowledge supplied by the various models, the crucial step of defining the control structure is further then taken for the fuel plant, specially for the coal gasifier.

Further detailed models for the combined cycle power plant subsystem will be presented in Part II, where a dynamic simulator DYSIM will be used.

CHAPTER 2. GASIFIER MODEL

The aim of this work is to investigate the dynamic behavior, controllability and operability of a combined cycle power plant fuelled by an air blown fluidized bed coal gasifier.

Professor Shinnar, Dr. A. Avidan, and Dr. Li Weng have developed a model suitable for the steady state simulation of such a gasifier, and have performed the overall system study for different configurations (R.Shinnar, et al, 1990).

However good steady state performance is not a sufficient criterion for designing power plants. Good controllability is essential and it is therefore important to understand the dynamic behavior of such a system. Of special importance is to understand the impact of the design itself on the dynamic behavior and controllability.

The present work extends the steady state model to a dynamic one with the same process, the same configuration, as well as the same model base.

1 BACKGROUND

First, let us describe the concept of the gasifier which will be the basis of our study. A generalized description of such a system is given in Fig.1. Granulated coal is fed to the gasifier together with dolomite by injection along with air and steam, and converted to fuel gas. The sulfur in the coal forms H_2S which is removed from the gas by the dolomite. The fuel gas is then fed to a combustion chamber to be combusted with compressed air and fed to a gas turbine. The gasifier, in essence, is the substoichiometric first stage of a two stage combustor.

A power plant is expected to respond readily to changes in the electric power demand. Such a plant should be designed with the prospect of a relatively fast rate of response for a possible load following service. But, as of now, there are no suitable

dynamic models available for an air blown fluidized bed gasifier, which would allow one to evaluate its dynamic behavior, and predict what control and operability problems might be encountered.

If an IGCC system is used for utility power generation applications, it is critical that the gasifier be able to handle a wide variety of coals while maintaining a gas quality that is acceptable to the cleanup equipment and, after that, to the power plant. From a broader

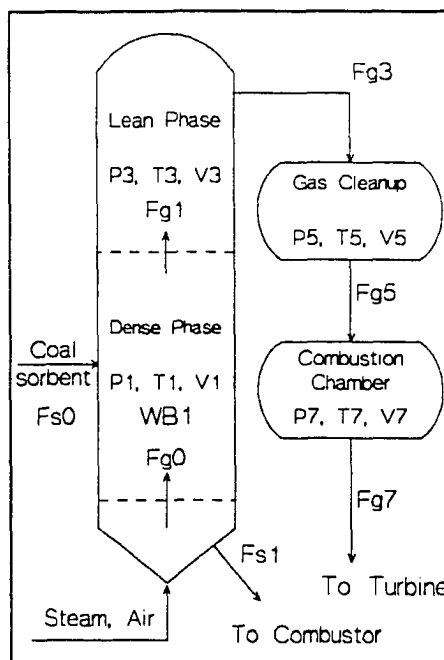


Figure 3 Gasifier

perspective, it is critical to show that the IGCC system can operate successfully in a controlled manner and supply variable load demands. The dynamic performance characterization is critical if the overall system is to be applied successfully.

For the last quarter century, DOE and other organizations have vigorously pursued R&D activities aimed at advanced coal technologies that are economically attractive and environmentally sound. As a result of these programs, the IGCC technology has developed a solid technical foundation. This technology is now ready for large scale commercial demonstrations. An excellent overview of IGCC related projects funded by DOE was given at the 9th EPRI conference on gasification power plants (Bajura, 1992), where the main thrust is on air blown systems with hot gas clean up. The air blown IGCC offers a number of advantages:

- * Potential for lower capital cost as an oxygen plant is eliminated.
- * Higher plant efficiency and lower operating costs as an oxygen plant consumes up to 9 % of produced power.
- * Hot gas clean up makes the fuel gas maintain all or most of its sensible heat.

This contributes to higher thermal efficiency and minimizes the number of heat exchangers.

On a commercial scale, such an IGCC plant is expected in the near future.

The importance of power plant dynamic modeling in relation to a safe and economical operation has been recognized throughout the industry. Transients which occur as a result of both planned and unplanned events can impact the reliability, availability, efficiency, and even the integrity of power plant systems. Computer modeling is an effective tool to characterize operating conditions, analyze performance, and understand and predict the dynamic behavior of systems under a variety of conditions. It is important that dynamic simulation be used routinely, beginning early in the planning stages of new power plant projects. A good example of DOW combined cycle process showed dynamic simulation had a dramatic impact on overall plant design and capability. The result was a plant with integrated control and equipment design to be optimized under reliability and performance in practice (EPRI GS-6173).

A number of experimentally verified steady state models have been reported for the fluidized bed gasifier. An extended review has been made by Weng and Shinnar (1990).

There exist relatively few published attempts at dynamic modeling of an air blown fluidized bed gasifier. Also the published data is limited. A short review has been given by Felder et al (1987).

Chang (1984) gives a two phase model for an isothermal manure gasifier with devolatilization developed from experimental data. Limited data was given to compare the results for the steady state case. The work of Weimer (1981) involved a bubbling fluidized bed model with temperature independent coal devolatilization. Kuten (1978) studied char gasification modeling with simple kinetics models. Goyal's (1982) work referred to a moderate pressure gasifier. The work of Franklin treated a bottom feed coal

gasifier. Most of them are qualitatively described in the cited publications. Felder's work (1987) was compared with experimental data, but, the data was shown in an unpublished way. None accounted for the hot gas cleanup or included a linear model analysis. Neither was the gasifier model merged to the power plant simulation to perform a load following control study in any of the above works.

Some models related to an entrained fluid bed (Texaco), moving bed have been developed to demonstrate integration effects (EPRI AP-2053, EPRI GS-6073) between fuel and power plants. Early simulation studies (EPRI AP-2053) showed controlled operation of IGCC without intervening gas storage buffers. That research also showed dynamic response of 10 % /min and helped in determining control strategies for the Texaco case (EPRI AP-4832). The work was updated for a commercial IGCC project (EPRI GS-6173). Corman (1983) developed a simulator based on component models for a moving bed case, where his main attention was for the gas resaturator.

Previous work indicates (Shinnar 1990) that fluidized bed gasifiers have a more complex dynamic response compared to that of an entrained bed, e.g., inverse response and overshoot. Kuttan's (1978) work was based on a fluidized bed gasifier using simple coal gasification kinetics. Felder's (1987) work involved coal reactivity and no sorbent. The transient responses of all process variables follow roughly exponential decaying paths over a 1 to 20 minutes period followed by a slow drift of up to 2 hours. Inverse response or overshoots were not mentioned.

A more realistic steady state model concerning sulfur capture for fluidized bed gasifier has been developed (Weng, 1990), which takes into account coal properties, and the decay of char reactivity with conversion. The model showed good agreement with published data-for different large fluidized bed gasifier pilot plants. It predicted well the impact of changes in operating conditions on temperature, carbon conversion, and gas composition, and is now used by DOE and IGT. We have already, as the first part of this

project, constructed a dynamic model, based on Weng's steady state one.

This research and development program was carried out by using computer simulation to evaluate the IGCC concept. The study of IGCC described here is focused on process understanding rather than on component configuration. The goals of this simulation are to see that the fuel plant subsystem is compatible with the combined cycle power plant and to study the integration and control of the overall system.

2 MODEL DESCRIPTION

A coal gasification subsystem has been investigated in detail. For simulation of the gasifier, we developed a detailed non-linear steady state and dynamic model. If a process is constrained by a slow reaction, coarse particles, and therefore a limited gas conversion and a big solid holdup, a detailed description of the flow model for the gas phase is not critical. The reason is that as the reaction rate is slow, its transient time is long compared to that of solid mixing in the fluidized bed, which means that reaction is taking place at an average gas composition. Purdy (1984) found that a well mixed CSTR model was as effective as the complex bubble model of Kato and Weng. Some comparisons have been given by Levenspiel (1991), where he showed that the CSTR model is a good assumption at low conversions. Dr. Weng and Prof. Shinnar (1991) used a CSTR model which greatly simplified the computations with a small percentage loss in accuracy. Here this simplification is adapted, which gives an advantage in reducing our model from partial differential equations (PDE's) to ordinary differential equations (ODE's) for the gasifier.

Some additional considerations for dynamic modeling, such as gas and solid holdups, are discussed in the following paragraphs:

For a dynamic study, especially for the IGCC system, a fast rate of fuel gas response is expected. This is important when the fuel gas is used for power generation

as it needs a fast rate control. For this reason models for the gas phase and gas holdup are included in the dynamic simulation. They are assumed as well mixed CSTR models both for the dense phase and for the lean phase.

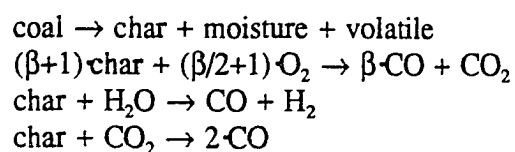
As the bed weight, or bed height is a control variable, it changes with time. In the dynamic simulation, in some cases the gas velocity will change by a large amount because of as much as a, say, 50% reduction in the gas feed during the transient. This means that the gas velocity will be halved and this changes the operating conditions tremendously, and hence the bed height and the gas voidage. A fluidization model is therefore applied to simulate these cases for the bed height, bed weight and gas holdup. This model can also be utilized to simulate solid mixing/segregation in the gasifier to see the effects of operation changes on the solid profile, when there are different solid particles by size and density.

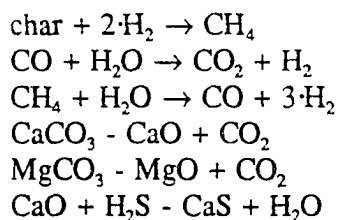
We also need to develop some relationships between state variables, measurable or not, and kinetic rate expressions, such as relations for carbon conversion and carbon content. Even the definition of carbon conversion needs to be redefined for the dynamic case when the coal properties change. This is not included in Felder's work (1987).

There are differences for the steady state simulation too. To meet the above requirements, modifications are made to the steady state model, both for the purpose of steady state simulation and for the case of initial value and final value checking for the dynamic simulation.

2.1 KINETIC MODELS

The important reactions occurring in a gasifier are:





As the gas turbine in the IGCC plant requires fast control, we have to consider the fuel gas response rate or a time constant, i.e., as the benchmark with which we need to compare the different reaction time constants and residence time for the modeling. The first judgement for kinetics, free or not, is its time constant. In pilot plant case, τ_g is about 10s. The magnitudes of the pertinent time constants involved in the model are listed below ($t=\tau$).

* solid RTD	1-10 hrs
* gas RTD	10 s
* devolatilization	1 s
* O ₂ disappear	0.01-0.1 s
* gasification	10-30 hrs
* calcination	10-200 s
* sulfur caption	1-20 min
* shift reaction	infinite fast
* coal burning	0.5-10 min
* solid movement	10-20 cm/s

We can see that the devolatilization, oxidation and shift reaction can be treated as infinitely fast compared to the gas τ_g . The solid mixing is also fast enough to compared to its residence time.

The reaction rate is determined by the state variables, as in the steady state and it leads to a dependency on the solid τ . But in the dynamic case, since the residence time distribution (RTD) changes with time, we have to relate the rate processes to the system's state variables, which are now time functions, and to a summation of their histories, which thus results in a distribution. The rate process is now weighted by these distributions functions and is therefore some kind of an average.

For the energy balance, we use the combustion heating value (HHV), like coal,

as a base, which includes the sensible heat.

2.1.1 GASIFICATION KINETICS

Many authors have observed the dependency of the overall gasification rate on conversion. Muhlen's (1985) result has been used in our model. It comes from combining the elementary reaction steps by Shaw (1977) and is the same as used by Johnson (1974) for reactivity decay. The equations are:

$$r = r_{x=0} (1-x)^{2/3} \exp(-b \cdot x^2 / RT) \quad (1)$$

$$r_{x=0} = \frac{k_1 P_{CO_2} + k_8 P_{CO_2}^2 + k_9 P_{H_2O} + k_{11} P_{H_2O}^2 + k_{12} P_{H_2O} P_{H_2} + k_4 P_{H_2}^2}{1 + k_2 P_{CO_2} + k_3 P_{CO} + k_{10} P_{H_2O} + k_5 P_{H_2}} \quad (2)$$

$$b = (25.55 - 0.36 P_{H_2O}) \quad (KJ/mol) \quad (3)$$

The predictions are good only for $X < 0.8$. For considering values of $X > 0.8$, which is generally the case, we take a corrected term developed from the data fitting as:

$$r_{x=0} = (1-x)^{2/3} \exp(-b \cdot x^2 / RT - 11.5x^{15}) \quad (4)$$

where, $k_i = k_{i0} \exp(-E_i / RT)$, the k_{i0} and E_i are the same as used by Weng (Muhlen, 1985).

It is noted that the gasification rate is a function of char conversion, which gives a reactivity decay during the reaction. With these kinetics applied to the flow model, a master equation was thus developed.

2.1.2 DEVOLATILIZATION PROCESS

The initial coal gasification stage of devolatilization (Anthony, 1976), is completed within seconds. This has been confirmed by experiments. The rate equation is:

$$\frac{dV}{dt} = 1.14 \cdot 10^5 \cdot \exp(-8900/T) \cdot (V^* - V) \quad (5)$$

where $\tau_r=0.03$ s under $T=1100$ °K. This is relatively small in comparison with $\tau_g=10$ s. It is therefore reasonable to assume that the rate of devolatilization is infinitely fast for dynamic simulation purposes.

The composition of volatiles chosen from the literature is from Laesione's (see Wen 1979) data that fixes ratios of CO/CO_2 , $\text{H}_2\text{O}/\text{CO}_2$ and H_2 together with material balance on the elements C, N, O, H, and S. The volatile yields in moles/gcoal are as following:

$$\begin{aligned} F_{\text{CO}} &= Y_{\text{O}} \cdot A_m \cdot 0.3 / M_{\text{O}_2} \cdot 2 \\ F_{\text{CO}_2} &= Y_{\text{O}} \cdot A_m \cdot 0.33 / M_{\text{O}_2} \\ F_{\text{H}_2\text{O}} &= Y_{\text{O}} \cdot A_m \cdot 0.37 / M_{\text{O}_2} \cdot 2 + Y_m / M_{\text{H}_2\text{O}} \\ F_{\text{H}_2\text{S}} &= Y_{\text{S}} \cdot A_m / M_{\text{S}} \cdot (1 + X_T) / 2 \\ F_{\text{CH}_4} &= Y_{\text{C}} \cdot A_m / M_{\text{C}} \cdot 0.08 \\ F_{\text{N}_2} &= Y_{\text{N}} \cdot A_m / M_{\text{N}_2} \cdot (1 + X_T) / 2 \\ F_{\text{H}_2} &= Y_{\text{H}} \cdot A_m / M_{\text{H}_2} \cdot 2 \cdot F_{\text{CH}_4} - F_{\text{H}_2\text{O}} - F_{\text{H}_2\text{S}} \quad (\text{maybe negative}) \end{aligned}$$

It is assumed that in the first step of devolatilization sulfur and nitrogen have devolatilized to the extent of 40%, and CH_4 is formed from getting H_2 in the gas phase. This may cause F_{H_2} to be negative. After devolatilization, the rest of the carbon will be treated as char for the beginning of char gasification and oxidation reactions. It is obvious that the char generated is a function of time in the dynamic case, and is also a function of the coal properties. As one knows coal properties change from time to time, and we will have to find a way to treat this at a later stage.

2.1.3 OXIDATION

Compared to coal gasification, combustion is much faster. If we follow O_2 into the bed, we would like to know how much O_2 reacts with char or with the volatile and how fast. The disappearance of oxygen is estimated here as in Field's work for carbon oxidation where the surface reaction rate and the diffusion rate are considered where K_s is a surface reaction rate constant and K_d a diffusion coefficient,

$$r = k \cdot P \cdot Y_{O_2} \quad \frac{1}{k} = \frac{1}{k_s} + \frac{1}{k_d} \quad (6)$$

$$K_s = 8710 \exp(-35000/RT) \quad (7)$$

$$K_d = 0.292 \frac{D_e}{d_p T} \quad D_e = \frac{4.26}{P} \left(\frac{T}{1000} \right)^{1.75} \quad (8)$$

$$\frac{dY_{O_2}}{dt} = - \left(\frac{K}{12} \right) \left(\frac{6}{d_p} \right) \cdot RT \cdot \frac{Y_{O_2}}{2} \quad (9)$$

For an oxygen conversion of 99.5%, at $d_p = 0.1$ cm under $P = 15$ atm and $T = 1200$ °K, the time required will be $t = 0.07$ s. If gas velocity $U_g = 2$ ft/s and the bed voidage $\varepsilon = 0.5$, the corresponding disappearance distance will be 0.3 ft, which is certainly negligible when compared with the bed height of 15 ft or more.

The O_2 reacts with the combustive gas and the solid. As the combustion is limited by the O_2 feed, and as the devolatilization process is assumed to be evenly distributed in the whole bed, the volatiles combust partially in the bottom of the bed. We assume 10% of the volatile to be combusted to H_2O , and CO_2 . The rest of O_2 reacts with char to form CO and CO_2 . The intrinsic ratio of $\beta = CO / CO_2 = 500 \exp(-12400/RT)$ has been given by Arthur (1951). It is noted that the particle size has an effect on this ratio, and we use a correlation by Wen et al (1978),

$$\begin{array}{ll} 1/f = (2Z+2)/(Z+2) & d_p < 0.005 \text{ cm} \\ 1/f = [(2Z+2) - Z \cdot (d_p - 0.005)/0.095] / (Z+2) & 0.005 < d_p < 0.1 \text{ cm} \\ 1/f = 1.0 & d_p > 0.1 \text{ cm} \end{array}$$

which is important in the dynamic case as the particle size or conversion changes. It is noted that this ratio has a big effect on the char gasification rate and so on the bed temperature. It is this big effect which induces it to be used as a model parameter for data fitting as in the work of Smith (1989) and Felder (1899). It can be shown that

variations in β can lead to maximum differences in T of 50 °K, and in X of 5% (Fig.30). The dynamic responses will be affected in the same way for both the gain and the response time constant parameters (Fig.36).

2.1.4 SORBENT CALCINATION ($\text{CaCO}_3 \rightarrow \text{CaO} + \text{CO}_2$)

This reaction is reversible. Its rate depends on the particle size, P_{CO_2} and T (Bhatia 1983), which has been studied by Beruto (1984) and by Borgward (1985). They showed first order behavior and the same activation energy E for different particle sizes. Combining their data by model fitting, we develop a model which covers particle sizes d_p from 1 μm to 1 mm:

$$r = \frac{dx}{dt} = \frac{10^5 x}{1.4d_p^2 - 0.0052d_p + 0.000012} \left(1 - \frac{P_{\text{CO}_2}}{EK}\right) \exp\left(-\frac{26600}{T}\right) \quad (10)$$

where d_p is in cm and $EK = 10^{(7.079 - 8388/T)}$ is an equilibrium constant. For 90% calcination, it spends 5 mins for $d_p = 0.05$ cm, and 10 s for $d_p = 0.01$ cm under $T = 1200^\circ\text{K}$ and $P_{\text{CO}_2} = 0$. Noticeably, it is a slow process when compared with gas $\tau_g = 10$ s. For this reason, the sorbent feed position has a negligible effect on the gas profile.

2.1.5 WATER-GAS SHIFT REACTION

This is one of the most important reactions which determines the gas composition distribution from coal gasification. It has been considered that the reaction is catalyzed by ash (Wen, 1979).

$$r = 2.88 \cdot 10^5 \cdot (Y_{\text{CO}} - Y_{\text{CO}}^*) \exp(-14000/T) \cdot P_f \cdot R_a \quad (11)$$

$$Y_{\text{CO}}^* = \frac{Y_{\text{CO}_2} \cdot Y_{\text{H}_2}}{Y_{\text{H}_2\text{O}} \cdot EK} \quad EK = e^{-3.968 + 7234/RT} \quad (12)$$

where R is the total reaction rate, and $f = 0.5$ is a correct factor. By calculations, it has

$$P_f = P^{0.5 - P/250} \quad \text{and} \quad R_a = \exp(-8.91 + 5553/T) \quad (13)$$

$$R = r \cdot W_s \cdot Y_{Fe_2O_3} \cdot f \quad (14)$$

been shown that the shift reaction is almost under equilibrium conditions, $(Y_{H_2} \cdot Y_{CO_2}) / (Y_{CO} \cdot Y_{H_2O}) = (0.99 - 1.01) \cdot K_E$. It is this close approach to equilibrium which causes a stiff problem in solving the ODE's.

2.1.6 METHANATION ($CH_4 + H_2O \rightarrow 3 \cdot H_2 + CO$)

Methane is mainly formed during coal devolatilization. After that it is formed by the hydrogen carbon reaction and consumed by the methane-steam reforming reaction. It was found by Blackwood and McGrory that the methane concentration can be larger than the equilibrium value in accordance with the experiments (1958). The total effect after devolatilization is for methane to get consumed. Based on Zahradnic and Grace's work (1974), the following simple expression was adopted:

$$\frac{dCH_4}{dt} = -K \cdot CH_4 \quad K = K_0 e^{-30000/RT} \quad (15)$$

The constant, k_0 is adjustable by fitting the methane yield to pilot plant data.

2.1.7 DESULFURIZATION AND LUMPING SIZE ($CaO + H_2S \rightarrow CaS + H_2O$)

By fitting IGT data to the shrinking core model, and assuming the dominant effect to be diffusion control, we get a kinetic model for the sulfur capture as:

$$r = \frac{dx}{dt} = \frac{1}{\tau \cdot f'(x)} \quad Y_{H_2S}^* = \frac{Y_{H_2O}}{Ek} \quad \log Ek = -0.268 + \frac{3519.2}{T} \quad (16)$$

$$t = \tau \cdot f(x) = \frac{(0.01d_p + 0.5d_p^2) \cdot \sqrt{T}}{P(Y_{H_2S} - Y_{H_2S}^*)} \cdot [1 - 3(1-x)^{2/3} + 2(1-x)] \quad (17)$$

where d_p in cm. The average reaction rate when the RTD is also linked will be:

$$\bar{r} = \int_0^{\tau} r \cdot E(t) \cdot dt = \int_0^1 E[f(x)] \cdot dx = \int_0^1 \frac{1}{\bar{t}} \exp\left(-\frac{\tau}{\bar{t}} f(x)\right) dx \quad (18)$$

which can be written as

$$\bar{r} \cdot \bar{t} = \int_0^1 \exp\left(-\frac{\tau}{\bar{t}} f(x)\right) dx = F\left(\frac{\tau}{\bar{t}}\right) \quad (19)$$

Plotting the above equation in a range of $0.001 < \tau/\bar{t} < 5$, we get (Fig.2)

$$\log_{10}(1 - \bar{r} \cdot \bar{t}) = \log_{10} a + b \log_{10}\left(\frac{\tau}{\bar{t}}\right) \quad a = -6.22 \quad b = -0.92 \quad (20)$$

If the system has a particle size distribution, the total reaction rate will be

$$R = \sum w_i \bar{r}_i = \frac{w \cdot \sum w_i a(\tau/\bar{t})^b}{\bar{t}} \quad (21)$$

The average reaction rate based on the average particle size can be written as

$$R = w \bar{r} = w \cdot \frac{1 - a\left(\frac{\tau}{\bar{t}}\right)^b}{t} \quad (22)$$

where $w = \sum w_j$. Letting $f_j = w_j/w$,

$$\tau^b = \sum f_j \tau_j^b \quad (23)$$

$\therefore \tau \propto d_p^n$, $n=1$, kinetic control, $n=2$, diffusion control. Therefore we get a lumping method to calculate the average

particle size from the size distribution for the sulfur capture as follows

$$\bar{d}_p = [\sum f_j d_{pj}^{nb}]^{1/nb} \quad (24)$$

which is significantly different from that used in the fluidization model.

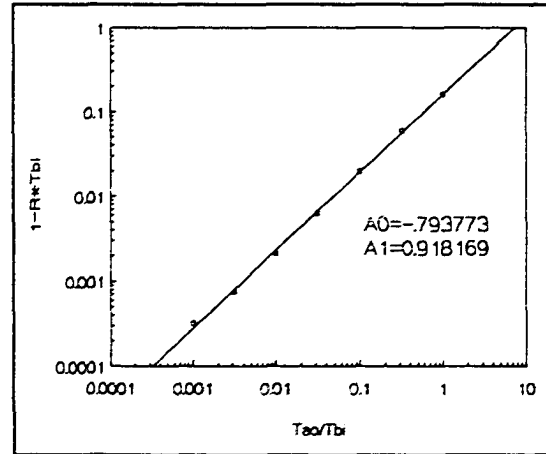


Figure 4 Sulfur Capture

If $b=1.0$, the average reaction time constant τ for all particles will be just the average of τ_i by weight fraction.

2.2 X-DISTRIBUTION (X-D) AND REACTIVITY

In the steady state case, we compute two conversions, one total carbon conversion, and another char conversion,

$$X_T = \frac{\text{carbon converted}}{\text{carbon feed}} \quad X_{\text{char}} = \frac{\text{char converted}}{\text{char generated}} \quad (25)$$

The char reactivity is a strong function of X_{char} . This gives rise to a problem for the dynamic model, where we have to know X_{char} in the bed. In the dynamic case, the definition of conversion cannot follow the steady state one due to changes in the coal feed and solid withdrawal rates with time. A way of obtaining dynamic conversion based on the information inside the bed has been developed. For example, in the model the total carbon conversion in the bed is determined by the use of ash content as a reference:

$$1-x = \frac{(\text{Carbon/Ash})_{\text{bed}}}{(\text{Carbon/Ash})_{\text{coal}}} \quad (26)$$

We still have a problem with the above definition, which is that the coal properties, such as the coal reactivity, the coal volatility content, change with time and so does the ratio of carbon/ash. The rate of char generated is also a time function. This means that the information in the bed is a summation of their histories. At any point, we are required to find the char conversion distribution, and its history. We record these histories with respect to time.

A more complex case is that of the solid RTD as a function of time or coal feed rate caused by changes in load and coal property, because the solid RTD in reactor is affected by the feed rate and the reaction rate.

As the oxidant feed is limited for partial oxidation, char must pass through the combustion zone several times to obtain a certain conversion. The question is how to treat the solid residence time in the combustion and the gasification zones when it goes from one place to another.

We have developed some ways to solve these problems, where the relative reaction rate is related to the average carbon conversion. It is independent of RTD for a 1st order reaction. The X-D is used to treat reactivity decay for different reactions on the same solid, where the X-D in the bed is fixed and unique for different reactions on the same solid if the average conversion is given. The rate constants only give indications of how fast to get a certain conversion distribution (X-D) or average conversion. On the base of this idea, we can derive an expression for the X-D in the following way. In general, the rate of a gas solid reaction can be expressed as

$$r = r_x \cdot r_g(P, T) \quad (27)$$

where r_x is a function of the solid conversion and r_g a function of gas composition and temperature. When the solid RTD or X-D ($E_x(x)$) is taken into account, the weighted reaction rate is

$$\bar{r} = \int_0^\infty r(t) \cdot E_t(t) \cdot dt = \int_0^1 r(x) \cdot E_x(x) \cdot dx \quad (28)$$

Putting $r \cdot dt = dx$ in above equation, we get the general formula:

$$\bar{r} = \int_0^1 E_t[t(x)] \cdot dx = \int_0^1 E_r(x) \cdot dx \quad (29)$$

The average reaction rate depends on the X-D, and can be obtained from probability theory as:

$$E_x(x) = E_t[t(x)] / r \quad (30)$$

The fraction of reactivity between x to $x+dx$ for any reaction becomes

$$E_x(x)dx = \frac{E_t[t(x)]}{r} dx = E_t(t) dt = E_\theta(\theta) d\theta \quad (31)$$

and for multiple reactions will be

$$E_x(x)dx = \sum E_j(x)dx = \frac{\sum E_j(\theta)d\theta}{\int \sum E_j(\theta)d\theta} = E_\theta(\theta)d\theta \quad (32)$$

where $E_\theta(\theta)$ is the dimensionless RTD of the solid in the gasifier. Now, the relative reaction rate can be obtained from one of the kinetic models. It is just by calculating the dimensionless $\tau\tau_g$ from that model with a combined τ for reaction time, which is in correspondence to the total char conversion by that reaction,

$$\bar{X} = \int_0^1 X E(X) dX = \int_0^1 X(t) \frac{E[t(X)]}{r_g r_x} dX = \int_0^1 X \cdot \frac{1}{\tau} e^{-\frac{1}{\tau_0} \frac{dx}{r_g r_x}} \frac{dx}{r_g r_x} \quad (33)$$

$$\therefore \quad \bar{X}_i = \bar{X}_j \quad \therefore \quad (\tau \cdot r_g)_i = (\tau \cdot r_g)_j \quad (34)$$

$$E(X) = \frac{E(t)}{r_g r_x} = \frac{\frac{1}{\tau} \exp\left(-\frac{1}{\tau_0} \frac{dx}{r_g r_x}\right)}{r_g r_x} \quad (35)$$

which means

$$E_i(X) = E_j(X) \quad \text{if} \quad \bar{X}_i = \bar{X}_j \quad (36)$$

The X-D is uniquely determined by the average carbon conversion, and is independent of the reactions, and so is independent of the gas phase in the kinetic models. It is only determined by the solid itself. The above master equation relates the average rate to the average char conversion. Now a new question is that since there are many paths (different $E(t)$ by feed rate change and/or flow model) to get the same average carbon conversion, does it have any effect on the average reaction rate? Let's now assume the solid RTD is given by:

$$E(t) = U(t-t_0) \cdot E(t-t_0) \quad (37)$$

where $U(t)$ is a step function. Then we have

$$\bar{X} = \int_0^{\infty} X(t) \cdot U(t-t_0) \cdot E(t-t_0) \cdot dt = \int_0^{\infty} X(t+t_0) \cdot E(t) \cdot dt \quad (38)$$

$$\bar{r} = \int_0^{\infty} r(t+t_0) \cdot E(t) \cdot dt \quad (39)$$

If we use average carbon conversion as a base, we can write

$$\bar{X}^* = \int_0^{\infty} X(t) \cdot E^*(t) \cdot dt \quad \bar{r}^* = \int_0^{\infty} r(t) \cdot E(t) \cdot dt \quad (40)$$

$$\bar{r}^* = \int_0^{\infty} r(t) \cdot E^*(t) \cdot dt \quad (41)$$

Now we have a condition:

$$\bar{X} = \bar{X}^* \quad (42)$$

It is obvious that, if r is a linear function of X , or $r = \text{const.}$, we will always have

$$\bar{r} = \bar{r}^* \quad (43)$$

The gasification kinetics shows almost linear behavior, its order being between (2/3 and 1.0). It can be approximately said that the average rate is determined by the average conversion alone, and is independent of its path or the RTD.

Any property change with time in a known flow model can be expressed as

$$\bar{M} = \int_0^{\infty} M_{\text{feed}}(t) \cdot E(t) \cdot dt \quad (44)$$

It changes with time and needs to be recorded as a history, and to be filtered. Here the CSTR model is adopted as a filter, which is

There are two reactivities. One is the relative reactivity which depends on the char conversion, as treated above, and another is the intrinsic reactivity which depends

$$\tau \cdot \frac{dy(t)}{dt} + y(t) = f(t) \quad (45)$$

on coal.

$$r = a_i r_g r_x \quad \bar{r} = a_i f(\bar{X}) \quad (46)$$

where a_i stands for intrinsic reactivity. For different coal in the bed, we can write

$$\bar{r} = \sum w_i^* a_i f(\bar{X}) = \bar{a} f(\bar{X}) \quad \tau \frac{d\bar{a}}{dt} = a_{in} - \bar{a} \quad (47)$$

where * is the state in the bed for $x=0$. As the coal properties change with time, the char generated and converted in the gasifier also change, as given by the expression,

$$(Carbon)_{bed} = (1 - \bar{X}_T) \cdot W_c^* - (1 - \bar{X}_{char}) \cdot W_{char}^* \quad (48)$$

$$\frac{1 - \bar{X}_{char}}{(1 - \bar{X}_T)} = \frac{W_c^*}{W_{char}^*} = \delta_c^*(t) \quad \tau \frac{d\delta_c^*}{dt} = \delta_{in}^* - \delta_c^* \quad (49)$$

where W_{char}^* is the char content in bed for $x=0$, and which is related to the history of the coal devolatilization. The total carbon conversion is defined as

$$1 - X_T = \frac{W_c}{W_c^*} \quad W_c^* = \left(\frac{Carbon}{Ash} \right)_{coal} (Ash)_{bed} = CTOA^* (Ash)_{bed} \quad (50)$$

As the CTOA is a time function, it can be expressed as

$$\frac{d}{dt}(CTOA^*) = (CTOA_{in} - CTOA^*)/\tau \quad (51)$$

So as to in the model, we record δ_c , CTOA, and a_i by the filter. By this way, we can follow the state variables in the bed to get the reaction rates for any case of changes.

For sulfur capture, the same master equation with the method of lumping sorbent size was developed too. By these equations, a large saving in CPU time was achieved.

2.3 FLOW MODEL

Any simulation will need making some reasonably simplifying assumptions. In addition to Dr. Avidan's and Dr. Weng's models (1991), we make an important assumption, which is the gas holdup in the dense phase is assumed to be a well mixed CSTR, and whose volume correspond to the average bed height.

2.3.1 FLUIDIZATION MODEL

The bed height in a fluidized bed can be calculated from an empirical formula which was proposed by Babu et al (1978) specially for coal and related materials. The gas holdup can be calculated from an assumption of ideal gas behavior in the gasifier under high temperature and moderate pressure (10-20 atm):

$$\frac{1}{\phi \cdot e_{mf}^3} - 8.81 \quad \frac{1 - e_{mf}}{\phi^2 \cdot e_{mf}^3} - 5.19 \quad e_{mf} - 0.5175 \quad (52)$$

$$\frac{L_f}{L_{mf}} - 1 + \frac{1.95 \cdot (U - U_{mf})^{0.738} \cdot D_p^{1.006} \cdot \rho_s^{0.376}}{U_{mf}^{0.937} \cdot \rho_g^{0.126}} \quad (53)$$

where the minimum fluidization gas velocity U_{mf} is calculated by

$$Re_{mf} = \sqrt{(25.25^2 + 0.0651 Ar)} - 25.25 \quad (54)$$

$$Re_{mf} = \frac{U_{mf} \cdot \rho_g \cdot d_p}{\mu} \quad Ar = \frac{d_p^3 \cdot \rho_g \cdot (\rho_s - \rho_g) g}{\mu^2} \quad (55)$$

So, the bed height is known. The gas holdup is calculated as

$$V_g = \frac{P}{R \cdot T} \cdot \frac{W_s}{\rho_s} \cdot \left(\frac{L_f / L_{mf}}{1 - e_{mf}} - 1 \right) \quad (56)$$

The average particle size used in equations is

$$\frac{1}{d_p} - \sum \frac{f_j}{d_{pj}} = d_p - d_{pn}(1-x)^{1/3} \quad (57)$$

Here we assume that the char diameter changes with the conversion, and the reaction behavior of coal is given by the shrinking core model. By this definition, d_p in the bed is a function of the carbon conversion.

2.3.2 PARTICLE SEGREGATION

Particle mixing is good and rapid in a bubbling fluidized bed. But a high degree of segregation may occur in a gasifier as it holds particles of coal, char, ash, and sorbent, all having different physical properties, such as particle size, and densities. They are the most powerful factors causing segregation, especially when the operating gas velocity is not far from U_{mf} in the dynamic case. A model is used to calculate the solid mixing state. This model was applied to the continuous operation case with different solid withdrawal positions. The result will help in the design of the gasifier.

Rowe's model for a flotsam rich system with weight fraction of flotsam >25% or more is used to predict the solid mixing/segregation profile for different particles in the gasifier. The model can be expressed by the following two ODE's:

$$\frac{dC_B}{dz}(w+k-2KC_B)+qH(C_w-C_B)=0 \quad \frac{dC_w}{dz}-\frac{qH}{w}(C_B-C_w)=0 \quad (58)$$

Nienow et al (1985) gave some methods to estimate parameters for above equations. The Runge-Kutta method can be used to solve the above equations with an integral condition and two boundary conditions at the top of the bed as following

$$f_i = Z^* + \int_0^{1-Z^*} \bar{C} d(Z-Z^*) \quad (59)$$

A good way to solve the above problem is to assume C_B at the top of bed to match the integral condition by an one dimensional optimization. This program has been

$$w \cdot C_w - w C_B + k \cdot C_B (1 - C_B) = w \cdot \frac{dC_w}{dZ} + qH(C_B - C_w) \quad (60)$$

merged in the steady state simulation and the dynamic simulations.

A pseudo-steady state method is used to calculate the continuous case where we assume that the solid mixing is relatively fast compared to the total particle concentration change, which means that the solid mixing is very fast and tends to attain an equilibrium profile for any time.

A simpler model (Nienow, 1980) can be used for this case to predict a mixing index M for the continuous operation case. Let X be the weight fraction of jetsam, in steady state. Then one has $X_{in} = X_{out}$, and $X_{out} = X_{top}$. The mixing index is defined as $M = X'/X = M(X)$, and can be calculated. So, one gets

$$\bar{X} M(\bar{X}) = X' - X_{out} \quad (61)$$

This implies M is determined only by X_{in} for a given system.

In general when $U/U_{mf} > 3$, the solid is nearly well mixed. So in the most of the cases we switched off this calculation in the dynamic simulation to save CPU time.

3 SIMULATION RESULTS

After having a predictive dynamic simulation model, we are able to evaluate the effect of feed rate changes of coal, air, steam, and sorbent, for different coal properties to see what happens during the dynamic transition, and to discuss the system stability and controllability aspects.

3.1 CALCULATION METHODS

The water-gas shift kinetics are extremely fast. This lead to stiff ODE's so we cannot directly use the rate model in the dynamic calculation. In the literature, one way

is to use an iteration method with all the ODE's together. Another way is to leave the shift reaction out of the bed as a second block, thus making unsynchronous. Here we use a perturbation method to generate an approximate solution. The idea is that the system is under equilibrium at time t_N , and during Δt , the reaction occurs and some reactants are consumed, which breaks the equilibrium. This upset is compensated to a new equilibrium state by the gas feed and by the shift reaction; and further since Δt is small, we may assume:

$$Ek = \frac{\Delta a \cdot \Delta c}{\Delta b \cdot \Delta d} \quad (62)$$

From $\Delta_i = F_i + \sum \alpha_i R_i - R_s$, the shift amount of R_s can then be calculated. We call this method as a shift inside method, as opposed to the shift outside method, which separates the shift reaction as a second block.

Fig.33 shows the results for different treatments of the shift reaction. It shows that the shift outside and the shift inside methods give the same results. The reason for this is that the step length is small, and is limited only by the gas τ_g .

Fig.32 gives effects of step length on convergency. The problem is stiff because of two small τ 's, one being the gas τ_g , and another the sulfur capture τ , where the driving force is very small in ppm, and tends to equilibrium, its effect on the reaction rate becomes big. As soon as the step length satisfies the above two τ 's, it is acceptable to the others. In general the step length was about $1/10 \tau_g$.

Adaptive iteration and annealing process methods are developed and applied in solving the steady state model equations.

3.2 DYNAMIC RESPONSES

Now we show some responses for selected state variables in dynamic simulation,

to understand the concept of transient behavior for such a gasifier (Fig.1-Fig.25).

3.2.1 SIMULATION OF FLOW PATTERN

First let us see the flow pattern in the fluidized bed and the solid mixing characteristics. Fig.27 is the simulation result for the data from the Foster Wheeler pilot plant, where the model gave good predictions.

Fig.28 is for the case of the continuous operation simulation with a solid withdrawal position at the bottom. Fig.29 is another case where the withdrawal position is at 30% of the bed height from the bottom. It is seen that there is a big difference between the two cases. The fluidized bed would have good mixing if the solid withdrawal were not at the bottom. Fig.26 is for the dynamic response of the solid concentration. Tab.1 gives the char profile in the bed for a pilot plant case simulation. It shows that the particles are almost well mixed in

Table 1 X_{Char} vs H

Top	W_i
100	.262
95	.262
90	.261
85	.260
80	.259
75	.258
70	.256
65	.255
60	.254
55	.252
50	.250
45	.248
40	.246
35	.243
30	.239
25	.235
20	.230
15	.222

the bed, so that the perfect mixing assumption is valid enough for our model. In most of the cases, the solid is well mixed by the gas feed, so that for the dynamic simulation we close the calculation of the solid mixing to save on CPU time.

An important conclusion from the above results, is to keep the system well mixed by increasing gas velocity or by reducing the particle size. If some constraints exist, it is better not to use the bottom as a withdrawal site, otherwise the efficiency of sulfur capture will go down to a great extent. Maybe one could design two withdrawal holes for the solid out flow in case solid segregation does occur.

3.2.2 DYNAMIC SIMULATION OF GASIFICATION

Typical dynamic responses for two cases have been plotted, one for a fixed $\beta = \text{CO}/\text{CO}_2 = 0.3$ (Fig.25), and another for a variable β , vs. temperature and particle size (Fig. 36), where the final values are given by the steady state simulation. It validated that the dynamic computation is exactly convergency to the steady state variables.

One can see that the gas composition change is quite small during the dynamic response, but the gas yield changes by a large amount as the gas throughput changes.

The responses can be divided into three sections. All the variables related to the gas, such as gas composition, gas flow rates, fuel gas HHV, and its heating value, have jumps in the beginning caused by the gas feed. After that, those pass through a two time-constant process. The whole response may have overshoot, inverse, or over damped behavior. The long term time constant has the same magnitude as the solid time constant process, which induces a drift in the system response for long times. In between there is a smaller time constant. We will show that it is the thermal time constant caused by heat replacement.

3.2.3 STEADY STATE SIMULATION OF GASIFICATION

The modified steady state model was then used to predict the Foster Wheeler's pilot plant data. The results (Tab.5-6, Fig.34-35) showed that the modified model did a very good job. In the process of comparisons, the experimental and operating data were analyzed. This gave some guidance as to analysis of mass balance. Note that in a simulation, no model parameters are adjustable. The heat loss by heat transfer is simulated by a built-in model.

3.3 CASE STUDY

A case study provides more information for different disturbance with different step change heights. The typical responses (Fig.25) mentioned above tell us that the most

important things are carbon conversion (Fig.1-6), temperature (Fig.7-12), fuel gas HHV (Fig.13-18), and its heating value (Fig.19-24). These state variables are listed for comparison and discussion.

The dynamic responses show there are three time constants involved. One is for long term effects, such as all the aspects of solid. These are caused by a long solid replacement action and the slow gasification rate. The second one is for immediate response related to the gas phase. This is caused by gas replacement, and has a small gas τ_g . The third one is a short term for the temperature transient, which represents the thermal- τ of the gasifier caused by heat holdup of solids. It is clearer if we use a log plot for the x-axis, where the τ 's can be identified. Later, we use a simplified model to evaluate these τ 's, which leads to a physical understanding of the system.

The rate of dynamic response at the beginning is determined by the τ_g , such as the fuel changes immediately. It is good for a load-following fuel control. This means the fluidized bed gasifier is the same as an entrained bed with regard to the fuel rate response which can be adjusted rapidly by gas feed. This is a key result for the use of fluidized bed gasifier in the IGCC system. The difference is that the fluidized bed has long time effect due to solid holdup, the solid feed response is, of course, a slow process.

Although the model is non-linear, it can be found from the step responses that the carbon conversion is nearly linear for different step disturbances and manipulated variables. In the most of cases, the gain of conversion showed a near linear behavior. Almost all of the responses of conversion are monotonic, except for the steam response. The reason is the steam increases the gasification rate at the beginning but a long term, it decreases the temperature. These are two opposite effects on the carbon reaction. It is also true that most of the temperature responses are nearly linear.

The HHV is affected by both the gas feed and the solid feed. The response of HHV to a gas feed change is linear, being almost symmetric. For the solid, it looks

like linear in gain but it was asymmetric. The same results as the HHV applied for the fuel rate. This symmetry is mainly determined by carbon conversion state. One might expect that it will be more symmetric at low conversion than that at high conversion.

Based on the manipulated variables, we could say that disturbances in gas feed and dolomite feed give linear and symmetric performance while the feed rate and the coal feed give a nearly linear but asymmetric performance.

It can be found that the fuel change is proportional to the feed rates of gas and solid over the entire response range. This is a good for the fuel demand control. Steam just has a big effect on the fuel response at the beginning.

The residence time effects were evaluated by the solid feed or gas feed rates step change as shown in Fig.1-24. A long residence time of solid led to a high conversion, low temperature, and high HHV, but a low fuel rate. A long residence time of gas, led to low conversion, low temperature, high HHV, and low fuel rate.

3.4 EFFECTS OF PARAMETERS

In our simulation studies, we introduced two important parameters. One is the combustion ratio $\beta = \text{CO}/\text{CO}_2$, which has a large effect on the temperature and the carbon conversion. It also affects the gas fuel value. Another is methane yield which is determined by kinetics both in the coal devolatilization and the char gasification. As the methane has a big heating value, it has effect on the product gas, the HHV and fuel value.

3.4.1 β EFFECT

β is a ratio of CO/CO_2 during the carbon combustion. As the carbon combustion is fast and carbon conversion is affected by this ratio, β is a very important parameter in coal gasification simulation. In the steady state the β effect is plotted at Fig.30 where the difference may be 50 °K for temperature, and 0.05 for the carbon conversion. Fig.36

shows the effect of β on the dynamic response. It is obvious that the β only give the effect on the steady state gain, especially on the short response term. As the β increases, the effect becomes small.

3.4.2 METHANE KINETICS EFFECT

The Fig.31 shows the effect of methane yield on the other variables. They show almost linear behavior to the methane fraction. The big effects are, as we expect, the temperature about 50°K, conversion about 0.05, and the HHV about 10 Btu/scf. It is clear that the methane is desired product due to the high HHV value, and high conversion.

For the above two reasons, we recommend that the model parameters should be estimated based on the plant data. In the present simulation, β is between 1.0 and 2.0 and is changed with temperature and particle size. The fraction of methane in the gas is fixed by kinetics, which is about 2.0% in fuel gas as reported in pilot plant for the fluidized bed.

3.4.3 GAS FLOW MODEL EFFECT

We only evaluate the gas composition and yield for different gas flow models. We assume devolatilization is evenly distributed in whole bed and the combustion region occupies 10% of the bed volume. Tab.2 is a comparison of gas yields on the base of the same operation conditions such as T, P, and solid composition ($X_T=0.9$).

Now we know that the difference is small, even at the X_T as high as to 0.9, since the char gasified is small as cited before. But the CSTR model is ODE's, and is easy to integrate. It is noted the sulfur capture has a sensible difference for two models. Considering with pilot plant data, we prefer the CSTR model to the plug flow model in safely estimating sulfur capture.

Fig.38 shows the gas yields vary with bed height. It is obvious that the system

is far from equilibrium of the gas solid reaction, so the well mixed gas model gives a good approximation.

Table 2 Effects of gas flow model

Model	CO	CO ₂	H ₂	H ₂ O	H ₂ S	CH ₄	N ₂	F _g
CSTR	3.85	1.67	3.51	2.68	0.0126	0.0276	10.86	22.61
Plug	4.53	1.51	3.88	2.29	0.0085	0.0085	10.85	23.06

3.4.4 OPERATION MAPS AND CONSTRAINTS

Two numerical methods have been developed for steady state simulation. One is a multi-variable adaptive iteration method; the other is an annealing process simulation method. After building a predictive simulation model, we are able to study the operating ranges. The manipulated variables for a gasifier are the feed rate of coal, air, steam and sorbent. These variables can be also expressed as ratios of O₂/C, H₂O/C, τ_F , and Ca/S (Tab.4).

From the operation maps of previous works (Weng,1991), we know that for the carbon conversion the major constraint is the air to carbon feed ratio (Tab.3), while steam to carbon ratio in the range of H₂O/C=0-1.2 gives small effect. The main purpose of steam is to keep temperature down, but this also decreases the HHV. At low conversion, and so low temperature, the steam feed is not needed, such as for the Foster Wheeler case. It is obvious, the model parameter $\beta = \text{CO}/\text{CO}_2$ has a bigger effect than that of steam (Fig.30). It is seen that a lower conversion in gasifier is better for plant efficiency when a char combustor is applied (Tab.3).

The Foster Wheeler pilot plant case has been simulated (Fig.34-35) using our's model. When compared with experimental data, the results show that a good prediction

has been made (Tab.5-6). The simulation results are then applied to experimental data analysis, and to a guide of operating strategy as well as design modifications.

When a gasifier is connected to a power plant, the pressure and the bed weight will be the time variables to control during load changes or coal properties changes. The next chapter is turn to whole coal process operation evaluation and plant control based on a load following turbine lead model.

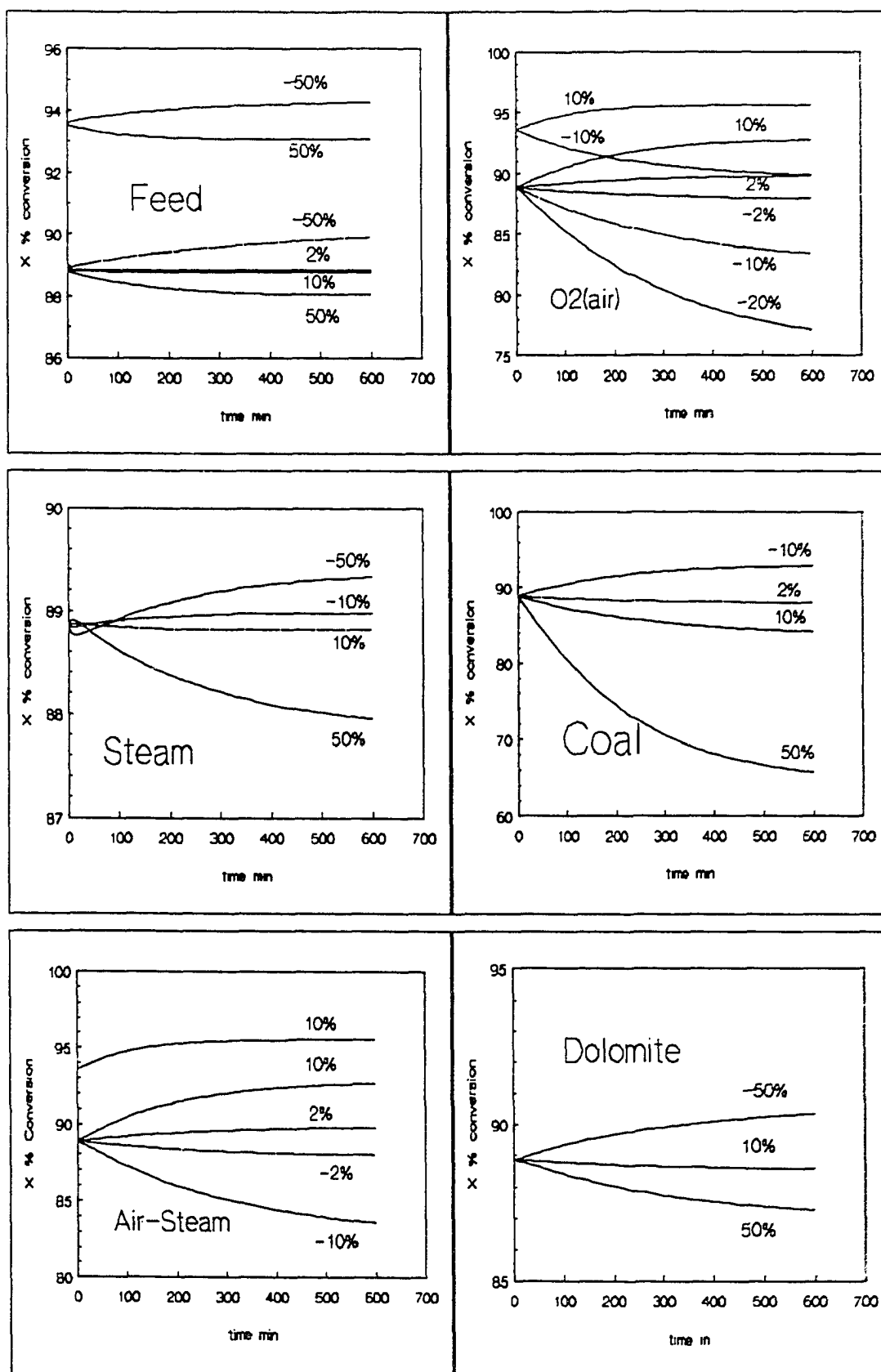


Figure 1 to 6 Feed effects on carbon conversion

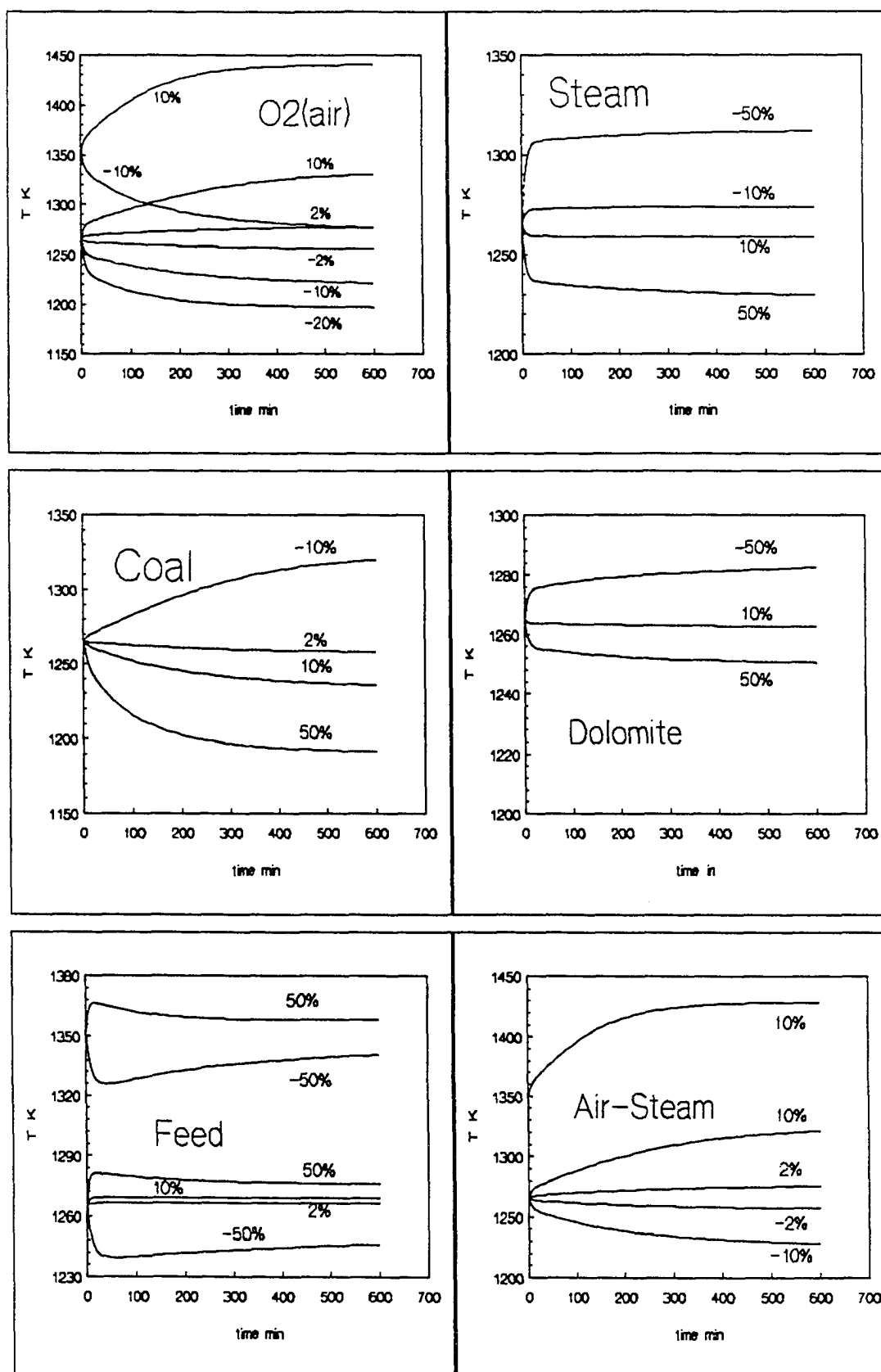


Figure 7 to 12 Feed effects on gasifier temperature

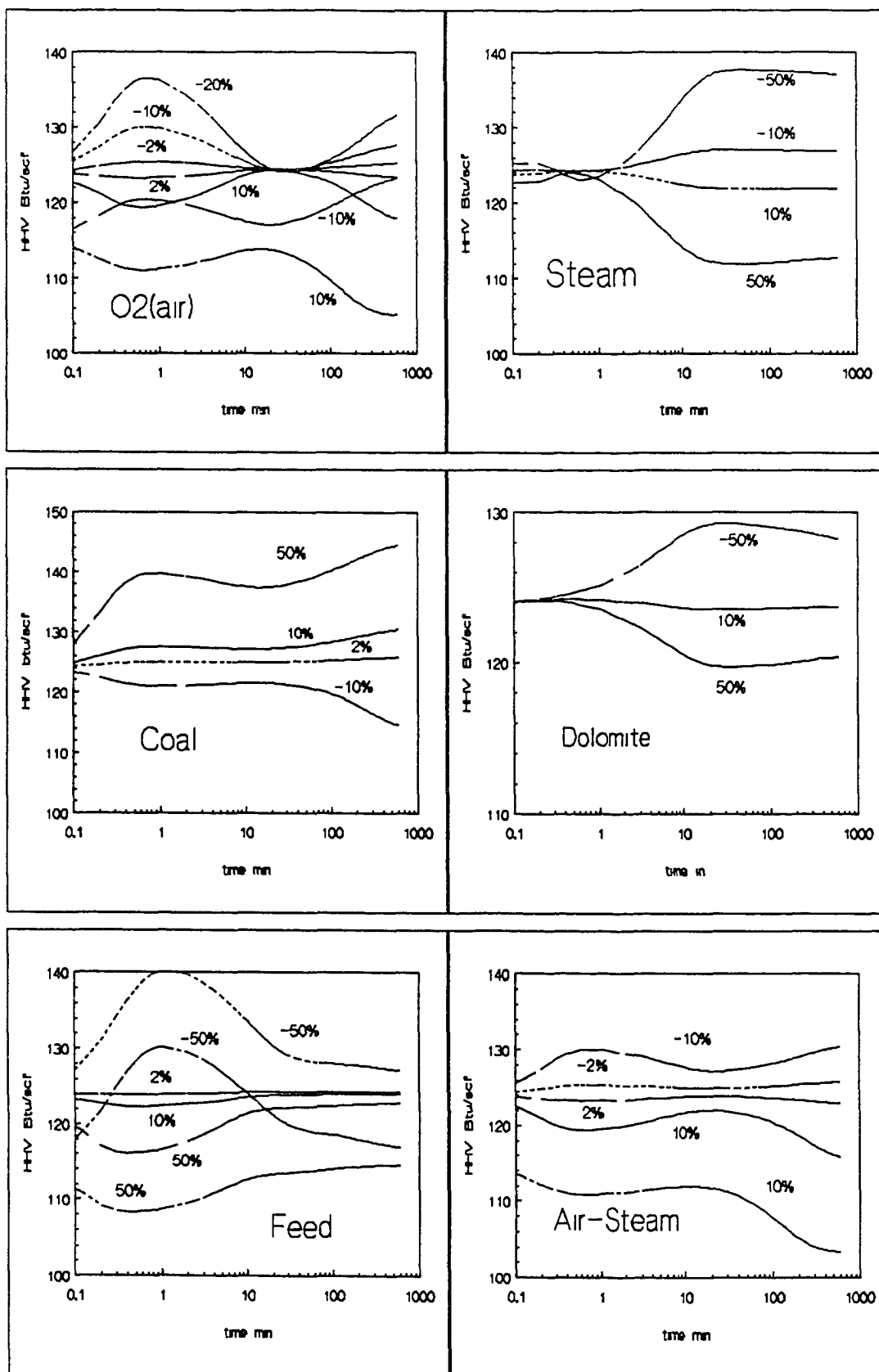


Figure 13 to 18 Feed effects on fuel gas HHV

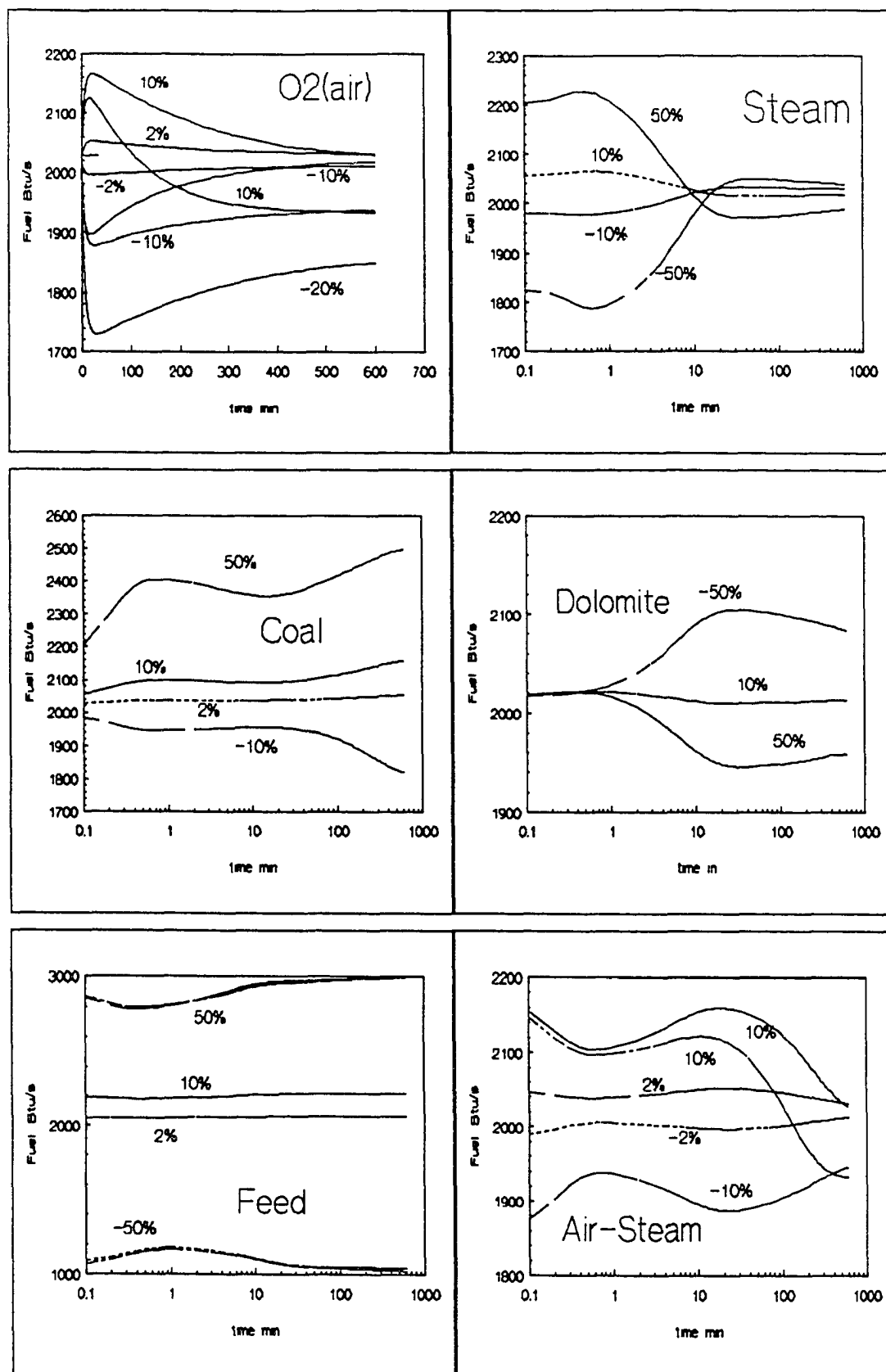


Figure 19 to 24 Feed effects on fuel yield

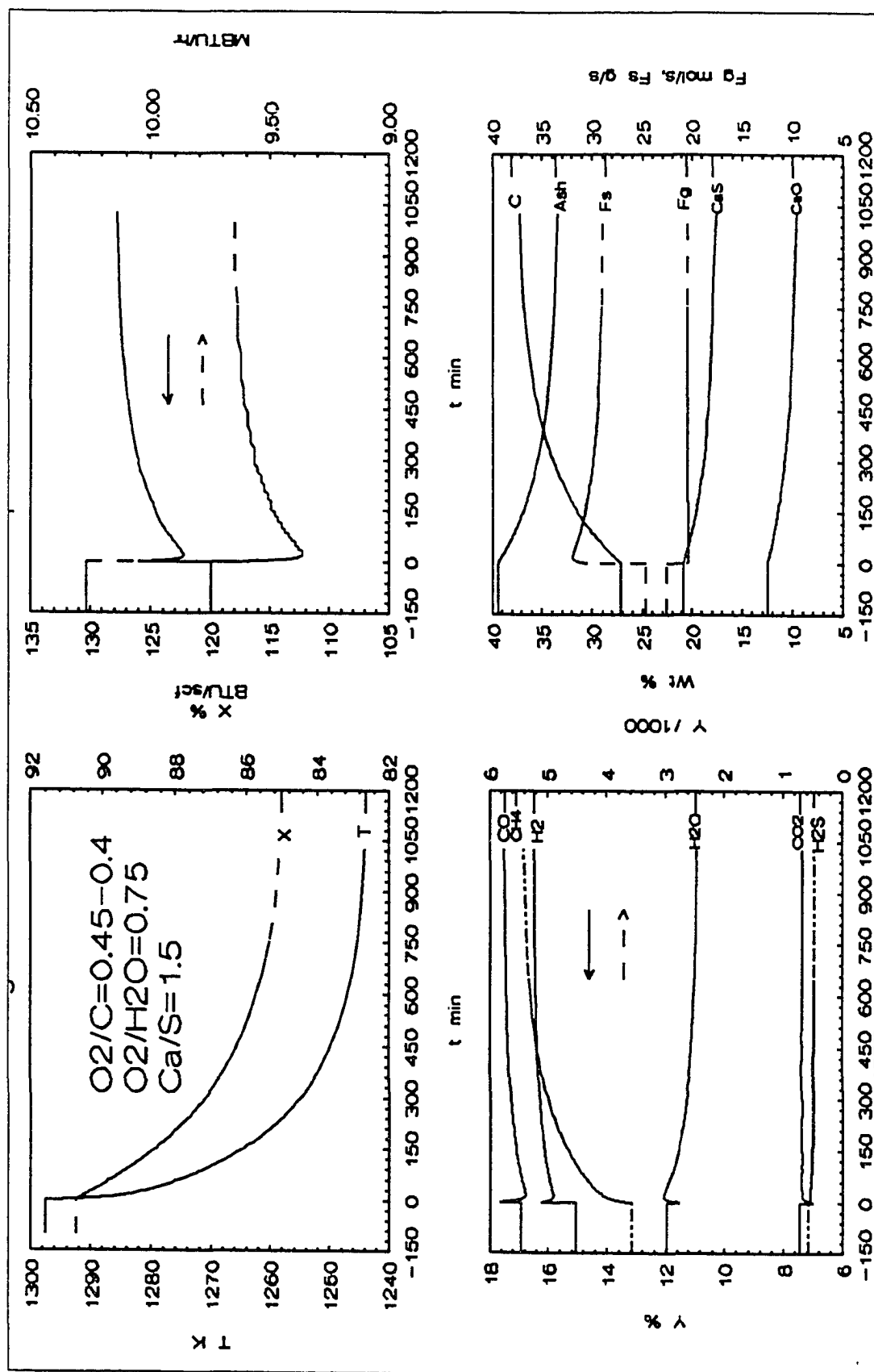


Figure 25 Dynamic Response of One Case ($\beta=0.3$)

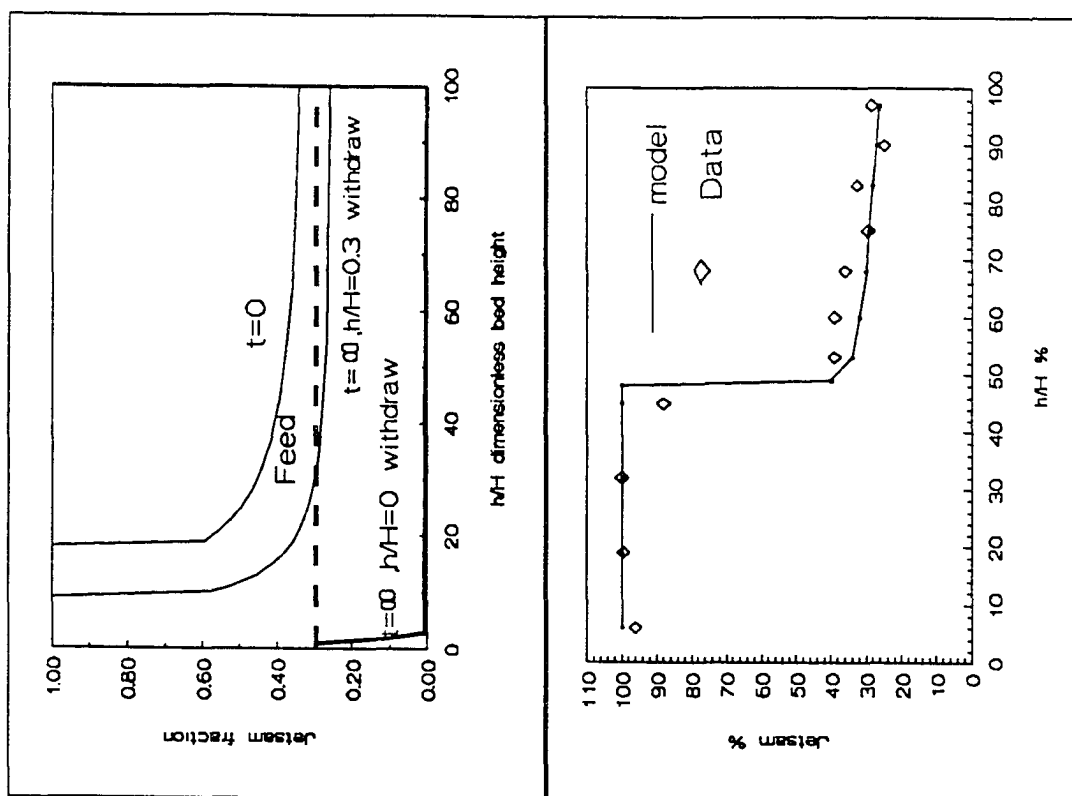


Figure 26 Solid profile vs. t Figure 27 FW plant data fit

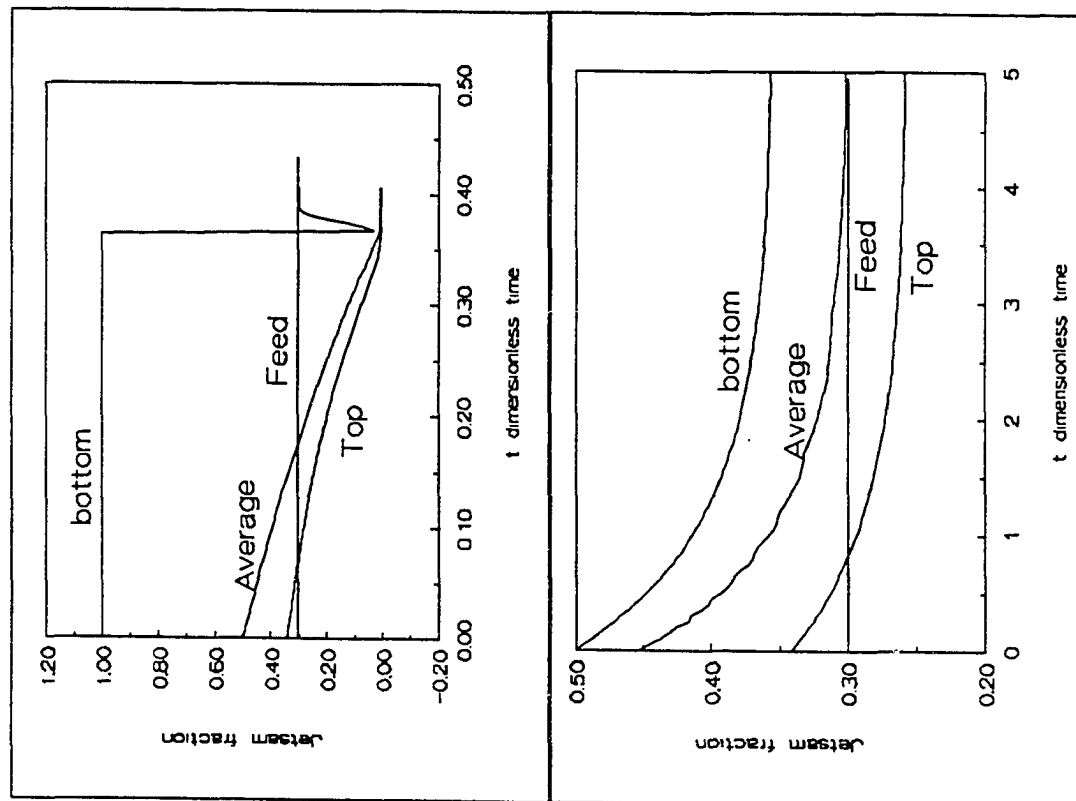


Figure 28 Withdraw from bottom Figure 29 Withdraw from $h/H=0.3$

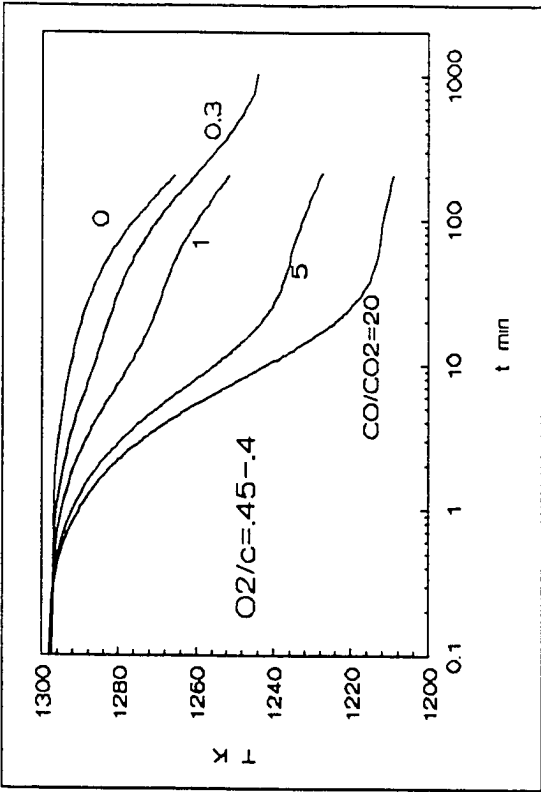


Figure 30 β effect on dynamic

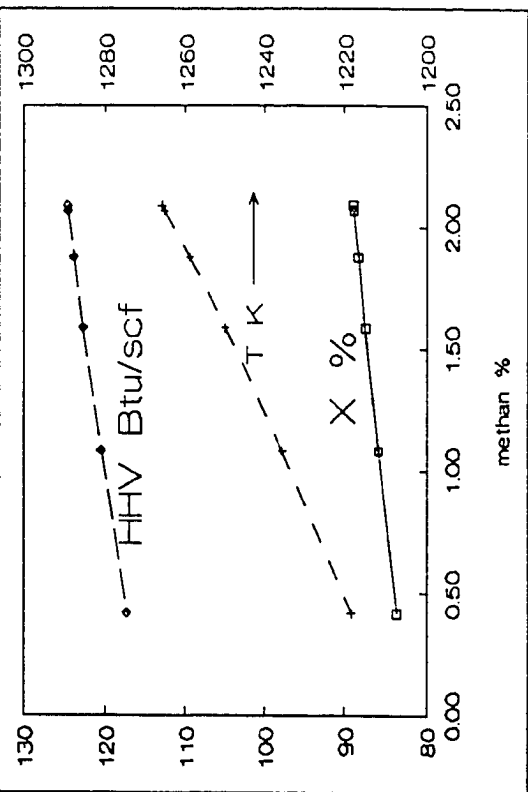


Figure 31 Methane effects

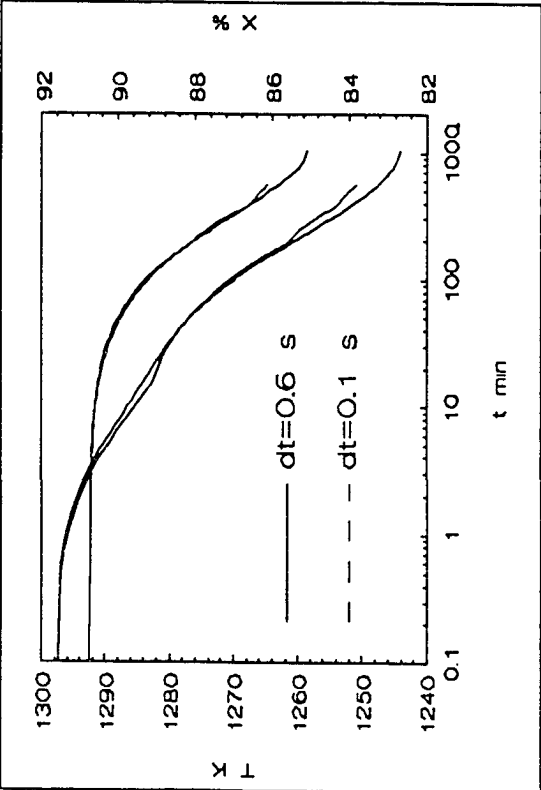


Figure 32 Step effect

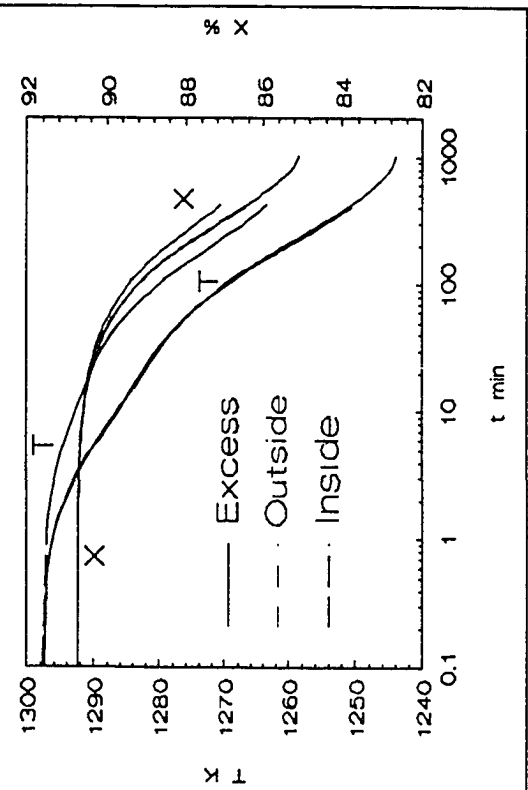


Figure 33 Shift Treating

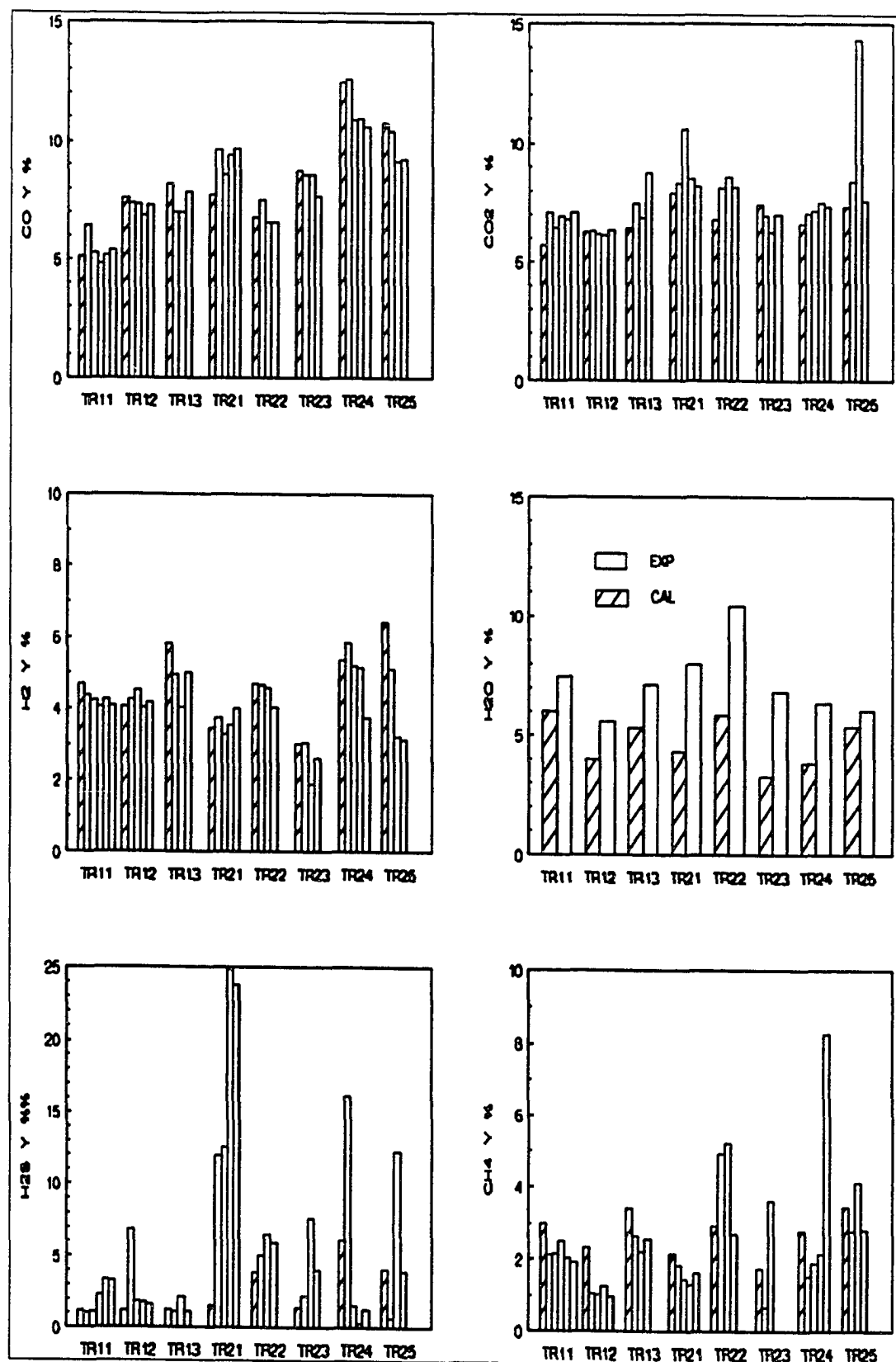


Figure 34 Foster wheeler data simulation by version 2

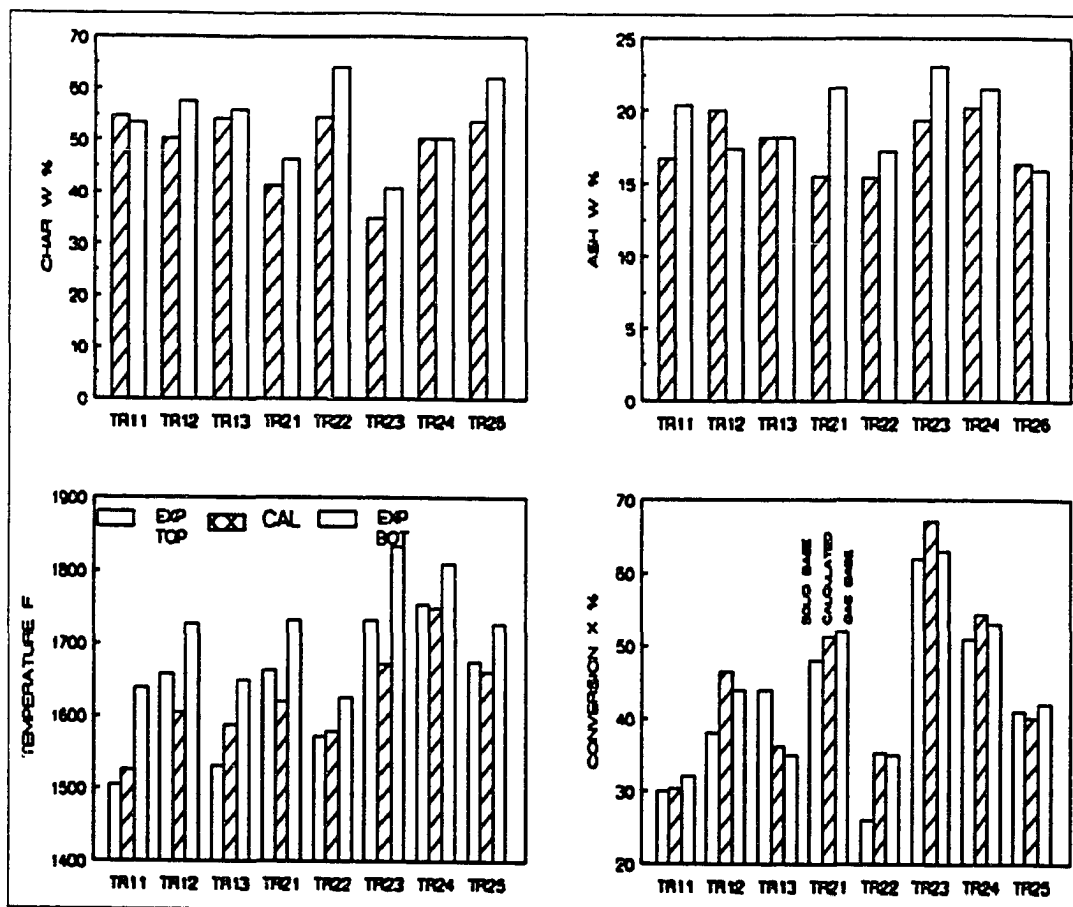


Figure 35 Foster Wheeler data simulation by version 2

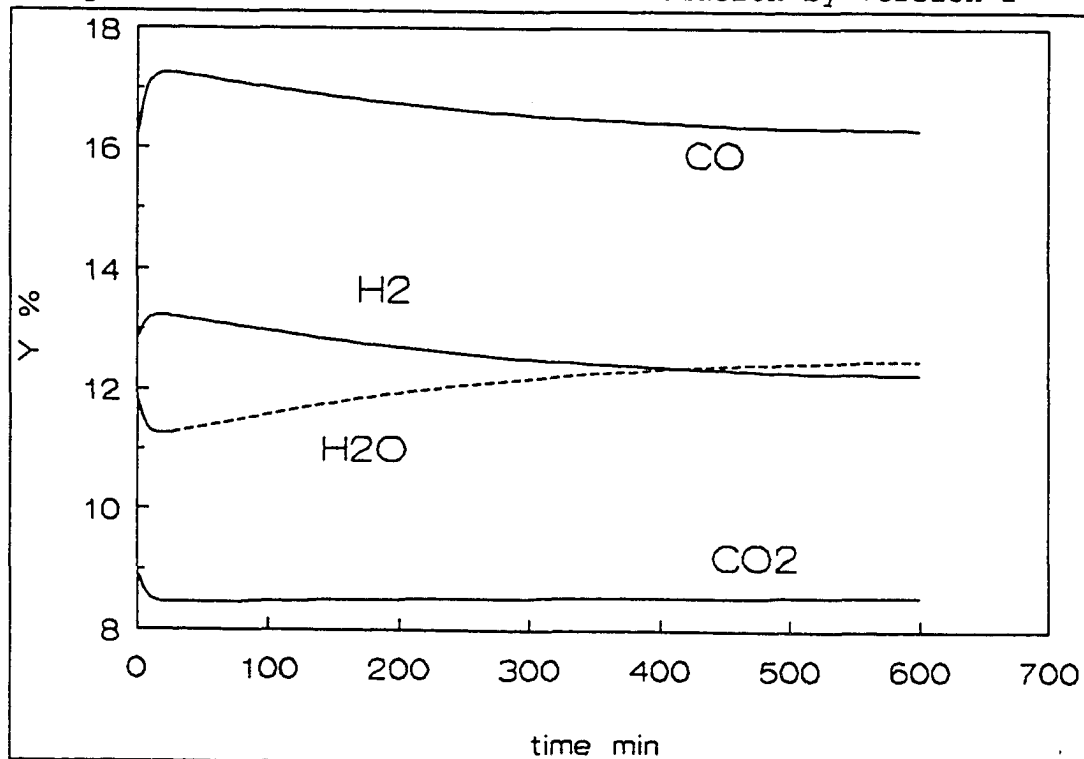


Figure 36 Typical response of Y (β variable)

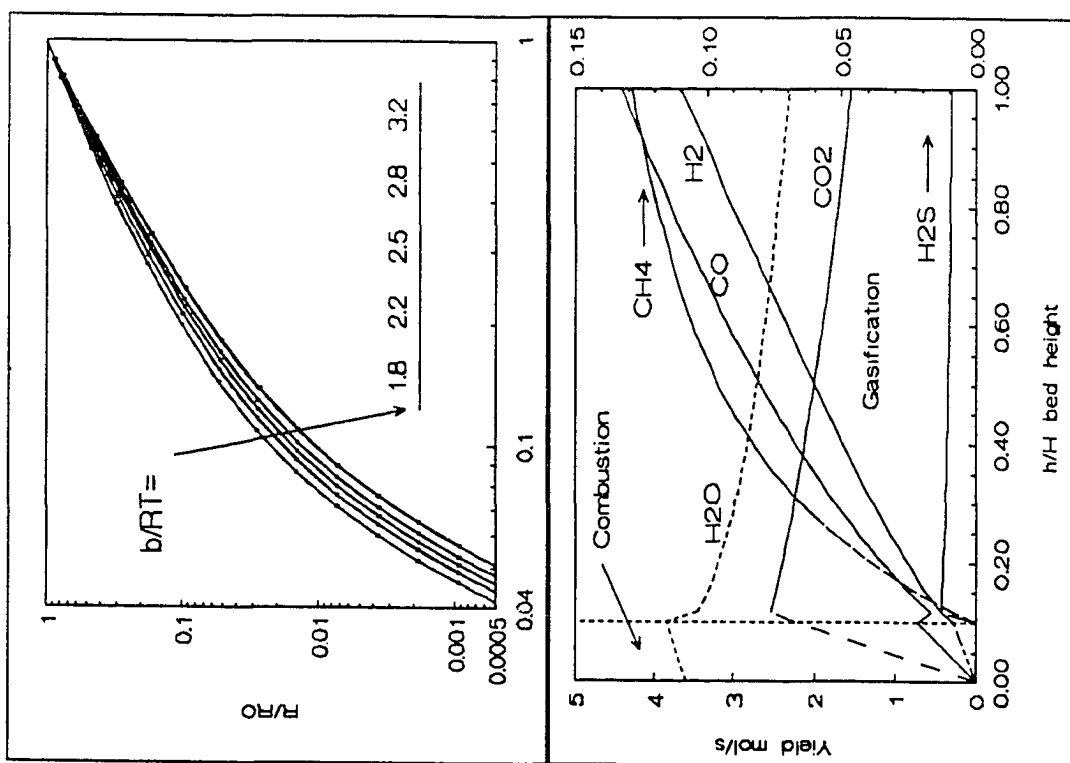


Figure 37 Reactivity vs 1-X **Figure 38** Gas Plug Flow

Table 3 Operation Effects on Fuel Gas and Plant Efficiency

GAS FEED EFFECT:					
O ₂ /C	H ₂ O/C	X %	T(K)	HHV	Eff.%
0.30	0.2	71.1	1222	152	46.00
	0.4	71.0	1194	141	46.86
	0.6	72.3	1192	135	45.20
0.35	0.2	81.9	1273	149	43.77
	0.4	81.7	1237	139	43.74
	0.6	80.9	1211	129	43.73
	0.8	80.9	1202	122	43.89
0.40	0.4	90.3	1313	134	43.24
	0.6	89.7	1279	125	43.21
	0.8	89.0	1250	117	43.19
	1.0	88.0	1225	110	43.19
0.45	0.6	94.1	1373	116	42.47
	0.8	93.9	1348	110	42.56
	1.2	92.8	1283	98	42.57
LOAD EFFECT ON ELECTRIC EFFICIENCY					
load			100%	90%	T(K)
co-shaft			43.15		1600
co-shaft fix T				38.69	1600
co-shaft fix air				42.65	1550
separated shaft			42.19		1600
separated shaft				42.16	1600

Table 4 IGCC plant simulation

##### IGCC POWER PLANT SIMULATION #####					
			GASIFIER	COMBUSTOR	TURBINE
OUT	GAS	MOL/S	21.810959	16.159882	65.788
	CO		0.140734		
	CO2		0.095792	0.040095	0.122
	H2		0.144022		
	H2O		0.157947	0.000000	0.113
	H2S		0.000660		
	CH4		0.019566		
	N2		0.441287	0.806758	0.725
	O2		0.000000	0.153081	0.066
OUT	SOLID	G/S	30.660538	27.233856	
	C		0.253802		
	ASH		0.316368	0.356174	
	CACO3		0.001750		
	CAO		0.101430	0.115296	
	MGCO3		0.001474		
	MGO		0.163330	0.184674	
	CAS		0.161846		
	CASO4		0.000000	0.343861	
BED	TEMPERATURE	K	1251.941160	1050.000000	1567.773
	PRESSURE	ATM	15.000000	13.000000	12.000
	CONVERSION		0.891739	1.000000	
	WEIGHT	KG	810.000000	360.000000	
	GAS-HOLD	MOL	121.896988		
	DIAMETER	CM	61.000000	42.000000	
	HEIGHT	M	4.464082		
	UMF	CM/S	10.722934	14.423788	
	U	CM/S	46.877228	81.481934	
	HEAT LOSS	CAL	5000.000000	2000.000000	
CHAR	FRA TOP-AVE-BOT		0.268447	0.241863	0.253
FEED	O2 IN AIR	MOL/S	2.393984	3.259277	6.270
	STEAM	MOL/S	4.787968		
	COAL	G/S	100.000000	DP (CM) =	0.080
	DOLOMITE	G/S	23.009674	DP (CM) =	0.050
SOME	RATIOS	IN MOLE BASE			
	N2/O2	4.000000	MG/CA IN DOLOMITE		1.000
	O2/C	0.400000	CO/CO2 IN COMBUST		1.389
	STEAM/C	0.800000	CHAR/FIX C		0.891
	CA/S	1.500000	VOLATILE COMBUST		0.100
SOME	EFFICIENCY			TEMPERATURE (K)	
	GAS TURBINE INNER	0.8700	AIR FEED		450.0
	STEAM TURBINE	0.2800	STEAM FEED		450.0
	HEAT EXCHANGER	0.9500	SOLID WITHDRAW		900.0
	PLANT CONSUM	0.0500	EXHAUSTED GAS		874.4
	AIR COMPRESSOR	0.8200	STACK GAS		410.0
	MECHANIC-ELECTRIC	0.9700	TURBINE STEAM		800.0
	PLANT EFFICIENCY	0.4287	HHV (BTU/SCF)		117.8
COAL	PROPERTY		C	0.826200	
	FIX CARBON	0.497000	H	0.057530	
	MOISTURE	0.033000	O	0.067740	
	VOLATILE	0.373000	N	0.016670	
	ASH	0.097000	S	0.032410	

Table 5 SUMMARY OF SIMULATION REPORT FOR FW PLANT

PROPERTY		C	H	O	N	S	ASH	MOIST	
CASE 11-13		67.495	4.578	6.302	1.854	3.174	14.425	2.173	
21-25		68.192	4.587	6.055	1.923	4.505	12.452	2.285	
CASE		11	12	13	21	22	23	24	25
P PSI	EXP	153	133	147	141	132	138	195	199
T F TOP	EXP	1505	1658	1531	1663	1572	1733	1755	1675
BOT	EXP	1640	1728	1650	1732	1625	1834	1810	1727
AVE	V1	1550	1675	1600	1683	1590	1763	1773	1675
	V2	1512	1597	1551	1650	1570	1692	1739	1646
HEIGHT FT		26.0	17.8	25.8	15.5	16.7	16.9	15.4	22.3
DIAMET FT		1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
COAL	LB/HR	329	235	354	234	315	182	376	508
AIR	LB/HR	494	523	568	609	546	615	884	881
	O2/C	.193	.287	.207	.332	.221	.431	.300	.221
	V1	.188	.320	.225	.338	.230	.481	.287	.227
PURGE	LB/HR	549	472	441	443	445	435	430	449
	O2/N2	.109	.122	.132	.137	.129	.139	.163	.160
	T F	800	800	800	800	800	800	800	800
SORB	LB/HR	89	67	98	118	94	83	99	160
	CA/S	1.55	1.63	1.59	2.04	1.20	1.84	1.06	1.27
H2O/C	EXP	0.	0.	0.	0.	0.	0.	0.	0.
FG	EXP	47.6	41.5	45.1	42.4	41.0	41.5	56.0	56.7
LBMOL/HR	V1	44.7	44.4	47.0	45.4	44.5	46.4	56.3	62.8
	V2	43.3	41.2	43.5	43.9	42.1	42.7	57.3	59.9
FS	EXP	280	166	234	192	243	111	240	319
LB/HR	V1	273	152	264	174	233	108	226	370
	V2	288	170	294	182	258	114	234	393
X	EXP	30-32	38-44	35-44	48-52	26-35	62-63	51-53	41-42
%	V1	32.7	52.7	37.9	53.5	38.5	70.9	55.5	41.5
	V2	29.4	46.0	33.4	53.3	34.8	68.5	53.8	39.0
HHV	EXP	75.6	66.6	86.4	77.2	111.4	73.2	110.7	99.6
BTU/SCF	V1	72.4	66.7	78.9	64.0	74.8	55.5	96.6	96.0
	V2	63.0	63.8	75.8	65.7	69.8	61.4	90.1	93.1
CGE	V1	28.0	35.5	30.0	34.6	29.7	39.2	40.5	33.2
%	V2	24.0	32.3	27.0	35.2	26.7	41.2	39.2	31.3

H2S	EXP	180	160	130	1060	510	420	500	580
ppm	V1	100	100	140	120	180	130	320	260
	V2	120	120	130	150	380	140	570	390

Table 6 **SUMMARY OF SOLID COMPOSITION**

		Unit = weight %							
CASE		11	12	13	21	22	23	24	25
C	EXP	53.4	57.7	55.7	46.2	64.0	40.6	50.4	62.2
	V1	58.5	53.0	60.2	45.5	60.3	35.7	53.8	58.4
	V2	54.6	50.3	54.2	40.9	54.3	34.2	50.6	53.8
ASH	EXP	20.4	17.4	18.1	21.6	17.2	23.1	21.5	15.9
	V1	17.4	22.4	19.4	16.8	16.8	21.0	20.7	17.1
	V2	16.5	20.0	17.4	16.0	15.2	19.8	20.0	16.1
CAS	EXP	5.34	6.22	4.81	6.10	4.14	9.56	9.33	5.03
	V1	3.37	6.24	4.18	9.41	8.11	13.5	11.5	8.44
	V2	3.15	5.11	3.62	9.05	7.05	12.6	10.6	7.85
CAO	EXP	0	0	0	0	0	0	0	0
	V1	0	9.50	8.80	14.0	6.32	13.6	4.77	7.00
	V2	5.70	7.73	6.52	12.6	4.56	12.8	4.90	5.90
CACO3	EXP	1.95	5.82	4.59	10.9	2.87	9.03	2.97	4.28
	V1	14.2	0	0	0	0	0	0	0
	V2	3.02	1.43	2.27	1.80	2.76	0.79	0.63	1.69
MGO	EXP	1.75	5.62	2.93	9.29	2.38	10.9	5.68	3.68
	V1	6.56	8.91	7.48	14.3	8.48	16.2	9.23	9.11
	V2	0	4.18	1.76	9.33	2.82	11.8	7.42	5.76
MGC03	EXP	12.9	2.56	9.15	2.83	4.64	3.60	6.71	5.12
	V1	0	0	0	0	0	0	0	0
	V2	12.8	7.50	10.1	7.24	9.14	5.34	1.95	4.76
Note:	EXP,	experiments							
	V1,	version 1							
	V2,	version 2							

CHAPTER 3. COAL GASIFICATION PROCESS SIMULATION

1 INTRODUCTION

The work discussed in this chapter is an investigation of the load following performance of an IGCC plant using a coal gasifier. The system studied is composed of a fluidized bed gasifier, a fluidized bed combustor, and a simplified combined-cycle turbine units.

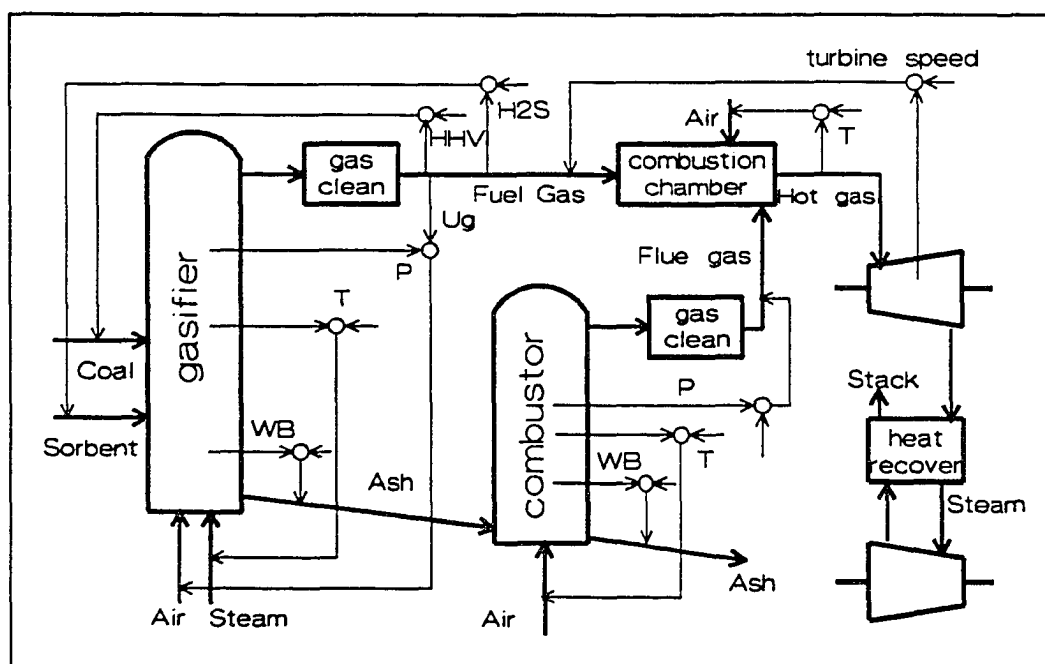


Figure 1 Scheme of an IGCC plant configuration

A typical system is shown in Fig.1. Granulated coal is fed to the gasifier along with dolomite and is injected with air and steam to produce a low BTU fuel gas. The sulfur in the coal forms H_2S which is removed from the gas by dolomite as CaS . The unconverted char is sent together with the dolomite to a fluidized bed combustor, where in addition to char combustion, the CaS formed in the gasifier is converted to $CaSO_4$. Fuel gas from the gasifier, and flue gas from the combustor are then fed to a gas turbine and combusted with compressed air to drive an electric generator. The use of the fluidized bed combustor depends on the carbon conversion in the gasifier.

The use of fluidized bed gasifiers results in more difficult control problems than those encountered in entrained bed gasifiers. This is due to their large holdups of both carbon and heat, which result in time lags and in slow and inverse responses. Startup and shutdown are also governed by these large capacitances.

In the IGCC system, the fuel is generated by coal gasification. One must know how fast the gasifier responds to changes in feed conditions. For the turbine control, we have to find a way to maintain the gas BTU content (LHV) above a critical value, and preferably keep it constant. For this we need to know what the response of the LHV of the gas will be to changes in the gasifier operation.

The linear gas velocity in the gasifier has to be maintained within relatively narrow limits (1.8-2.5 ft/s). Otherwise fluidization will be affected or there will be excessive entrainment. These are narrow constraints on control action. The main manipulated variables are the steam and air feeds. Since the LHV is strongly affected by changes in the ratio of steam to air. As the main variable which allows fast control of the bed temperature is the steam feed rate, the need to maintain LHV strongly limits our ability for effecting fast changes. We are therefore dealing with a control problem with strong constraints on the manipulated and the controlled variables. This is a challenging problem in multivariable control.

We developed dynamic models both for the coal gasification and for the control of the IGCC power plant. The models involve a set of ordinary differential equations for the gasifier, combustor, and cleanup, where the reaction kinetics combined with residence time distributions result in highly non linear equations.

The model description and development for the gasifier have been written in Chapter 2. In this chapter, the gasifier model is used as a fuel plant model to evaluate the plant open loop and closed loop (for a load following case, see Fig.1). It concentrates on the coal gasification plant's operation and control itself. The model embraces the

gasifier, the combustor, and the combined-cycle system. A quasi-steady state model is used for the combined cycle subsystem for the purpose of the IGCC process load following simulations. A linearized model developed for the gasifier will be discussed to understand the gasifier's operation. The next part deals with the simulation of the combined cycle subsystem.

2 MODEL DEVELOPMENT

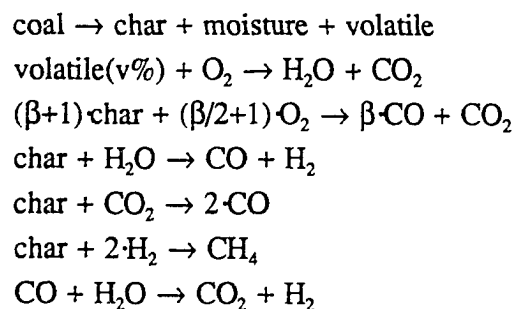
2.1 MODEL ASSUMPTIONS

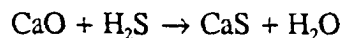
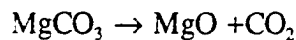
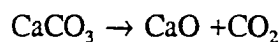
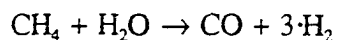
The dynamic models differ from the steady state models in that the feed conditions and operations are functions of time. For modeling, we need information on the kinetics, the flow patterns, and on the some state variables, such as bed temperature, pressure, carbon conversion, char reactivity, particle size, and solid composition. For the simulation and control of the bed weight and the gas velocity, a fluidization model is also needed. A particular problem is that the coal quality changes with time and it can not be measured rapidly on time. In addition, the coal reactivity is related to the coal feed properties and char conversion.

2.1.1 Gasifier

The main features of the model developed to meet the above requirements are listed as in the following:

(1) Reactions considered in the gasifier





Process reaction kinetics were developed, and time constants were calculated based on the data from KRW and FW pilot plants, as well as from the literatures. Some of the models are kinetics free if the time constants are very small compared to the gas time constant, τ_g . Char reactivity was also considered in the model. The char combusts with oxygen to yield CO and CO₂ at a molar ratio of β . It depends upon the temperature and the particle size where it is treated as a function of char conversion in the bed. The kinetics and carbon reactivity are then integrated with the flow model and the carbon conversion distribution.

(2) It is assumed that the solid mixing is fast compared to the solid residence time τ . So the flow model is set to the well mixed CSTR model for both the gas and the solid phases, similar to Aviden (1986) and Weng (1991). Also, using a CSTR model reduces the PDE's to ODE's, which simplifies subsequent simulation and linearization. The justifications are given in the above reference.

The gas in the lean phase is also assumed to be well mixed, where the lean phase volume is the difference between the gasifier and the dense phase volumes, and is therefore affected by the bed height.

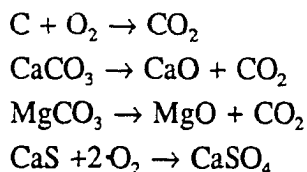
2.1.2 Fluidized Bed Combustor Model

It is difficult to achieve high conversion in fluidized bed gasifiers. In order to get a high conversion of carbon, and to oxidize CaS to CaSO₄, a pressurized fluidized bed combustor in tendency with the gasifier is employed.

The flow model is CSTR, the same as the gasifier for both the gas and the solid phases. In order to control the bed weight, the temperature, and the pressure at their set

points, a fluidization model for the combustor is implemented.

The reactions in the combustor are assumed as



Since at normal operation conditions, a large amount of excess air is used, the combustion rate is fast, and since the solid in the bed has a long residence time, all of the carbon and the CaS are assumed to be completely combusted, so that a kinetics free model can be used for the combustor.

The air feed rate is manipulated to limit the bed temperature. The bed weight is controlled by solid withdrawal, and the pressure is controlled by the output valve.

One advantage when a combustor is applied is that carbon conversion in the gasifier can be regulated. This regulation has two effect, one is on plant efficiency (chapter 2), and another one is on control. It was shown that in the gasifier stage, a carbon conversion of $X < 85\%$ is better than $X > 85\%$, for higher thermal efficiency. The fuel production behavior based on dynamic simulation of the gasifier showed that $X < 0.9$ is better than that $X > 0.9$. The reason for this is that the higher the conversion, the higher the gasifier temperature and so the deeper, more complete the combustion will be. As there is a temperature limit to downline equipment such as valve, hot gas filter, the high temperature gas should be cooled down to an acceptable temperature. This results in reducing plant efficiency.

The results show that in the gasifier stage, a carbon conversion of $X < 85\%$ is better than $X > 85\%$, either for fuel gas control or for plant efficiency.

2.1.3 Combined Cycle Module

The model for the combined cycle module for the IGCC is a steady state model,

which means that the time constants are ignored when we concentrated only on the coal process. As the space time of the gas passing through the turbine is very small, this assumption is reasonable when it is compared to the coal gasification process, where the gas time constant, τ_g is much large. The model is used to calculate power plant efficiency, and to evaluate control configurations for the fuel plant.

To simplify the model, the following assumptions have been made

- (1) Ideal gas behavior for a high temperature and low pressure case;
- (2) Pressure drops by mechanical friction are small and ignored;
- (3) Mechanical losses are small and ignored;
- (4) Turbine efficiency is constant.

As the space time of the gas passing through the turbine is very short, it is assumed that the output state is at equilibrium with hot gas inlet conditions, which means all the holdups are set to zero in considering the dynamic response for the gas turbine.

The steam turbine is simply modeled. Some extra assumptions are made as follows:

- (1) Constant back pressure;
- (2) Negligible heat losses.

The same assumption as for the gas turbine is made in that the outlet state is in equilibrium with the inlet state. The steam tables have been programmed as a part of the work, so that the steam properties can be calculated directly (User's manual).

2.2 MODEL EQUATIONS

The flow patterns for the gas and the solid can be expressed

by Fig.2. There are six vessels,

all of those are well mixed holder. The model equations are as follows.

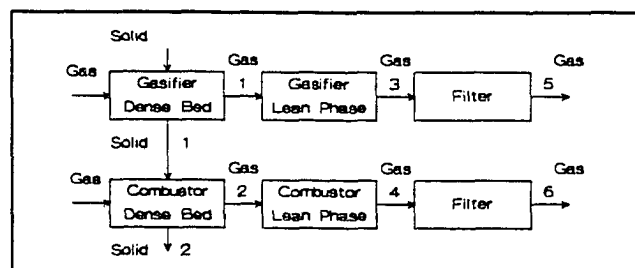


Figure 2 Models of gas & solid flow

2.2.1 Reaction Zone

Gasifier

$$\frac{dn_{il}}{dt} - \frac{dY_{il}}{dt} \left(\frac{V_1 P_1}{RT_1} \right) = F_{g0} \cdot Y_{i0} - F_{g1} \cdot Y_{i1} + R_{il} \quad (1)$$

$$V_1 = V_{Bed1} \cdot \epsilon_{f1} = S_{Bed1} \cdot H_{f1} \cdot \epsilon_{f1} \quad (2)$$

$$\frac{dW_{il}}{dt} - \frac{d(W_1 X_{il})}{dt} = F_{s0} \cdot X_{i0} - F_{s1} \cdot X_{i1} - R_{il} \quad (3)$$

$$\frac{dW_1}{dt} = F_{s0} - F_{s1} - \sum R_i \quad (4)$$

$$\frac{d}{dt} [T_1 (C_{vgl} \cdot V_1 + C_{psl} \cdot W_{B1})] = \sum Q_{1inlet} - \sum Q_{1out} + Q_r + Q_{devo} - Q_1 \quad (5)$$

$$Q_1 = hA\Delta T \quad (6)$$

where F_{g0} is the bed gas flow rate including devolatilization, combustion and the gas feed, and Q is the heat content. From the heating value of coal, and the devolatilization products, the heat of devolatilization is obtained based on HHV.

$$H_{devo} = Dulong_{HHV} - \sum H_{HHV} \quad (7)$$

Combustor

$$\frac{dn_{i2}}{dt} - \frac{dY_{i2}}{dt} \left(\frac{V_2 P_2}{RT_2} \right) = F_{air} \cdot Y_{i2} - F_{g2} \cdot Y_{i2} + R_{i2} \quad (8)$$

$$V_2 = V_{Bed2} \cdot \epsilon_{f2} = S_{Bed2} \cdot H_{f2} \cdot \epsilon_{f2} \quad (9)$$

$$\frac{dW_{i2}}{dt} - \frac{d(W_2 X_{i2})}{dt} = F_{s1} \cdot X_{i1} - F_{s2} \cdot X_{i2} - R_{i2} \quad (10)$$

$$\frac{dW_2}{dt} = F_{S1} - F_{S2} - \sum R_i \quad (11)$$

$$\frac{d}{dt}[(C_{vg2} \cdot V_2 + C_{ps2} \cdot W_{B2}) \cdot T_2] = \sum Q_{2inlet} - \sum Q_{2out} + Q_r - Q_2 \quad (12)$$

2.2.2 Lean Phase

Gasifier

$$\frac{dn_{13}}{dt} = \frac{dY_{13}}{dt} \left(\frac{P_3 \cdot V_3}{RT_3} \right) = F_{g1} \cdot Y_{11} - F_{g3} \cdot Y_{13} \quad (13)$$

$$F_{g3} \approx F_{g5} \quad (14)$$

$$\frac{d[V_3 \cdot C_{vg3} \cdot T_3]}{dt} = Q_{g1}(T_1) - Q_{g3}(T_3) - Q_3 \quad (15)$$

Combustor

$$\frac{dn_{14}}{dt} = \frac{dY_{14}}{dt} \left(\frac{P_4 \cdot V_4}{RT_4} \right) = F_{g2} \cdot Y_{12} - F_{g4} \cdot Y_{14} \quad (16)$$

$$F_{g4} \approx F_{g6} \quad (17)$$

$$\frac{d[V_4 \cdot C_{vg4} \cdot T_4]}{dt} = Q_{g2}(T_2) - Q_{g4}(T_4) - Q_4 \quad (18)$$

2.2.3 Cleanup

The gas cleanup equipments were treated as gas holder,

$$\frac{dn}{dt} = \frac{d}{dt} \left(\frac{P_1 V_1}{RT_1} + \frac{P_3 V_3}{RT_3} + \frac{P_5 V_5}{RT_5} \right) = F_{g0} - F_{g5} \quad (19)$$

2.2.4 Power Plant

Combustion chamber

$$Q_{out} = \sum Q_{jin} + Q_r \quad (20)$$

Gas turbine

$$W = Q_{in} - Q_{out} \quad (21)$$

$$T_{exha} = (1 - a + a(P_1/P_2)^{-\gamma}) \cdot T_{inlet} \quad \gamma = 1 - C_p/C_v \quad (22)$$

where $a=0.87$ is turbine inner efficiency depending on the scale.

Steam turbine

For the steam turbine system, a shaft power generated is assumed to be proportional to heat input to the system, that is

$$W = Q_{in} a_{steam} \quad (23)$$

where Q_{in} comes from heat exchanger. Here we assume that steam turbine system can be described by a total system efficiency a_{steam} and it is set as 0.28.

$$(Power)_{out} = W \cdot a_m \cdot a_E$$

where $a_m=0.97$ and $a_E=0.97$ are the mechanical and the electrical efficiencies. For the air compressor, we assume the efficiency as 0.82.

2.3 Model Solutions

Some of numerical methods used to solve the simulation equations are given below.

For the steady state calculation, which was used to initialize the dynamic simulations, an adaptive iteration method was developed. It considers the precision of different loops in a cascaded and synchronous way. The fourth-order Runge-Kutta

method was used for solving the differential equations. The step length was selected to achieve numerical stability and acceptable accuracy. A quasi-steady state method was adopted for some components in the dynamic simulations.

For a frequency domain analysis, a linear model has also been developed for the coal gasifier.

3. FUEL PLANT CONTROL

A control scheme has been developed to satisfy load following requirements as well as to maintain steady state operations. Fig.1 is for the whole fuel plant, and Fig.3 concentrates on the gasifier.

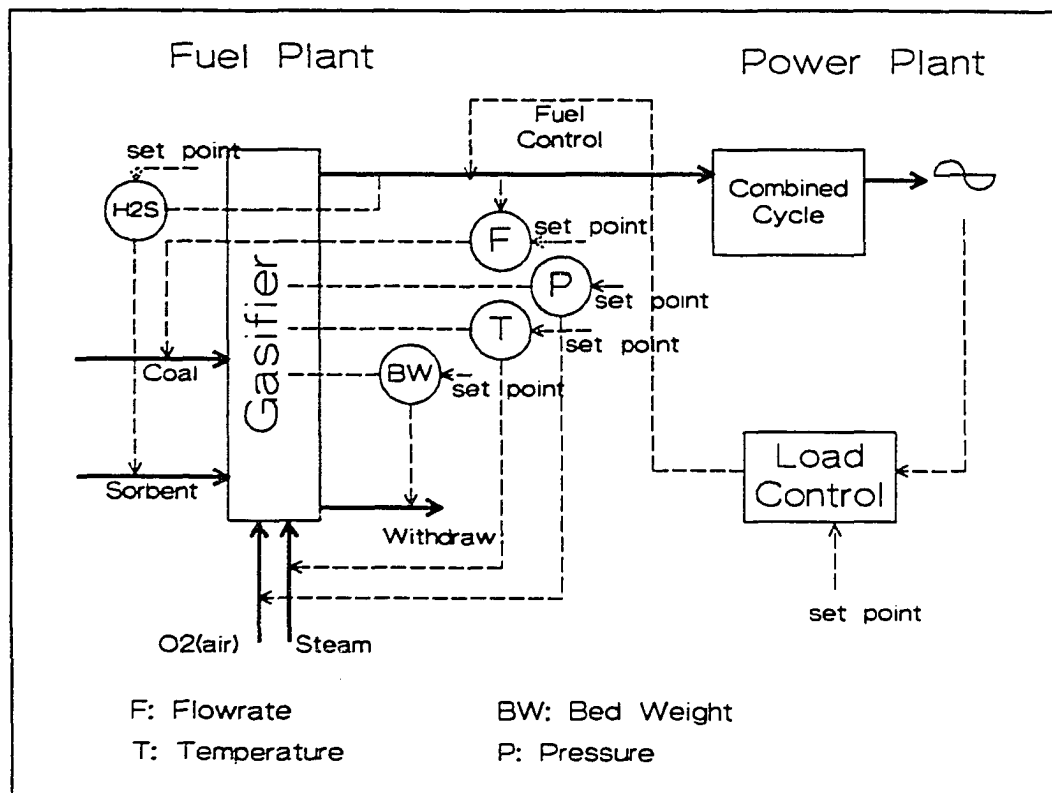


Figure 3 Gasifier control for a load following operation

3.1 CONTROL CONFIGURATION

The following control configurations (Fig.1 & Fig.3) have been selected on the

basis of the dynamic behavior of the gasifier as discussed in Chapter 2.

Let us first describe the control demands. The fuel gas flow rate will increase with the load as a forcing function. This discharge causes a pressure drop in the fuel plant. Gas feed rate, air or steam, is then manipulated to maintain the pressure at a set point. When this gas feed rate, or coal property changes, the ratio of air to carbon is affected, which causes the fuel gas LHV to change. This results in a control action to manipulate the coal feed rate. The temperature will follow these variations. All these changes in the solid feed rate, or in the reaction rate, cause a bed weight change, which is designed to be controlled by the solid withdrawal rate. Next in this sequence, the combustion duty to the combustor is affected, where the temperature is controlled at a setpoint by adjusting the excess air feed. This then effects the flue gas generation which effects the fuel demand for the gas turbine, and hence, in turn, the gasifier control.

It must be pointed out, based on simulated results of the gasifier that changing the gas feed rate does not guarantee a fuel rate change in the same direction as negative gain, inverse phenomena, as well as overshoots occur in some operating regions. For example, at high carbon conversions, as increase in the air feed will cause a decrease in the fuel production and the fuel LHV as more combustion occurs, which is different from that of the low conversion case. This means that the sign for the gain of the fuel product rate changes with an increase in the carbon conversion. It has been shown (chapter 2) that the fuel gas LHV decreases when the carbon conversion increases, but the gas rate increases as conversion increases. These two opposing effects cause the gain to change its sign. As a result, the gasifier should be carefully designed and the fuel quality LHV should be controlled.

The control configuration for the gasifier was shown in Fig.3, where it involved controls for the gas flow rate, bed weight, fuel quality, gas supply rate, pressures, temperatures, and the gas quality (such as H_2S content). Among these, the most

important ones are the fuel gas rate and quality controls.

The fuel gas rate is controlled through a valve to meet the forced demand from the power plant. As the fuel is generated by coal gasification, a coal feed control is needed for the load or the coal property changes. It is therefore needed to know how fast the control action should be set, and what the response performance will be. In general coal quality can not be measured fast on line, a feedback or other control schemes is needed to handle the solid coal feed rate. At this control is connected to the fuel gas quality, LHV through the coal devolatilization in an immediate way, the reset time is very important here.

If fuel gas LHV can be measured rapidly on line, there will be no problem in controlling coal feed rate directly by its measurement. If it can not be done, an inferential control scheme (Fig.3, Fig.4) is needed, where the coal feed rate

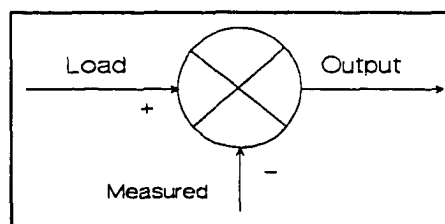


Figure 4 LHV control

is manipulated by controlling the fuel gas mass flow rate. Its setpoint is determined from the load and the fuel gas LHV value. The control action is so set that it is positive for a load change and negative for others.

The sorbent feed rate is manipulated by measuring the sulfur content of the exit gas. As the sorbent calcination reaction is not fast compared to coal devolatilization where nearly half of the sulfur releases, desulfurization is a slow process. Thus makes it hard to measure and control H_2S rapidly.

Temperature response in the gasifier is affected by heat replacement in minutes. It is designed to be controlled by adjusting the steam feed rate to the gasifier. The reset time may be in minutes. But, as the air feed rate to gasifier is manipulated by pressure control, the pressure control should be fast when the fuel plant is connected to the gas turbine in the power plant. This fast control induces a variation in the air/carbon ratio,

which causes temperature changes in a fast way, so that the reset time for steam feed should be as fast as that for the air feed. There is a strong interaction between them.

The gasifier design pressure set point should consider the linear gas velocity in the bed to suffer a large turndown. In case this happens, the pressure set point can be varied to maintain a certain gas linear velocity and a good fluidization operation. The pressure set point is of cascade control, and is included in the model (Fig.1).

The bed weights and hence the bed heights are controlled by the solid withdrawal rate both for the gasifier and for the combustor. This control requirement is easy to fulfill as we have a long solid residence times. The reset time for this loop should be large to avoid fast changes in solid flow rates, and to reduce the interaction between the gasifier and the combustor.

The temperature of the combustion chamber is controlled by the air feed, and this is done by adjusting the compressor inlet guide vane. It can only be adjusted in a range from 80% to 100%. When the partial load is less than 80%, the firing temperature will go down.

It is assumed that the temperature control in the fluidized bed combustor is achieved by adjusting the air throughput. In some of the designs, the compressor is fixed and the air flow rate can not be varied. When such a case is met, a waste heat boiler should be installed to take the heat out, and to maintain the temperature setpoint.

Of course, many more control strategies can be made up, and the configuration shown in Fig.1 or Fig.3 is just an example.

3.2 CONTROL ANALYSIS

Before we set the controller parameters, let us discuss controller design of the gasifier based on a linear analysis of the gasifier. There two methods were used, the state space method, and a simple linearized model.

3.2.1 State Space Analysis

The state space method as used here is limited only to the gasifier. As one can see, the gasifier is a MIMO system (Fig.5).

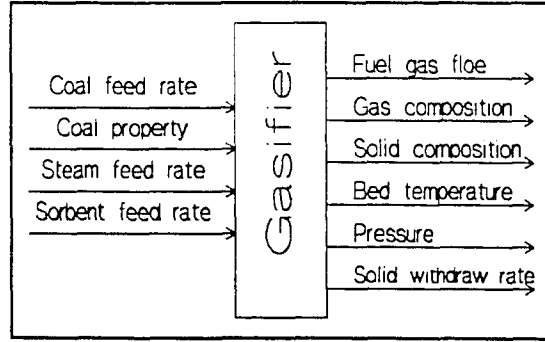


Figure 5 Gasifier MIMO

The relative gain array (RGA) is applied for such a MIMO system to evaluate its controllability. These gains can be obtained either from steady state simulations or from the final value of the dynamic responses.

There are four feed rates to be used as manipulated variables, those of air, steam, coal, and sorbet. The output variables are more than four. We selected carbon conversion, bed temperature, fuel gas HHV (or LHV) and fuel gas product rate as the control variables. Here we concentrate on the effect of the sorbet on the fuel gas response owing to the large effect of the sorbet feed on the mass and heat balances. As an example, we fix the step change of the input to +10% at a conversion of $X=0.89$ for the case of $O_2/C=0.4$ (in mole). The gain matrix becomes

$$\begin{bmatrix} \Delta X \\ \Delta T \\ \Delta HHV \\ \Delta F \end{bmatrix} = \begin{bmatrix} -0.53 & 0.44 & -0.004 & -0.031 \\ -0.23 & 0.514 & -0.052 & -0.019 \\ 0.55 & -0.47 & -0.144 & 0.0024 \\ 0.60 & -0.025 & -0.098 & -0.114 \end{bmatrix} \begin{bmatrix} \Delta F_{Coal} \\ \Delta F_{Air} \\ \Delta F_{Steam} \\ \Delta F_{Sorbent} \end{bmatrix} \quad (24)$$

All of the gains are in % relative change. These gains showed the relative magnitude of the effect of input variable changes on output variables. Of course, different operation conditions will show different gain matrix, even and gain with different signs.

The RGA for the above gain matrix becomes

$$RGA = \begin{bmatrix} 1.63 & -1.01 & 0.02 & 0.36 \\ -0.43 & 1.52 & 0.12 & -0.21 \\ -0.51 & 0.50 & 1.00 & 0.00 \\ 0.30 & -0.00 & -0.14 & 0.85 \end{bmatrix} \quad (25)$$

It shows the diagonal elements to be dominant, which suggests a way of coupling the control loops for steady state control.

It can be seen from the gain matrix and the RGA that carbon conversion, fuel gas LHV, and bed temperature are mainly determined by the air/coal feed ratio. They also show that the fuel gas product rate is strongly affected by the coal feed and the sorbet feed rates, and weakly affected by the air and steam feed rates. As the sorbet is used for desulfurization, the fuel gas rate has to be controlled by the coal feed rate. This is true for a given carbon conversion. In spite of the dominant effects of diagonal elements, almost all four of the output variables are affected to some extent by the coal feed rate, which can be seen from β_{j1} . The air feed rate has the same tendency β_{j2} as the coal feed except for its effect on the fuel rate.

The steam feed rate effects the fuel gas temperature, LHV, and the fuel gas product rate. As the LHV has the same tendency as the fuel gas product rate (chapter 2), and the fuel is mainly controlled by the coal feed rate, the steam is designed to be used as a coolant. For a high conversion case, the steam acts mainly as a diluent for the fuel gas HHV ($\beta_{33}=1.0$) and as a coolant for the bed temperature.

As demonstrated by the RGA, there are interactions to some degree between the four feeds, and even more strong interactions in the dynamic case (Chapter 2). For the sake of fast control, the initial rate of response is much more important than the steady state gain because of overshoots with large initial gain observed in the dynamic responses as shown in Chapter 2. However, the present control structure design involving multi-single loop simple feedback PID controllers, with no dynamic decoupling was found to

give a satisfactory enough performance.

3.2.2 Linearized Model Analysis

The purpose of developing simplified linear model was to simulate the gasifier in the frequency domain, and use the results in designing the controllers by the classical frequency domain transfer function analysis.

The model development details are given in the appendix. The outlines are as following:

In the solid phase, the most important reaction is carbon conversion by devolatilization, gasification, and combustion. In a linear model, we only concentrate on the carbon reaction. As one finds in the dynamic response studies, the gas residence time is very short, about 6-10 seconds. Based on this fact, we assume gas compositions to be in equilibrium with the reacting solid. This means that the gas residence time or gas holdups can be neglected, and so we cancel all the ODE's for the gas phase.

The bed weight and the operating pressure are assumed as constants for the gasifier. $F_{gin}=F_g$ is assumed for an air blown fluidized bed.

Then the model can be written as

$$W \frac{dx}{dt} = F_c - F_s x - R_c \quad (26)$$

It can be written for the heating value as

$$\frac{d}{dt}(W \cdot x \cdot H_c + W \cdot C_p \cdot T) = F_{coal} \cdot Du + F_s \cdot H_s - F_s x \cdot H_c - F_g \cdot H_g - F_g \cdot C_p \cdot (T - T_l) - r_s \cdot H_s \quad (27)$$

$$r_s \cdot H_s = F_{dola} \cdot H_{dola} - \frac{Y_s \cdot amf \cdot F_{coal}}{M_{H2S}} \cdot H_{desulf} \quad (28)$$

where H is the heat of combustion, or the latent heat, Du is the coal heating value based

on the Dulong equation, $r_s H_s$ includes the heat of the dolomite calcination and desulfurization. One more assumption, is that the heat of combustion of CO and H_2 are taken to be equal. The carbon reaction rate is assumed as

$$R_C - kW_C P_{H_2O} + \frac{\beta + 1}{\beta/2 + 1} F_{O_2} + (CH_4 + CO + CO_2)_{devola} \quad (29)$$

which includes gasification, combustion and devolatilization teams. The final result is as following:

$$\frac{d\Delta X}{dt} - c_1 + a_1 \Delta X + b_1 \Delta T \quad \frac{d\Delta T}{dt} - c_2 + a_2 \Delta X + b_2 \Delta T \quad (30)$$

$$G(s) = \frac{K(\tau_d s + 1)}{(\tau_L s + 1)(\tau_s s + 1)} \quad (31)$$

where a and b are related to the state variables, and c is to the manipulated variables. It is a 2nd order system with a zero, determined by the value of c.

The long term and the short term response time τ 's can now be evaluated (appendix). The Bode plots (Fig.9) show that:

- (1) the carbon conversion shows a first order behavior to a step change in air, coal, or dolomite feed rates, and a second order behavior to a steam feed rate change;
- (2) for a temperature change, all of the feeds give a first order lag appearance. In addition, the dolomite feed has a lead effect at low frequencies.

The time constant are summarized in following table.

Table 1 τ_a min (where $\tau_L = 158$ min; $\tau_s = 14.5$ min)

	air	coal	steam	sorbet	τ_{feed}
for X	14.9	14.5	NA	13.8	14.1
for T	73.8	17.7	164	524	-378

If $\tau_a \approx \tau_s$, or $\tau_a \approx \tau_L$ the system will behaves like a first order lag. The results show that

for conversion, τ_a values nearly equal to τ_s for all the cases; and for temperature, for the case of coal feed, $\tau_a \approx \tau_s$. This induces a slow temperature response as seen in chapter 2 where the air/coal ratio change cause a long time response in temperature as in conversion.

The open loop dynamic responses based on the linear model are plotted in Fig.10, where the four feed rates were varied by 10% each. The comparison between the linear and the non-linear models are given in Fig.11 for temperature and Fig.12 for conversion. One can see that the linear model shows the same tendency as the non-linear model for most of the ranges, except for some difference in the initial rates for temperature.

The property of additivity has been checked (Fig.13) by the so called "in-out" model. The model for temperature and conversion are as following:

$$\Delta T = G_{air} \cdot \Delta_{air} + G_{ss} \cdot \Delta_{ss} + G_{coal} \cdot \Delta_{coal} + G_{dolo} \cdot \Delta_{dolo} \quad (32)$$

$$\Delta X = G_{air} \cdot \Delta_{air} + G_{ss} \cdot \Delta_{ss} + G_{coal} \cdot \Delta_{coal} + G_{dolo} \cdot \Delta_{dolo} \quad (33)$$

where G is a transfer function. and Δ is a disturbance. This model says that the effects of different disturbances are additive. Fig.13 is the result where the solid line is the multivariable response, and the dashed line is the summation of the different responses by different disturbances. It is obvious that the effect is additive for the gas feeds for some operation points but it is not additive for all of the feed rate range (10%). It tells us that the total multivariable effect is not a linear combination of separate multivariable effects, and that interaction exists between these variables, as has been shown by the RGA analysis. From these results, it appears that a non-linear analysis might be helpful in control design.

4. RESULTS AND DISCUSSION

The results obtained from the linear model have been mentioned above. Here we give some of the dynamic simulation results.

The simulation results for different disturbances have been plotted in Fig.5-Fig.8, where they include changes in coal property. In spite of some oscillations, the outcomes show that load following control can be achieved in a fast way, and the pressure can be controlled smoothly. In most of the cases, power can be controlled in a reset time of less than 1 min, and the pressure was kept at its setpoint value within an error of less than 0.1 atm.

Fig.5 shows the responses to coal property changes from #1 (Pit.#8) to #2 (Ill.#6) both for the open loop and for the closed loop cases. Owing to the coal property changes, air/carbon and fuel gas LHV change, and result in an increased coal feed rate and reduced air and steam feed rates, as well as gas output rates. As the load is constant, the fuel feed rate is kept unchanged. The results showed that pressure and temperature were well controlled at their set points. In this example, the gas flow rate was uncontrolled and changed as the gas feed rate changed.

Fig.6 shows a load change of up to 20%. It includes two cases, one for constant pressure setting (U_g uncontrol), and another for varied pressure operation (U_g control) in order to keep constant gas linear velocity. In both the cases, fuel gases showed a fast response, and pressures and temperatures were well controlled in a fast way. The feed manipulation of coal, air, and steam were under damping and reset time was near 4 min. which is different from that for coal properties change.

In general, fuel HHV or LHV changes fast with changes in the gas feed rate, such as air or steam, or the coal feed rate. But because the coal feed rate can not be adjusted so fast, filter or a slow reset time constant is required for the coal feed rate control, other than, under damping response occurs as shows in Fig.6. Fig.7 shows the results for the case where a filter was utilized to suppressed the fluctuations for both the above cases,

i.e. a load change of 20% or a coal feeds changes from #1 to #2. The results showed that the damping was reduced for the case of load change, and a little enhanced for the coal change case.

Fig.5 was another example where slow control for the coal feed rate, and the gas feed rate were used to try to limit the damping. The power and pressure are well controlled in spite of the long response time required to reach the new steady state ($t > 30$ min). In Fig.6, the time reach the new steady state was less than 10 min. The coal feed transient was much smoother than that in Fig.5 to Fig.7.

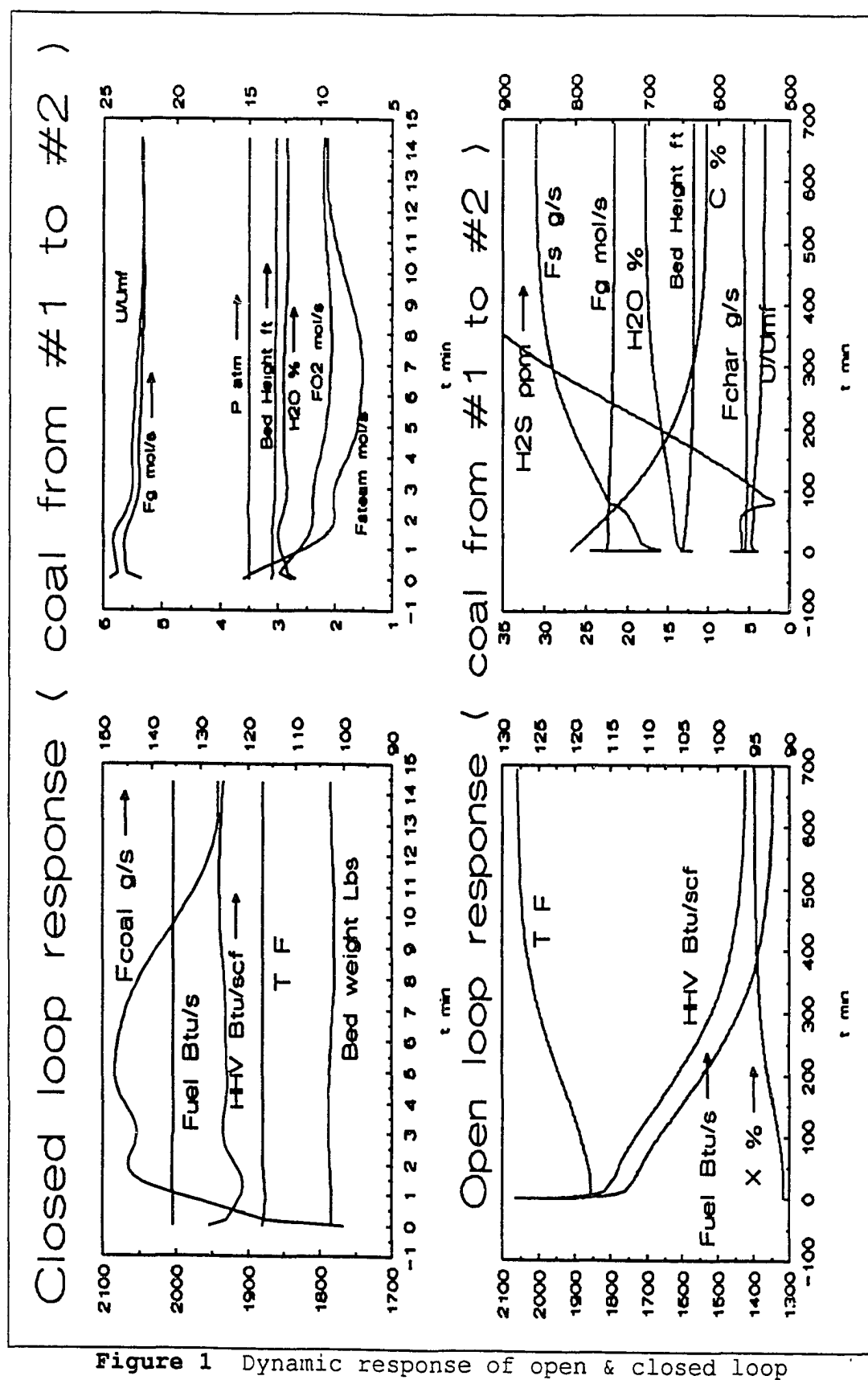
A point worth mentioning is that the fuel gas BTU (LHV) has a maximum value which occurs at a conversion of about 0.9. Above this point, the fuel gas Btu has a negative gain as X increases. It is obvious that better operation from the control view point is under $X < 0.9$. In this region, the fuel product rate will increase as air feed increases. This will give an advantage in fuel quality control. It is seen once again that a good design can eliminate some of the control problems as mentioned before. However, if the fuel gas is send directly to the gas turbine, without any cooling to the fuel gas, the sensible heat should be counted in the fuel gas heating value. In this case, there is no negative gain.

There is still a problem in that H_2S can not be controlled fast because of two factors, one being fast sulfur release during devolatilization, and the another relatively slow process of calcination. One way to solve this problem is to store a great amount of CaO , that is increase the ratio of CaO/S in the gasifier to suppress the fluctuations. And then control the sorbet feed by a ratio control with the coal feed rate, the ratio being set by measuring H_2S in the gas phase. As the replacement of CaO in the gasifier would be a slow process, this measurement can now be done off line. This then forms a loose control loop.

From Fig.6 to Fig.8, one can see that the coal feed transient in load following is

like a step function with an overshoot. This suggests that it be controlled by load directly during load changes, which is called as gasifier lead model. Combined with LHV measurement, its ratio, the set point thus be adjusted. However when the load is constant, it should be manipulated by the fuel gas flow rate control. This is an inferential control for the fuel gas LHV. If coal feed is controlled by the load, it is preferable to use fast controls for air and steam feed rates as shown in Fig.5 to Fig.8.

In order to decouple the interaction between the gasifier and the combustor, changes in the solid withdrawal rate should be as slow as possible. This also gives an advantage in decoupling the interaction between the combustor and the gas turbine, which otherwise because of a fast change in the combustor duty would effect the turbine fuel control, and in turn the gasifier. A good design is to use the gasifier itself as a filter to reduce this coupling effect. This once again brought out the importance of, and the need to, closely integrate process design and process control.



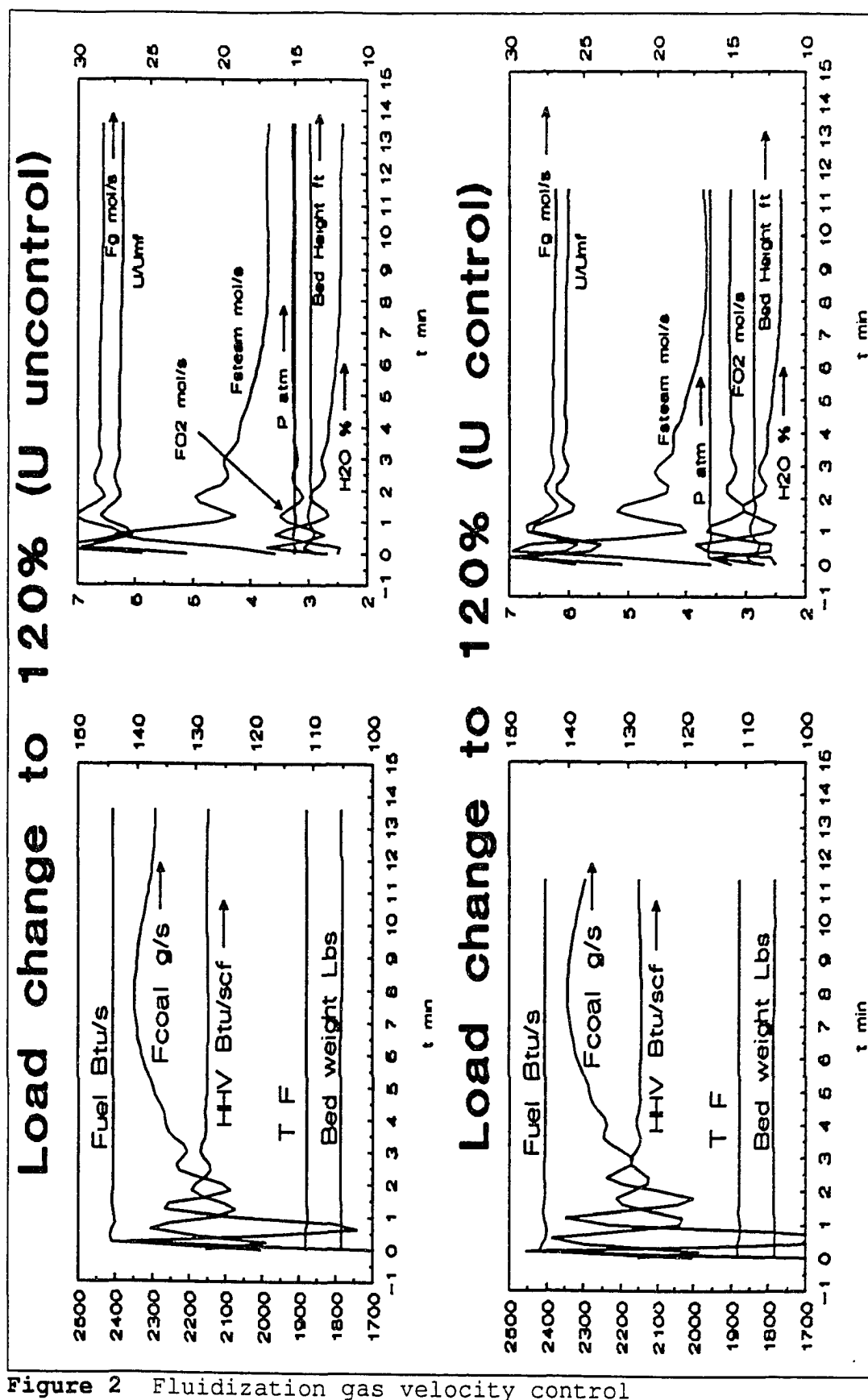


Figure 2 Fluidization gas velocity control

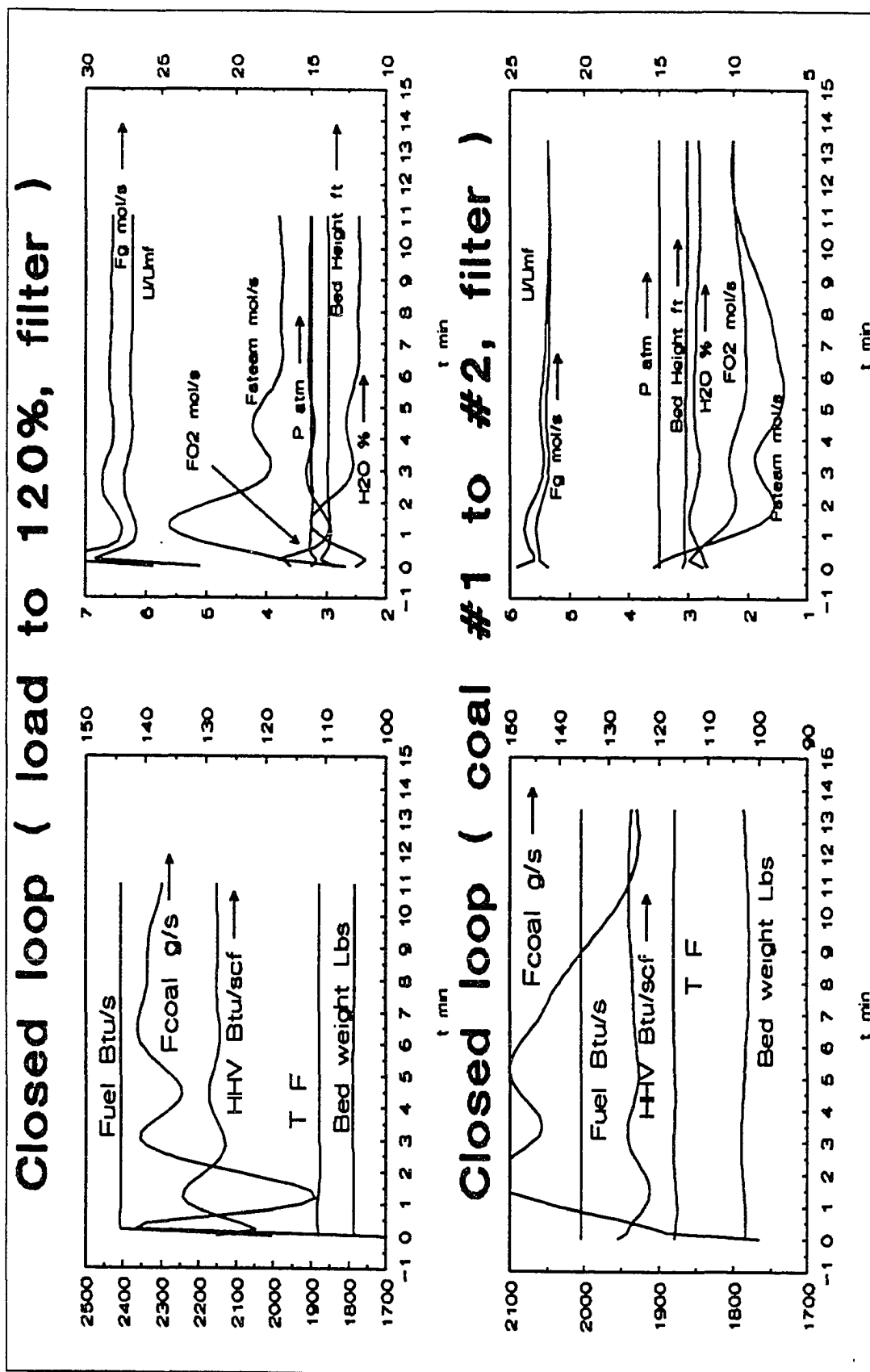


Figure 3 Filter application to control

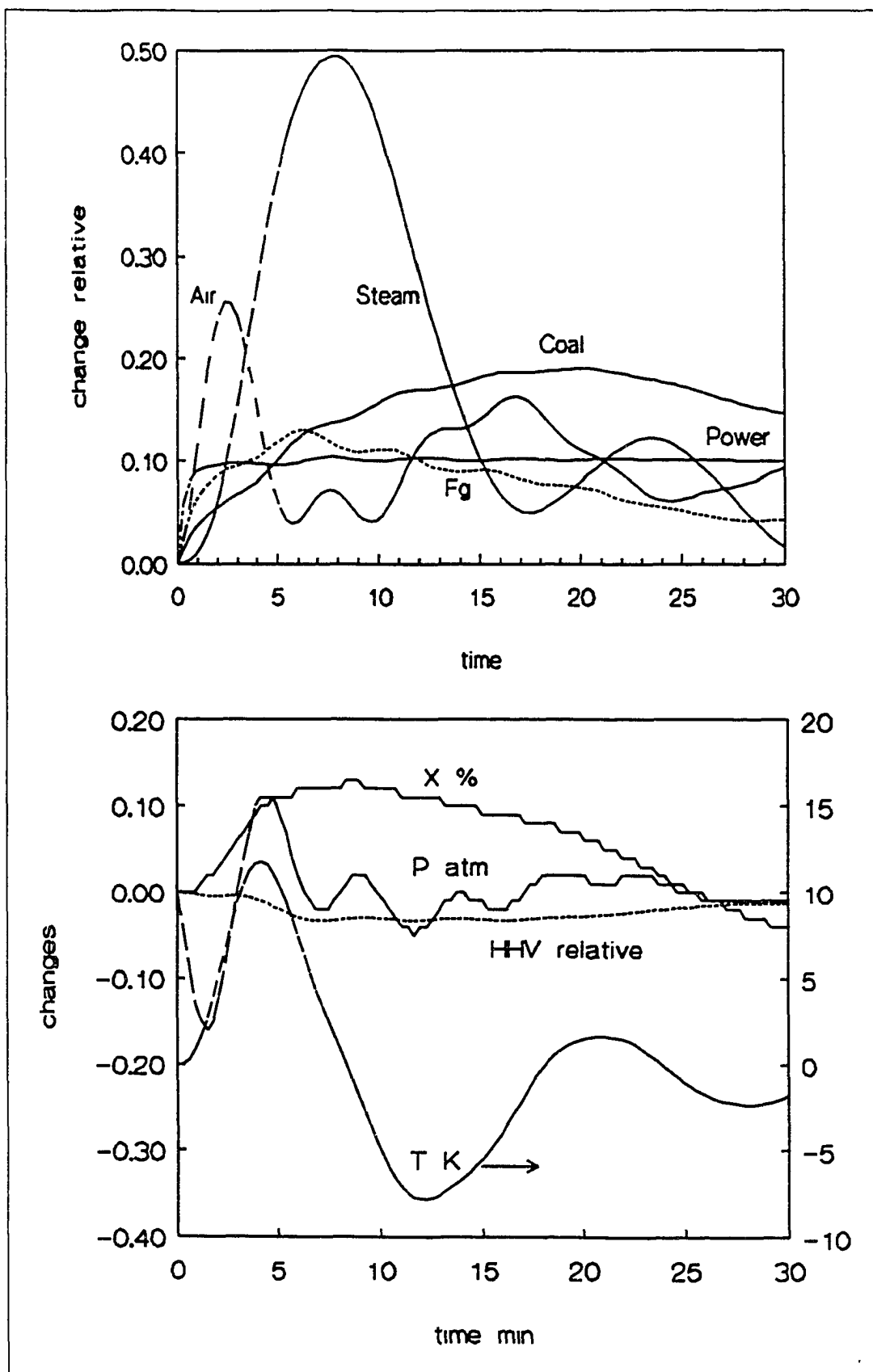


Figure 4 Control for load 10 % change in a slow way

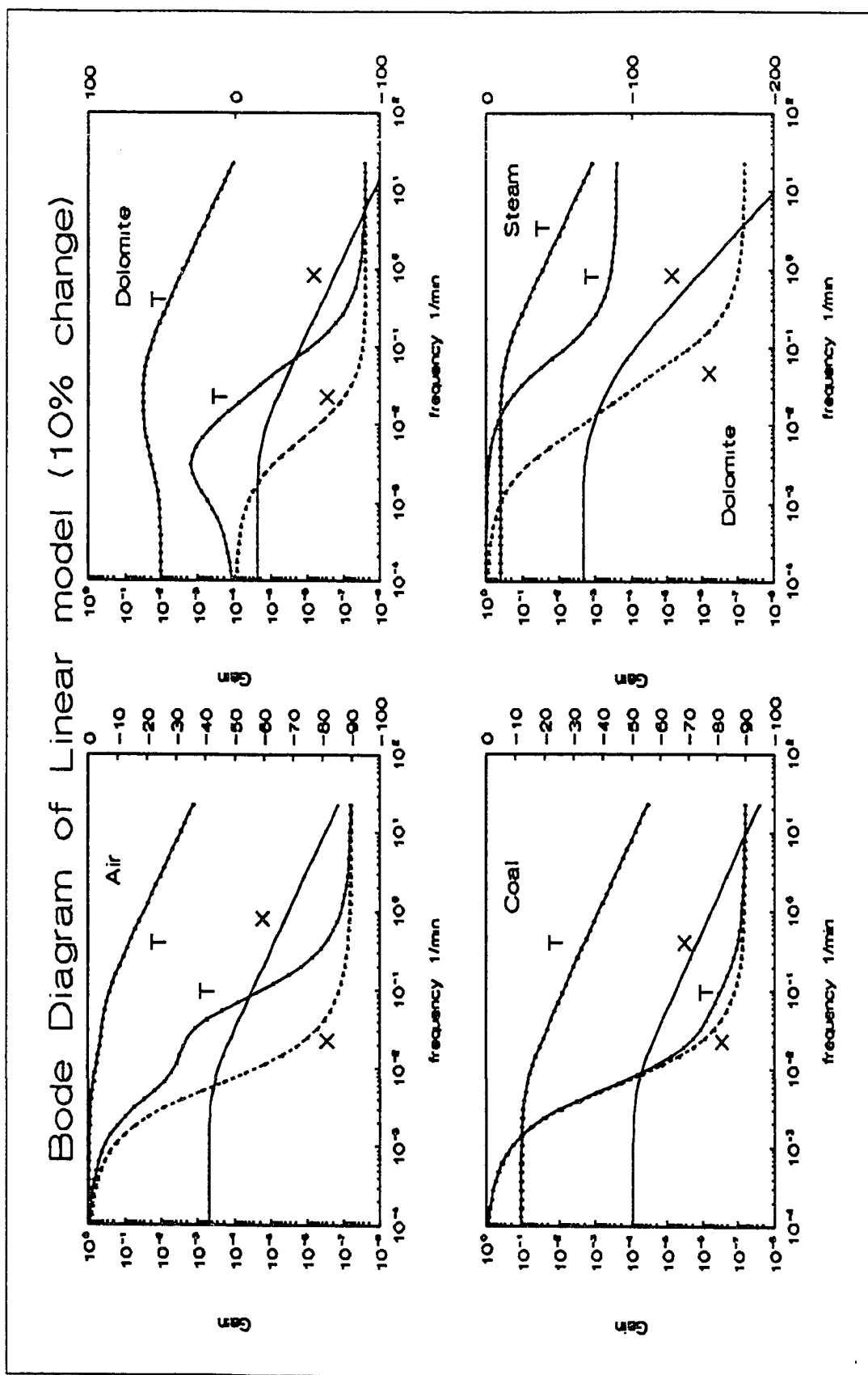


Figure 5 Bode Diagram of Linear Model

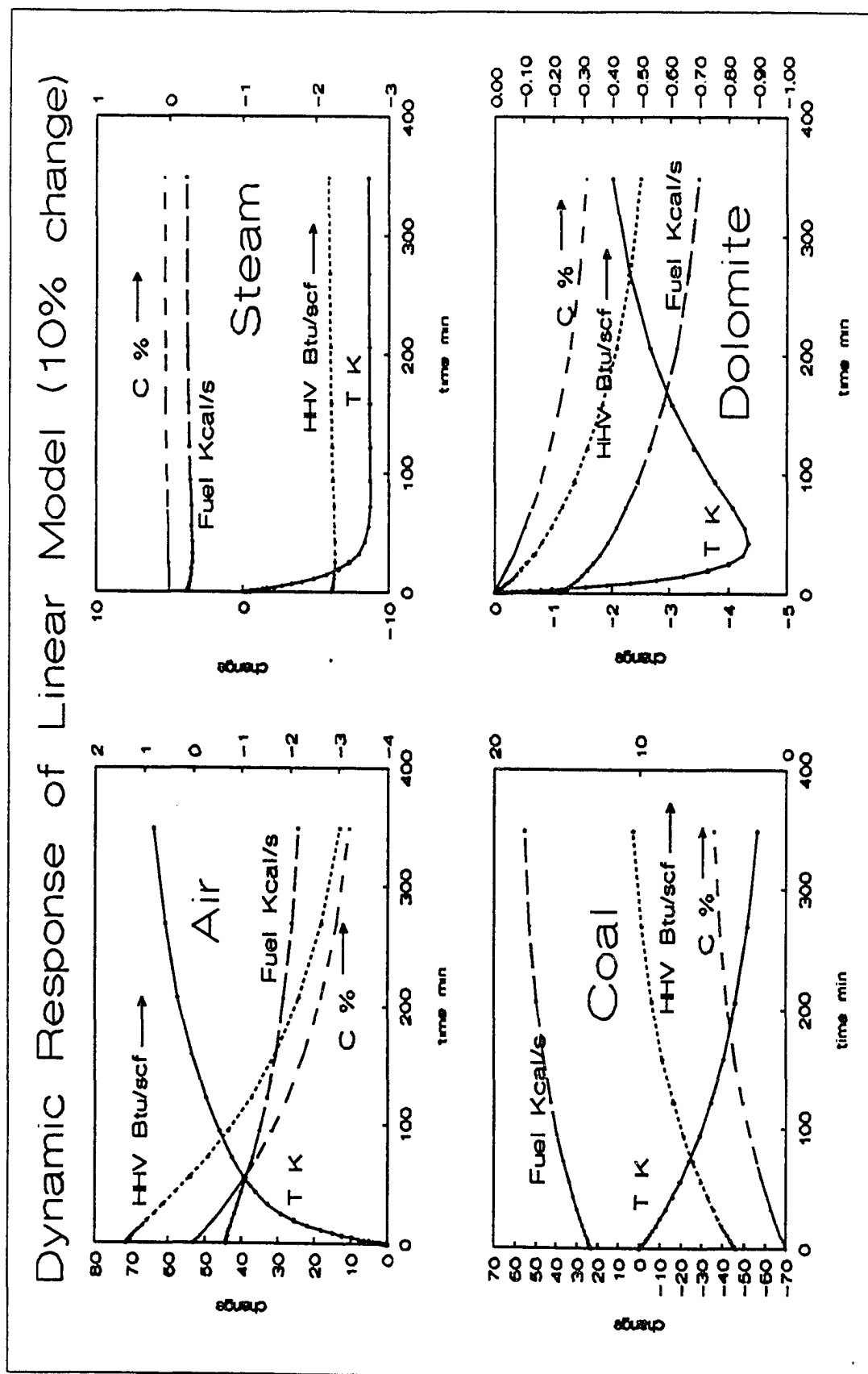


Figure 6 Dynamic Response by Linear Model

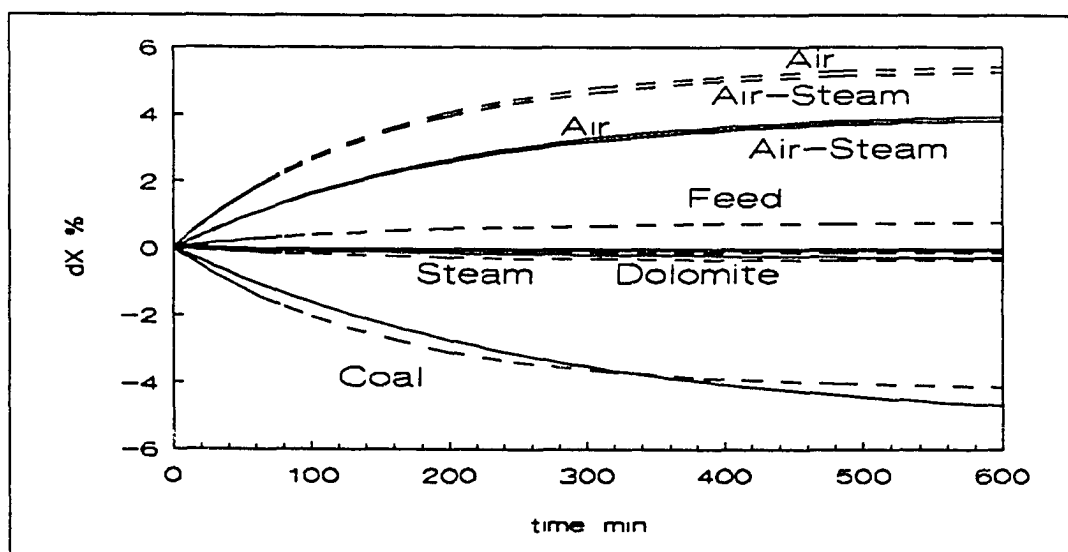
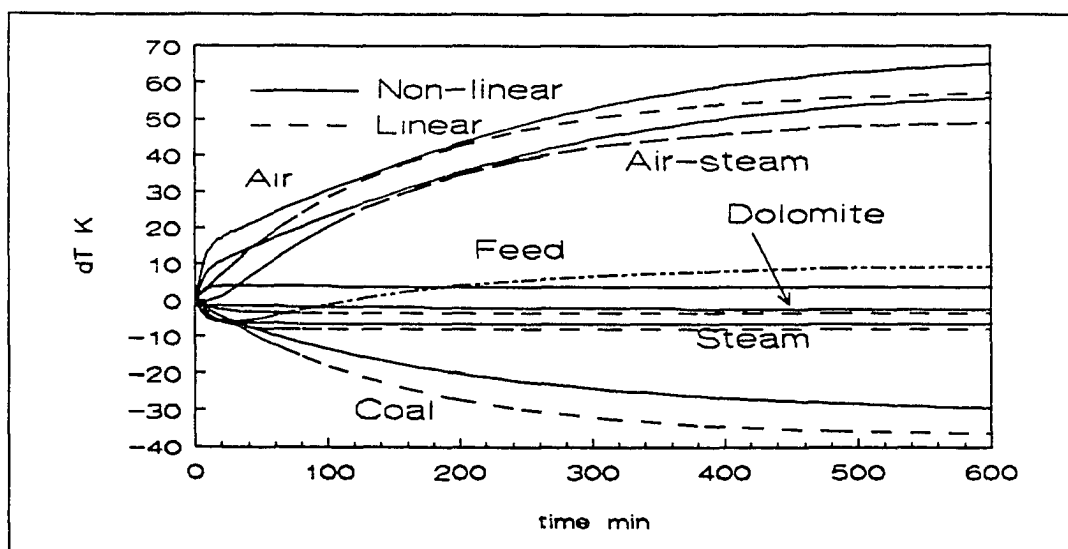


Figure 8,9 Comparison of Non- and Linear Model ($\Delta=10\%$)

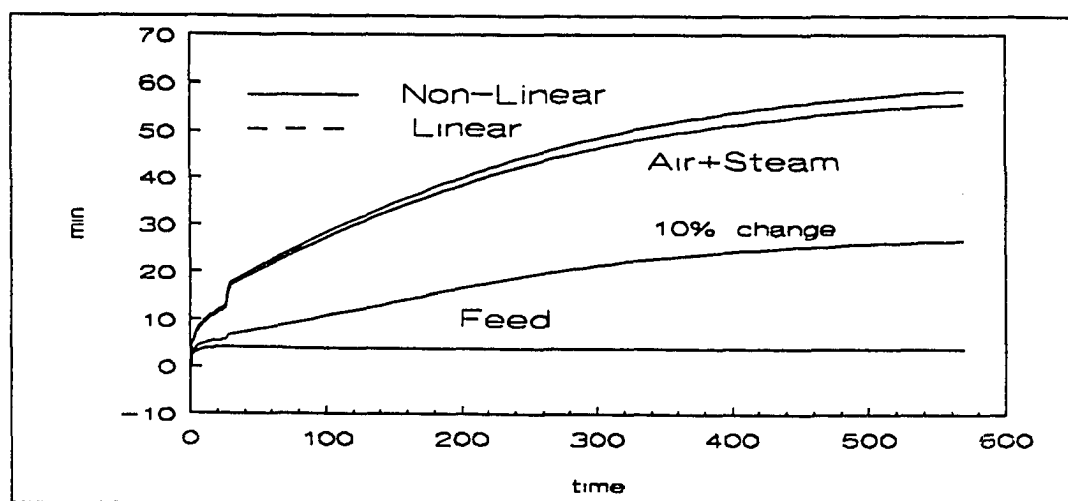


Figure 9 Test of Additive

5. SUMMARY OF WORKS TO DATE (PART I)

The conclusions from part I are listed as following;

(1) Time Constants

The Coal gasifier has been simulated in detail. Three time constants characterizing the dynamic response of the gasifier were found. The first one for short term response in seconds, corresponds to gas replacement. The second one in minutes, corresponds to the thermal capacitance of the gasifier. And the last one for long term response in hours i.e. due to the solid displacement and the coal gasification reaction. These time constants have been evaluated by developing a linear model, which gives a good explanation on the τ behavior of these different time constants. And the transient behaviors can be well understood to the limit amount of dynamic data available in literature (Rhinehart,1987; Kutten,1974). These time constants are useful in setting the controller parameters, and in designing the configuration and last but not the least in the design of the power plant.

On the basis of the linear model, the gasifier can be treated simply as an overdamped second order lag first order lead system. The zero τ_z for the lead gives an effect to thermal time constant, that is heat replacing. If the feed change causes a large change in the conversion, the thermal τ becomes large, such as in the cases of air or coal feed rate changes.

As the short term time constant for the gas flow is very small, 6-10 s, it appears that the fluidized bed gasifier can be directly integrated to the power plant because of its fast response. This is similar to the entrained fluidized bed gasifier in which the gas and solid residence times are short and this gives a fast response for a load following control. The simulation results show that this fast response has been well applied to the IGCC system.

But because of a big solid holdup and a long solids turn over time, the fluidized

bed gasifier is not suitable to on-off operation. It is only suitable for base load or a load following operation in a certain range. If we could have a way to keep the bed in a warm state for some carbon conversion, we might be able to use it for an on-off operation.

It is noted that the initial response and its gains are quite different when compared to the steady state gains. Under some conditions the simulations even showed inverse responses and overshoots.

(2) Control and Design Aspects

Though the heat and mass holdups of the coal gasifier are big so that the reset times are big too, the gas residence time is small, about 10 s for the FW case. This gives an advantage for fuel control for the fast response required for load following operation. The dynamic simulation validates this point, and the fuel supply can be controlled very well.

Dynamic results for the open-loop and closed-loop operations for a coal quality change, and for load changes have also been studied. The control for coal quality changes is done quite satisfactorily by using the model, where it tells us that the fuel gas quality is sensitive to the coal quality.

The fuel gas heating value has a maximum value that occurs at a conversion of about 0.9, which induces a gain sign change for the fuel gas heating value as X increases. It is obvious that better operation for control is under $X < 0.9$. In this region, the gain is positive, and the fuel product rate will increase as air feed increased. This gives an advantage in fuel quality control. In addition, when the operating X is low, its effect on the following combustor unit will be small. Thus, we see that a good design can eliminate some of the control problems as mentioned before.

(3) Plant Operation

The application of a fluidized bed combustor adds a degree of freedom to the IGCC system by allowing changes in carbon conversion in the gasifier. The plant

efficiency will be high if the conversion in the gasifier is relatively low when the fluidized bed combustor is used. This corresponds to a low O_2/C ratio operation, low temperature, and a high fuel gas BTU, which is better for the IGCC operation. Materials constraints limit the temperature of the fuel gas produced and so cooling the gas down will reduce the thermal efficiency. A low conversion operation is also being an advantage in the control of the IGCC system as mentioned before.

(4) Model Validation

The model has been applied for the pilot plant simulation, for Foster Wheeler Inc., for the steady state case, and good predictions were obtained. Some of the dynamic transients of the pilot plant show the same results as obtained from the simulation, when switching from one steady state operation to another.

The prediction of solid mixing/segregation profile also showed a good agreement between the simulated and the pilot plant data.

6. APPENDIX - LINEAR MODEL

Though the globe model is non-linear, but the dynamic response shows that it looks like linear behavior with the first order processes in parallel. It is specially clear from a log plot for the x-axis. it acts as a case of single zero 2nd order system. It hints us to develop a 2nd order linear model to study system analytically from frequency domain. What we are interested now is that, could we develop a 2nd order model taking account of the dominant effects to simulate the gasifier, that could help us in control design for the stability.

There are three ways to create linear model. One is linearized non linear model in details. One is by neglecting some unimportant factors or by use of simple model to erase nonlinear term and then linearizing them. The last one is to use input-output response data to figure out the linear model. The first one will give more information

than others, but the second one is more useful in understanding parameters physically. In general the last one is of more precision in a wide range since it comes from the data fitting. As one already know that our model was a little complex by non linear kinetics, reactivity decay, equilibrium constraints, and the RTD's of the gas and the solid, our selection was last two methods by which we either understand the system performance or get a good linear model.

6.1 MODEL DEVELOPMENT

As the shift reaction relates to the gas composition, it gives a trouble in solving ODA's (the ordinary differential algebraic equations). They can be solved numerically, but it is unsuitable in the Laplace and frequency domains. What we treat here is we neglect the gas holdup. By this way, we cancel all the ODE's from the gas phase, which is the same as Weng's model (1991) in the steady state case. We also found that in different dynamic cases, gas composition changes only a little, that means we can neglect the effect of the gas composition on the kinetics. In solid, the most of important reactions are carbon converted by devolatilization, gasification, and combustion. So as to in a linearizing model, we concentrate to the carbon reaction, assume all of the others are in equilibrium or full completed.

Model assumptions:

- (1) the gas holdup is neglected;
- (2) bed weight, pressure are kept as constants;
- (3) gasification kinetics is 1st order to carbon in bed.

The kinetics of gasification is taken as simple one:

$$R_c = k \cdot W_c \quad (1)$$

where W_c is the carbon content in bed. The models for the mass and energy balances are

as follows

$$W \frac{dx}{dt} = F_c - F_s \cdot x - R_c \quad F_s = F_{st} - R \quad (2)$$

$$\frac{d}{dt}(W \cdot x \cdot H_c + W \cdot C_p \cdot T) = F_{coal} \cdot Du + F_{st} \cdot H_{st} - F_s \cdot x \cdot H_c - F_g \cdot H_g - F_g \cdot C_p \cdot (T - T_l) - r_s \cdot H_s \quad (3)$$

$$r_s \cdot H_s = F_{dola} \cdot H_{dola} - \frac{Y_s \cdot amf \cdot F_{coal}}{M_{H2S}} \cdot H_{desulf} \quad (4)$$

where H is heating value, Du is coal heating value on the base of the Dulong equation for the coal, $r_s H_s$ includes the heat of the dolomite calcination and the desulfurization process, where we assumed the calcination and the sulfur capture are almost 100% for the material and energy balances. We further made an assumption of $F_{gin} = F_{gout}$ for the air operation. Based on the constant bed weight, one has

$$R = R_c + F_{coal} [Y_M + amf(1 - Y_C)] + F_{dola}/2 \quad (5)$$

$$R_c = k \cdot W \cdot x + \frac{\beta + 1}{\beta/2 + 1} \cdot F_{O2} \cdot M_C + (CH_4 + CO + CO_2)_{devola} \quad (6)$$

$$(CH_4 + CO + CO_2)_{devola} = F_C (0.07 + 0.93 \cdot \frac{Y_O}{Y_C} \cdot \frac{M_C}{M_{O2}}) \quad (7)$$

where x is carbon weight fraction in bed, $\beta = CO/CO_2$ is a parameter for the C combustion.

$$F_C = F_{coal} \cdot amf \cdot Y_C \quad F_{st} = F_{coal} + F_{dola} \quad (8)$$

$$F_{H2} = F_{st} + \frac{Y_M \cdot F_{coal}}{M_{st}} + \frac{Y_H \cdot amf \cdot F_{coal}}{M_{H2}} \quad (9)$$

$$F_O = 2 \cdot F_{O2} + F_{st} + \frac{Y_M \cdot F_{coal}}{M_{st}} + \frac{Y_O \cdot amf \cdot F_{coal}}{M_O} + 2 \cdot \frac{F_{dola}/2}{M_{CO2}} \quad (10)$$

The material balances for C, H₂, and O are as

$$C: R_C^* - CO + CO_2 + CH_4 \quad \Delta R_C^* - \Delta CO + \Delta CO_2 + \Delta CH_4 \quad (11)$$

$$F_{H_2} - H_2 + H_2O + 2 \cdot CH_4 \quad \Delta F_{H_2} - \Delta H_2 + \Delta H_2O + 2 \cdot \Delta CH_4 \quad (12)$$

$$F_O - CO + 2 \cdot CO_2 + H_2O \quad \Delta F_O - \Delta CO + 2 \cdot \Delta CO_2 + \Delta H_2O \quad (13)$$

In the model, based on the result of the dynamic simulation, we assumed the methane yield is given as $CH_4 = 0.072 \cdot F_C / M_C$, that is $\Delta CH_4 = 0.006 \cdot \Delta F_C$. The shift equilibrium can be written as $CO_2 \cdot H_2 = K \cdot CO \cdot H_2O$, which gives

$$CO_2 \cdot \Delta H_2 + H_2 \cdot \Delta CO_2 - K (CO \cdot \Delta H_2O + H_2O \cdot \Delta CO) + CO \cdot H_2O \cdot \Delta K \quad (14)$$

$$\Delta H_2O = \frac{CO_2(\Delta F_{H_2} - 2\Delta CH_4) + (\Delta F_O - \Delta R_C^*)(H_2 + K \cdot H_2O) - K \cdot H_2O \cdot \Delta R_C^a - CO \cdot H_2O \cdot \Delta K}{K(H_2O + CO) + (H_2 + CO_2)} \quad (15)$$

$$\Delta R_C^a - \Delta R_C^* - \Delta CH_4 \quad (16)$$

$$\Delta F_g - \Delta H_2O + \Delta CO_2 + \Delta CO + \Delta H_2 + \Delta CH_4 + \Delta N_2 - \Delta R_C^* + \Delta F_{H_2} + \Delta F_{N_2} - 2 \cdot \Delta CH_4 \quad (17)$$

$$\therefore \Delta H_2O - \Delta(F_g \cdot Y_{H_2O}) - F_g \cdot \Delta Y_{H_2O} + Y_{H_2O} \cdot \Delta F_g \quad (18)$$

$$\Delta Y_{H_2O} = \frac{\Delta H_2O - Y_{H_2O} \Delta F_g}{F_g} - \frac{Y_{H_2O}}{F_g} (\Delta R_C^* + \Delta F_{H_2} + \Delta F_{N_2} - 3 \cdot \Delta CH_4) + \frac{CO_2(\Delta F_{H_2} - 2\Delta CH_4) + (H_2 + K H_2O) \Delta F_O - (H_2 + 2K H_2O)(\Delta R_C^* - \Delta CH_4) - CO \cdot H_2O \Delta K}{F_g [K(H_2O + CO) + (H_2 + CO_2)]}$$

$$\therefore \Delta Y_{H_2O} - A_1 - A_2 \cdot \Delta R_C^* - A_3 \cdot \Delta K - \frac{Y_{H_2O}}{F_g} \cdot \Delta R_C^* - A_4 - (A_1 - A_4) - (A_2 + \frac{Y_{H_2O}}{F_g}) \Delta R_C^* - A_3 \cdot \Delta K \quad (19)$$

Linearizing for R_c , we had

$$\begin{aligned} \Delta R_c - W(k \cdot \Delta x + x \cdot \Delta k) + \frac{\beta + 1}{\beta/2 + 1} \cdot M_c \cdot \Delta F_{O_2} + 0.1 \cdot \Delta F_c + M_c \cdot F_{O_2} \cdot \frac{1/2}{(\beta/2 + 1)^2} \cdot \frac{d\beta}{dT} \cdot \Delta T \\ - A_6 + A_7 \cdot \Delta x + A_{89} \cdot \Delta T \end{aligned} \quad (20)$$

$$\Delta k = \frac{k \cdot E}{RT^2} \cdot \Delta T \quad \Delta K = \frac{K \cdot \Delta H}{RT^2} \cdot \Delta T \quad (21)$$

$$\therefore \Delta R = \Delta R_c + \Delta(F_{coal} \cdot [Y_M + amf(1 - Y_c)] + F_{dold}/2) = \Delta R_c + B_1 \quad (22)$$

$$\begin{aligned} \therefore W \frac{d\Delta x}{dt} - \Delta F_c - x \cdot \Delta F_s - F_s \cdot \Delta x - \Delta R_c - \Delta F_c - x(\Delta F_s - \Delta R_c - B_1) - F_s \cdot \Delta x - \Delta R_c \\ - B_2 + (x - 1)\Delta R_c - F_s \cdot \Delta x - B_2 + (x - 1)(A_6 + A_7 \cdot \Delta x + A_{89} \cdot \Delta T) - F_s \cdot \Delta x \\ - B_2 + (x - 1) \cdot A_6 + [(x - 1)A_7 - F_s] \Delta x - (x - 1)A_{89} \cdot \Delta T \end{aligned} \quad (23)$$

$$\frac{d\Delta x}{dt} = c_1 + a_1 \cdot \Delta x + b_1 \cdot \Delta T \quad (24)$$

Linearizing for the heat balance equation, we got

$$\begin{aligned} W \cdot C_p \frac{d\Delta T}{dt} - \Delta(F_{coal} \cdot Du + H_x \cdot F_x - r_s \cdot H_s) - H_c(\Delta F_c - \Delta R_c) - \Delta(F_g \cdot H_g) - \Delta[F_{gC_p}(T - T_i)] \\ - B_3 + H_c \cdot \Delta R_c - \Delta(F_g \cdot H_g) - \Delta[F_g \cdot C_p(T - T_i)] \end{aligned} \quad (25)$$

If we take $H_{CO} \approx H_{H_2}$, we had

$$\begin{aligned} \Delta(F_g \cdot H_g) = H_x \cdot \Delta H_2O + H_{CO} \cdot \Delta CO + H_{H_2} \cdot \Delta H_2 + H_{CH_4} \cdot \Delta CH_4 \\ - H_x \cdot \Delta H_2O + H_{CH_4} \cdot \Delta CH_4 + H_{CO}(2 \cdot \Delta R_c^* - \Delta F_O + \Delta F_{H_2} - 4 \cdot \Delta CH_4) - H_x \cdot \Delta H_2O + 2 \cdot H_{CO} \cdot \Delta R_c^* \end{aligned} \quad (26)$$

$$W \cdot C_P \frac{d\Delta T}{dt} - B_3 + H_C \cdot \Delta R_C - H_{st} \cdot \Delta H_2 O - 2 \cdot H_{CO} \cdot \Delta R_C^* - B_4 - F_{gC_P} (\Delta T - \Delta T_i) - C_P (T - T_i) \Delta F_g$$

$$- B_5 + [H_C + (H_{st} \cdot F_g \cdot A_2 - C_P (T - T_i) - 2 \cdot H_{CO}) / 12] \cdot \Delta R_C + H_{st} \cdot F_g \cdot A_3 \cdot \Delta K - F_g \cdot C_P \cdot \Delta T$$

$$- (B_5 + A_6 \cdot B_6) + A_7 \cdot B_6 \cdot \Delta x + (A_{89} \cdot B_6 - F_g \cdot C_P + H_{st} \cdot A_3 \cdot F_g \frac{K \cdot \Delta H}{RT^2}) \Delta T \quad (27)$$

$$\frac{d\Delta T}{dt} - c_2 + a_2 \cdot \Delta x + b_2 \cdot \Delta T \quad (28)$$

Taking the Laplace transfer for two equations, we got

$$s\Delta \bar{x} - c_1 + a_1 \Delta \bar{x} + b_1 \Delta \bar{T} \quad s\Delta \bar{T} - c_2 + a_2 \Delta \bar{x} + b_2 \Delta \bar{T} \quad (29)$$

$$G(s) = \frac{K \cdot (\tau_a s + 1)}{(\tau_L s + 1)(\tau_s s + 1)} \quad (30)$$

where τ_L and τ_s related to the model and state variables, τ_a to manipulated variables. It has a solution for a step input:

$$G(t) = K \cdot M \cdot \left(1 + \frac{\tau_a - \tau_L}{\tau_L - \tau_s} e^{-t/\tau_L} + \frac{\tau_a - \tau_s}{\tau_s - \tau_L} e^{-t/\tau_s} \right) \quad (31)$$

If $c_1=0$ the solution will be reduced to

$$G(t) = K \cdot M \cdot \left[1 + \frac{1}{\tau_L - \tau_s} (\tau_s e^{-t/\tau_s} - \tau_L e^{-t/\tau_L}) \right] \quad (32)$$

6.2 RESULTS

For the case of $X=0.89$, $T=1280^\circ K$, ($O_2/C=0.4$, $Steam/C=0.6$, $Ca/S=1.5$, and $\tau_{feed}=2.25$ hr), the result is $\tau_L=158$ min, and $\tau_s=14.5$ min. The τ_a 's are related to the manipulated variables, and are listed bellow:

Table 1 τ_a min (where $\tau_L=158$ min; $\tau_S=14.5$ min)

	air	coal	steam	sorbent	τ_{feed}
for X	14.9	14.5	NA	13.8	14.1
for T	73.8	17.7	164	524	-378

It is clear that the τ_{ax} is near τ_S , so the x response is almost determined by τ_L , while the τ_{at} is big, the temperature response is mainly determined by τ_S . These are the facts what we have seen in dynamic simulation.

Above studies are based on the individual feed manipulated variable. In fact, sometimes one likes to use ratios as manipulated variables, such as O_2/C , H_2O/C , Ca/S , and τ . The results are listed as τ_{feed} , where the other ratios are kept as constants. It was noted that the $\tau_{At}<0$, which means for a τ change, the temperature has an inverse phenomena. While in a non-linear dynamic simulation, it behaviors as an overshoot phenomena with the same tendency as the non-linear case.

The Bode diagrams for the feed rate change of coal, air, steam etc are shown in Fig.9. It was found that most of cases like the first order behavior, as the τ_a shows.

Simulation in time domain is shown in Fig.10. As gas τ_g is very small, say 10 s, any changes will cause gas composition change immediately, after that it will pass a short and a long term changes with change in temperature and solid composition.

The comparison of the temperatures by the linear and the non-linear model is given in Fig.12. The program for the responses of the linearized model, and the Bode diagram have been produced in QPRO by the spreadsheet and graph together.

After we get the δT and δX , the fuel value of the product gas and its HHV can be calculated from above equations. All of them are set up in the QPRO.

6.3 τ 's EVALUATION

Now we would like to know τ 's more physically and directly. The response τ 's

can be evaluated based on the two assumptions:

- (1) R_C is constant in short time, and gasification rate is first order for long time;
- (2) a pseudo-steady state is used for the gas phase, that is neglect of the gas holdup.

Now material balances are

$$W_s \cdot \frac{dx_A}{dt} = w_A - w \cdot x_A \quad W_s \cdot \frac{dx_C}{dt} = w_C - w \cdot x_C - R_C \quad (33)$$

Linearizing equations by $\Delta x = x_i - x_i(0^+)$, one can get

$$W_s \cdot \frac{d\Delta x_i}{dt} = -w \cdot \Delta x_i \quad (34)$$

They are linear and easy be solved out. The energy balance is

$$W_s \cdot C_{ps} \cdot \frac{dT}{dt} = H_f - F_g \cdot C_{pg} \cdot (T - T_r) - F_g \cdot \sum Y_i \cdot H_i - w \cdot C_{ps} \cdot (T_o - T_r) - w \cdot \sum x_i \cdot H_i \quad (35)$$

Linearizing it by $\Delta T = T - T(0^+)$

$$W_s \cdot C_{ps} \cdot \frac{d\Delta T}{dt} = 0 - F_g \cdot C_{pg} \cdot \Delta T - F_g \cdot \sum \Delta Y_i \cdot H_i - 0 - w \cdot \sum \Delta x_i \cdot H_i \quad (36)$$

Here we use $t=0^+$ as a base is trying to get rid of the jump in gas composition induced by gas feed jump. Under equilibrium conditions,

$$Ek = \Pi Y_i \quad (37)$$

by the use of perturbation and letting $Ek \approx \alpha \cdot \beta \cdot T$, the Δ can be solved out as

$$Ek = \Pi[Y_i(0^+) + \Delta] = \Pi Y_i(0^+) \cdot \Pi[1 + \frac{\Delta}{Y_i(0^+)}] = Ek(0^+) \cdot [1 + \sum \frac{\Delta}{Y_i(0^+)}] \quad (38)$$

$$\Delta = \frac{Ek/Ek(0^+) - 1}{\sum \frac{1}{Y_i(0^+)}} = \frac{-\beta \cdot \Delta T}{Ek(0^+) \cdot \sum \frac{1}{Y_i(0^+)}} \quad (39)$$

and $\therefore \Delta Y_{CO} = \Delta Y_{H_2}$ and $H_{CO} = H_{H_2}$, therefore

$$\sum \Delta Y_i \cdot H_i = \Delta Y_{CO} \cdot H_{CO} + \Delta Y_{H_2} \cdot H_{H_2} + \Delta Y_{H_2O} \cdot H_{H_2O} - \Delta Y_{H_2O} \cdot H_{H_2O} = -\Delta \cdot H_{H_2O} \quad (40)$$

So, we get the thermal- τ in short time as

$$W_s \cdot C_{ps} \cdot \frac{d\Delta T}{dt} = -[C_{pg} + \frac{\beta \cdot H_{H_2O}}{Ek(0^+) \cdot \sum \frac{1}{Y_i(0^+)}}] \cdot F_g \cdot \Delta T - w \cdot \sum \Delta x_i \cdot H_i \quad (41)$$

$$\tau_{thermo} \approx \frac{W_s \cdot C_{ps}}{F_g \cdot C_{pg}} \quad (42)$$

Long term response τ_L corresponding to long solid residence time τ which is for a low gasification rate. One will have $\tau_L = \tau_{RTD} = W_B / F_s$ in a kinetics free model. The more precision method can be developed by a first order kinetics for gasification

$$\frac{dx}{dt} = k(1-x) \quad (43)$$

In general gas composition only be changed a little under different cases. If it is taken as a constant and combined with RTD, we get

$$\bar{X} = \int_0^\infty X(t) \cdot E(t) dt = k \frac{\tau_{RTD}}{1 + k \cdot \tau_{RTD}} = k \tau_L \quad (44)$$

while the $K \cdot \tau_{RTD}$ can be calculated from $k\tau = \Delta x / (1-x)$. let $\Delta x = 0.25$ and the $x = 0.9$ for char, then $K \cdot \tau_{RTD} = 2.5$, and $\tau_L = \tau_{RTD} / 3.5$. So the τ_L will vary with τ in the same way. It is confirmed by the dynamic simulations. The τ_{thermo} and the τ_L are calculated to give 16 min and 160 min respective, they are the same as those in the dynamic simulation and in the linearized model.

Now we know the system responses are main determined by the following three facts: one is gas velocity change which cause the system jump immediately, it is determined by gas τ_g (about 10 s); one is the thermo- τ (about 15 min), which is mainly

determined by the thermo holdup and the sensible heat release; and the last one is combined the solid τ and the reaction relax time (in hours). It is noted that the thermo- τ is affected by the solid τ , as the heat release is main related to the solid throughput, and a little to the carbon conversion.

The first two facts are important for stable operation and process control, special for power plant control when the fuel plant is connected to the power plant.

6.4 IN-OUT MODEL

The in-out model for the temperature is as following:

$$\Delta T = G_{air} \cdot \Delta_{air} + G_{ste} \cdot \Delta_{ste} + G_{coal} \cdot \Delta_{coal} + G_{dolo} \cdot \Delta_{dolo} \quad (45)$$

where the G is a transfer function. Δ is disturbance. The same for the conversion can be written as:

$$\Delta X = G_{air} \cdot \Delta_{air} + G_{ste} \cdot \Delta_{ste} + G_{coal} \cdot \Delta_{coal} + G_{dolo} \cdot \Delta_{dolo} \quad (46)$$

From the non-linear dynamic response. We might say that temperature and conversion can be expressed by the first order model with different time constant τ and gain for different disturbance. Two τ 's had been used,

$\tau=15$ min, steam and dolomite in temperature;

$\tau=150$ min, all others.

The in-out model says the effect of different disturbances is additive. But the real response is additive. Fig.13 is the result where solid line is multivariable response, dashed line is the summation of multi-response by different disturbances. It is obvious that the effect is additive for the gas feeds but it is not additive for the gas and solid feeds together. It tells us that the total effect by multivariable is not a linear combination of multivariable effects. The interaction effect exists between the variables.

7. REFERENCE

- Anthony,D.B. & J.B.Howard, "Coal devolatilization and hydro-gasification", AIChE J., 22(1976), 625
- Bhattacharya,A., L.Salam, M.P.Duduković & B.Joseph, "experimental and modeling studies in fixed-bed char gasification" I&EC Pro.Dev. Des., 25 (1988), 988
- Babu,S.P., B.Shah & A.Talwalkar, "fluidization correlations for coal gasification materials-minimum fluidization velocity and fluidized bed expansion ratio",AIChE Sym.Ser. No.176, 64(1978), 176
- Beruto,D. & W.Searcy, "Use of the Langmuir method for kinetic studies of decomposition reactions:calcite(CaCO_3)",
- Bhatia,S.K. & D.D.Perlmutter, "Effect of the product layer on the kinetics of the CO_2 -lime reaction", AIChE J., 29(1983), 79
- Borgward,R.H., "calcination kinetics and surface area of dispersed limestone particles", AIChEJ,31(1985),10
- Bortolozzi,R.A. & J.A.deiber, "mass transfer between growing air bubbles and an emulsion of coal particles in fluidized gasification and combustion", Chem.Eng.Sci.,45(1990), 1189
- Chang,C.C., L.T.Fan & W.P.Walawender, AIChE Symp.Ser. 1984
- Denn,M.M., C.Y.Wen & J.Wei, "Parameter sensitivity and kinetics-free modeling of moving bed coal gasifiers", I&EC Fund., 18(1979), 286
- Deen,M.M. & R.shinnar, "coal gasification reactors", in "Chemical reaction and engineering", edit by J.J.Carberry & A.Varma, Marcel dekker, Inc., New York, 1987
- Elliott,M.A., "Chemistry of coal utilization', 2nd, Wiley Interscience, 1981
- EPRI AP-4832, "Coal water coal gasification program: 4th annual progress report", Oct. 1986
- EPRI AP-3980, " Evaluation of the british gas corporation/lurgi slugging gasifier in GCC power generation", March 1985
- EPRI AP-2487, "Cool water coal gasification program-first annual progress report", July 1982
- Gavalas,G., "Non-catalytic gas-solid reactions with application to char combustion and gas desulfurization" in "concepts and design of chemical reactions",
- Gibilaro,L.G. & P.N.Rowe, "A model for segregating gas fluidized bed ", Chem.Eng.Sci.,

29(1974), 1403

Goyal,A. et al, "Support studies for the U-gas coal gasification process", AIChE Annual meeting, Los Angeles, 1982

Gulf Pub. Co., "Compressor Handbook", Houston, 1979

Hills,A.W.D., "The mechanism of the thermal decomposition of calcium carbonate", chem.Eng.Sci., 23(1968), 297

Ho,T.C., M.Kirkparrick & J.R.Hopper, "Mixing and combustion in a shallow coal-limestone fluidized bed combustor", Canadian J.Chem. Eng., 67(1989), 207

Kunii,D. & O.Levenspiel, "Fluidized reactor models. 1. for bubbling beds of fine, intermediate, and large particles. 2. for the lean phase: freeboard and fast fluidization", Ind.eng.Chem.res., 29 (1990), 1226

Kutten,M., "Modeling and control of a fluidized bed gasifier", Doctoral dissertation, City College of CCNY, 1978

Kunii,D. & O.Levenspiel, "Fluidization engineering", 2nd edition, Butterworth, Boston, 1991

Morell,J.I., N.R.amundson & S.K.Park, "dynamics of a single particle during char gasification", Chem.Eng.Sci., 45(1990), 387

Muhlen,H.J., K.H.van Heek & H.Juntgen, "Kinetic studies of steam gasification of char in the presence of H₂, CO₂ and CO", fuel, 64 (1985), 944

Naimier,N.S., T.Chiba & A.W.Nienow, "Parameter estimation for a solids mixing/segregation model for gas fluidized beds", Chem.Eng. sci., 37 (1982), 1047

Nienow,A.W. & N.S.Naimier, "continuous mixing of two particulate species of different density in a gas fluidized bed", Trans.ICE, 58 (1980), 181

Oguztoreli,M.N., A.Chambers & B.Ozum, "analysis of mass and energy balance equations in gas-solid reactions", Canadian J.Chem.Eng., 66 (1988),767

Pichot,Pierre., "Compressor application Engineering", 1 & 2, Gulf Pub. Co., Houston, 1986

Quentin,G.H., "Control analysis of gasification/combined cycle (GCC) power plants", Chemical Process Control II",

Rhinehart,R.R., R.M.Felder & J.K.Ferrell, "dynamic modeling of a pilot-scale fluidized-bed coal gasification reactor" Ind.Eng.Chem. Res., 26(1987), 738

- Rinard, I.H., "A roadmap to control-system design" in "Practical process instrumentation & control" V.2, edit by Matley, J., McGraw-Hill, New York, 1986
- Ross, I.B. & J.F. Davidson, "The combustion of carbon particles in a fluidized bed", Trans. IChE, 59(1981), 108
- Schouten, J.C., P.J.M. Valkenburg & C.M. Van Den Bleek, "segregation in a slugging FBC large-particle system", Powder Tech., 54(1988), 85
- Shaw, J.T., "Theoretical work on reaction sequences in the gasification of coke by carbon dioxide and by steam in conditions remote from equilibrium", Fuel, 56(1977), 134
- Shinnar, R., "System assessment of novel concept in coal gasification", final report, DE-AC21-86Mc23280, 1990
- Shinnar, R., C. Tsiligiannis & I.H. Rinard, "Modeling, operability analysis and control of fluidized bed coal gasifier", research proposal, City College of CCNY, 1990
- Shinnar, R. & I.H. Rinard, "partial control of complex systems with large model uncertainties", research proposal, city college of CUNY, 1990
- Singh, C.P.P. & D.N. Saraf, "Simulation of high-temperature water-gas shift reactors", I&EC Proc. Dev. Des., 16(1977), 313
- Silverstein, J.L. & R. Shinnar, "Effect of design on the stability and control of fixed bed catalytic reactors with heat feedback 1. concepts", I&EC Proc. Dev. Des., 21(1982), 241
- Smith, I.W., "The combustion rates of coal chars: a review", 19th symp. on combustion/the combustion institute, 1982, 1045
- Wei, J., "A stoichiometric analysis of coal gasification", I&EC Proc. Dev. Des., 18(1979), 554
- Weimer, A.W. & D.E. Clough, AIChE Symp. Ser. 1981
- Wen, C.Y. & T.Z. Chuang, "Entrained-bed coal gasification modeling", I&EC Proc. Dev. Des., 18(1979), 684
- Weng, L., "Modeling and analysis of the design constraints for IGCC system", Doctor proposal, City College of CCNY, 1990
- Yo, H., J. Wei & M.M. Denn, "A model for moving-bed coal gasification reactors", AIChE J., 24(1978), 885
- Yoshida, K. & D. Kunii, "Complex reaction in fluidized beds", J. Chem. Eng. Japan, 7(1974), 34

PART II COMBINED CYCLE POWER PLANT

The purpose of this project was to investigate in detail the operation of the combined cycle power plant subsystem of the IGCC system in both the steady state and the dynamic modes. Various configurations and operating conditions were examined. Of particular interest were load following and part load operation. Published strategies for load following control were evaluated.

The tool these studies is a computer based dynamic simulation model of the combined cycle power plant using a DYSIM simulator. DYSIM is a dynamic simulator for the process developed by Professor Irven H. Rinard (Rinard 1992,1993) at the Department of Chemical Engineering, The City College of City University of New York. It is intended to be both a working simulator for the process system engineer and a research tool for dynamic simulator development. Additional dynamic models for key components in the combined cycle plant were developed and integrated into DYSIM. The simulator for the combined cycle power plant allowed prediction of plant performance for equipment configurations, different ambient conditions, and different control strategies. Only limited amount of model validation was possible.

There are two chapters involved in this part: Chapter 4 for the model development, including a linearized model for control system analysis, and Chapter 5 for control strategy evaluation and simulation results.

CHAPTER 4 MODEL DEVELOPMENT

1. INTRODUCTION

There are many reasons why a combined cycle power plant might operate in the load following mode. Since the demand for power changes daily and weekly, the power plants in a utility grid must operate at part load to match the fluctuation in power requirements. In such a load following mode, the power plant might run at less the full capacity, or beyond the rated conditions at extreme weather and heavy load. In the control of a plant engaged in load following, the dynamic or transient behavior is very important (Weber 1983).

It was therefore necessary to develop detailed dynamic models for the key components of such systems which could be used to predict the performance of the whole power plant under various operating conditions during the load following.

In this section we present a more detailed study of the dynamics and control of the combined cycle subsystem. We note that this is a problem of interest even if the fuel is not supplied by the gasifier. A typical combined-cycle system is shown in Fig.1. There are a growing number of such the combined cycle systems in service which use either natural gas or fuel oil as energy source. Most of these units are used to provide intermediate power, i.e., they are brought on line each weekday morning and are shut down the following evening. It is specially suitable for the gas turbine operation because of its fast response and ease of start up and shut down. Therefore, as more of other types of plants are committed to base load service, the need to use combined cycle systems for load following will increase.

The control systems of a power station must cope with the normal changes in demand for power in a load following mode, as well as with fault transients that arise locally or in the grid network. To achieve the designed life and to prevent damage to

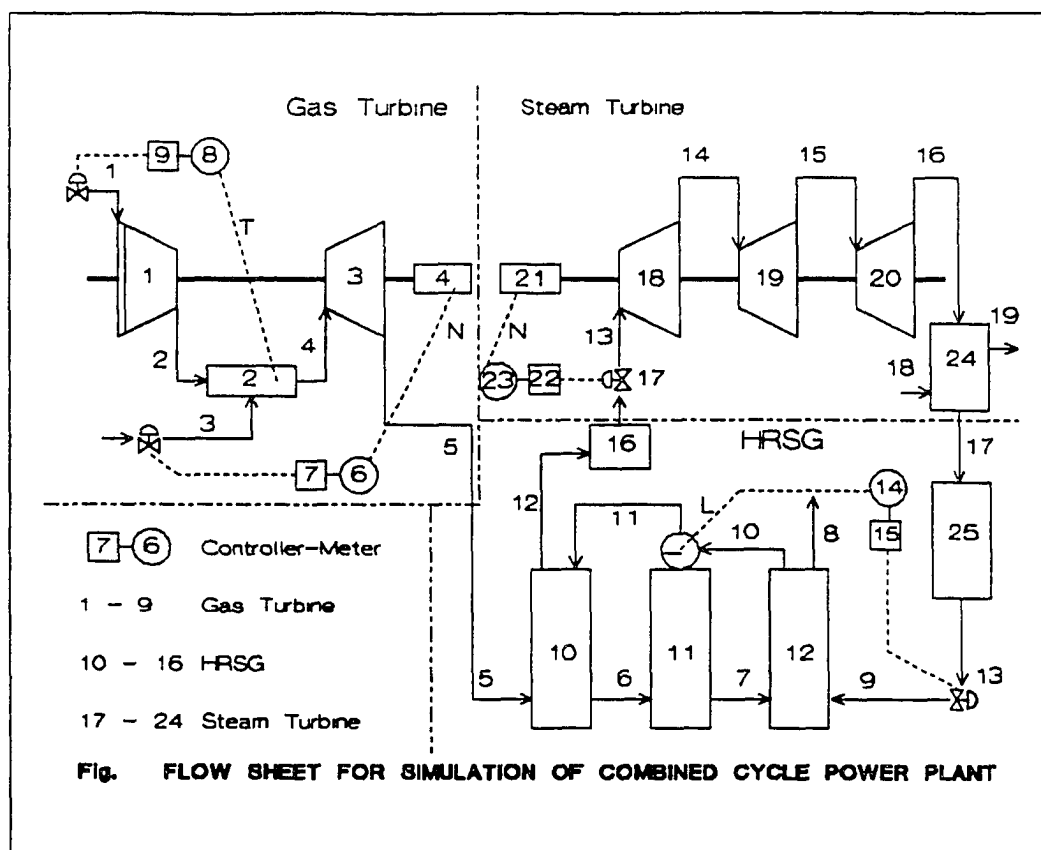


Figure 1 Simulation scheme of combined cycle power plant

equipment, the individual control loops must be co-ordinated so as to prevent the violation of operational constraints during these induced manoeuvre.

The major emphasis is the dynamic simulation and control system analysis for load following plants. A short review of this area follows:

Combined cycle power plants are commercially available. In a typical plant, multiple gas turbines are connected each through its own HRSG to a steam header. The steam is used to run steam turbine. Such a combined cycle system is shown as in Fig.1.

Much work has been done related to power station simulation and control, which includes fossil plants, combined cycle systems, HAT's (humid air turbines), IGCC integration, as well as some co-generation processes. The combined cycle technology has been successfully developed and widely applied in the world for more than twenty years. The IGCC concept is been actively researched, and some pilot plants have been developed

to test this idea. A commercial scale plant is expected in the near future (Holt, 1992).

The importance of power plant dynamics for safe and economic operation has been recognized throughout the industry (EPRI EL-6659). The major application of power plant computer modeling has been developed for engineering simulation studies and operator training simulators. Typically, engineering process simulation is performed using detailed models, since its purpose is to do an in depth analysis of complex process problems. A simulator based on engineering analysis supports a variety of plant design activities which include developing system configuration, and control philosophy. This analysis also aids in finding alternative solutions prior to implementation in the plant.

IGCC plant modeling can be applied to do in depth engineering studies. Overall design, planning, and training costs will be minimized by the availability and effective use of a simulator. Recognizing the potential benefits of developing of a powerful dynamic simulator to support commercialization of the IGCC concept, the work of simulator development has been taken up by EPRI (EPRI CS-6074).

In previous research projects, some simulation models related to IGCC plant have been developed. A paper discussing with the simulation objectives and the requirements of advanced cool gasification based power plants has been given by Weber, Quentin, and McDaniel (1983).

Computer simulation has played a unique role in the evolution of IGCC plant technology. Early simulation studies (EPRI AP-891, AP-892) of IGCC with moving bed and entrained fluidized bed gasifiers led to three control strategies for the IGCC process without requiring any buffers between the gasifier and the gas turbine. These control strategies are called as the turbine lead mode, the gasifier lead mode, and the mixed modes. These ideas were tested (EPRI AP-2503), and research showed dynamic response rates as high as 10% per minute. It was helpful in determining the functional requirements and the control strategies for the Cool Water IGCC demonstration plant

(EPRI AP-4832).

Later a MMS simulator (Modular Modeling System) was developed for dynamic simulation of fossil and nuclear power plants (EPRI CS-3016). It was implemented using forty component models including pipes, pumps, heat exchanger, superheaters, flash tanks, condenser, valves, control elements, furnaces etc. Each model is self-contained and may be interconnected with others in numerous arrangements to represent desired configurations for the power plant system.

A work for the IGCC system simulation based on early simulation studies (EPRI AP-2503) by applying the MMS has been done for the simulation and development of functional specifications for further use in all phases of a commercial 600 MW IGCC project (EPRI GS-6173) based on the Texaco gasifier. As these investigators recommended, further progress in the IGCC project needs an improvement in simulation capabilities.

For STAG combined cycle plant controls, flexibility and reliability were shown by Baker (1973, 1975). A paper related to start up based on overall control of multi-stage combined cycle plants was given by Takaoka (1983). Chu (1983) discussed configuration and sequence control for a combined cycle plant where four gas turbines were involved. ABB has declared a guarantee for high availability for combined cycle plant controls (1989).

Some of the work related to the component models has been widely published. Studies on dynamic models for steam and hydro turbines in power system have been reported in the IEEE committee report (1973), where typical time constants were also given. Fawke showed the use of digital computer methods for the prediction of gas turbine response. Lausterer (1984) gave a modular modelling approach applied to a Benson boiler. Recently, a state space boiler model for process optimization has been developed (Dieck-Assad, 1990). A paper discussing the multivariable robust control of

a boiler turbine system was given by Kwon (1989).

It is obvious that the detailed dynamic and control simulation models result in a simulator tool with high flexibility since the user can configure a specific system by using different combinations of the component models.

2. MODEL DESCRIPTION FOR DYSIM

In this section, we describe the computer based, dynamic simulation model, using the DYSIM simulator. The model is developed to simulate the dynamic performance of the combined cycle power plant. Detailed models are developed on the basis of basic physical, chemical, and engineering principles as well as data available in the literature.

DYSIM was used as the basic framework to integrate individual component models and form a complete simulation structure for the combined cycle power plant. In the following section, an overview of DYSIM is given. Following that are descriptions of the in-house models developed, and the power plant control strategies based on the linear model analysis.

2.1 DYSIM OVERVIEW

There are many programs that have been developed for the dynamic simulation of a specified process. As specified, they are of high complexity, have fixed structures or configurations, and are hard to be modified by the user.

In today's software market, more and more programs tend to provide for flexibility in system configuration by the user, such as ASPEN and MMS, where the user determines a scheme, sets the equipment types and their parameters, and then interconnects these by their material or electrical power signal flows.

DYSIM (Version 0A) is a dynamic simulator to provide such flexibility. It is intended to be both a working simulator for the process system engineer and a research

tool for dynamic simulator development. It consists of an equipment model library, a physical properties system, and a user interface for conducting a simulation. The processes are simulated by interconnecting the appropriate equipment models through an input file. The following models available in Version 0A:

TANK01:	Well-Mixed Tank
DRUM01:	Gas Drum
XC01:	Two-Fluid Heat Exchange
SPOOL1;	Liquid Flow Spool Piece
SPOOL2;	Vapor Flow Spool Piece
SPOOL3;	Flow Splitter Spool Piece
METER:	Generic Meter
CONPID:	Continues PID Control

Also because of the modular structure of the DYSIM, new models can be added to the model library in a straight forward manner. Different process configurations can be simulated by the user in a fast and easy way (Rinard 1992,1993).

2.2 IN-HOUSE COMPONENT MODELS

An extended set of models have been developed and incorporated into DYSIM Version 0A model library for those components specific to the power plant simulation:

GTURB:	Gas Turbine
COMP;	Air Compressor
GENER:	Generator
COMBUS:	Combustor
BOIL;	Boiler module
STURB:	Steam Turbine
COND:	Steam Condenser
MULT:	Signal Multiplier

The following is a brief description of each of these models.

COMP: Compressor

This equipment model simulates the fluid flow through a axial flow turbo

compressor.

A compressor is referenced to the stream flowing through it. The stream outlet pressure is determined by background pressure and flow resistance. The outlet temperature depends upon the number of stages of turbo fans as well as the turbine efficiency. The compressor may operate

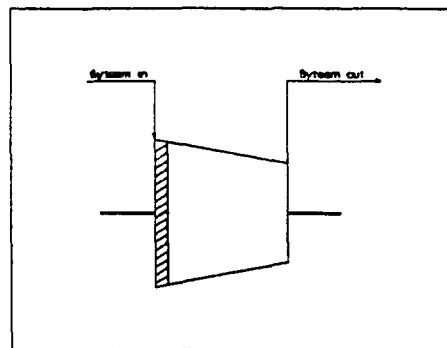


Figure 2 Air Compressor

isolated or co-shaft with a gas turbine. For a multi-stage operation, an inter-stage cooling system is designed for cooling gas temperature to inlet case, which is assumed.

An inlet guide vane is used to adjust the inlet gas flow rate in a range of 80-100% of maximum flow as suggested by the literature (EPRI AP-5027). This range can be modified by the user.

The flow rate is also affected by electrical frequency or rotational speed when the compressor is connected to the power station or electric generator. The model is based on a performance map selected from Laval's handbook (1970). The performance map shows the relation of flowrate-speed-pressure. It follows the turbofan law in the normal operating range.

$$F = f\left(\frac{N}{\sqrt{T}}, \frac{P_2}{P_1}\right) \quad (1)$$

By the data fitting, the following equation was determined:

$$m^* = N^*(A + B \cdot N^*) + C \cdot P_2^{*\frac{1}{\gamma-1}} \quad P_2^* = \frac{P_2}{P_{surge}} \quad (2)$$

Where $A=1.036$, $B=0.0738$, $C=-0.1174$. Fig.3 shows the resultary map.

Let m_o , $T1_o$, $P1_o$, N_o , and $P2_o$ be the design conditions. The dimensionless groups are

$$m^* = \frac{m}{m_o} \sqrt{\frac{TI}{TI_o}} \sqrt{\frac{PI_o}{PI}} \quad (3)$$

$$N^* = \frac{N}{N_o} \sqrt{\frac{TI_o}{TI}} \quad (4)$$

$$P_2^* = \frac{P_2/P_1}{P_{20}/P_{10}} \quad (5)$$

where $N^*=0.75-1.1$, $P_2^*=0.20-1.4$.

The surge pressure is given as

$$P_{surge} = (P_{20} - P_{10}) N^{*2.8} + P_{10} \quad (6)$$

where the index 2.8 came from data fitting (Perry's, 1963, Fig.6-42).

$$\text{For } 0 \leq \left(\frac{P_2}{P_{surge}} \right)^{\frac{\gamma}{\gamma-1}} \leq 1, \quad (7)$$

assume that the developed equation can be applied over a range of $N^*=0-1.3$.

GENER: Electric Generator

This equipment model simulates an electric generator when it is connected to gas turbine, steam turbine, or the other drivers. The number of drivers is limited to eight.

It becomes a motor in inverse operation where electric power is input.

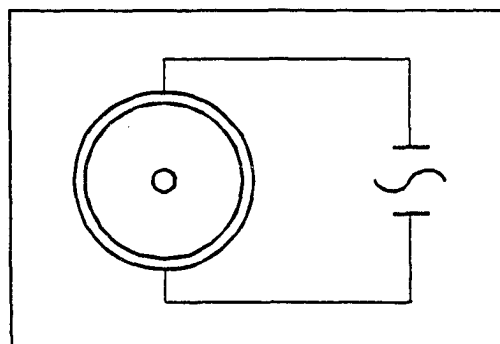


Figure 4 Electric Generator

The mechanical equations of a synchronous machine are very well established. The basic assumptions involved are: (1) rotor speed does not vary greatly from synchronous speed (this allows per unit power to be equated with per unit torque); (2) rotational

power losses are ignored. The basic equation governing rotational acceleration is

$$Mg \frac{d\omega}{dt} = P_s - P_e, \quad \text{or} \quad (8)$$

$$\tau \frac{d}{dt} \left(\frac{\omega}{\omega^s} \right) = \frac{P_s - P_e}{P^s} \quad \tau = \frac{Mg \omega^s}{P^s} \quad (9)$$

where Mg is the angular momentum, which is relatively constant (Cohen 1991).

Eddy currents induced in the rotor iron produce torques which oppose the rotation.

A deceleration is introduced to account this damping effect

$$\tau \frac{d}{dt} \left(\frac{\omega}{\omega^s} \right) = \frac{P_s - P_e}{P^s} - \beta \left(1 - \frac{\omega}{\omega^s} \right) \quad (10)$$

where β is so called as autodamping coefficient. In general it is about 1.5 to 2.0.

The τ is a summation of all of inertial momentum caused of the rotor parts in system. It is related to Mg/P^s . In general, it can be considered as a constant. The reason is that the bigger the P^s , the bigger the Mg will be. But in the modern turbines, τ becomes small as the P^s grows faster than Mg . Thus τ will become large when the station is on network.

GTURB: Gas Turbine

This equipment model simulates the fluid flow through a gas turbine. The gas turbine is treated as a spool piece, so that pressure drop varies with flow rate.

Turbine efficiency is assumed to be constant. The expansion of gas is treated as an isentropic process with inner efficiency. Combining the efficiency and pressure drop, the outlet temperature can be calculated.

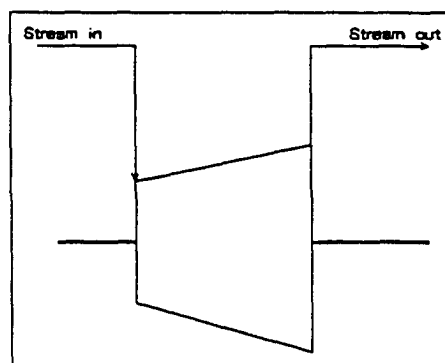


Figure 5 Gas Turbine (GT)

For the most part, gas flow is in the choked condition, so that the flow rate is determined by the choked flow relation

$$F_g = C_{vx} \frac{P_{in}}{\sqrt{T_{in}}} \quad (11)$$

We assume this choked flow model applied to the entire range. The power generated is calculated from enthalpy of the fluid

$$P_e = F_g (H_{in} - H_{out}) \quad (12)$$

Here, the exit temperature is calculated from

$$T_{out} = T_{in} [1 - \eta_{GT} (1 - \gamma)] \quad \gamma = \left(\frac{P_{in}}{P_{out}} \right)^{-\alpha} \quad \alpha = \frac{C_p - C_v}{C_p} \quad (13)$$

The turbine efficiency is assumed as constant. The data of turbine rotational inertia will be transferred to the electrical generator for the rotational speed simulation.

For the case of warm up operation, there is heat transfer between hot gas and turbine metal. This was simulated, where it was assumed that the gas exit temperature is under equilibrium with that of metal. The time constant of this process at rated conditions is about 0.1 min. At start up conditions it is about 0.3-0.4 min (EPRI AP-5975), and is affected by the gas flow rate.

BOIL: Drum Boiler for Steam Generation

This equipment model includes the steam drum and recirculating boiler piping. A forced circulation pump is assumed. The flow rate of circulated fluid is specified by the user. It should be about 10 times rated steam rate.

In the heated piping, the circulated liquid

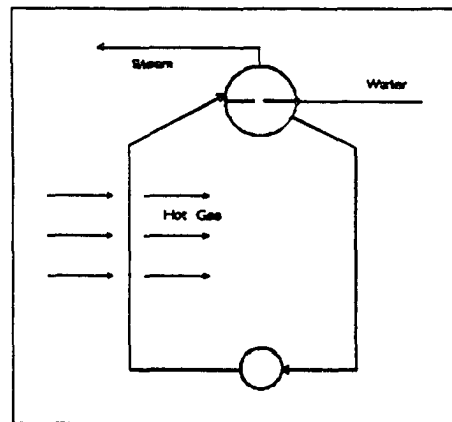


Figure 6 Boiler

is first heated to boiling temperature, and then vaporized to steam. The void fraction in the heated pipes is a state variable, so that it varies with time. There are two model modes. One is pure liquid heat up during start up. The other is two phase boiling at operating conditions. Liquid level and pressure inside drum change with time.

These should be controlled. The model also can be used for start up operation. The model demonstrates the inverse response of level to heat input.

A two-phase model is applied inside the boiler tubes. The steam quality is assumed to vary linearly along the tubes. The void fraction is assumed to correspond to no-slip at the average steam quality. As the boiling process has very big heat transfer coefficient, the resistance inside tube is neglected. The heat content in downcomer is neglected too, as its specific surface is small comparing to heating pipes.

Due to the static head of the external downcomer, the liquid leaving circulating pump is subcooled. It must first be heated to the boiling point up and then be vaporized.

The model equations include steam and liquid in drum, fluid in heating pipes. The overall balances are:

Mass balance

$$\frac{dV\rho}{dt} - \Sigma F \quad V\left(\frac{\partial\rho}{\partial T}\right)\left(\frac{dT}{dt}\right) + V\left(\frac{\partial\rho}{\partial P}\right)\left(\frac{dP}{dt}\right) + \rho\left(\frac{\partial V}{\partial Z}\right)\left(\frac{dZ}{dt}\right) - \Sigma F \quad (14)$$

Energy balance

$$\frac{d(V\rho\cdot U)}{dt} - \Sigma(F\cdot H) \quad (15)$$

$$U\frac{d(V\rho)}{dt} + V\rho\left(\frac{\partial U}{\partial T}\right)\left(\frac{dT}{dt}\right) - \Sigma(F\cdot H) \quad (16)$$

$$V\rho\left(\frac{\partial U}{\partial T}\right)\left(\frac{dT}{dt}\right) - \Sigma[F\cdot(H-U)] \quad (17)$$

Followings are balances for two phases:

Liquid in drum

$$\frac{dT}{dt} = \frac{\Sigma F(H-U)}{V \cdot \rho \left(\frac{\partial U}{\partial T} \right)} \quad \frac{d\rho}{dt} \approx 0 \quad (18)$$

Vapor in drum

$$\frac{dT}{dt} = \frac{\Sigma F(H-U)}{V \cdot \rho \left(\frac{\partial U}{\partial T} \right)} \quad \left(\frac{dZ}{dt} \right)_V = \left(\frac{dZ}{dt} \right)_L \quad (19)$$

$$\frac{dP}{dt} = \frac{\Sigma F - V \left(\frac{\partial \rho}{\partial T} \right) \left(\frac{dT}{dt} \right) + \rho \left(\frac{\partial V}{\partial Z} \right)_L \left(\frac{dZ}{dt} \right)}{V \left(\frac{\partial \rho}{\partial P} \right)} \quad (20)$$

To Heat transfer

$$q = F_g \cdot C_{Pg} \cdot \Delta T_g = h \cdot A \cdot \Delta T_{LMTD} \quad (21)$$

$$T_{g2} = T_m + (T_{g1} - T_m) e^{-\frac{h \cdot A}{F_g \cdot C_{Pg}}} \quad (22)$$

$$q = F_g \cdot C_{Pg} \cdot (T_g - T_m) (1 - e^{-\frac{h \cdot A}{F_g \cdot C_{Pg}}}) \quad (23)$$

where $T_m = T_B$ for boiling process, $T_m = (T_{in} + T_{out})/2$ for heating up.

Voidage

It is assumed that the steam quality X varies linearly along the tube and the void fraction can be evaluated at the average value of vapor fraction.

Assuming no-slip two phase flow, we get

$$\bar{\epsilon} = \frac{X/2}{X/2 + (10X/2) \frac{\rho_v}{\rho_l}} \quad \frac{d\bar{\epsilon}}{dX} = \frac{\frac{1}{2} \frac{\rho_v}{\rho_l}}{[X/2 + (1-X/2) \frac{\rho_v}{\rho_l}]^2} \quad (24)$$

where the X is steam fraction by wt at the exit of the boiler tubes. It is a state variable.

Heating pipe

$$\frac{d}{dt}(V\bar{e}\cdot\rho_v+(1-\bar{e})\cdot V\cdot\rho_L)-\Sigma F \quad (25)$$

$$\frac{dX}{dt} = \frac{\Sigma F}{V(\rho_v - \rho_L)(\frac{d\bar{e}}{dX})} \quad (26)$$

$$\frac{d}{dt}[V(1-\bar{e})\cdot\rho_L\cdot U_L + V\bar{e}\cdot\rho_v\cdot U_v + V_w\cdot\rho_w\cdot U_w] - \Sigma F\cdot H + q \quad (27)$$

where W means tube wall. The equation can be written as

$$J_e(\frac{d\bar{e}}{dt}) + J_H(\frac{dT}{dt}) - \Sigma F\cdot H + q \quad (28)$$

The conditions are

$$\frac{dT}{dt} - (\frac{\partial T_B}{\partial P})(\frac{dP}{dt}) \quad (X>0) \quad (\text{trackboilingpoint}) \quad (29)$$

$$\frac{d\bar{e}}{dt} = 0 \quad (X=0) \quad (\text{subcoolesliquidnoboiling}) \quad (30)$$

One of two variables above can be solved out when X is given.

As the liquid level in drum is dynamic variable, the steady state operation of boiler model need a flow rate controller for the liquid feed. For the drum pressure control, a valve is also needed in output stream.

COND: Steam Condenser

This equipment model simulates the behavior of steam condensation on tube bank. There are two input streams and two output streams.

An important assumption is that the vapor and the condensed steam are in equilibrium.

The heat transfer coefficient is calculated as a built-in model for the water-steam system. The user can give it as input value for the other fluids.

The model equation for the vapor phase

$$\frac{d(V \cdot \rho)}{dt} = \Sigma F - F \cdot X - \Gamma \quad (31)$$

For the coolant

$$\frac{d(V \cdot \rho \cdot U)}{dt} = \Sigma F \cdot H - F \cdot \Sigma H \quad (32)$$

The equations can be rewritten as

$$V \cdot \left[\left(\frac{\partial \rho}{\partial T} \right) \left(\frac{dT}{dt} \right) + \left(\frac{\partial \rho}{\partial P} \right) \left(\frac{dP}{dt} \right) \right] = F \cdot X - \Gamma \quad (33)$$

$$V \cdot \rho \cdot \left(\frac{\partial U}{\partial T} \right) \left(\frac{dT}{dt} \right) = F \cdot \Sigma H \quad (34)$$

$$\frac{dT_s}{dt} = \left(\frac{\partial T_s}{\partial P_s} \right) \left(\frac{dP_s}{dt} \right) \quad (35)$$

The amount of steam condensed is

$$\Gamma = \frac{Q}{\lambda} \quad (36)$$

$$Q = U \cdot A \cdot \Delta T_{LMTD} \quad (37)$$

$$U = 0.725 \left[\frac{g \cdot \lambda \cdot \rho_L^2 \cdot k_L^3}{\mu_L \cdot N \cdot d \cdot (T_s - T_w)} \right]^{1/4} \quad (38)$$

where the SI units are applied, λ is the heat of condensation, Q is the total heat transferred, N is the number of tubes and d is tube diameter.

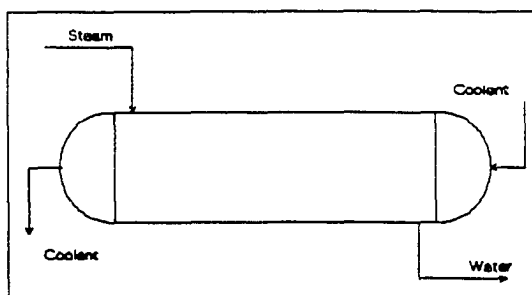


Figure 7 Condenser

STURB: Steam Turbine

This equipment model simulates the steam flows through a steam turbine. In considering the cases of multistage, reheat, and two steam drums, the pressure is set as a dynamic variable. The fluid flow through the turbine is treated as through a vessel, (including

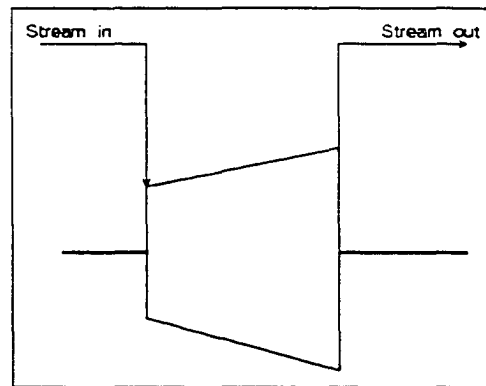


Figure 8 Steam turbine (ST)

connecting piping), so that pressure is a state variable, and steam flow rate is related to pressure drop. The volume should be chosen to avoid stiffness in the dynamic pressure calculation.

There are two model options, one for HP, where the flow through the turbine is treated as spool piece, and another for LP, where the flow is choked flow (Jian 1985).

The exit temperature is related to exit pressure and inner efficiency. For the last stage in most of cases, steam is partially condensed. This is identified by a built-in model. The exit pressure and temperature are in equilibrium. The wet fraction is calculated based on expansion condition and inner turbine coefficient, which is a function of pressure and temperature.

The turbine efficiency is function of inlet steam P and T. Here we assumed it is

$$\Delta \eta_t = \alpha \Delta T + \beta \Delta P \quad (39)$$

Some of properties for steam and water are treated as

$$C_p = C_p(T, P) \quad H = H(T, P) \quad S = S(T, P) \quad (40)$$

The pressure drop is fixed by

$$\frac{d(V \cdot p)}{dt} = \sum F \quad (41)$$

$$F_o = C_{vx} \sqrt{\rho \cdot \Delta P} \quad \text{or} \quad F_o = C_{vx} \frac{P_{in}}{T_{in}} \quad (42)$$

where one is for HP, and another for LP.

The calculation procedure is (1) to get reversible exit T from exit P, (2) to get real exit T from the reversible T and the inner efficiency, and (3) to calculate changes.

Mult Signal Multiplier

This is an utility feature used for multiplying mass flow or electrical signals. It is used for the case of parallel operation if all gas turbines are assumed to operate identically.

3. LINEAR MODEL DEVELOPMENT

In this section a linear model is developed to provide the basis for the analysis of the combined cycle (CC) system control. Even in a reduced form, it will be seen that the system is multivariable. It is known that multivariable system, can exhibit complex and interacting behavior. Controllability and stability analysis are therefore included in following section. A diagonal control structure (multiloop SISO) and the BLT (Bigest Log Tuning, Luyben 1990) method will be used for the controller design. Matrix or state space methods will be applied in this part.

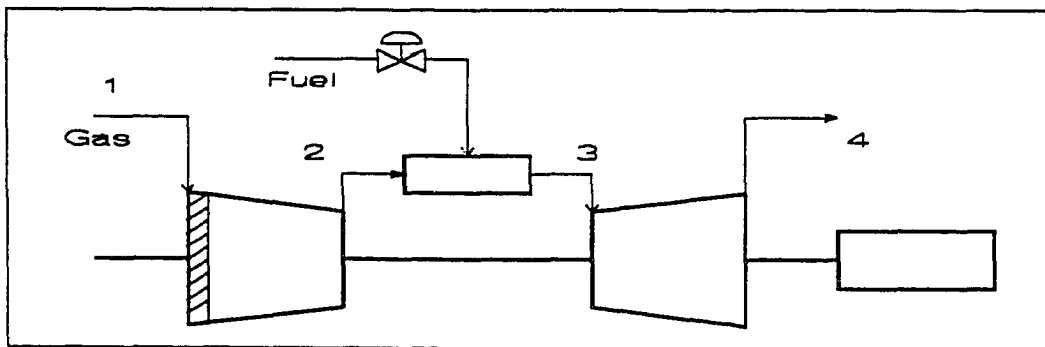


Figure 9 Scheme of Gas Turbine System

The gas turbine system (Fig. 9) includes an air compressor, a combustion chamber, the gas turbine, and an electrical generator. Two major control loops are of interest. One

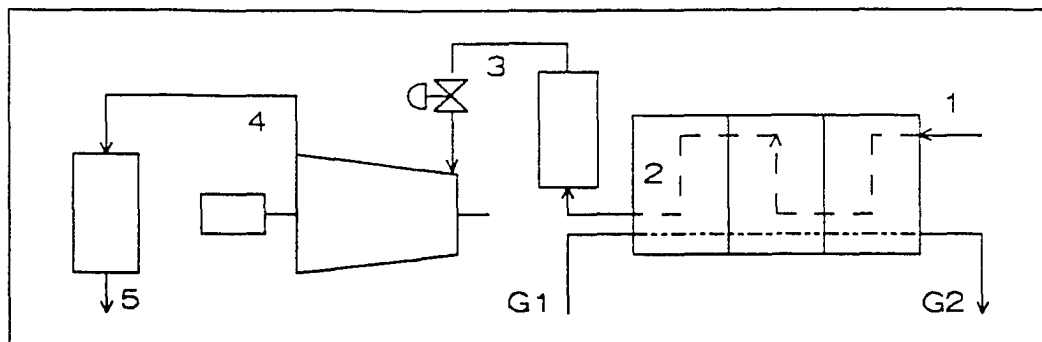


Figure 10 Model base for Steam Cycle Linear Model

is the electrical frequency/power control by fuel valve adjustment; the other, temperature control by air feed rate through an air inlet guide vane (IGV) for gas turbine high temperature limitation. They are the real case in power plant.

The steam turbine system (Fig. 10) includes a heat recovery steam generator (HRSG), the steam turbine, a steam condenser, and an electrical generator, as shown in Fig.1. The model is a simple no-reheat and no-extraction steam turbine system. A very important assumption is that the system is under sliding pressure operation, which means different pressures, as well as different temperatures, corresponding to different loads. Special attention has been paid to boiler level model due to its response characteristic.

3.1. Basic Equations, Assumptions and Simplifications

There are state equations relating electrical frequency N , heat rate Q , pressure P , temperature T , liquid level Z , vapor fraction ϵ or x , and valve position X to the system input variables. All of them are balance equations on mass and energy.

Given the very small time-constant, quasi-steady state is assumed for the air compressor, the combustion chamber, the gas turbine, and the steam turbine. These equipment items are treated as spool pieces. For example, for the gas turbine, the flow equation becomes

$$F = C_X \sqrt{P^2 - 1} \approx C_X P \quad C_X = F^*/P^* \quad (43)$$

Electrical Generator

$$\frac{d}{dt} \left(\frac{1}{2} I \omega^2 \right) = W_{GT} - W_{GE} \quad N = \frac{\omega}{2\pi} \quad W = M \omega \quad (44)$$

$$I \omega \frac{d\Delta \omega}{dt} = \frac{\partial W_{GT}}{\partial \omega} \Delta \omega + \frac{\partial W_{GT}}{\partial W_{in}} \Delta W_{in} - \frac{\partial W_{GE}}{\partial \omega} \Delta \omega \quad \frac{\partial W_{GT}}{\partial W_{in}} = \frac{W_{GT}^*}{W_{in}^*} \quad (45)$$

$$\tau_\omega \frac{d}{dt} \left(\frac{\Delta \omega}{\omega} \right) = -\beta \frac{\Delta \omega}{\omega} + \frac{\Delta W_{in}}{W} \quad \tau_\omega = \frac{I \omega^2}{W} \quad \beta = \frac{\left(\frac{\partial W_{GE}}{\partial \omega} - \frac{\partial W_{GT}}{\partial \omega} \right) \omega}{W} \quad (46)$$

where τ_ω is assumed a constant, (7-10 s), although it depends on turbine size and load, as well as load type; β is called autodamping coefficient, which is caused by generator; W_{in} is shaft work input, it becomes $W_{GT} - W_{GC}$ for gas turbine cycle; M is the momentum of rotation.

Combustion Chamber

Assumption is $F_g \gg F_{gc}$. By which, gas exit temperature after combustion chamber can be expressed as

$$(F_g + F_f) C_{Pg} T_3 \approx F_g C_{Pg} T_3 = F_g C_{Pg} T_2 + F_f (C_{Pf} T_f + Q_{LHV}) \approx F_g C_{Pg} T_2 + F_f Q_{LHV} \quad (47)$$

$$\Delta(F_g C_{Pg} T_3) = \Delta(F_g C_{Pg} T_2) + Q_{LHV} \Delta F_f \quad \underline{\underline{(48)}}$$

Gas Turbine and Air Compressor

Temperature changes and energy balances for air through the compressor and gas turbine when thermal inner efficiency η_g and η_c used are as

$$W_{GT} = F_g C_{Pg} (T_3 - T_4) = F_g C_{Pg} T_3 (1 - \gamma) \eta_g \quad (49)$$

where spool piece character equation $\Delta F_g = C_X \Delta P$ is used. T_1 is air inlet temperature, it

$$W_{GC} - F_g C_{Pg} (T_2 - T_1) - F_g C_{Pg} T_1 \left(\frac{1}{\gamma} - 1 \right) \frac{1}{\eta_c} \quad (50)$$

$$\gamma - P^{-\alpha} \quad \alpha = 1 - \frac{C_v}{C_p} \quad \Delta \gamma = -\alpha P^{-\alpha} \frac{\Delta P}{P} = -\alpha P^{-\alpha} \frac{\Delta F_g}{F_g} \quad (51)$$

is a constant in a short time. At the front of compressor, there is an inlet guide vane (IGV) to control air feed rate in the range of 80-100%. We treat it as a valve. The shaft work in the gas cycle is

$$W_{GT} - W_{GC} - F Q_{LHV} (1 - \gamma) \eta - F_g C_{Pg} T_1 \left(\frac{1}{\gamma} - 1 \right) \left(\frac{1}{\eta} - \eta \right) \quad (52)$$

$$\Delta W_{in} - Q_{LHV} F_f \eta \left[(1 - \gamma) \frac{\Delta F_f}{F_f} - \Delta \gamma \right] - F_g C_{Pg} T_1 \left(\frac{1}{\eta} - \eta \right) \left[\left(\frac{1}{\gamma} - 1 \right) \frac{\Delta F_g}{F_g} - \frac{\Delta \gamma}{\gamma^2} \right] \quad (53)$$

Valve

The valve is assumed as a first order lag as follows:

$$F - F^* X \quad \Delta F - F^* \Delta X \quad \tau_x \frac{dX}{dt} = X^* - X \quad \tau_x \frac{d\Delta X}{dt} = \Delta X^* - \Delta X \quad (54)$$

When the fan law is applied to the air compressor, where is the ϕ adjustable range of the IGV, it becomes

$$\frac{F_g}{F_g^*} = \frac{N}{N^*} (X\phi + 1 - \phi) \quad \frac{\Delta F_g}{F_g^*} = \frac{N}{N^*} \left[\phi \Delta X + (X\phi + 1 - \phi) \frac{\Delta N}{N} \right] - \phi \Delta X_g + \frac{\Delta \omega}{\omega} \quad (55)$$

Condenser

The model consists of cooling water and vapor phase. For a vapor phase

$$\frac{d(V\rho_v \lambda)}{dt} = Q_4 - Q_v = F_4 x \lambda - hA(T_v - T_w) \quad Q_v^* = Q_4^* \quad T_v^* = T_v - T_w \quad (56)$$

where A comes from the Antoine equation $\ln P = A/T + B$ for the vapor-liquid

$$\tau_v \frac{d}{dt} \left(\frac{\Delta T_v}{T_v^*} \right) - \left(\frac{\Delta F_4}{F_4} + \frac{\Delta x}{x} \right) - \left(\frac{\Delta T_v}{T_v^*} - \frac{\Delta T_w}{T_w^*} \right) \quad \tau_v = \frac{V \rho_v}{F_4 x} \left(\frac{A}{P} - 1 \right) \frac{T_v^*}{T_v} \quad (57)$$

$$\frac{\Delta x}{x} = \frac{1 - \eta_{ST}}{x} \frac{\partial(S_v/S_{VL})}{\partial(T_v/T_v^*)} \frac{\Delta T_v}{T_v^*} - k \frac{\Delta T_v}{T_v^*} \quad (58)$$

equilibrium, λ is assumed as a constant. For the cooling water phase where i is for feed in

$$V \rho_L C_P \frac{dT_w}{dt} - Q_i - Q_o + Q_r - F_w C_P (T_i - T_w) + Q_r \quad T_w^* = T_w - T_i \quad (59)$$

$$\tau_w \frac{d}{dt} \left(\frac{\Delta T_w}{T_w^*} \right) - \left(\frac{\Delta F_w}{F_w} - \frac{\Delta T_i}{T_w^*} - \frac{\Delta T_w}{T_v^*} \frac{T_v^*}{T_w^*} \right) + \left(\frac{\Delta T_v}{T_v^*} - \frac{\Delta T_w}{T_w^*} \right) \quad \tau_w = \frac{V \rho_w C_P T_v^*}{Q_r} \quad (60)$$

Steam Turbine

The energy balance for steam turbine is

$$W_{ST} - Q_3 - Q_4 - F_3(H_3 - H_4) \quad F_4 - F_3 = C_X X P \quad H_4 - H_L + \lambda x = \lambda x \quad (61)$$

$$\Delta W_{ST} - Q_4 \frac{\Delta F_3}{F_3} - F_3 \Delta H_4 \quad \frac{\Delta F_3}{F_3} = \frac{\Delta P}{P} + \frac{\Delta X}{X} \quad (62)$$

where x is vapor fraction of exit steam. Considering the steam turbine inner efficiency,

$$\eta_{ST} = \frac{\Delta H_r}{\Delta H_i} = \frac{\lambda(\Delta x_i + \Delta x)}{\lambda \Delta x_i} = 1 - \frac{\Delta x}{\Delta x_i} \quad x_i = \frac{S_v - S_i}{S_v - S_L} = \frac{S_v - S_i}{S_{VL}} \quad (63)$$

where r means real case, i for ideal case. Combining above equations, it gives

$$\Delta H_4 - \lambda \Delta x - \lambda(1 - \eta_{ST}) \frac{\partial S_v}{\partial T} \frac{\Delta T}{T_v^*} \frac{T_v^*}{S_{VL}} = k \lambda x \frac{\Delta T_v}{T_v^*} \quad (64)$$

where T_v is temperature for condenser. It is almost constant if condenser is operated under constant pressure which is corresponding to a good control of condensation

temperature by adjusting feed of cooling water. As a big holdup of condenser, the operation is stable.

Steam Drum

As the pressure change is much faster than that of the temperature when the flow rate changes, the drum is treated as a mass holder, which includes the steam header and boiler drum together, then

$$\frac{d}{dt}(\rho V) = mF_2 - F_3 \quad \tau_P \frac{d}{dt}\left(\frac{P}{P^*}\right) = \frac{mF_2 - F_3}{F^*} \quad \tau_P = \frac{VP^*}{F^*} \frac{\partial \rho}{\partial P} \quad (65)$$

$$\tau_P \frac{d}{dt}\left(\frac{\Delta P}{P}\right) = \frac{m\Delta F_2 - \Delta F_3}{F^*} \quad F_3 = C_X XP \quad \Delta F_3 = C_X(X\Delta P + P\Delta X) \quad (66)$$

where m is the number of HRSG's in one combined cycle. One important thing is that the τ_P should include the phase change. For example, when the steam flow rate increases, it causes a pressure drop in the drum. This pressure drop reduces the boiling temperature, and more hot water will be evaporated. As pressure decreases, the hot water will gradually be vaporized. So that the real τ_P is much bigger than that of the drum vapor holdup itself. The C_X consists of the resistance for both the valve and the steam turbine.

HRSG

Here the HRSG model is a global model, which lumps superheater, boiler, and economizer together. The following assumptions are made for a sliding pressure operation, where the pressure changes with the load for a partial load operation.

$$\frac{d(\sum \rho VU)}{dt} = \tau_B \frac{dQ_2}{dt} = Q_1 - Q_2 + Q_r \quad (67)$$

$$\therefore Q_{G2} \propto Q_{G1} \quad \therefore Q_{G2} = \beta Q_{G1} \quad \text{and} \quad Q_r = Q_{G1} - Q_{G2} = Q_{G1}(1 - \beta) \quad (68)$$

as Q_1 is much less than Q_2 , and then by recalling the equation for heat transfer, we get

$$\tau_B \frac{d}{dt} \left(\frac{\Delta Q_2}{Q_2} \right) = - \frac{\Delta Q_2}{Q_2} + \frac{\Delta Q_r}{Q_r} \quad (69)$$

It is noted that as H is a function of T & P , the T varies with P for a sliding pressure operation, so that it is assumed H is function P only.

$$\Delta H_2 = \frac{\partial H}{\partial P} \Delta P = \frac{\partial \lambda}{\partial P} \Delta P \quad \lambda = 468.0 - 1.50P \quad (70)$$

$$Q_2 = F_2 H_2 \quad F_2 = F_B x \quad \Delta Q_2 = H_2 \Delta F_2 = H_2 (x \Delta F_B + F_B \Delta x) \quad (71)$$

where the λ is heat of evaporation, and can be well expressed by above formula.

Liquid Level

Let F_B be the two phase flow rate coming from boiler tubes. The liquid level model can be expressed as

$$\frac{d(\rho_L V)}{dt} = F_1 - F_C + F_B(1 - x) \quad (72)$$

$$\tau_Z \frac{d}{dt} \left(\frac{Z}{Z^*} \right) = \frac{F_1 - F_C + F_B(1 - x)}{F^*} \quad \tau_Z = \frac{AZ^* \rho_L}{F^*} \quad (73)$$

$$\tau_Z \frac{d}{dt} \left(\frac{\Delta Z}{Z} \right) = - \frac{\Delta F_1 - F_C \Delta x + (1 - x) \Delta F_B}{F^*} \quad (74)$$

If a two phase model does not apply to the boiler, the 'shrinking-swelling' phenomena will be missed. It is noted that F_B is different from F_C . The term $F_B(1 - x)$ composes two opposite effects, $d\epsilon/dt$ and $(1 - x)$, where when $(1 - x)$ begins to change, $d\epsilon/dt$ is maximum and when $(1 - x)$ tends to a new steady state point, $d\epsilon/dt$ equals to zero.

Vapor Fraction

The state equation developed here for vapor fraction inside the boiler tubes is to simulate an inverse phenomena for the liquid level in boiler, which is called a 'shrinking-swelling' effect. Any disturbance which causes ϵ to change by a pressure change, causes a two-phase flow rate change. These changes affect to liquid level inversely. This is important in design of boiler itself to limit inverse gain or in control of liquid level. Sometimes an advanced controller is desired.

When vapor density is constant in a certain time, and $x \ll 1$, we have

$$\frac{d}{dt}(\rho_v \epsilon V + \rho_L (1 - \epsilon) V) = F_C - F_B \quad -\tau_c \frac{d\epsilon}{dt} = \frac{F_C - F_B}{F_C} \quad \tau_c = \frac{V \rho_L}{F_C} \quad (75)$$

$$\frac{d}{dt}(\rho_v V \epsilon) = F_v - x F_B \quad \tau_\epsilon \frac{d\epsilon}{dt} = \frac{F_v - x F_B}{F^*} \quad \tau_\epsilon = \frac{\rho_v V}{F^*} \quad (76)$$

$$\tau_\epsilon \frac{d\Delta\epsilon}{dt} = \frac{\Delta F_v - x \Delta F_B - F_B \Delta x}{F^*} \quad \tau_c \frac{d\Delta\epsilon}{dt} = \frac{\Delta F_B}{F_C} \quad (77)$$

$$\frac{\Delta F_v}{F_v} = \frac{\Delta Q_g}{Q_g} - \frac{F_C C_p (T_C - T_w)}{\lambda F_v} \frac{\Delta T_C}{T_C - T_w} = \frac{\Delta Q_g}{Q_g} \quad (78)$$

Combining above two equations, one will get

$$(\tau_\epsilon F^* + x \tau_c F_C) \frac{d\Delta\epsilon}{dt} = \Delta F_v - F_C \Delta x \quad (79)$$

where F_C is for water forced-circulation rate, it is a constant. It is noted that the water feed rate change will affect the circulated water temperature if the water is directly feed to boiler, and so the Q/λ . But if a preheater such as economizer is used, the effect is filtered by preheater and boiler liquid, where the heat transfer is almost constant at preheater for deferent water feed rate.

One assumption is that the x in flow out is double to that in tubes. For no-slip

two-phase flow, we have

$$x = \frac{2e\rho_v}{2e\rho_v + (1-e)\rho_L} \sim \frac{2e}{1-e} \frac{\rho_v}{\rho_L} \quad (80)$$

$$\Delta x = \frac{2}{(1-e)^2} \frac{\rho_v}{\rho_L} \Delta e = b \Delta e \quad \frac{\Delta x}{x} = \frac{1}{1-e} \frac{\Delta e}{e} \quad \frac{\Delta e}{e} = \frac{2}{x\rho_v/\rho_L + 2} \frac{\Delta x}{x} \quad (81)$$

3.2. Transfer Function

Gas Turbine

The transfer functions for the gas turbine system are

$$F_g C_{Pg} \bar{T}_3 = H_f \left(\frac{\bar{F}_f}{F_f} - \frac{\bar{F}_g}{F_g} \right) - H_1 \frac{\bar{\gamma}}{\gamma^2} - H_f \frac{\bar{F}_f}{F_f} - (H_f - H_1 \frac{\alpha}{\gamma}) [\phi \bar{X}_g + \frac{\bar{\omega}}{\omega}] \quad (82)$$

$$H_f - F Q_{LHV} \quad H_1 - F_g C_{Pg} T_1 \quad \frac{\bar{F}_g}{F_g} = \bar{X}_g \phi + \frac{\bar{\omega}}{\omega} \quad (83)$$

$$\frac{\bar{\omega}}{\omega} = \frac{\frac{H_f}{W} (1-\gamma) \eta \frac{\bar{F}_f}{F_f} - [\frac{H_1}{W} (\frac{1}{\eta} - \eta) (\frac{1}{\gamma} - 1) - \frac{\eta \gamma^2 H_f - (1/\eta - \eta) H_1}{W} \frac{\alpha}{\gamma}] \phi \bar{X}_g}{\tau_\omega S + \beta + [\frac{H_1}{W} (\frac{1}{\eta} - \eta) (\frac{1}{\gamma} - 1) - \frac{\eta \gamma^2 H_f - (1/\eta - \eta) H_1}{W} \frac{\alpha}{\gamma}]} \quad (84)$$

Δ has been dropped on the perturbation variables; a bar over a variable indicates a Laplace transform for a perturbation variable.

Steam Turbine

$$\bar{F}_2 - \bar{F}_B + \bar{x} = e \tau_c \bar{S} \bar{e} + \frac{1}{1-e} \bar{e} = (e \tau_c \bar{S} + \frac{1}{1-e}) \bar{e} \quad (85)$$

$$\frac{\bar{e}}{e} = \frac{\frac{\bar{F}_v}{F_v}}{e(\tau_e + \tau_c) \bar{S} + b e/x} \quad \frac{\bar{Z}}{Z} = \frac{\frac{\bar{F}_1}{F_1} + \frac{(1-x) \tau_c \bar{S} - b}{2\rho_v/\rho_L + 1} \frac{\bar{e}}{e}}{\tau_z \bar{S}} \quad (86)$$

$$\frac{\bar{P}}{P} = \frac{m \frac{\bar{F}_2}{F_2} - \frac{\bar{X}_s}{X_s}}{\tau_P \bar{S} + 1} \quad \frac{\bar{\omega}}{\omega} = \frac{\frac{\bar{P}}{P} + \frac{\bar{X}}{X} - \frac{Q_4}{W} k \frac{\bar{T}_v}{T_v}}{\tau_\omega \bar{S} + \beta} \quad (87)$$

$$\frac{\bar{T}_w}{T_w} = \frac{-\frac{\bar{F}_w}{F_w} + \frac{\bar{T}_l}{T_l} + \frac{\bar{T}_v}{T_v}}{\tau_w S + T_v^*/T_w^* + 1} \quad \frac{\bar{T}_v}{T_v} = \frac{(\tau_w S + T_v^*/T_w^* + 1) \frac{\bar{F}_4}{F_4} - \frac{\bar{F}_w}{F_w} - \frac{\bar{T}_l}{T_l}}{(\tau_w S + T_v^*/T_w^* + 1)(\tau_v S + 1 - k) - 1} \quad (88)$$

The models have been developed and they will be used for the control study and dynamic evaluation for the next chapter.

CHAPTER 5 SIMULATION OF COMBINED CYCLE PLANT

In this chapter the results of the combined cycle power plant simulation runs which were used to investigate dynamic performance and system controllability are presented and analyzed. It includes controller configurations and their analysis based on the linearized model. The control structures are then applied to dynamic simulation by DYSIM. The results are divided into three categories: stand alone options of gas turbine, HRSG, and steam turbine subsystems; the combined cycle power plant option; and some start up options. Parameter settings and base case calculations for using DYSIM are evaluated and listed in appendix, where some off-design calculations are given for the study of plant performance at the steady state operations.

1. CONTROL ANALYSIS

The control analysis is based on the linearized model and transfer functions developed at Chapter 4. The system to be analyzed is shown in Fig.7, which is based on published control strategies (Knowles 1990, Kelhofer 1990). The system to be simulated is a multi-variable system as discussed in Chapter 4.

The basic procedure to be followed for such a multi-variable system is that of validation of controller configuration and evaluation of controller parameter settings. The former is done by an interaction analysis, the latter by Luyben's (1990) tuning method (BLT).

BLT means biggest log-modules tuning method and is developed for multivariable-input multivariable-output (MIMO) systems. The BLT method allows a MIMO system to be tuned by a classical tuning for each G_{ij} , a diagonal multi-loop SISO (single input single output) system. Then an F detuning factor is applied to the whole system tuning based on the BLT criteria. The BLT method has been successfully used for many cases

(Luyben 1990).

By means of the matrix method, the Relative Gain Array is used to check the control system configuration and interaction effects, and the Niederlinski Index is used to validate the controllability.

1.1 EVALUATION OF TRANSFER FUNCTION

Based on the linear model developed in last chapter, the transfer function can be evaluated by following data:

Gas turbine

$\tau_w = 20$ s	$\gamma = 0.518$	$H1/Wg = 1.76$	$Fg = 222$ kg/s
$\beta = 2.0$	$\alpha = 0.2857$	$Hf/Wg = 3.73$	$Ff = 3.5$ kg/s
$m = 4$	$\eta = 0.85$	$H3/Wg = 7.61$	$C_x = 22.2$
	$P = 10$ atm	$T1 = 300$ K	
	$P1 = 1$ atm	$T3 = 1300$ K	

Steam turbine

$\tau_p = 200$ s	$\varepsilon = 0.726$	$H4/Ws = 2.30$	$F3 = 92$ kg/s
$\tau_Q = 15$ s	$b = 1.11$	$H3/Ws = 3.30$	$Fs = 23$ kg/s
$\tau_e = 10$ s	$\beta = 2.0$	$T2 = 773$ K	
$\tau_c = 24$ s	$x = 0.1$	$P4 = 0.1$ atm	
$\tau_z = 100$ s	$\rho_l/\rho_v = 24$	$P2 = 100$ atm	
$\tau_w = 20$ s	$k = -0.06$	$T_v^* = 13$ K	
$\tau_v = 286$ s	$A = 924$ atm	$T_w^* = 22$ K	
$\tau_w = 20$ s	$\phi = 0.20$		
$\tau_B = 15$ s	$\rho_v = 0.08$ (at $P4$)		

Some of data come from steady state calculation (appendix), some of them from literature. By these data, the transfer function now can be evaluated (in **dimensionless** for a short writing). For the gas turbine, such as frequency ω , gas turbine inlet temperature $T3$, air feed rate F_g , they become

$$\bar{\omega} = \frac{1.26\bar{F}_f - 0.081\bar{X}}{20s + 2.4} \quad \bar{T}_3 = 0.49\bar{F}_f - 0.363(0.2\bar{X} + \bar{\omega}) \quad (1)$$

$$\bar{F}_g - 0.2\bar{X}_g + \bar{\omega} = \frac{0.525\bar{F}_f + 0.167(10s + 1)\bar{X}_g}{8.3s + 1} \quad (2)$$

for the steam turbine and HRSG, such as liquid level Z , steam void fraction ϵ in boiling tube, steam turbine frequency ω , boiler pressure P , steam rate F_2 , cooling water exit temperature in condenser T_w , and steam condensation temperature T_v , they become

$$\bar{Z} = \frac{\bar{F}_1 - 1.03(-19.5S+1)\bar{\epsilon}}{100S} \quad \bar{\epsilon} = \frac{\bar{F}_v}{25S+8} \quad \bar{F}_v = \bar{Q}_g \quad (3)$$

$$\bar{\omega} = \frac{\bar{P} + \bar{X}_s - 0.14\bar{T}_v}{20S+2} \quad \bar{P} = \frac{4\bar{F}_2 - \bar{X}_s}{200S+1} \quad \bar{F}_2 = (1.74S+3.62)\bar{\epsilon} \quad (4)$$

$$\bar{T}_w = \frac{-\bar{F}_w + \bar{T}_t + \bar{T}_v}{20S+1.6} \quad \bar{T}_v = \frac{(20S+1.6)(\bar{P} + \bar{X}_s) - \bar{F}_w + \bar{T}_t}{(20S+1.6)(286S+1.06)-1} \quad (5)$$

As one can see that all of them are simply reduced. Some of them has integer action, such as Z , some of them has zeros, such as Z and T_v , which may be positive and gives an inverse response, like liquid level in boiler drum.

1.2 CONTROL LOOP PARING

There are two operation modes for steam turbine system. One is constant pressure where a bypass valve for steam from drum is used. The other is sliding pressure where the heat can be adjusted either by refiring or by bypassing hot gas. The Interactions between the gas turbine (GT) and the steam turbine (ST) can be shown by the following transfer function

$$\begin{bmatrix} \bar{Y}_{GT}(S) \\ \bar{Y}_{ST}(S) \end{bmatrix} = \begin{bmatrix} G_{11} & 0 \\ G_{21} & G_{22} \end{bmatrix} \begin{bmatrix} \bar{u}_{GT} \\ \bar{u}_{ST} \end{bmatrix} \quad (6)$$

Where Y means output variables, u is for perturbation variable, and G is for transfer function. From above equation, it can be seen that since the cross effect of U_{GT} to Y_{ST} by G_{21} can be merged to U_{ST} by G_{22} , G_{11} and G_{22} are considered as two major state space transfer functions for the gas turbine and steam turbine respectively. State gains of

systems can be obtained by $K_p=G(S=0)$, where an integrator for liquid level is assumed as $K_p/\tau_p S$.

$$\begin{bmatrix} \bar{\omega}(S) \\ \bar{T}_3(S) \end{bmatrix} = \begin{bmatrix} G_{11} & G_{12} \\ G_{21} & G_{22} \end{bmatrix} \begin{bmatrix} \bar{X}_f \\ \bar{X}_g \end{bmatrix} \quad K_p = \begin{bmatrix} 0.525 & -0.014 \\ 0.30 & -0.06 \end{bmatrix} \quad (7)$$

$$\begin{bmatrix} \bar{\omega}(S) \\ \bar{P}(S) \\ \bar{Z}(S) \\ \bar{T}_v(S) \end{bmatrix} = \begin{bmatrix} G_{11} & G_{12} & G_{13} & G_{14} \\ G_{21} & G_{22} & G_{23} & G_{24} \\ G_{31} & G_{32} & G_{33} & G_{34} \\ G_{41} & G_{42} & G_{43} & G_{44} \end{bmatrix} \begin{bmatrix} \bar{X}_f \\ \bar{X}_s \\ \bar{X}_w \\ \bar{X}_c \end{bmatrix} \quad K_p = \begin{bmatrix} 0.615 & 0 & 0 & 0.1 \\ 1.81 & -1 & 0 & 0 \\ -0.13 & 0 & 1 & 0 \\ 4.14 & 0 & 0 & -1.43 \end{bmatrix} \quad (8)$$

The dependent and independent variables for the GT and the ST are

$$(\omega, T_3) - (X_f, X_g)$$

$$(\omega, P, Z, T_v) - (X_f, X_s, X_w, X_c)$$

where subscript f is for fuel, g for gas, s for steam, w for water, and c for cooling water.

Relative Gain Array

The RGA method is developed as an index of loop interactions. It was proposed by Bristol and has been extensively applied as it is easy to calculate and only needs steady state gain data. If $\beta=I$, the multivariable control can be designed as multi-loop SISO configurations. For GT and ST, matrix β is

$$\beta_{GT} = \begin{bmatrix} 1.48 & -0.48 \\ -0.48 & 1.48 \end{bmatrix} \quad \beta_{ST} = \begin{bmatrix} 0.68 & 0 & 0 & 0.32 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0.32 & 0 & 0 & 0.68 \end{bmatrix} \quad \beta_{ij} = [K_p]_{ij} [K_p]_{ji}^{-1} \quad (9)$$

The β_{GT} shows there are some extent interactions at GT, but the couples are still dominant as expected, where β_{ii} near to 1, the β_{ST} is very closed to I matrix, the interaction can be neglected. A multi-variable control system is reduced to multi-loop SISO system. It should be pointed out that the RGA is based on steady state operation

control, it does not account the transient behavior.

Niederlinski Index

The Niederlinski index (NI) is used for stability analysis and for the elimination of unworkable control pairs for a multivariable system. For a negative index, the system will be unstable for any controller tuning. When it is greater than zero, it tells us that the system is controllable. Based on the gain evaluated above, we have

$$NI = \frac{\text{Det}[K_p]}{\prod_{j=1}^N K_{pij}} \quad NI_{GT} = -0.68 \quad NI_{ST} = -1.0 \quad (10)$$

It is noted that NI is a reciprocal of β_{ij} for a 2X2 system.

From the above results of NI's, we know that the present system of the GT and ST are controllable.

1.3. CONTROLLER TUNING METHODS

One of the major questions in multivariable control is how to tune controllers in a diagonal multi-loop SISO system. Once the pairs have been chosen, the gain and reset time must be specified so that the overall system is stable and gives acceptable load response.

The BLT method will be used for controller tuning. For each G_{ij} tuning, some classical SISO methods are used, such as the critical frequency (Ziegler-Nichols) method for high order system, the first order system with dead time (Cohen and Coon) method for a system with big τ , specified damping decay factor $\zeta=0.8$ (optimum response time) method for a second order. All of these design results will be plotted by the BLT method to see the biggest log modules.

The process might be of high order system with many time constants. The control

design depends on what we are interested on those response times. For a fast control, the small τ in system is the key point in controller setting; vice verse, for a slow response process a big τ in system is important in suppressing the noise or the fluctuations during a closed loop dynamic response.

Different methods will be used to different cases as follows:

BLT Method

The biggest log-modules tuning (BLT) method is one tuning methodology for MIMO systems using a diagonal multi-loop SISO system (Luyben,1990). It involves classical SISO turning for each G_{jj} , and a F factor optimized for detuning. The detuning factor F is defined as

$$K_{cl} = \frac{K_{cl}}{F} \quad \tau_{cl} = \tau_{cl} F \quad (11)$$

where $F=1.5-4$ for the Ziegler-Nicholas method. The controller tuning by the Ziegler-Nicholas method is

	K_c	τ_I	τ_D
P	$K_u/2$	-	-
PI	$K_u/2.2$	$P_u/1.2$	-
PID	$K_u/1.8$	$P_u/2$	$P_u/8$

where u means under ultimate case. The objective function for the detuning is

$$L_{MAX} = -2N - 20 \log_{10} \left[\frac{W}{1+W} \right] \quad W_{(\omega)} = -1 + \det[I + GB] \quad (12)$$

where B is diagonal controller matrix. When $N=1$, it is one variable system. We get the familiar SISO controller tuning by $dB=2.0$ maximum closed loop log modules criterion. According to Luyben, this empirically determined criterion has been tested on a large number of cases and gives reasonable performance. Other SISO loop tuning methods can be also applied. It should be expected that different detuning factors will be required for each loop tuning method, but the final setting should be not much different.

Second Order System Tuning

Since Ziegler-Nichols and Cohen-Coon methods do not apply to second order system, we will tune for damping coefficient ζ of 0.8. The general transfer function for such a second order system with one LHP zero is

$$G = \frac{K_p(\tau_a S + 1)}{(\tau_1 S + 1)(\tau_2 S + 1)}$$

The characteristic equation is

$$\tau_1 \tau_2 S^2 + (\tau_1 + \tau_2 + K_c K_p \tau_a) S + (1 + K_c K_p) = 0 \quad \text{or} \quad S^2 + 2\zeta \omega S + \omega^2 = 0 \quad (14)$$

$$\omega^2 = \frac{1 + K}{\tau_1 \tau_2} \quad \zeta = \frac{\tau_1 + \tau_2 + K \tau_a}{2\sqrt{1 + K}\sqrt{\tau_1 \tau_2}} \quad K = K_p K_c \quad (15)$$

So the closed loop response will have ultimate frequency ω , and the decay coefficient ζ .

Both of them are affected by K_c .

Based on dynamic performance (Time-Integral Performance) criteria, for the most often used, we have

ISE	$\zeta=0.5$
ITSE	$\zeta=0.6$
IAE	$\zeta=0.7$
ITAE	$\zeta=0.7$

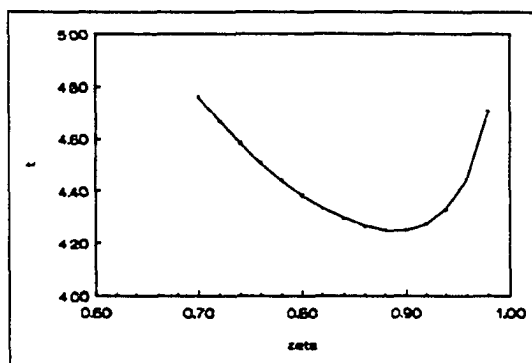


Figure 1 t vs. ζ

To suppress error that persist for a long times or to get a short response time, $\zeta=0.8$ is suggested for the control tuning (Shinnar, 1992). The transient time required to approach 95% of steady state is approximately

$$t_{95} = \frac{1}{\omega \cdot \zeta} \ln \left[\frac{20}{\sqrt{1 - \zeta^2}} \right] \quad (16)$$

It is plotted in Fig.1, where one can see that the minimum $t_{\min}$ occurs at $\zeta=0.9$.

Cohen-Coon Method (Stephanopoulos, 1984)

Cohen and Coon observed that the response of most processing units to a step change has a sigmoidal shape, which is composed of a 1st order lag and a dead time. The controller setting is for minimum offset, ISE criteria, and 1/4 decay ratio. It is originally done by graphic method. If one has transfer function, it can be done directly, such as to a 2nd order system, if $\tau_1 < \tau_2$, one can assume that τ_1 is a dead time, or if $\tau_1 < \tau_2 \ll \tau_2$, one can have mathematically

$$\frac{\tau_d s + 1}{(\tau_1 s + 1)(\tau_2 s + 1)} \approx \frac{e^{(\tau_d - \tau_1)s}}{\tau_2 s + 1} \quad (17)$$

Where τ_2 is a long term response. By this way, the Cohen-Coon method can be applied to high order system if we interest on a long term control.

1.4 CONTROLLER TUNING RESULTS

Some stand alone system or component controls based on the above methods are discussed as following:

GT Control

Gas turbine involves power/frequency and firing temperature controls.

As the turbine inlet temperature is mainly affected by the ratio of fuel to air, a ratio controller is indicated. If this can be done, the rest of the problem is the power/frequency (P-f) control. However, we have found that a simple PID is adequate. The P-f is maintained by a governor, where it translates rotating speed into a fuel feed valve position by a droop method. When droop is expressed in percent, it is the same as the proportional band in process control.

$$X_f = K_c \frac{\Delta \omega}{\omega} \quad K_c = \frac{1}{\text{droop}} \quad (19)$$

$$G_{\omega} = \frac{\bar{\omega}}{\bar{\omega}_f s} = \left[\frac{1.26}{\tau_{\omega} s + 2.4} \right] \left[\frac{1}{\tau_f s + 1} \right] \quad (18)$$

If the droop is 4%, then the control gain is 25. As the $K_{po}=0.525$ (eq.7), a 4% droop is to $0.52*60 *4\%=1.26$. Because the time constant of the governor τ_{ω} is very small, the system is near first order. The closed-loop transfer function will be of response time of $\tau_c=1.4$ s and gain of 0.93. For elimination of offset, $\tau_i=2.8$ by $\zeta=0.8$ may be added to form a PI control.

These setting parameters ($K_c=25$, and $\tau_i=2.8$) have been checked by the closed loop Bode plot where the BLT method is used. The result shows $dB=2.35$ (Fig.2E). As $dB=2.0$ is corresponding to about $\zeta=0.4$, this result is expected, which gives fast reset response.

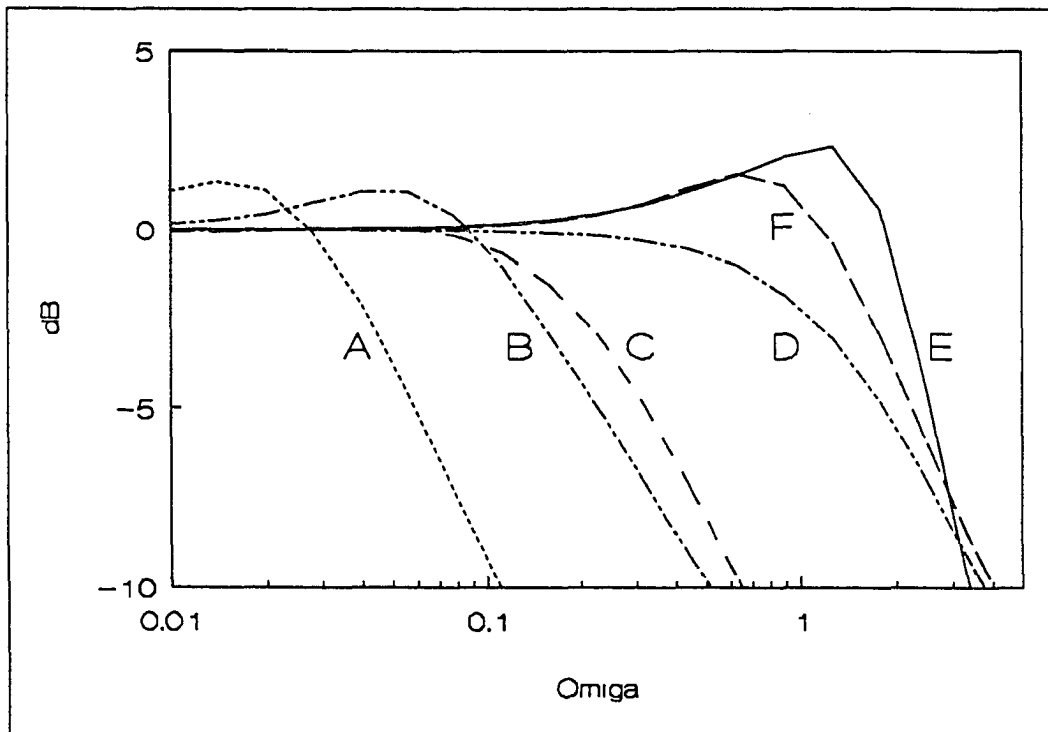


Figure 2 BLT plot (A- T_v , B-P, C-Z, D-ST, E-GT, F-ST)

Water Level Control for Boiler

The water level control is for the water holdup in boiler drum. It is done by

adjusting the water feed rate.

The transfer function of level Z to steam rate F_2 is given as follows

$$G_z = \frac{\bar{Z}}{\bar{F}_2} = \left[\frac{1}{(2\rho_v/\rho_L + 1)e} \right] \left[\frac{(1-x)\tau_c S - b}{\tau_z S[(\tau_c + \tau_e)S + b/x]} \right] \quad (20)$$

Note that the steady state gain is negative gain and infinite caused by $-b$ and $1/S$. There always exists the inverse phenomena because of a positive (RHP) zero in transfer function numerator. It can be limited by shift τ_e to near zero. It can be done by a good design to let $S_j \rightarrow \infty$ or $\tau_z \rightarrow \infty$, which lead S_j satisfies $G(S_j)=0$. That is reduce τ_c , which means increasing water circulation rate, or increase τ_z (increasing boiler cross section area). This is the effect of process design on the controller performance, so a good design is as important as control itself.

Most liquid levels represent mass inventory, and it is relatively unimportant where the level is as long as it is in a safe region. Therefore P or PI control is good enough to give a smooth changes in flow rate and to filter out fluctuations for up or down stream. For a system that has an inverse response, a slow response is desired. Assume a P control is applied, the characteristic equation is

$$\omega^2 = \frac{K}{\tau_p} \quad \zeta = \frac{K\tau_a + 1}{2\sqrt{K}\sqrt{\tau_p}} \quad K = K_p K_c \quad (21)$$

For a real ζ , the maximum K is 0.01. The solution shows:

$$\begin{array}{ll} \zeta=0.8 & K_c=30 \quad \omega=0.024 \quad \text{PI} \quad K_c=27 \quad \tau_i=218 \\ \zeta=2.8 & K_c=10 \quad \omega=0.014 \quad K_c=9 \quad \tau_i=370 \end{array}$$

One example of fast control action in literature is PI controller with $\tau_i=92$ s for the same τ_p , but a slow response is recommended (Polonyi 1991).

The biggest log modules $\text{dB}=0$ (Fig.2C) in terms of the BLT Bode plot, which is what we want. The fluctuations have been suppressed by this setting.

Condenser Temperature Control

The transfer function of temperature to steam flow rate as developed in Chapter 4 is

$$G_{TV} = \frac{\bar{T}_v}{\bar{F}_3} = \frac{20S+1.6}{(20S+1.6)(286S+1.06)-1} = \frac{2.3(12.5S+1)}{(12S+1)(667S+1)} \quad (22)$$

As $\tau_d=12.5$, $\tau_i=12$, then $\theta=25 < \tau_p=667$, the Cohen-Coon method can be applied to a PI controller design, the design result is $K_c=10$, and $\tau_i=77$ s. It belongs to a long term control and roughtness.

By the BLT Bode plot, we get a peak dB=1.34 (Fig.2A). It is less than dB=2.0, so the design is acceptable as it will give a smoother response. This is an advantage in steam turbine control since it reduces the effect of steam temperature in the condenser to exit steam temperature at steam turbine. The exit temperature has a big effect on the steam turbine and power generation as the water wet fraction will change with temperature.

Pressure Control

Here the pressure control means the boiler drum pressure or steam pressure. The pressure can be controlled by refiring fuel feed rate, or by the main steam valve. The gain matrix tells us that for long times, the P-f only can be controlled by fuel adjustment, the response is slow process. The preferred way to control P-f is by the steam valve. The pressure is controlled either by bypassing steam or by refiring fuel feed rate. The transfer functions are

$$\bar{P} = \frac{(70S+14.6)\bar{Q}_g - (25S+8)\bar{X}_s}{(200S+1)(25S+8)} \quad \frac{\bar{P}}{\bar{Q}_g} = \frac{1.83(4.8S+1)}{(200S+1)(3.3S+1)} \quad (23)$$

It is again a long term control problem.

By the Cohen-Coon method, where $\theta=7.1$, we get $K_c=14$ and $\tau_i=22$ s, which is in corresponding to $dB=1.1$ (Fig.2B) in a closed loop gain plot by the BLT method. The oscillation will be small as dB tends to zero.

Steam Turbine P-f Control

The rate of steam passing through the steam turbine is dependent upon the upstream pressure, so that this pressure is a key variable in load following control. This can be done either by steam valve control where the valve shares the pressure drop, or by sliding pressure before the high pressure steam turbine. For a big power plant, the sliding pressure operation is desirable.

The steam valve provides additional pressure drop to control the steam flow rate. It causes a pressure change of up stream. In the long run the system only can be controlled by the heat rate. If a refired HRSG used, the fuel rate is the manipulated variable. For a no-refired HRSG, steam bypassing should be used.

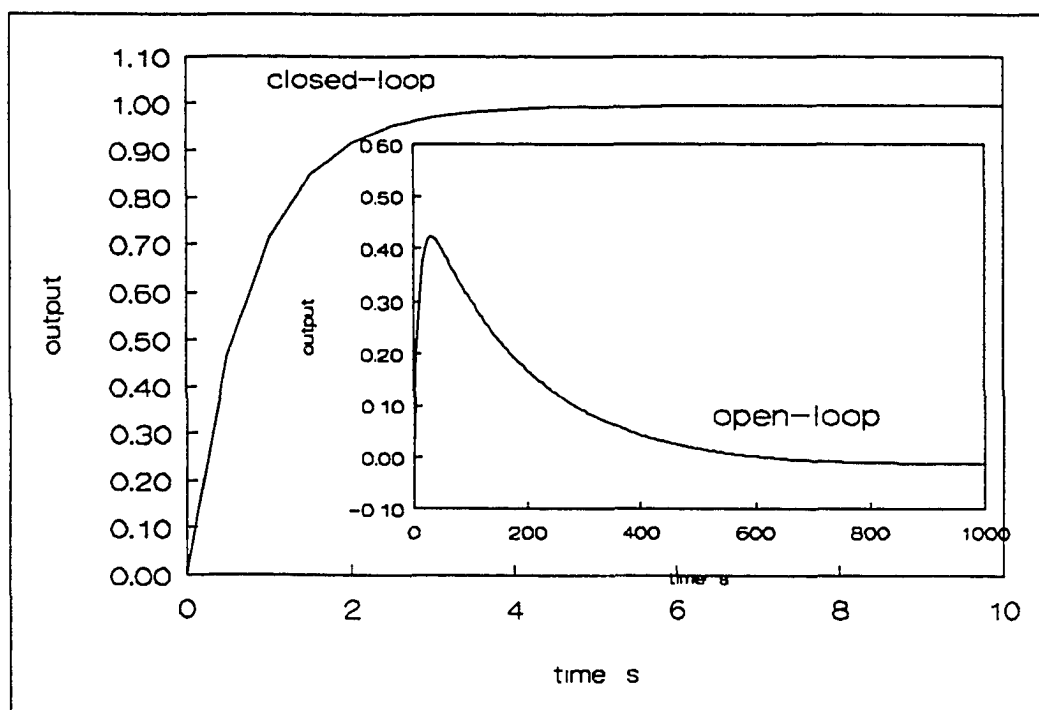


Figure 3 Power/frequency response to steam valve position

The transfer function of frequency to valve position for this case, based on the equations developed at Chapter 4, is

$$G_{\omega} = \frac{\bar{\omega}}{\bar{X}_v} = \frac{68S(980S+1)}{(666S+1)(200S+1)(10S+1)} \quad (24)$$

As expected of fast control, the small τ is key point. So as to the control is set as PI with a $\tau_i=10$ s, and $K_c=25$ for a 4% droop again. The open loop and closed loop are shown as Fig.3.

The BLT plot has a peak dB=0 (Fig.2D). If one want fast control, set $\tau_i=1.5$ s, which will give a peak dB=1.55 (Fig.2F).

Control of Entire Combined Cycle Plant

Here the plant control or ripple control means total power to the fuel feed. It is a secondary control loop for whole plant control at load following, where all of the generators are connected to the grid. On the basis of the model, we develop an overall response relation of ω to fuel valve position X_f .

$$G_s = \frac{\bar{\omega}}{\bar{X}_f} = \frac{2.72(1000S+1)(1.74S+3.62)}{(667S+1)(200S+1)(25S+8)} \quad (25)$$

This equation comes from following equations

$$(\tau_{\omega}S+2)\bar{\omega} - \varphi(1.28\bar{X}_f - 0.081\bar{X}_g - 0.4\bar{\omega}) + (1-\varphi)(\bar{P} + \bar{X}_s - 0.14\bar{T}_v) \quad (26)$$

$$\bar{P} + \bar{X}_s - 0.14\bar{T}_v - G_s(\bar{Q}_g + \frac{200S(25S+1)}{4(1.74S+3.62)}\bar{X}_s) \quad (27)$$

$$\bar{Q}_g = \frac{\bar{X}_f - \eta_{GT}\bar{W}}{1 - \eta_{GT}} - 0.9\bar{X}_f + 0.03\bar{X}_g + 0.15\bar{\omega} \quad (28)$$

$$\Delta Q_g = \Delta Q_f - \Delta W \quad W = \eta_{GT}Q_f \quad (29)$$

where $\phi=0.62$, is power ratio of GT to (GT+ST). Assuming that the primary loop is perfectly controlled, such as for a constant gas turbine inlet temperature, then $\Delta X_g/X_g = \Delta X_f/X_f$ and the frequency to fuel valve position becomes

$$\frac{\bar{\omega}}{\bar{X}_f} = \frac{0.74+0.35G_s}{20s+2.25-0.057G_s} \quad (30)$$

The power transfer function can be obtained by setting $\Delta\omega=0$

$$\frac{\bar{W}}{\bar{X}_f} = -0.74+0.35G_s \quad (31)$$

The open loop responses of P-f are shown in Fig.4, where they give out a fast jump at initial by GT, following a little overshoot by boiling tubing, and a long time small drift by HRSG. The initial fast response make the combined cycle system suitable to the load

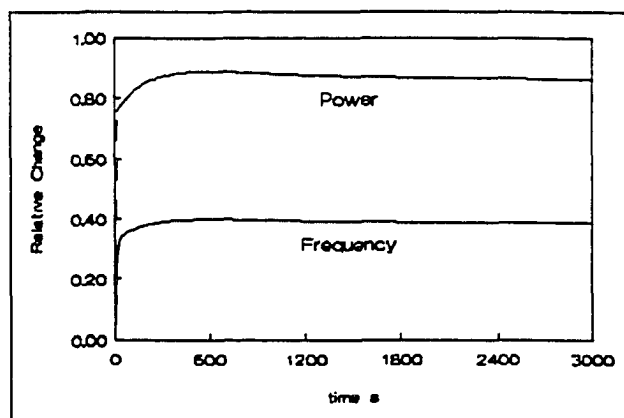


Figure 4 P-f response to step change in X_f

following control, especially by the gas turbine component. As the slow response of ST, the fuel feed at beginning will have an overshoot. For this reason, the preferred design is a response with $\zeta > 1$ to eliminate the oscillation.

2. DYNAMIC SIMULATION RESULTS

Using the controller tuning determined in the previous section, the dynamic behavior of the combined cycle power plant was simulated by using DYSIM. Dynamic responses were obtained for changes of load, feed or operating conditions. Both step and ramp responses were simulated. The purpose of these dynamic tests was to validate the

control system design.

The data for our simulation and control studies are based on the combined cycle units at the Gilbert Station of Jersey Central Power & Light. This unit uses four GE gas turbines, each with its own HRSG with supplementary firing, and a two-stage no reheat steam turbine.

The whole plant can be divided as three groups: gas turbine, HRSG, and steam turbine. We first show the closed loop responses for each system on a stand alone basis.

2.1 Gas Turbine System

Fig.5 is the schematic diagram for the gas turbine system simulation. It contains an air compressor with an IGV valve, a combustion chamber, a gas turbine, an electrical generator, and a fuel valve. Two control loops are configured for control of power/frequency and turbine inlet temperature.

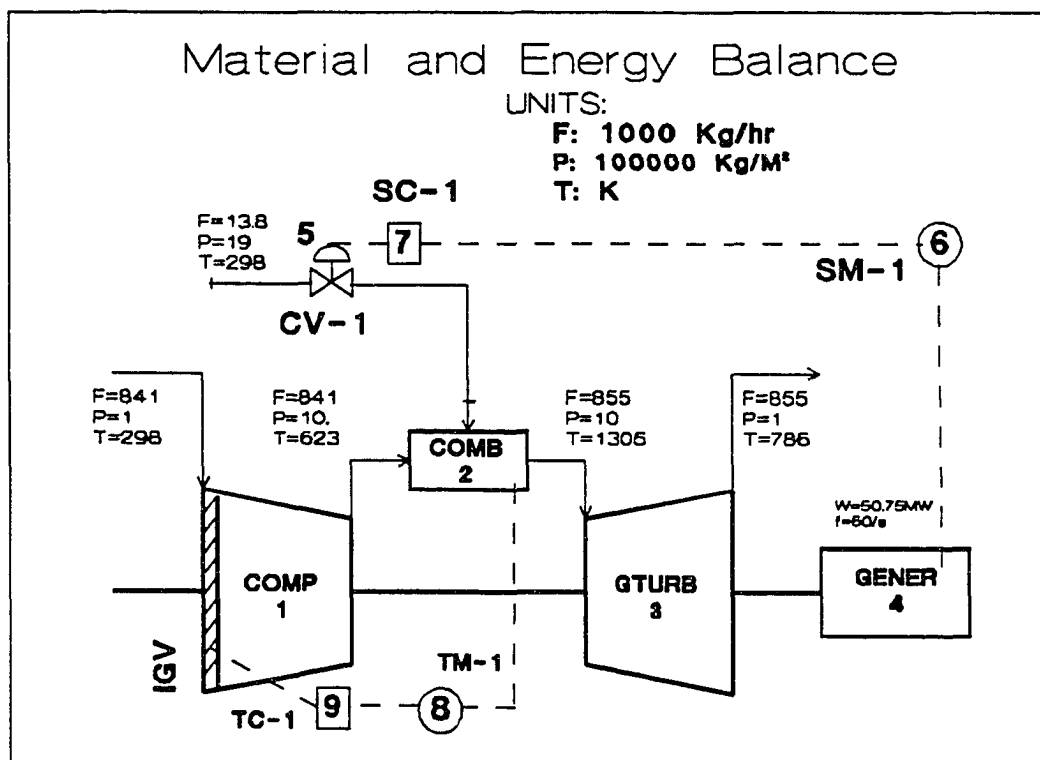


Figure 5 Gas turbine system

Fig.13 is the responses to a load change for the GT system for both a step change of -10% and a ramp changes of -10% per min for 2 mins ($t=1-3$ min). As a droop method, or a fast control is applied to the GT, the system shows under- damped behavior for a governor control with gain of 25 and reset time of 4 sec. The electric frequency is controlled well at a certain range as shown in Fig.13. It has a droop effect, 0.01 Hz, during a ramp change.

The model can represent the pressure and gas flow rate changes during a load change. It took about 1.0 min to get a new steady state for frequency, fuel rate, and firing temperature; 2.0 min for the pressure and the gas flow rate. The turbine inlet temperature and gas flow rate drift during the ramp time, after that, they tend to the new steady states in very short time.

2.2 Steam Turbine System

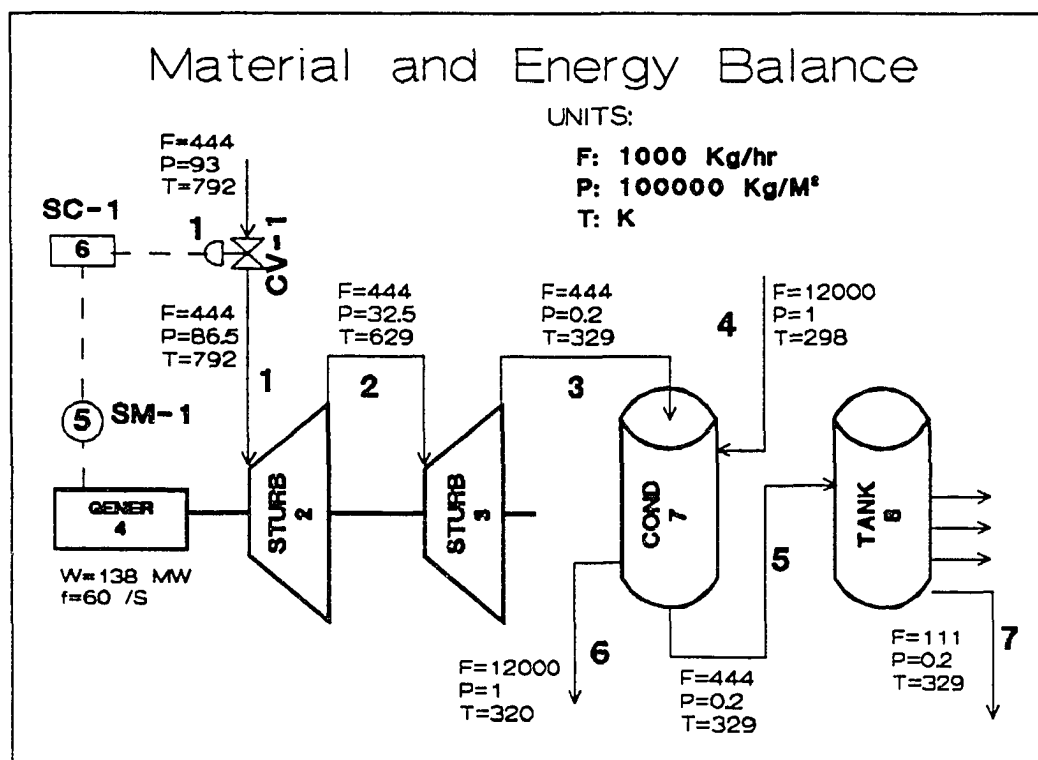


Figure 6 Steam turbine system

Fig.6 is the schematic diagram for the steam turbine system simulation. It contains a main steam valve, high pressure and low pressure steam turbines, a steam condenser, and an electrical generator. A control loop for the power/frequency is induced. As it is a stand alone mode, the upstream steam quality is assumed constant.

As the ST is in sliding pressure operation by a main steam valve with a controller of gain of 25 and reset time of 3.6 sec, the ST has different responses when compared to the GT. For example, the frequency droop is a little bigger than that of GT during the same ramp changes (Fig.15).

Most of the responses (Fig.13 & Fig.15) show step functions in the same direction following a step load change. The power response shows a little overshoot before it goes to a new steady state. All of the responses are overdamped, which is desired.

2.3 HRSG System

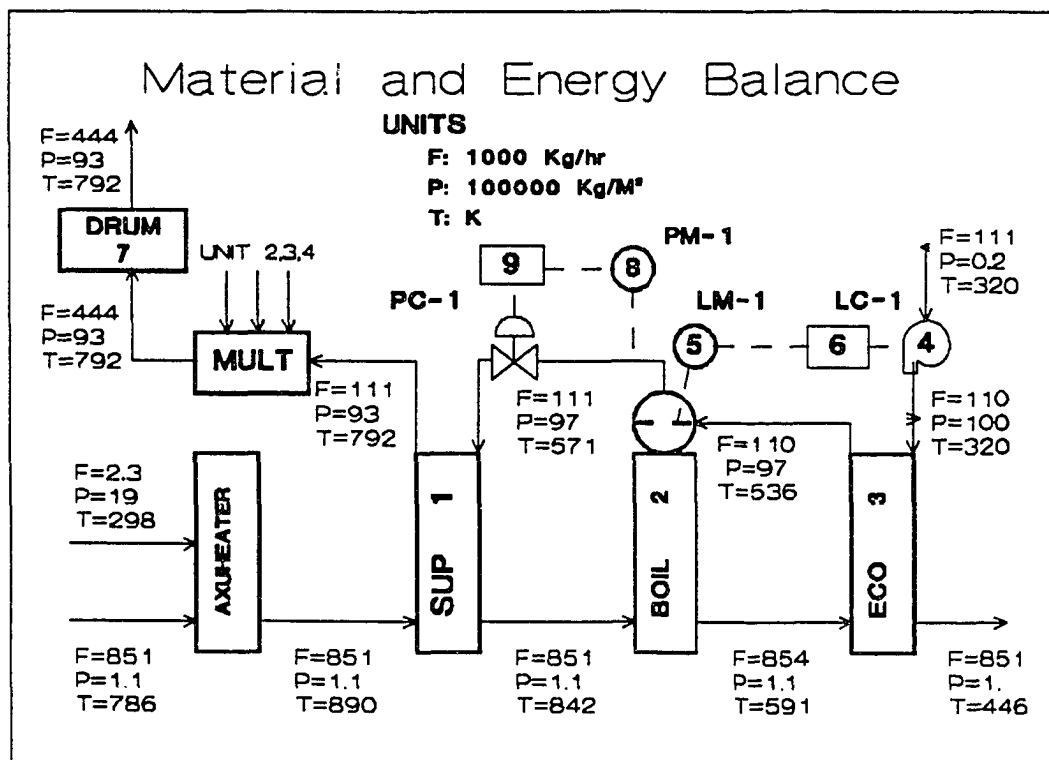


Figure 7 Heat recovery steam generation system HRSG

Fig.7 shows the schematic diagram for the HRSG system simulation. It consists of an auxiliary heater (refiring room), superheater, a forced circulation steam boiler with a drum, an economizer, a water feed pump, and a steam header. There are two control loops, one for pressure in drum, another for liquid level in drum.

The results for HRSG for a step change of 10% and a ramp change of 10% per min for 2 mins in hot gas flow in temperature are shown in Fig.14. The response time is greater than that of GT and ST. The response shows that there are overshoot for steam exit temperature, first order behavior for steam rate, and long drift effects exist for boiling and exit gas temperatures. For the present design, the inverse response of liquid level is limited to very small.

For all above three cases, the responses to ramp change show that they are smoother than the responses to step changes. In real operation, the rate for ramp change is limited to reduce the thermal stress, and extend the gas turbine's service life.

2.4 Combined Cycle system

Fig.8 is the schematic diagram for the combined cycle system simulation. It contains four gas turbines, four HRSG's, and one steam turbine. A feedforward control is applied to the power/frequency control for the steam turbine part by the refiring fuel manipulation. It is sliding pressure operation.

When the GT, HRSG, and ST are combined to form a combined cycle system (Fig.8), there are interactions between them. Fig.16 to Fig.18 are the results of the combined cycle system responses for a ramp change of -10% for 2 min in load to the GT.

As expected, the responses of the GT have the same results as shown in Fig.13 and Fig.16. The exhaust hot gas will give the effects on the following HRSG and ST. Fig.18 is the response of the ST in the combined cycle system. As one can see, when compared to the Fig.14, there are drift effects. Though the responses show the same

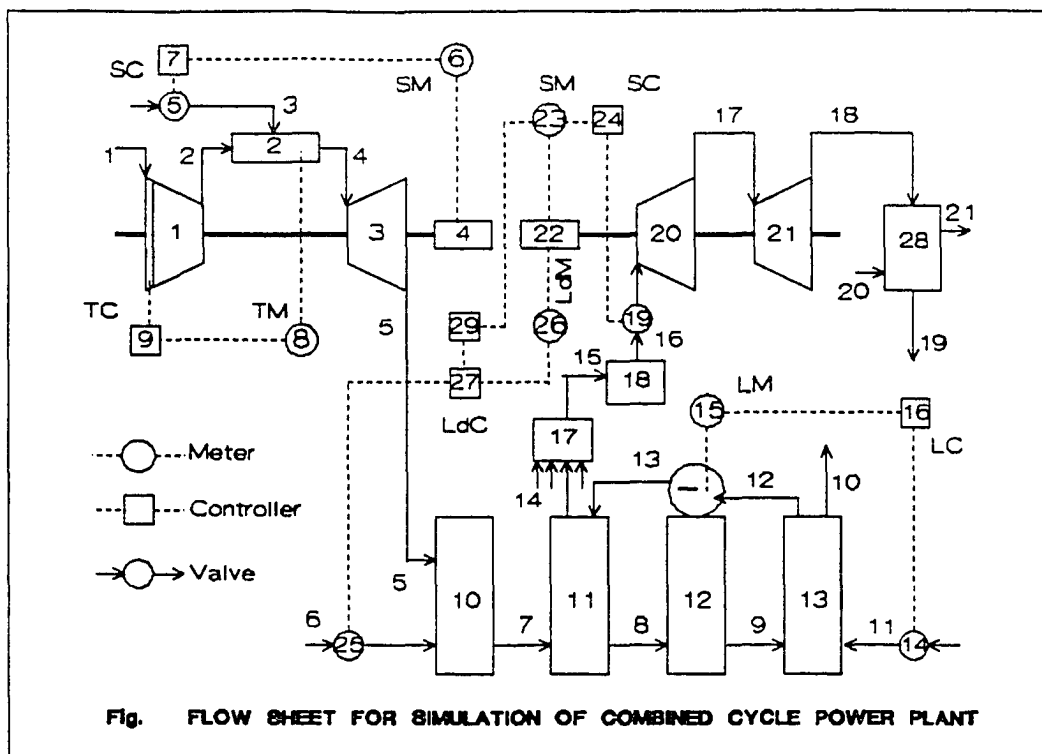


Figure 8 Combined cycle system

tendency as does in Fig.14, the responses are no long linear as temperature and flow rate of exit hot gas from gas turbine are ramp in a non-linear way (Fig.13). It takes more time to get a new steady state than does the stand-alone ST.

For the HRSG, we see the same tendency as in GT or ST, that is the response is smoother (Fig.17) than that in stand-alone case (Fig.14). There is a difference in stack gas temperature. It shows a little overshoot in Fig.17. The overshoot of steam temperature (Fig.14) is no long seen in the combined cycle case (Fig.17). The response times for the liquid level are also different. In the combined cycle case, the response is slow. These coupling effect is important in design of the controller.

All these results show that the HRSG and the ST are affected by the GT, that the steam cycle is affected by the gas cycle. Another effect is that the HRSG and the ST interact through the steam flow. One can expect that if the steam header is big, it will decouple these effects and the behavior of stand alone responses will be obtained.

2.5 Startup Operation

Start up was simulated as an exercise in model validation since typical start up transients are shown in the literature (EPRI AP-5975).

The start up operation for the HRSG is shown in Fig.19. The scheme is the same as in Fig.7.

Initially, it is a warm up process. The steam rate is zero, liquid level is constant and the steam pressure is also constant. After it reaches the boiling temperature, the system looks like it has a response of step change. A little inverse response for liquid level can be found at the initial of the boiling. The present design and control can make the liquid level to be well controlled as shown in Fig.13. It tells that a good design can lead a good control by simple controller setting.

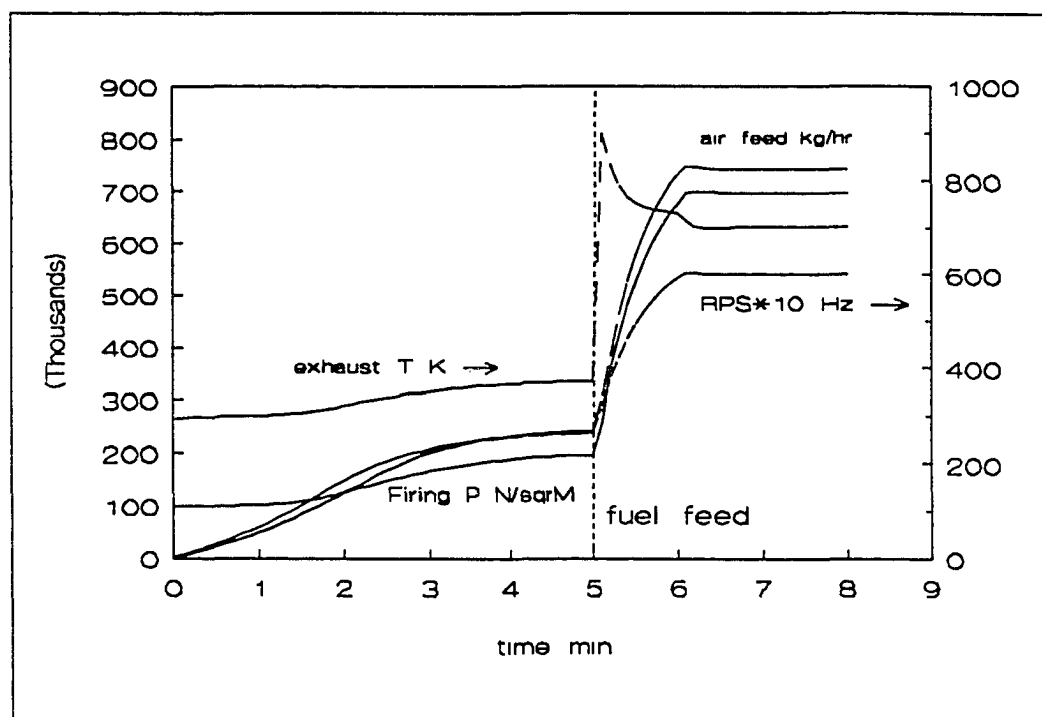


Figure 9 Start up option of GT

The scheme for the start up of the gas turbine is the same as in Fig.5. The results of the GT startup is shown in Fig.14.

Initially the air compressor is driven by an electrical motor for few minutes (about

5 mins) to get a certain frequency. At the same time, firing pressure and hot gas exhausted temperature are raised. Then fuel is injected into combustion chamber. After that, the air compressor is driven by the gas turbine. When the frequency is raised near to 60 Hz, the control action will be initialized to manipulate the fuel feed rate, and to let the system synchronized. Following is the system to be connected to grid and power generated is send out to the electrical network.

When compared to the work (EPRI AP-5975), the present simulation represents the startup option very well at frequency, pressure, and air feed rate, a little difference is at the temperature.

The results from Fig.18 and Fig.19 show that the DYSIM is a good tool for the dynamic simulation, even for the startup operation.

2.4 Steady State Results

The results discussed here are concerned to off design and rated operation design which will be used as data base for the off design and dynamic simulation.

Rated Operation Design

The input data file for a design under a rated operation condition is presented in Tab.1. The user should fix the data under user column. The data listed in default column are an example. Any efficiency under rated design conditions should be provided by the user. The computer result of rated case design is tabulated in Tab.2.

Fig.10 is a result of the HRSG temperature profile at a rated operation case.

In Tab.2, it reports power generated and thermal efficiency for gas turbine, steam turbine, and total plant. The wet fraction of steam exit from steam turbine is also included, it revealed a reasonable design. The temperature profiles were obtained for the GT, HRSG, and ST respectively. Based on the given data, the HRSG design was in an

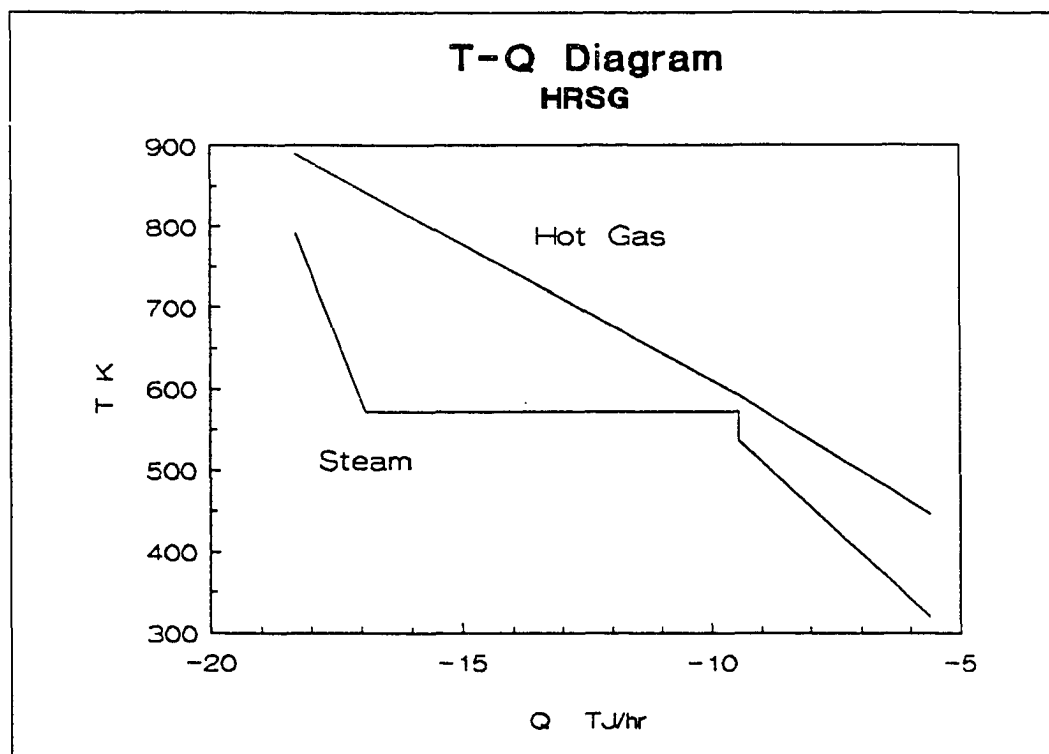


Figure 10 Q-T

acceptable range as shown by Fig.10. The gas flow rate is the sequence results from the fuel feed rate, and the steam flow rate is from the pinch temperature settings.

Off Design

The input data file for an off design operation case was demonstrated in Tab.3. The main variable is fuel feed rate for different load. Electrical frequency was assumed well controlled. Firing temperature was controlled by adjusting air feed rate through the IGV in a certain range.

The results listed in Tab.4 showed that as the fuel rate reduced, the plant efficiency goes down (Fig.11), or heat rate goes up (Fig.12). It is almost constant in a load range of 80-105%, which is the same as reported in literature.

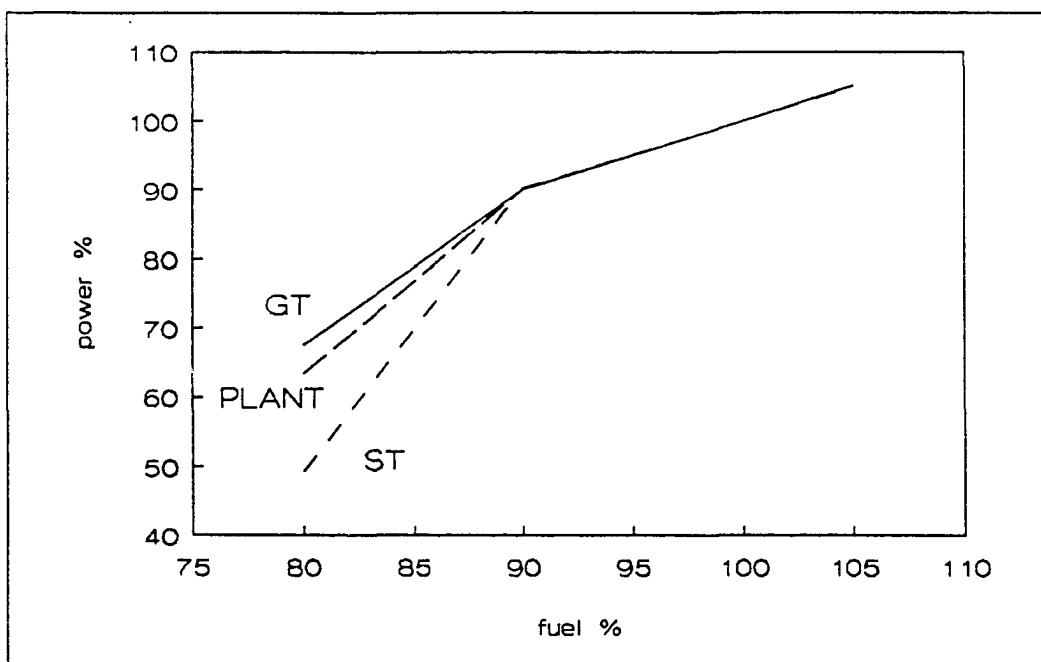


Figure 11 Load vs fuel

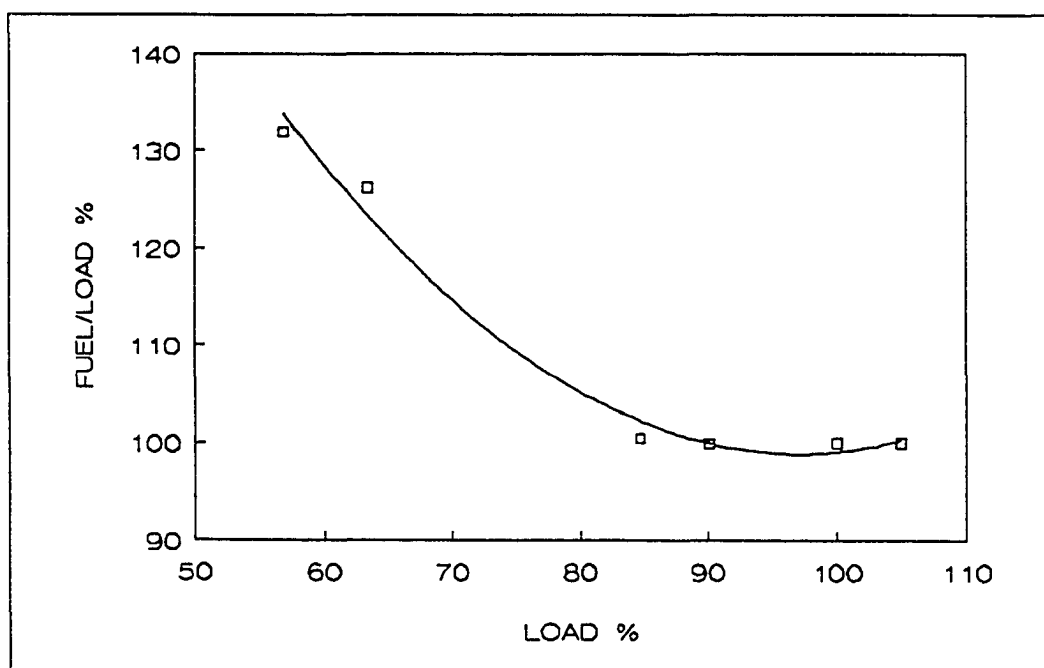


Figure 12 Heat rate vs load

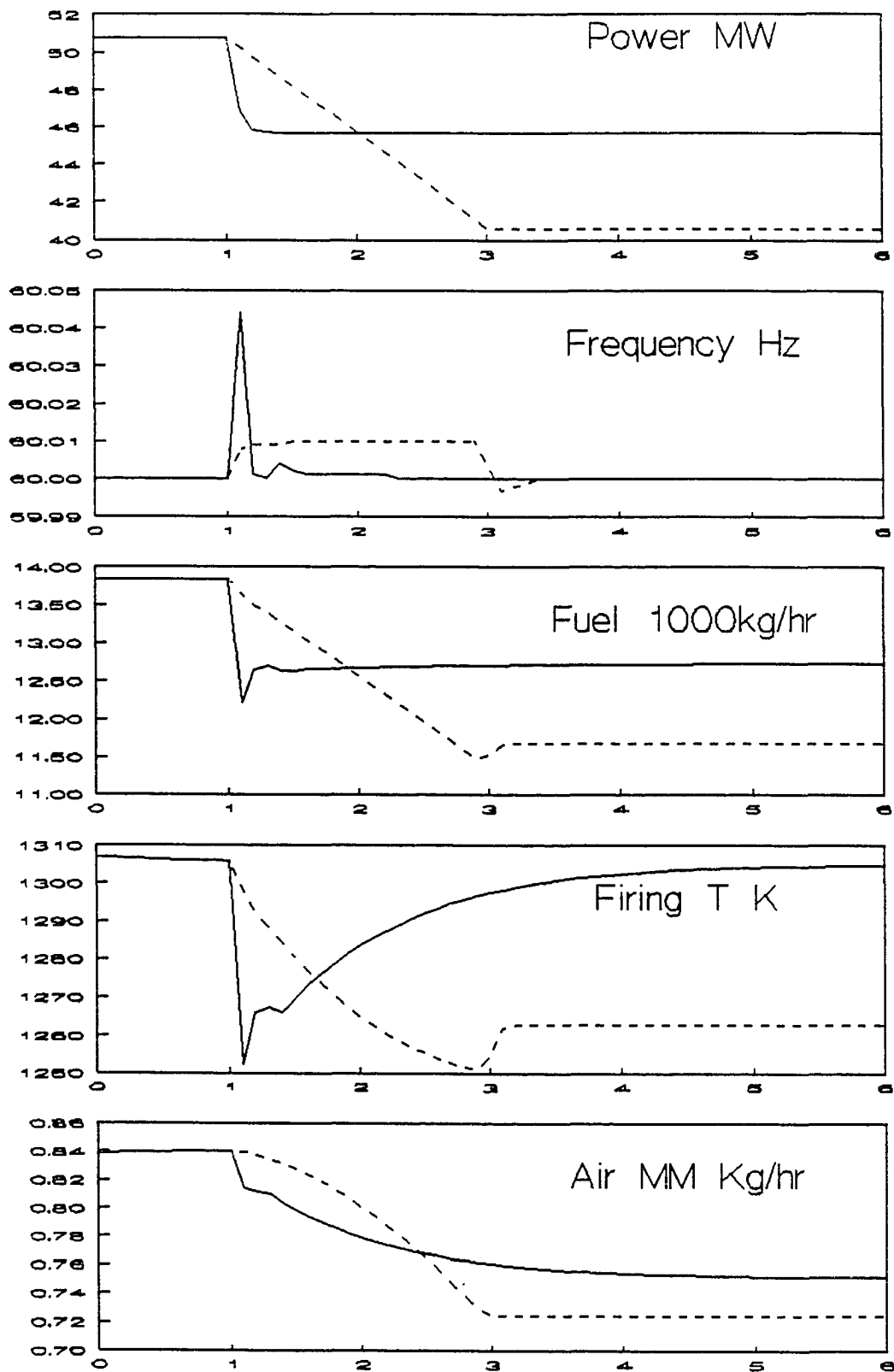


Figure 1 Gas Turbine Load Change

Dashed line (ramp): -10% per min; Solid line (step): -10%

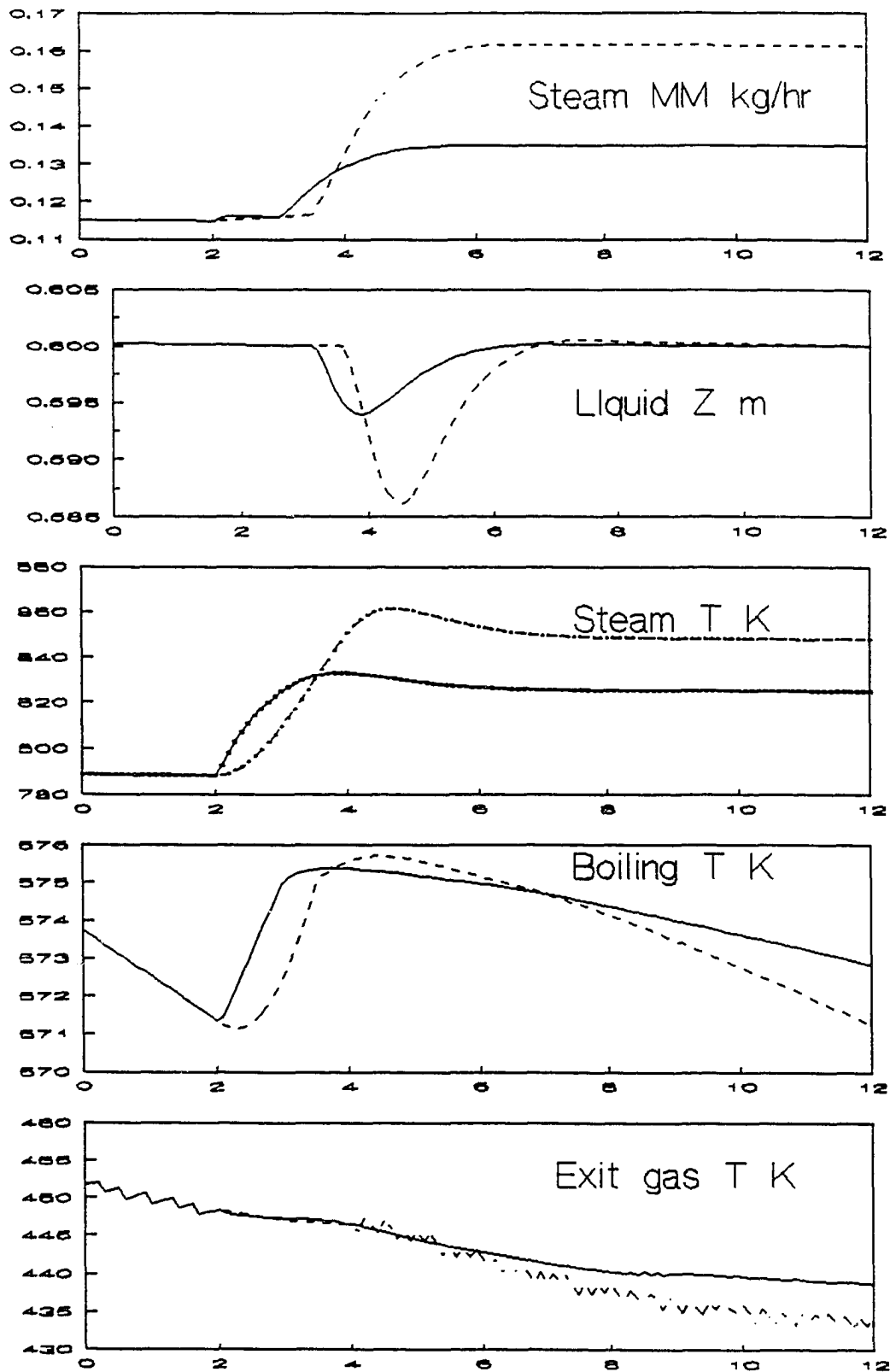


Figure 2 HRSG Hot Gas feed in Change in Temperature
 Dashed line (ramp): 10% per min; Solid line (step): 10%

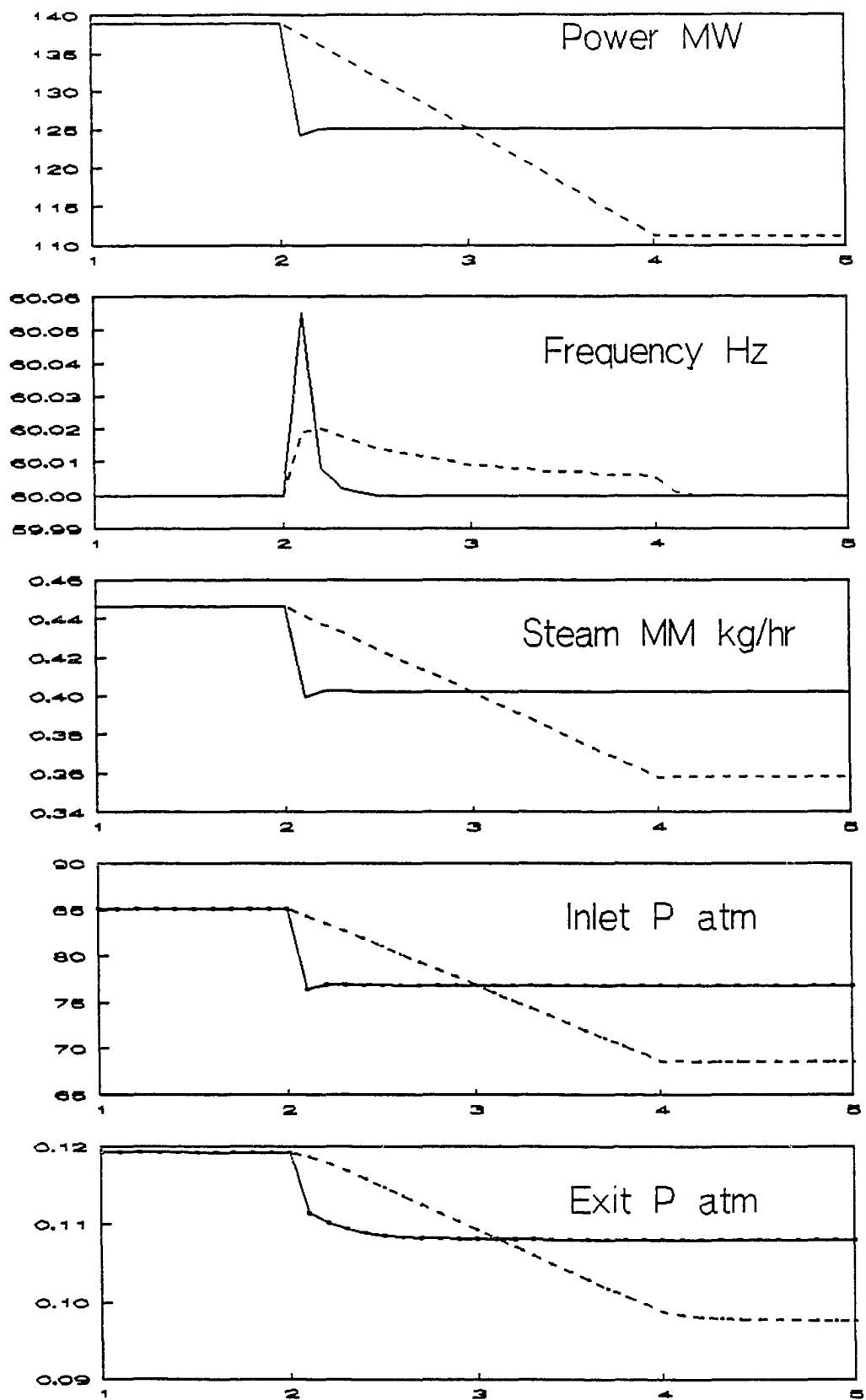


Figure 3 Steam Turbine Load Change

Dashed line (ramp): -10% per min; Solid line (step): -10%

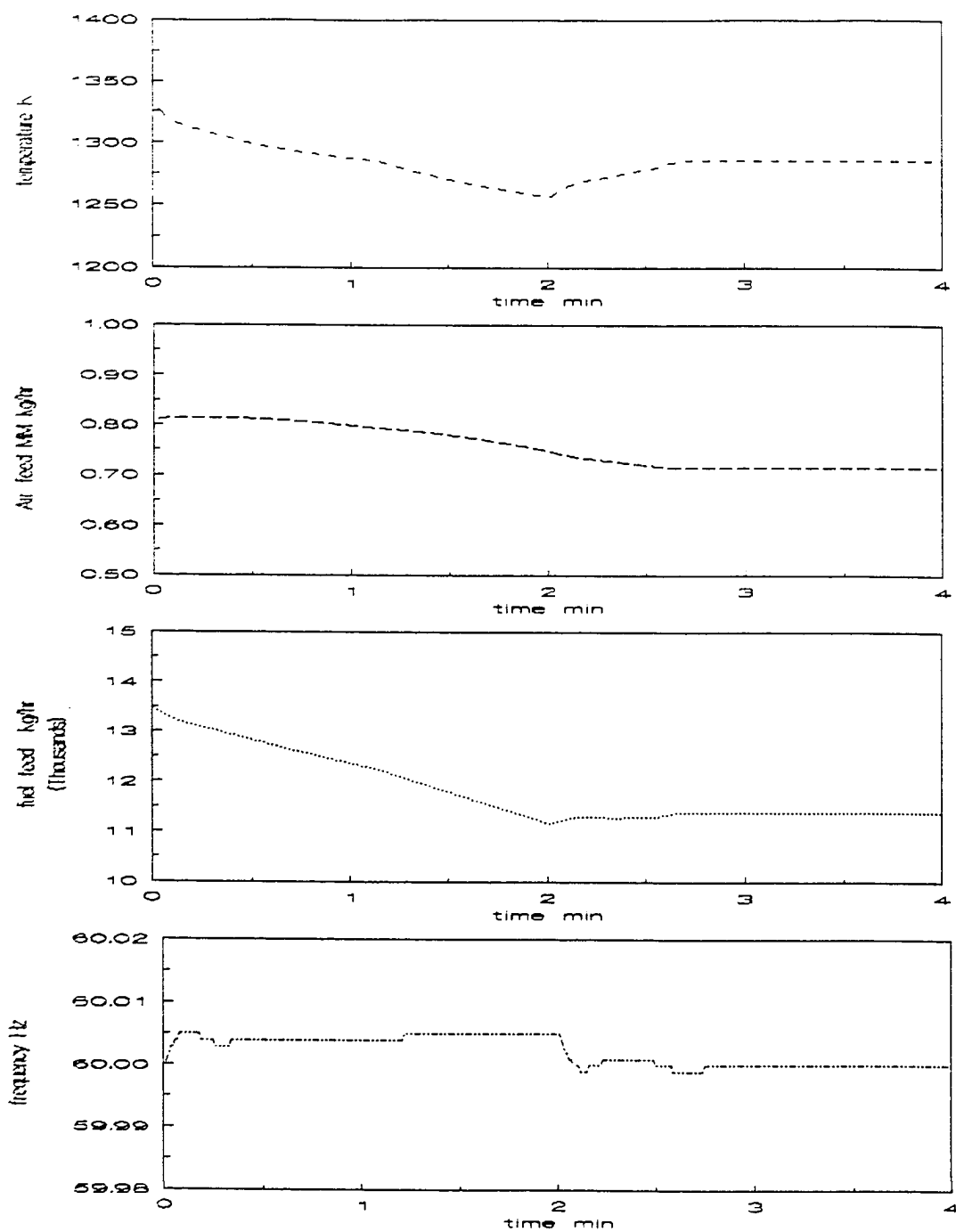


Fig.16 Combined Cycle load change, GT part

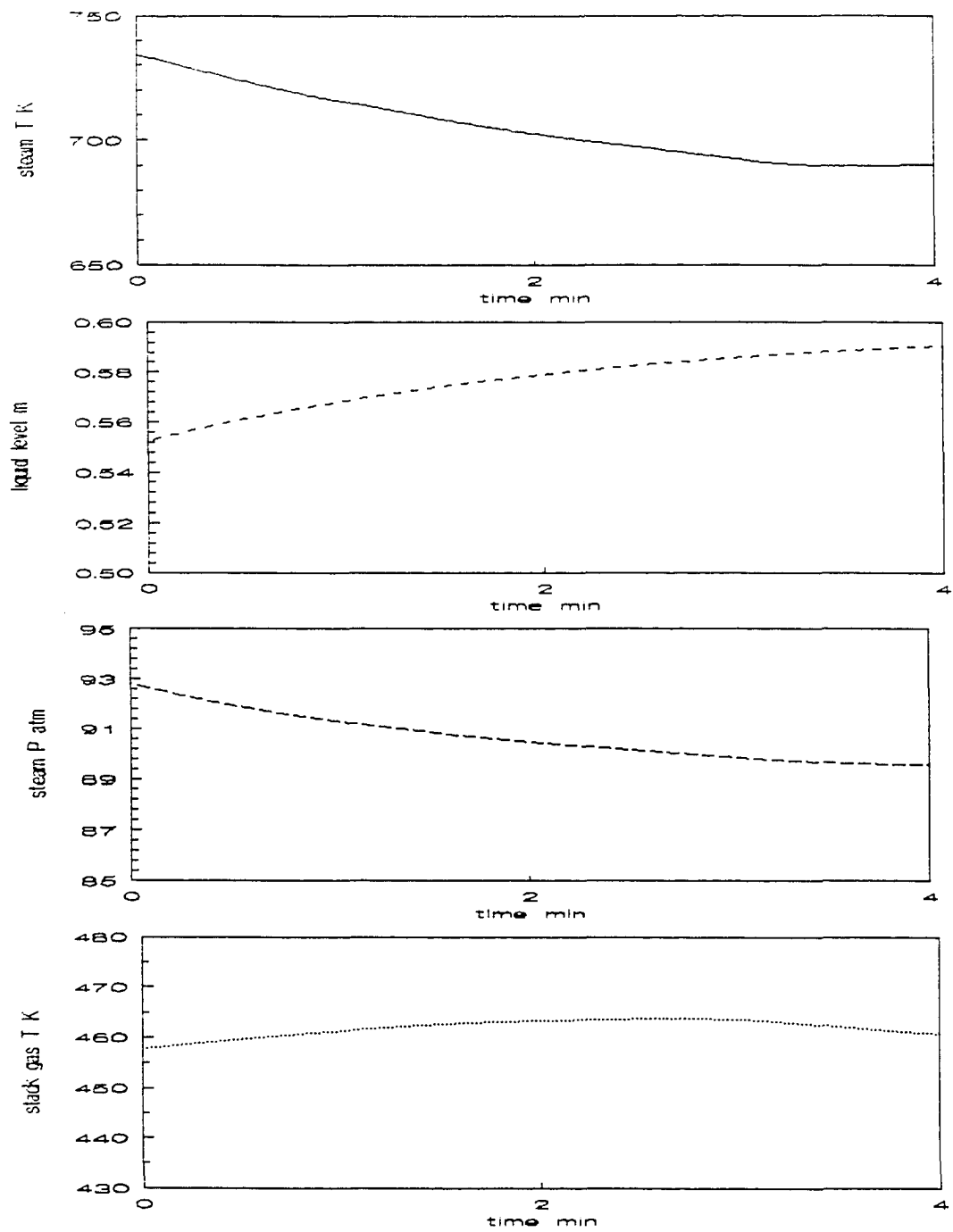


Fig.17 Combined Cycle load change, HRSG part

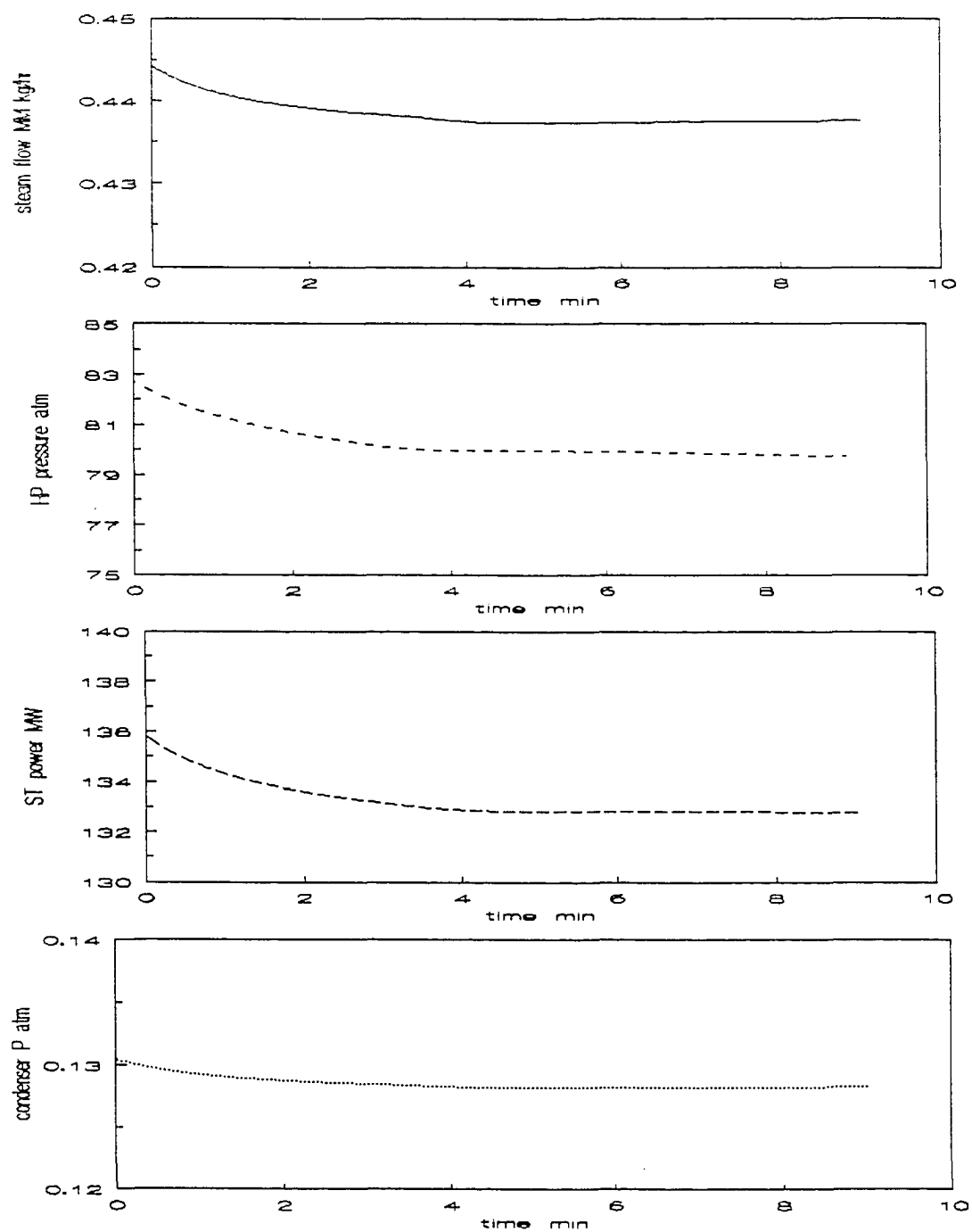


Fig.18 Combined Cycle load change, ST part

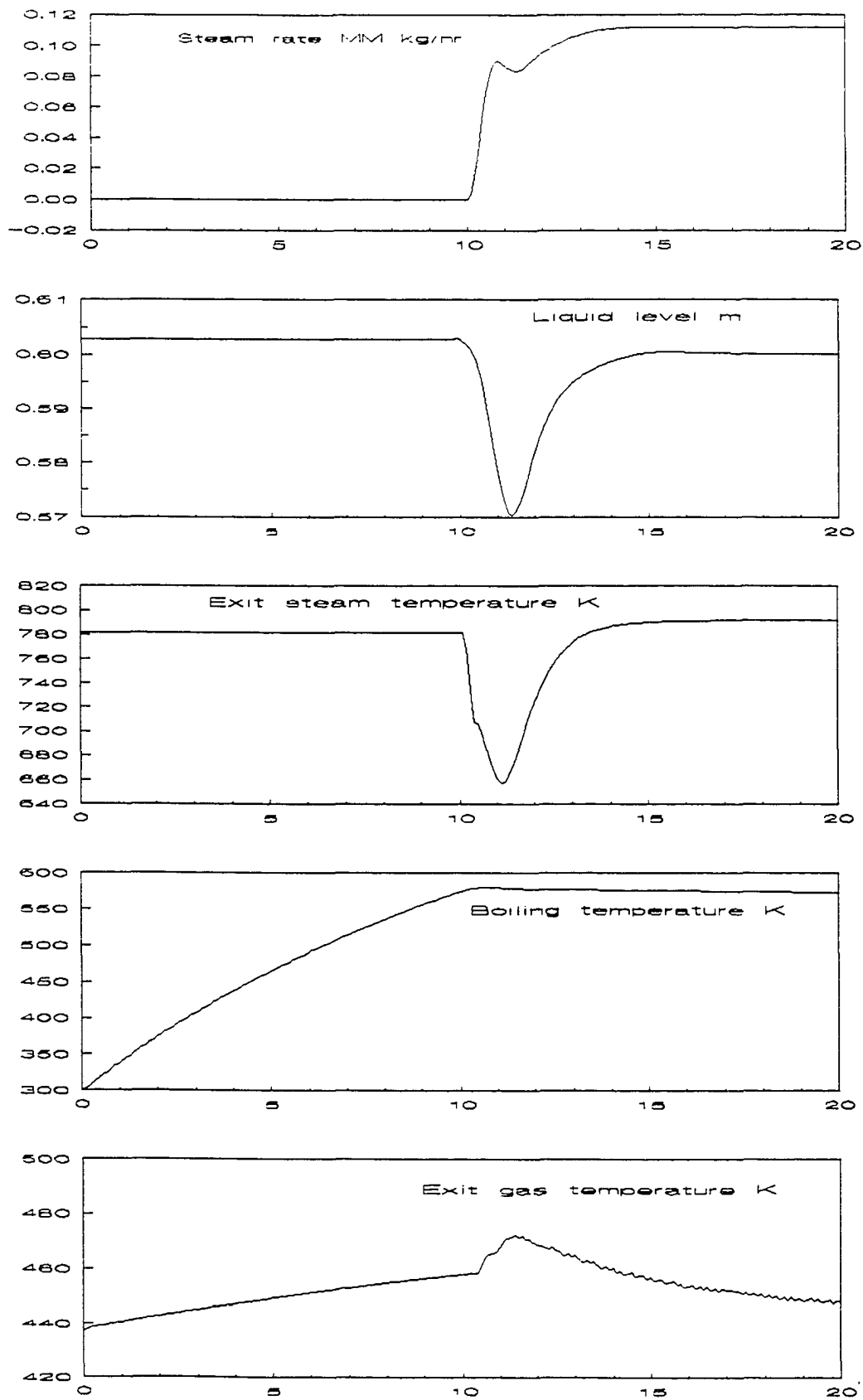


Fig.19 HRSG startup operation

Table 1 Input data for rated operation design

Definition			Default	User
NUMBER	GAS TURBINE (GT)		4	4.0
FUEL	FEED	KG/S	4.0	4.0
	LHV (CH ₄)	KCAL/KG	11875.	11875.
ELECTRIC FREQUENCY		REV/S	60.	60.
AIR COMPRESSOR	INLET P	ATM	1.0	1.0
	INLET T	K	300.	300.
	EXIT P	ATM	12.0	12.0
GAS TURBINE	INLET P	ATM	12.0	12.0
	DESIGN T	K	1500.	1500.
	EXIT P	ATM	1.1	1.1
STACK	EXIT P	ATM	1.0	1.0
T DIFFERENCE	HOTGAS-SUPERHEAT	K	40.	40.
	HOTGAS-BOILER	K	30.	30.
	BOILER-HOTWATER	K	25.	25.
HRSG SIZE	UG (273 K 1 ATM)	M/S	4.0	4.0
	HEIGHT/WIDTH		1.5	1.5
SUPERHEAT FIN	TUBE D	INCH	2.5	2.5
	TUBE CENT DISTANCE	INCH	6.0	6.0
	HEIGHT/DIAMETER		0.6	0.6
	DISTANCE/DIAMETER		0.5	0.5
	WIDTH/DISTANCE		0.25	0.25
REHEATER FIN	TUBE D	INCH	2.5	2.5
	TUBE CENT DISTANCE	INCH	6.0	6.0
	HEIGHT/DIAMETER		0.6	0.6
	DISTANCE/DIAMETER		0.5	0.5
	WIDTH/DISTANCE		0.25	0.25
BOILER FIN	TUBE D	INCH	2.5	2.5
	TUBE CENT DISTANCE	INCH	6.0	6.0
	HEIGHT/DIAMETER		0.6	0.6
	DISTANCE/DIAMETER		0.5	0.5
	WIDTH/DISTANCE		0.25	0.25
ECONOMIZER FIN	DRUM D	M	1.2	1.2
	TUBE D	INCH	2.0	2.0
	TUBE CENT DISTANCE	INCH	5.0	5.0
	HEIGHT/DIAMETER		0.6	0.6
	DISTANCE/DIAMETER		0.5	0.5
VALVE	WIDTH/DISTANCE		0.25	0.25
	HRSG WIDTH REDUCE		0.5	0.5
	POSITION		0.95	0.95
STEAM TURBINE	BEFORE VALVE	ATM	90.	90.
	HP INLET	ATM	85.	85.
	HP EXIT	ATM	25.	25.
	LP INLET	ATM	24.	24.
	LP EXIT	ATM	0.08	0.08
EFFICIENCY	GC INNER		0.85	0.85
	GT INNER		0.90	0.90
	HP INNER		0.85	0.85
	LP INNER		0.89	0.89
	MECHANICAL-ELECTRICAL		0.97	0.97

Table 2 Output of rated operation design

FUEL(CH4)	KG/S	GT	HRSG	ST	TOTAL
POWER	MW	4.0	.0	.0	16.0
EFFICIENCY	%	76.6	.0	89.0	395.3
WET FRACTION	%	38.50	66.93	34.67	49.69
				8.3	
P-T-PROFILE	1	2	3	4	5
P(GT) ATM	1.00	12.00	12.00	1.10	1.00
T(GT) K	300.00	450.41	1500.00	813.74	497.78
P(ST) ATM	80.00	75.00	20.00	19.00	.08
T(ST) K	773.74	771.35	661.49	773.74	314.14
T-PROFILE	GAS: IN	OUT	STEAM/WATER: IN	OUT	
REHEAT	813.7	736.2	661.5	773.7	
SUPERHEAT	813.7	736.2	564.1	773.7	
BOILER	736.2	594.1	564.1	564.1	
ECONOMIZER	594.1	497.8	314.1	539.1	
		SUPERHEAT	REHEAT	BOILER	ECONOMIZER
HRSG-HEIGHT M		6.8	6.8	6.8	6.8
HRSG-WIDTH M		4.5	4.5	4.5	2.3
PIPE SIZE IN		2.5	2.5	2.5	2.0
CENT DIST IN		6.0	6.0	6.0	5.0
FIN HEIGHT H/D		.6	.6	.6	.6
FIN INTERV L/D		.5	.5	.5	.5
FIN WIDTH B/L		.3	.3	.3	.3
SURFACE M ²		260.1	188.5	1008.2	603.2
UH CAL/M ² /K		101.0	102.0	82.7	66.5
VOLUMN M ³		3.1	2.2	12.0	5.7
RESID TIME S		4.6	1.1	492.8	293.2
ROW IN LINE		15	15	30	18
TUBES PER ROW		10	7	19	35
VELOCITY M/S		14.0	44.9	2.6	.5
DRUM D-L-H M		1.2	7.1	.6	
GAS-STEAM KG/S		158.6	18.1		

Table 3 Input data for steady state simulation

DEFINITION			DEFAULT	USER
ELEMENTS OF ARREY	GAS TURBINE		5	5
	SUPERHEAT		5	5
	REHEAT		5	5
	BOILER		5	5
	ECONOMIZER		5	5
	STEAM TURBINE		5	5
	CONTROL		5	5
GAS TURBINE	NUMBER		4	4
FUEL	FEED	KG/S	4.0	4.0
AIR COMPRESSOR	INLET P	ATM	1.0	1.0
	INLET T	K	300.	300.
	EXIT P	ATM	12.0	12.0
GAS TURBINE	INLET P	ATM	12.0	12.0
	EXIT P	ATM	1.1	1.1
STACK	EXIT P	ATM	1.0	1.0
VALVE	POSITION		0.95	0.95

Table 4 Summary of off design cases

	GT	ST	TOTAL
TURBINE	4	1	5
FUEL(CH4) KG/S	4.2	.0	16.8
POWER MW	80.4	93.2	414.8
EFFICIENCY %	38.50	35.34	49.66
LOAD %	105.00	104.94	104.99
	GT	ST	TOTAL
TURBINE	4	1	5
FUEL(CH4) KG/S	4.0	.0	16.0
POWER MW	76.6	89.0	395.3
EFFICIENCY %	38.50	34.67	49.69
LOAD %	100.00	100.00	100.00
	GT	ST	TOTAL
TURBINE	4	1	5
FUEL(CH4) KG/S	3.6	.0	14.4
POWER MW	68.9	80.2	355.8
EFFICIENCY %	38.50	35.23	49.70
LOAD %	90.00	90.26	90.06
	GT	ST	TOTAL
TURBINE	4	1	5
FUEL(CH4) KG/S	3.2	.0	12.8
POWER MW	51.7	43.8	250.6
EFFICIENCY %	32.49	32.07	39.38
LOAD %	67.52	49.29	63.42

3. SUMMARY OF PART II

The results of the work described in Chapters 4 and 5 can be summarized as follows:

- 1) A suite of dynamic models have been developed for the various equipment items used in combined cycle systems which represent these components with somewhat more fidelity than those published in the literature. This is particularly true of the HRSG models. Without models of this level of detail, many of the important dynamic effects (such as the inverse response of steam drum level) and process constraints (such as the approach temperature pinch) would not be accounted for.
- 2) Unfortunately, no detailed transient response data for combined cycle plants was available. The only data available is that for the start-up sequence which our model was able to predict with reasonable accuracy.
- 3) A multivariable control system analysis of the performance of control configurations for load following given in the literature showed these configurations to be reasonable. Multivariable tuning of the system based on a decentralized (multiple SISO loop) strategy gave reasonable controller responses.

This work is subject to some uncertainty, however, since it is based on a combined cycle plant design reverse-engineered from a limited amount of process data.

Further work is required, however, to better understand the dynamics and control of combined cycle systems. Some topics to be considered are:

- 1) One or more plant designs based on actual process data should be evaluated. Detailed transient response data for model validation is also desirable.
- 2) Some model improvements are indicated. A more detailed model of the gas turbine which includes a better representation of the thermal response should be developed. A more realistic void fraction representation for the boiler would give a more reliable prediction of the dynamics of this component.

- 3) Time did not permit the evaluation of the sensitivity of the controller tunings to equipment model parameter uncertainty. This should be done to assess the robustness of the control strategy that was studied.
- 4) Other MIMO tuning strategies should be evaluated. In particular, one based on Morari's robust design concepts should be studied.
- 5) Other control strategies should be evaluated, particularly those based on model predictive control, either Dynamic Matrix Control (DMC) or Internal Model Control (IMC).

4. Appendix - Combined Cycle Power Plant Model Parameter Evaluation

Our simulation and control studies are based on the combined cycle unit at the Gilbert Station of Jersey Central Power & Light. This unit uses four GE gas turbines, each with its own HRSG with supplementary firing, and a two-stage no-reheat steam turbine.

A limited amount of information concerning the design and operation of the plant was provided by Gilbert Station personnel. This information was insufficient to allow the direct evaluation of all of the parameters in our dynamic model. As a result, it was necessary to reverse engineer the plant design, i.e., to infer a design that meets all of the conditions set by the available information. This design was then used to evaluate most of the model parameters. The remainder were based on typical values available in the literature.

1. Plant information

The following information was provided:

1.1 Gas Turbine

Air compressor:	1.9 MM lb/hr at 14.6 Psia and 80 F
Fuel:	Feedrate of natural gas equivalent to 571 MM BTU/hr based on the LHV of 18900 BTU/lb (Flowrate of 30180 lb/hr)
Gas turbine:	Inlet conditions - 146.7 Psia at 1888 F Outlet conditions - 14.6 Psia at 960 F
Power generated:	50.75 MW

1.2 HRSG

Bypass leakage:	1 % of hot gas.
Refire fuel:	86 MM BTU/hr based on LHV of 18170 BTU/lb (temperature rise to about 1142 F)
Water feed:	0.261 MM lb/hr at 272 F
Steam generated:	1362 Psia and 951 F

1.3 Steam Turbine

Steam feed:	1.04 MM lb/hr at 1265 Psia and 951 F
Steam last exit:	0.9 MM lb/hr at 1.83 Psia and 123 F
Condenser:	56000 GPM (28 MMlb/hr) cooling water at inlet T=84 F and outlet T=116 F.
Power generated:	135 MW + 2 MW losses

2. Base Case Design

The base case design was developed as follows.

2.1 Gas Turbine

2.1.1 Equipment configuration and description

The configuration for this item of equipment is shown in Fig.5, where the air compressor, gas turbine and electrical generator are in co-shaft operation.

The unit is a GE MS-7000 gas turbine. The compressor has 17 stages; the gas turbine, 3 stages. The unit has a design ratio of rated/peak load equal to 90% with the exhaust temperature at 960 F and 1020 F respectively. The rotational speed is 3600 rpm. The inlet pressure to gas turbine is about 10 atm (EPRI RP-1048-3). The air flowrate through the compressor can be adjusted between 8% and 100% of maximum by opening or closing the inlet guide vane (IGV).

2.1.2 Assumptions

IGV is treated as a valve. All valves are assumed to have a linear valve character. The inner efficiencies of air compressor and gas turbine are assumed as constants. The gas flowrate is simulated based on compressor performance map for the compressor (De Laval,1970). Choked flow was assumed for the gas flow through the gas turbine. The inertial momentum of rotational parts is expressed by a time constant τ (for details, refer to model development), which is assumed constant.

2.1.3 Equipment parameters

The air flow rate under rated conditions, that is $P_{in}=1$ atm, $P_{out}=10$ atm was set the same as plant data, 0.85 MM kg/hr. The inner efficiency is selected as 0.86 from a reasonable range of 0.8-0.9. The discharge temperature for this design is 620 K, comparable to Weber's (1983) data at 605 K. The time constant of generator is assumed to be in the range of 15-30 s (Polonyi,1991).

For design flow rate, and pressure drop across the gas turbine, the flow performance parameter C_x was obtained as 30 where it satisfies the choke flow model.

$$F_g = \frac{C_{vx} \cdot P_{in}}{\sqrt{T_{in}}} \quad (120)$$

The inertia time constants for rotational parts were assumed as 3 s for the air compressor, 3 s for gas turbine, and 15 s for electrical generator. No effect of those settings on steady state simulation. In the dynamic case, it depends both on gas turbine itself and the load type if the motor is connected. For gas turbine itself, the summation of these τ is between 10-20 s (Polonyi,1991). The self-damping coefficient is about 1.5-2.

No data is available on the size of the combustion chamber, its volume was set to 20 m³ which gives a time constant 0.3 s at full flow.

Follow is a summary of the design parameters for the gas turbine.

AIR COMPRESSOR

Option:	co-shaft
Stage:	1
τ for inertia:	3 s
Designed condition:	$P_1=1$ atm, $T_1=300$ K, $P_2=10.2$ atm, $f=60$
Flowrate at designed case:	0.85 MKg/hr
IGV range:	80-100 %
τ of IGV:	3 s
Inner efficiency:	0.86

COMBUSTION CHAMBER

Option:	dynamic pressure
---------	------------------

Combustion chamber:	20 m ³ ($\tau=0.3$ s)
GAS TURBINE	
Option:	2 (co-shaft with compressor)
Time Constant:	3 s
Flow C_x :	30. ($F \approx C_x \Delta P$)
Inner efficiency:	0.85
ELECTRICAL GENERATOR	
Time constant:	15 s
Self-damping coefficient:	0.0005
Load:	50.75 MW
FUEL VALVE	
Option:	Calculated flowrate
Time constant:	0.
Valve C_v :	5.
Fuel upstream:	298 K & 19 atm

2.2 HRSG

2.2.1 Equipment configuration and description

The configuration for this item of equipment is shown in Fig.7, where it consists of refiring room, superheater, boiler, and economizer in sequence. The heat recovery steam generator is a forced circulation waste heat recovery boiler. The hot gases exiting the gas turbine refired or not flow past the HRSG giving up energy for steam generation. After that is a steam header to mix four streams together. The flow of steam from each HRSG is combined in the steam header. In the model, a drum pressure is maintained by output valve which is merged in the superheater model, where the flow resistance is adjustable. The user can set a valve between boiler and superheater. This control is necessary for a startup operation. After that, it may keep full open to limit the flow resistance.

As the temperature is not high enough, the tube with extended surface by fin was used. For the dynamic simulation, the most of the effect mass is due to the metal, it was included in the model.

2.2.2 Assumptions

The refiring combustion efficiency is 100%. On the base of plant data, the refiring temperature is about 890 K. The size of refiring room is set to 300 m³ to give a time constant of 0.6 s in saving computer CPU.

The HRSG is a box with a ratio of 1.5 of height to width (Walas,1988). The size of HRSG is designed based on the reasonable hot gas flow rate. The heat exchangers are so set that give the conventional cross flow. There is no heat loss through the wall. The descriptions of heat exchanger and boiler are referred to the user's manual (Rinard,1992).

The predominant heat transfer mechanism in the HRSG is convection. The input heat transfer coefficient is based on bare tube, it includes convection and radiation heat transfer outside tube, and convection inside tube. The coefficient of radiation accounts the effect of gas composition in a design model. Here we will give an approximately method to estimate the heat transfer coefficient and tube surface as well as fin design.

All of the valve are demonstrated by linear behavior.

2.2.3 Equipment parameters

Similar to the same hot gas velocity as the example (Babcock,1975), we assumed $U_g=5$ m/s under standard conditions. That is $F_g=\rho*U*A$, or $0.85E6=1.3*5*A*3600$. Therefore $A=36.3$ m², which gives $H=7.5$ m and $W=5$ m respectively. The tube length in vertical will be set as 7 m.

On the base of temperature profile and duty of heat transfer, approximate $U*A$ are estimated as

$$\begin{aligned} Q_s &= F * \Delta H = 0.111E6 * (-180.721 + 187.677)E6/18 = 0.043E12 \text{ J/hr} \\ \Delta T &= [(890-792) - (842-571)] / \ln[(890-792)/(842-571)] = 170 \text{ K} \\ UA &= 253E6 \text{ J/hr/K} \end{aligned}$$

$$\begin{aligned} Q_b &= F * \Delta H = 0.111E6 * (-187.667 + 224.492)E6/18 = 0.227E12 \text{ J/hr} \\ \Delta T &= [(842-571) - (591-571)] / \ln[(842-571)/(591-571)] = 96.3 \text{ K} \\ UA &= 2876E6 \text{ J/hr/K} \end{aligned}$$

$$Q_E = F \cdot \Delta H = 0.851E6 \cdot (9.2474 - 5.0195)E6 / 28.01 = 0.128E12 \text{ J/hr}$$

$$\Delta T = [(591 - 536) - (446 - 320)] / \ln[(591 - 536) / (446 - 320)] = 85.7 \text{ K}$$

$$UA = 1494E6 \text{ J/hr/K}$$

Taking the bared tube diameter and center distance as 2.5 and 12 inch for superheat, 2.5 inch for boiler, 2 and 5 inch for economizer, then based on the equation (Perry's, 1963)

$$U = 0.17 \frac{(U_{gmax})^{0.6}}{d^{0.4}} \quad (121)$$

where d in inch, U_g in ft/min, U in BTU/hr/ft²/F, the results are obtained as

$$U_S = 18.9 \quad U_B = 16.5 \quad U_E = 13.9$$

or in SI units J/hr/m²/K as

$$U_S = 3.6E6 \quad U_B = 0.314E6 \quad U_E = 0.264E6$$

They can be converted to a value based on the bare surface by

$$U' = U \frac{A_T}{A_o} - \beta \cdot U \quad \beta = 1 - \frac{H/D(H/D+1)}{L/D} \quad (122)$$

The corresponding total surface areas in m² are as

$$A_S = 703 \quad A_B = 9159 \quad A_E = 5659$$

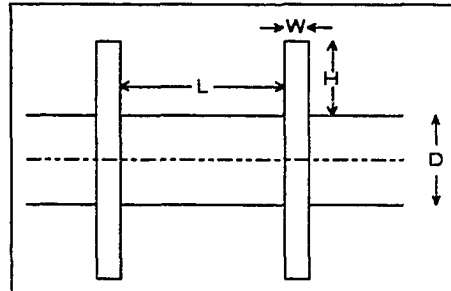
Based on tubes used in example, we assumed heat transfer enhancement by fin surface is $\beta=4$ for superheat and economizer, $\beta=5$ for boiler, which gives

$$A = 176 \text{ m}^2 (\approx 16 \cdot 8 \cdot 7 \cdot A_L) \quad A_L = 0.2$$

$$A = 1850 \text{ m}^2 (\approx 30 \cdot 44 \cdot 7 \cdot A_L) \quad A_L = 0.2$$

$$A = 1420 \text{ m}^2 (\approx 44 \cdot 30 \cdot 7 \cdot A_L) \quad A_L = 0.16$$

The fin can be designed assuming $H/D=0.6$. The results are as: $L/D=0.14$ for the superheat and economized, $L/D=0.11$ for boiler. The fin thickness is assumed as 1.5 mm for all of fin tubes.



The volume of steam header is designed based on 6 s replacement time. This

gives a lag between steam generated and steam turbine, which is needed in modeling.

The pump for water feed was assumed a constant pressure head. Then design flow rate based on the equation will be

$$F = C_X \cdot X \sqrt{\rho_L (P_{head} - P_{down})} \quad (123)$$

0.2E6 kg/hr for a $C_X=4$. The pump for water circulation was assumed to give constant flow rate, which was set 1.1E6 kg/hr, 10 times steam rate at rated conditions to limit the inverse response. PI control was applied to boiler liquid level with a reset time of 180 s to limit the inverse response and to give a smooth adjustment of water feed (Polonyi, 1990). The steam drum has the same length as HRSG width, the diameter is 1.2 m. The reset time for the drum pressure was set to 36 s for a smooth control.

Followings are the summation of setpoints or design points for the HRSG.

HRSG

Size: 7.5 m height, 5 m width
Gas velocity: 5 m/s at standard conditions

REFIRING

Refiring room: 300 m³ ($\tau=0.6$ s, assumed to avoid stiff)

SUPERHEATER

Tube area(bare): 176 m²
Heat transfer(bare base): 1.5 MJ/m²/K/hr
Rows and tubes/row: 16 X 8
Tube diameter and length: 2.5 in, 7 X 8 m
Fin size: H/D=0.6, L/D=0.14, W=1.5 mm

BOILER

Option: forced circulation
Drum D and L: 1.2 m; 5.0 m
Tube area(bared base): 1850 m²
Heat transfer(fin base): 1.25 MJ/m²/K/hr
Circulated flowrate: 1.1 Mkg/hr
Rows and tubes/row: 30 X 44
Tube diameter and length: 2.5 in, 7.0 m
Fin size: H/D=0.6, L/D=0.11, W=1.5 mm

ECONOMIZER

Tube area(bared base): 1470 m²
Heat transfer(fin base): 1.00 MJ/m²/K/hr
Rows and tubes/row: 44 X 30
Tube diameter and length: 2 in, 7 X 30 m
Fin size: H/D=0.6, L/D=0.14, W=1.5 mm

STEAM DRUM	
Volume:	20 m ³ ($\tau=6.5$ s assumed)
FEED WATER PUMP	
Pump head:	1200 mH ₂ O
VALVE	
Valve Cv:	4.
Time constant:	3.6 s

2.3 Steam turbine

2.3.1 Equipment configuration and description

The steam turbine system to be simulated here is illustrated in Fig.6. This system consists of main steam valve, high pressure steam turbine, low pressure turbine, steam condenser, and a reservoir for recycle water.

For a dynamic calculation of pressure, an intercomponent holder is applied for the steam turbine model. The steam turbine model has volume function to simulate pressure dynamically. Each turbine has its designed flowrate, inner efficiency and inertia time constant based on its rated conditions. The rated conditions come from the plant data. The pressure between two turbines is fixed as 32 atm (EPRI AP-2053). When the operation conditions change, the inner efficiency and the steam flow rate will change too.

The cooling water feeding into condenser is under constant feed rate. The condenser has a very big size for the steam condensation.

2.3.2 Assumptions

The main steam valve is linear performance, it is controlled by governor, so that the time lag is set to zero.

The steam turbine is assumed as a spool piece. The flow performance of the HP turbine follows the valve equation, the LP turbine follows the choked model. No extraction function for the LP turbine at present time. This induces different design in heat exchange (economizer).

2.3.3 Equipment parameters

The steam turbine flow time constant was set to 0.36 s at full flow rate. The inertia time is 15-20 s for generator. The auto-damping coefficient was set as 2s.

The inlet steam conditions to HP is determined from the steam header. Assumed LP inlet pressure is 32 atm, 630 K under rated conditions (EPRI AP-2053), then the flow coefficient at choked condition is 3.4 by eq(1). The calculation procedure of the flow coefficient is the same as in the gas turbine part, the result showed $C_x=115$ for main steam valve, $C_x=49$ for HP steam turbine.

On the base of temperature profile and duty of heat transfer, approximate $U \cdot A$ are estimated for the condenser as

$$\begin{aligned} Q_c &= F \cdot X \cdot \Delta H = 0.444E6 \cdot 0.877 \cdot (-177.62 + 238.98)E6/18 = 1.34E12 \text{ J/hr} \\ \Delta T &= [(320-298) + (320-319)]/2 = 11.5 \text{ K} \\ UA &= 0.117E12 \text{ J/hr/K} \end{aligned}$$

where X is exit steam wet fraction. The U is calculated based on condensation process assuming tube diameter is 1.0 inch

$$U = 0.725 \cdot 4187 \left[\frac{\lambda \rho_L^2 k^3}{\mu N d \Delta T} \right]^{1/4} \quad \text{J/hr/m}^2/\text{K} \quad (36)$$

where N is for the number of tubes in series (set $N=1$), d for tube diameter, λ for evaporation heat, μ for viscosity, k for heat conduct, which gives (Walas,1988)

$$U = 18E6 \text{ J/m}^2/\text{K/hr}.$$

The corresponding total surface areas becomes

$$A = 6000 \text{ m}^2$$

the volume tube occupied is $V=38 \text{ m}^3$. If tube center distance is 1.25 d , then the shell volume $V_{\text{shell}}=1.73V=65 \text{ m}^3$, which is corresponding to tube length $L=10 \text{ m}$, tube number is 7500, and shell diameter $D=2.7 \text{ m}$.

The volume of steam header is designed based on 6 s replacement time. This

gives a delay between steam generated and steam turbine.

Follow is a summary of design parameters for the steam turbine.

MAIN STEAM VALVE

τ of valve: 0.
Valve Cv: 115.

HP STEAM TURBINE

Option: HP turbine
 τ for inertia: 3.6 s
Design inlet condition: 792 K, 86 atm
inner efficiency: 0.85
Turbine C_x : 49
 τ for holdup: 0.36 s (assumed)

LP STEAM TURBINE

Option: LP turbine
 τ for inertia: 3.6 s
Design inlet condition: 630 K, 32 atm
inner efficiency: 0.89
Turbine C_x : 3.4
 τ for holdup: 0.36 s (assumed)

ELECTRICAL GENERATOR

τ for inertia: 15 s
Self-damping coefficient: 2
Load: 139 MW

CONDENSER

Option: vapor-liquid equilibrium
Total tube surface: 6000 m²
Shell size: D=2.7 m, L=10 m
Tube diameter and length: 1 in, 10 m
Tube center distance: 1.25 d
Number of tubes: 7500

3. Base Case Simulation Model

The dynamic simulation of the combined cycle unit was done using Version 0AX of the DYSIM dynamic process simulator developed at City college. For the details, refer to the DYSIM/VERSION 0A User Manual (Rinard,1993).

The base case system being simulated is shown in Fig.5-Fig.8.

The base case simulation uses only one gas turbine/HRSG combined with the steam turbine. To match the conditions of the Gilbert plant, it is assumed that the four turbines are operating identically in parallel. Thus the output of the single gas turbine/

HRSG is manipulated by a factor of 4 to generate the input to the steam turbine. There are two ways to configure the simulation. One is using 4 GT/HRSG's, which will form a big system or a big start file. It has an advantage in running GT in sequence. Another way is using a mixer to enlarge steam rate by 4 at exit of one HRSG. It is easy to handle. But it can not run sequence operation. As the first test, here is the second case results.

The flow sheets include equipments, controllers and streams. The data bus, as well as operating conditions (Fig.5 to Fig.8). Temperature profile and pinch points of the HRSG operation are shown in Fig.10.

3.1 Model Description

When the DYSIM is applied to plant case simulation, the feed conditions are set as given as the plant, but the operation are controlled to setpoints by manipulated variables such as valve position, feed rates. As the lack of detailed data, some of parameters have been estimated in above sections in order to run the DYSIM for the combined cycle power plant.

On the base of parameter settings, the steady state results are obtained after running the DYSIM dynamically. They have been done in two ways, One is running GT, HRSG, ST in sequence, that is a steady state data of GT was fed for HRSG, and so on. Another is running whole system simultaneously.

Here, some of steady state results are listed. The reansient behavior will be discussed some place else.

3.2 Results

Gas Turbine

Air feed to compressor:	0.841 MM Kg/hr at 300 K & 1 atm
Air flow out:	0.841 MM Kg/hr at 623 K & 10 atm

Fuel feed:	13834 Kg/hr at 298 K & 10 atm
Combusted gas out:	0.855 MM Kg/hr at 1305 K & 10 atm
Exhausted gas out:	0.855 MM Kg/hr at 786 K & 1.1 atm

HRS

Fuel feed:	2800 Kg/hr at 298 K & 1 atm
Exhaust gas in:	0.848 MM Kg/hr at 786 K & 1.1 atm
Hot gas after re-fired:	0.851 MM Kg/hr at 890 K & 1 atm
Hot gas after super:	0.851 MM Kg/hr at 842 K & 1 atm
Hot gas after boil:	0.851 MM Kg/hr at 591 K & 1 atm
Hot gas after econ:	0.851 MM Kg/hr at 446 K & 1 atm
Water before pump:	0.111 MM Kg/hr at 320 K & 1 atm
Water after pump:	0.111 MM Kg/hr at 320 K & 100 atm
Water after econ:	0.111 MM Kg/hr at 536 K & 97 atm
Steam after boil:	0.111 MM Kg/hr at 571 K & 97 atm
Steam after super:	0.111 MM Kg/hr at 792 K & 97 atm
Steam after drum:	0.444 MM Kg/hr at 792 K & 93 atm

Steam Turbine

Power generated;	139 MW
Steam before valve:	0.444 MM Kg/hr at 792 K & 93 atm
Steam before HP turbine:	0.444 MM Kg/hr at 792 K & 84.5 atm
Steam before LP turbine:	0.444 MM Kg/hr at 625 K & 33.8 atm
Steam before condenser:	0.444 MM Kg/hr at 320 K & 0.11 atm
Water after condenser;	0.444 MM Kg/hr at 320 K & 0.11 atm
Cooling water in:	12 MM Kg/hr at 298 K & 1 atm
Cooling water out:	12 MM Kg/hr at 319 K & 1 atm

Some comparisons between plant data and simulated data are listed in following two tables:

The difference is at a big size economizer design as for the no extraction steam turbine. As the gas exit temperature is reduced, the efficiency is a little higher than the plant case. Fig.10 is a result of Q-T for the HRS.

Tab. 5 Gas turbine

CASE	AIR KG/HR	FUEL KG/HR	T EXIT K
PLANT	863000	13702	789
SIMULATED	841000	13834	786

Tab.6 Steam turbine

CASE	POWER MW	T IN K	T OUT K	STEAM KG/HR
PLANT	137	797	323	472000
SIMULATED	139	792	320	444000

5. REFERENCE

ABB manual, "ABB control systems for combined cycle powerplants", ABB review, 1989.

Anderson K., "Advanced control techniques reduce power plant stress", Power, May 1991, 53

Babcock & Wilcox, Steam/Its Generation and Use, 38th edition, Babcock & Wilcox Co., (p.14-2), 1975

Baker J. et al, "Automation of Multi-generator Combined Cycle Plants", ASME, 1975

Baker J.M. et al, "Control and Automation of Single generator combined cycle plants", ASME-IEEE joint power generation conf., Portland, Sept. 1975

Baker, J.M. & A.C. Dolbec, "Stag combined cycle plant controls-flexibility and reliability", GE, April, 1973

Cohen H. et al, Gas Turbine Theory, Longman, (Wiley & Sons), New York, 1991

Corman J.C., "Experimental simulation and advanced control aspect of the IGCC concept: part I-simulation results", EPRI 3rd Annual Contractors' Conf. on coal gasification, Nov. 1983

De Laval's Engineering Handbook, McGraw-Hill, New York, 1970

Dieck-Assad, G. "Development of a state space boiler model for process optimization", Simulation, Oct. 1990

Fan Z. & I.H. Rinard, "Dynamic Simulator for Combined Cycle Power Plants - User Manual", City College of CUNY, New York, NY, Jan. 18, 1993

Fawke A.J. & H.I.H. Saravanamuttoo, "Digital computer methods for prediction of gas turbine dynamic response"

Ganapathy V., "HRSG design method optimize power plant efficiency", Power Eng., P40, May 1991

Gunn, D. & R. Horton, Industrial Boilers, Longman and John Wiley & Sons, New York, 1989

Holt, N.A. (editor), "Proceedings, 9th EPRI Conference on Gasification Power Plant", July, 1992

IEEE committee report, "Dynamic models for steam and hydro turbines in power system studies", IEEE PES Winter Meeting, New York, Jan 28, 1973

- Jian T.C., Principle of Steam Turbines, Power Pub., Peking, 1985
- Kehlhofer R., Combined-Cycle Gas & Steam Turbine Power Plants, The Fairmont Press, Lilburn, GA, 1990
- Knowles J.B., Simulation and Control of Electrical Power Stations, Wiley, New York, 1990
- Kwon W.H. et al., "On the multivariable robust control of a boiler-turbine system", IFAC, Power system and power plant control, Seoul, Korea, 1989
- Lausterer G.K. et al., "Modular modelling applied to a Benson boiler", Proc. 1984 ACC, June 6-8, 1984
- Leznov S. & Taits A., Power Station and Substation Maintenance, Mir, Moscow, 1983
- Li, K.W. & A.P. Priddy, Power Plant System Design, John Wiley & Sons, New York, 1985
- Luyben, W.L., "Process modeling, simulation and control for chemical engineering", 2nd edit, McGraw Hill, 1990
- Perry, J.H. Chemical Engineer's Handbook, 4th edit, McGraw-Hill, New York, 1963
- Polonyi, J.G. Power & Process Control Systems, McGraw-Hill, New York, 1991
- Quentin G.H., "Control analysis of gasification/combined cycle (GCC) power plants", Chem. Process Control II, 415, 1983
- Reason J., "Why high-performance controls are vital for gas turbines", Elec. World, April 1989, 71
- Rinard, I.H., "Process Systems Simulation: Its Use in Design Education for Engineers", Proceedings on Engineering Education, Curriculum, Innovation and Integration, Aunle and S. Carmi, Edit, United Engineering Trustess, Inc., New York p.321-326, 1992
- Rinard, I.H., "User Manual for DYSIMO - Version 0A", City College of CUNY, New York, NY 10031, Jan. 11, 1993
- Sawyer's Gas Turbine Engineering Handbook, V1, V3, Turbo-machinery Int. Pub., Connecticut, 1985)
- Takaoka H. & Tanaka S., "Overall control system of multi-stage gas/steam combined cycle plants", IFAC Modelling & control of Electric power plants, Como, 1983
- Walas, S.M., Chemical Process Equipment: Selection and Design, Butterworths, Boston, 1988

Weber,D.S., G.H.Quentin & J.E.McDaniel, "simulation objectives and requirements for advanced gasification based power plant", 3rd EPRI conference on coal gasification, 1983

Weber D.S., "Improved economic operation of power systems", EPRI topical report, Aug. 1978

Yang,S.M., Heat Transfer, University Pub., Beijing, 1980

EPRI AF-880, "Preliminary design study for an IGCC power plant" Aug. 1978

EPRI AF-911, "Application survey/evaluation of gas turbine needs", Nov. 1978

EPRI AP-2487, "Cool water coal gasification program - first annual progress report", July,1982

EPRI AP-2053, "Operation of gasification combined cycle (GCC) power plants on a utility network-a simulation analysis", Oct. 1981

EPRI AP-5027, "Analysis of the Off-design Performance and Phased Construction of IGCC power Plants, V1: Results and Conclusions; V2:Models and Procedures", Feb., 1987

EPRI AP-5975, "Soft-start fuel control modifications for large heavy-duty combustion turbines", Oct.1988

EPRI CS-3016, "Modular modeling system (MMS)", April 1987

EPRI CS-6047, "Dynamic Simulation and Performance Evaluation of Fossil Power Plants", Dec., 1988

EPRI EL-6659, "Proceedings: power system operations-research needs and priorities", Jan. 1990

EPRI GS-6173, "Development of an IGCC power plqnt simulator", 1989