Computer Modeling and Simulation of Coal fired Cogeneration Power Plant

By

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Acknowledgment

'… *not by might but by my Spirit*'. Thank you Lord; You bring me to the end of this study.

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Table of Contents

Abstract

This thesis describes a power plant simulator which has been developed using MATLAB code.

The basis of the power plant modeled here is the design data of Coal-Phosphate Fertilizer Complex Project (COFCOP), and the type of boiler and turbine are taken as selected by the project. Therefore, the boiler modeled here is a circulating fluidized bed coal combustor type, and the turbine is an extraction condensing type at heat to power ratio.

Conservation equations and characteristic curves are used to model each component of the power plant. From these mathematical model equations, the component modules are developed using MATLAB code. Output parameters are obtained from input data to the component modules. Simulation of systems, subsystems and components are accomplished using the Newton-Raphson technique.

The total system of the power plant is simulated by a program called *powerplant_main*. In this program there are four basic solvers. These are: the turbine solver, the boiler solver, the water flow solver and the steam flow solver. Each solver has its own residue generator function and Jacobean generator function. The Gaussian elimination function is called by any solver during simulation.

The power plant in COFCOP is still in design stage, and it cannot be used for program verification in this paper. Hence, a hypothetical power plant data is taken and the output of the simulation closely approximates the real case.

CHAPTER 1

Introduction

During the last two decades, there has been a tremendous development of software for modeling and simulation of energy systems. The need for precise information about the behavior of such systems has increased in connection with a higher degree of integration of processes, optimization of processes, getting closer to the limits of strength of the material due to temperature, pressure, or dimensions, or a need for better control of the processes due to for example quality of the product. The development of system simulator is also important in the training of operators, since it's difficult to train them by performing experiments on actual plant.

Based on this need of simulation of different systems, in this thesis it is tried to prepare a steam power plant simulator using MATLAB code. The simulator is developed by modeling each component of the power plant using conservation equations and its characteristic curve given by manufacturers. The program is designed to give the necessary output of the simulation at any intermediate point.

1.1 Aim and Scope of the Thesis

This thesis mainly focuses on the development of a computer simulation program module of a coal-fired cogeneration power plant.

The simulation program is aimed to be applied for the non-design conditions, such as part-load, or full load conditions of the power plant. It helps to investigate off-design operation to be sure that pressures, temperatures, or flow rates will not be too high or too low. It can also be applied to existing systems when there is an operating problem or a possible improvement is being sought. The effect of changing a component on the system can be examined before the actual change to ensure that the operating problem will be corrected and to find the cheapest means of achieving the desired improvement. Moreover, the program will give complete mass and energy balance of the cogeneration plant which will assist in planning of inputs and refuse disposals, and estimation of losses.

1.2 Modern Power Plant Analysis

Steam power plants, based on the Rankin cycle, are widely used to generate large amounts of power using coal, gas, or oil. The efficiencies of steam plants are quite high, more than 40 percent. Because of the large number of steam power plants and the accumulated experience and knowledge, their reliability is high and repair facilities are widely available. With improved combustion and technology, pollution, one of the concerns of using fossil fuels, has been significantly reduced, although at some cost. In some cases where steam is required for processing, power is a secondary consideration.

The major components of a fossil-fueled steam power plant are the boiler unit, the turbine unit, heat exchangers and different pumps. Here the boiler and turbine are discussed.

1.2.1 Power Boilers

Coal-fired boilers are one and most used worldwide. These types of boilers generate approximately 38% of the electric power generation worldwide and will continue to be major contributors in the future [33].

Coal achieved this major role because the worldwide resource is large, evenly distributed and easily extracted leading to low and stable prices. These factors are also resulting in a major role for coal in meeting the rapidly increasing electricity demand in the developing areas of the world as they strive to quickly industrialized and improve their standard of living.

Coal firing equipments can be classified into three major types according to different combustion modes:

1) Chamber Combustion Firing

In chamber combustion firing, coal is crushed into fine particles in pulverizers and injected into a chamber where combustion is taking place. Pulverized coal fired boiler uses this combustion principle.

2) Mechanical Grate Firing

Hand firing of coal is definitely excluded for all but the smallest furnaces. Boilers of moderate and high capacity cannot be fired physically and economically by hand and the uniformity of operation and control achieved by mechanical firing. The different types of equipment used for firing coal mechanically on grates are classified as:

- Chain grate and traveling grate stokers
- Underfeed stokers

- Spreader stokers
- Water cooled, vibrating grate stokers

The characteristics of coal that influence stoker selection are volatile matter content, caking qualities, ash content, and the softening and fusion temperatures of the ash.

3) Fluidized-Bed Coal Firing

These boilers are different in combustion mode and adoptable to different coal types, and are, therefore, used for different applications.

1.2.1.1 Pulverized Coal Boilers

Pulverized coal fired boiler technology is a major contribution to meeting worldwide electrical power generation requirements with approximately 970,000MW_e of capacity in operation 2000 [33].

The overall thermodynamic cycle efficiency of pulverized coal-fired systems increased continuously until peaking in the late 1950s and early 1960s as technical and economic factors made it unattractive to push the envelop further.

Increasing environmental awareness, in the 1970s, focused pulverized coal-fired power system research on reducing air pollution from coal-fired units so that today, a typical new coal-fired boiler system eliminates up to 97% of the combined particulate, sulfur dioxide and nitrogen oxide emissions [33].

The principle of combustion in a PC firing boiler is that coal is smashed into powder of magnitude in micrometers in pulverizers and mixed with air. Then, the mixture of coal powder and air is blown into the combustion chamber for combustion. Within the furnace, the radiant energy absorbed by the coal particles effects distillation of the volatile matter, which subsequently burns as a gas. The remaining coke particles burn with diffusion of oxygen through the layer of CO_2, CO, and N_2 that surrounds each particle. In order to achieve good combustion, the burners should provide for proper initial mixing of the fuel and air, followed by adequate turbulence to assist in the penetration of oxygen to the solid coke. Its applicable coal types include bituminous coal, anthracite coal and lean coal. It is small in load regulation range and fast in load change. It is the highest boiler thermal efficiency, normally greater than 90%, and the boiler capacity ranges from 35t/h to 2000t/h.

1.2.1.2 Circulating Fluidized Bed Boilers

Circulating fluidized bed (CFB) technology has emerged as an environmentally acceptable technology for burning a wide range of solid fuels to generate steam and electric power. CFB, although less than 20years old, is a mature technology with more than 400 CFB boilers in operation worldwide, ranging from $5MW_e$ to $250Mw_e$ [33]

The CFB technology utilizes the fluidized bed principle in which crushed (6-12mm*0size) fuel and lime stone are injected into the furnace or combustor. The particles are suspended in a stream of upwardly flowing air (60-70% of the total air) which enters the bottom of the furnace through air distribution nozzles. The balance of combustion air is admitted above the bottom of the furnace as a secondary air. While combustion takes place, the fine particles (<450microns) are elutriated out of the furnace with the flue gas. The particles are then collected by the solids separators and circulated back into the furnace. This combustion process is called circulating fluidized bed (CFB). The particles' circulation provides efficient heat transfer to the furnace walls and longer residence time for carbon and lime stone utilization.

The modern way of burning solid fuels requires fuel flexibility and reliable technology, plus good combustion efficiency with low emissions. CFB can handle a wide range of fuels such as coal, waste coal, anthracite, lignite, petroleum coke and agricultural wastes, with low heating value (>1500kcal/kg), high moisture content (<55%), and high ash content (<60%).

1.2.1.3 CFB Versus PC Technology

CFB technology brings the capability of designs for a wide rage of fuels from low quality to high quality fuels, lower emissions, elimination of high maintenance pulverizes, low auxiliary fuel support and reduce life cycle costs.

The lower combustion temperature of CFB highly reduces the formation of NO_x formation and the ability to capture SO_2 with lime stone injection in the furnace. Even though the combustion temperature of CFB is low, the fuel residence time is higher than PC, which results in good combustion efficiencies comparable to PC. The PC pulverizes, which grind the coal to 70% less than 75 microns, require significant maintenance expenses. These costs are virtually eliminated in CFB because the coal is crushed to 12-6mm*0 size [33]. Even though CFB boiler equipment is designed for relatively lower flue gas velocities, the heat transfer coefficient of the CFB furnace is nearly double that of PC which makes the furnace compact. One of the most important aspects is that CFB boilers release very low level of SO_2 and NO_x pollutants compared to PC. PC units need a scrubber system, which requires additional maintenance [33].

Table 1-1 Benefits of a CFB boiler over a PC fired boiler [33]

Description	CFB boiler	PC boiler	Benefits of CFB
• Fuel size	6-12mm*0	>70% <75microns	Crushing cost is reduced
• Fuel range(ash+moist.)	up to 75%	up to 60%	Accepts wider range
• Higher sulfur fuels(1-6%)	limestone injection	FGD* plant require	Less expensive SO_2 removal sys.
• Auxiliary fuel support	up to 20-30%	up to 60%	Less oil/gas consumption
• Auxiliary power cons.	Slightly higher	lower	If FGD is used in PC, CFB power is lower
Emissions			
SO_2, ppm	<200	<250 with FGD	Lower emissions in process, Less expensive
NO_x , ppm	<100	<100 with SCR**	No SCR system is required
Boiler efficiency,%	same	same	No difference
O&M cost	5-10%lower	5-10% higher	Lower because of less moving equipment
Capital cost	5-10% higher	5-10% lower w/o FGD[1] & SCR[2]	
	8-15% lowers	8-15% higher with FGD & SCR	

As described above, for Ethiopian coal, which has low calorific value, high ash and moisture content, it is wise to select CFB boilers which are fuel flexible.

1.2.2 Power Turbines

Steam turbines are the most commonly employed prime movers for cogeneration applications, particularly in industries and for district heating. The technology is well proven in sugar industries of this country which have high demand for both electricity and large quantity of steam at high and low pressures.

[1] **Flue Gas Desulphurization**
[2] **Selective Catalytic Reduction**

1.2.2.1 Extraction Back Pressure Turbine

A cogeneration system using a backpressure steam turbine consists of boiler, turbine, heat exchanger and pump. In this steam turbine, the incoming high pressure steam is expanded to a lower pressure level, converting the thermal energy of high pressure steam to kinetic energy through nozzles and then to mechanical power through rotating blades. Thermal energy of the turbine exhaust steam is then transferred to another fluid, water, air, etc., in a heat exchanger, providing heat to the processes. For instance, the air heated in the heat exchanger can be used to dry products in food processing industries.

Depending on the pressure (or temperature) levels at which process steam is required, backpressure steam turbines can have different configurations.

In extraction and double extraction back pressure turbines, some amount of steam is extracted from the turbine after being expanded to a certain pressure level. The extracted steam meets the heat demands at pressure levels higher than the exhaust pressure of the steam turbine.

The backpressure steam turbine has a higher heat to power ratio and gives higher overall efficiency. Furthermore, back pressure turbine cogeneration systems need less auxiliary equipment than condensing systems, leading to lower initial investment costs.

The efficiency of a backpressure steam turbine cogeneration system is the highest. In cases where 100 per cent backpressure exhaust steam is used, the only inefficiencies are gear drive and electric generator losses, and the inefficiency of steam generation. Therefore, the thermal efficiency of the turbine could reach as much as 90 per cent.

1.2.2.2 Extraction Condensing Turbine

The extraction condensing turbines have higher power to heat ratio in comparison with backpressure turbines. Although condensing systems need more auxiliary equipment such as the condenser and cooling towers, better matching of electrical power and heat demand can be obtained where electricity demand is much higher than the steam demand and the load patterns are highly fluctuating.

The overall thermal efficiency of an extraction condensing turbine cogeneration system is lower than that of back pressure turbine system; basically because the exhaust heat cannot be utilized (it is normally lost in the cooling water circuit). However, extraction condensing cogeneration systems have higher electricity generation efficiencies.

The advantages of extraction condensing turbine over extraction back pressure turbine are, in extraction condensing turbine:

- It is possible to extract steam from the intermediate stages of this turbine for process heat requirement.
- In case of a sudden drop in steam load needed by steam users, the surplus steam can continue to expand in the turbine stages after the extraction point for power generation.

A reheat cycle may exist in both types of turbines. In the reheat cycle, steam is extracted from the turbine at the exhaust of high pressure turbine and reheated in the boiler. Reheat cycles improve the overall thermal efficiency and eliminate any moisture that may form as the steam pressure and temperature are lowered in the low pressure turbine. Steam turbines may also include a regenerative cycle where the steam is extracted from the turbine and used to preheat the boiler feed water. The turbine type which is selected to be modeled here is of the extraction condensing one.

CHAPTER 2

Modeling & Simulation of Thermal Systems

The thermal engineer had, due to economic and environmental demands during the last decades, to focus on improving efficiency and reducing pollutants discharge from installations. Computer simulation is only one of the tools that may be applied in search for optimal solutions.

The plants that may be characterized as thermal systems are numerous. *A thermal system is a technical installation*, *which employs fluids as energy carriers between its components and is the subject of a thermodynamic analysis.*

The function of the installation may be to convert energy from one form to another and deliver it to consumers (such as power plants), to transport energy from one place to another (such as district heating), or to deliver a product which may be a working fluid at a given state (industrial process). The processes in the equipment may be of mechanical, thermo-dynamical or chemical nature.

Models of thermal systems are developed by modeling the devices or components of the system as separate subsystems by considering a control volume for each. The system is built by connecting these component models at the points where flow crosses the boundary of the control volume.

Modeling deals primarily with establishing valid quantitative relationships between real systems and model of real systems. This means, models are mathematical equations that represent physical systems. Simulation is concerned with implementing, via computer codes, the models in such a manner that the resulting computations have a high degree of fidelity to the real system.

System simulation is often used to denote the output of the modeling and simulation of real systems. Hodge [1] divides this process into four basic tasks. These are:

1) Preparation of mathematical model for the system
2) Preparation of mathematical model for the simulation
3) Preparation of simulation computer program
4) Verifying the simulation

These four tasks should be accomplished in order to achieve fidelity in the system simulation.

As discussed earlier, the mathematical formulation for most thermal processes rests on the concept of conservation of physically meaningful quantities as applied to control volumes. So, the development of many models in thermal systems involves control volume considerations.

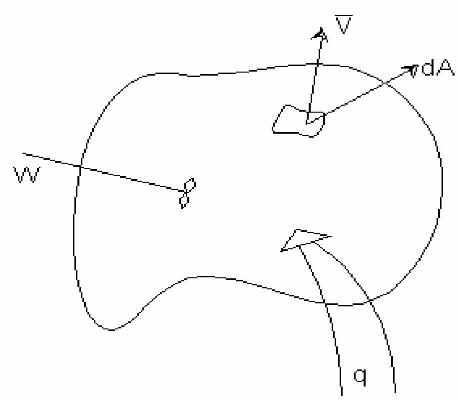

Fig.2.1 Control Volume

Considering Fig.2.1, if X is an extensive property then:

$$X_i - X_o = X_s \tag{2-1}$$

where: X_i = quantity entering control volume

X_o = quantity exiting control volume

X_s = quantity stored in control volume

The rate form of Eqn.(2-1) can be:

$$\dot{X}_i - \dot{X}_o = \frac{d\dot{X}_s}{dt} \tag{2-2}$$

For steady state behavior, $\frac{d\dot{X}_s}{dt}$ becomes zero. Hence:

$$\dot{X}_i - \dot{X}_o = 0 \tag{2-3}$$

These equations are useful in developing conservation of mass and energy statements for components and systems.

The flux of a number of quantities can be expressed using the concept of a forcing potential, *W* and a resistance, *R*.

$$\dot{X} = \frac{W}{R} \tag{2-4}$$

This concept can be used for heat flux computation.

$$q = \frac{\Delta T}{R} \tag{2-5}$$

2.1 Curve Fitting

Curve fitting is widely used in component modeling since much quantitative component information is expressed in graphical or tabular form, neither of these modes is convenient for computer based application.

The most crucial thing in the curve fitting process is the selection of the functional form of the fit. The general expression of the curve fit is given by the polynomial expression:

$$y = a_0 + a_1 x + a_2 x^2 + ... + a_n x^n \tag{2-6}$$

Quadratic polynomials are frequently used for modeling components of which their characteristic curves are available, such as pumps. Here, in this paper the model of pumps are curve fitted by quadratic polynomial.

2.2 *Interpolation*

Interpolation is a useful alternative to curve fitting when information are available in tabular form. Interpolation methods are available in widely varying accuracy and complexity. Of these, the *Lagrange interpolating polynomials* are the most used.

In this thesis, tabulated data, such as steam tables, fluid physical property tables, and so on, are represented by interpolation of the data. The *Matlab* programming code has a built-in function for interpolating data in multiple degrees. So, these functions, especially *interp1* and *interp2,* can be directly used for parametric representation of data by interpolation.

2.3 *System Simulation Technique*

The most common steady state simulation procedures used for thermal systems are the multivariable *Newton-Raphson* and the *Hardy-Cross methods.*

2.3.1 Hardy-Cross Method

This method is generally an acceptable simulation technique for fluid flow in pipes or piping network. Hodge [1] has detail explanation and use of this method for complex piping net works.

2.3.2 Newton-Raphson Method

This method is more general for steady-state simulation and its detailed explanation is presented here. In this paper, it is used as the main technique for the simulation of the power plant system and other subsystems.

The Newton-Raphson method uses information flow diagram to represent each system component, links in an approximate manner all the component information flow diagrams, and develops equations to represent the system. In this technique of simulation of systems, the number of unknowns must be equal to the number of equations, and the equations must be independent; otherwise convergence will not be achieved.

The multivariable Newton-Raphson methodology is basically developed by considering a *Taylor Series expansion*. Hodge [1] explains the details of the development.

The basic procedure to implement this technique is best outlined by Stoecker [2]. These are:

1) Rewrite all equations in the form $R_i(x_1, x_2, ... x_n) = 0$

where $i = 1,2,...n$ and n is the number of variables.

2) Assume initial guess values for the variables, which can be $x_{1t}, x_{2t}, ..., x_{nt}$

3) Evaluate *residues* at these trial values of the variables, $R_i(x_{1t}, x_{2t}, ... x_{nt})$. This step is the generation of *residues*

4) Compute the *Jacobin matrix* of the system of equations, which is the partial differential form of the residue equations at the trial values.

$$J = \begin{bmatrix} \frac{\partial R_1}{\partial x_{1t}} & \frac{\partial R_1}{\partial x_{2t}} & . & . & . & \frac{\partial R_1}{\partial x_{nt}} \\ \frac{\partial R_2}{\partial x_{1t}} & \frac{\partial R_2}{\partial x_{2t}} & . & . & . & \frac{\partial R_2}{\partial x_{nt}} \\ . & . & & & & \\ . & . & & & & \\ . & . & & & & \\ \frac{\partial R_n}{\partial x_{1t}} & \frac{\partial R_n}{\partial x_{2t}} & . & . & . & \frac{\partial R_n}{\partial x_{nt}} \end{bmatrix} \tag{2-7}$$

5) Put the system of equations in the matrix representation of the form:

$$\begin{Bmatrix} \delta x_1 \\ \delta x_2 \\ . \\ . \\ . \\ \delta x_n \end{Bmatrix} = \begin{bmatrix} \frac{\partial R_1}{\partial x_{1t}} & \frac{\partial R_1}{\partial x_{2t}} & . & . & . & \frac{\partial R_1}{\partial x_{nt}} \\ \frac{\partial R_2}{\partial x_{1t}} & \frac{\partial R_2}{\partial x_{2t}} & . & . & . & \frac{\partial R_2}{\partial x_{nt}} \\ . & . & & & & \\ . & . & & & & \\ . & . & & & & \\ \frac{\partial R_n}{\partial x_{1t}} & \frac{\partial R_n}{\partial x_{2t}} & . & . & . & \frac{\partial R_n}{\partial x_{nt}} \end{bmatrix}^{-1} \begin{Bmatrix} R_1 \\ R_2 \\ . \\ . \\ . \\ R_n \end{Bmatrix} \tag{2-8}$$

6) Solve the set of equations in step 5 using *Gaussian Elimination*.

7) Update the solution for the next iteration.

$$x_i^{(J)} = x_i^{(J-1)} - \delta x_i^{(J)} \qquad i = 1 \; to \; n \tag{2-9}$$

8) Check the value identified in step 7 converge with the guessed value in step 2. If convergence is achieved, stop iteration. If not, take the new values of the variables to be trial values and continue iteration starting from step 3

All models of components, sub-systems and systems in this paper are simulated applying the Newton-Raphson simulation technique discussed above.

CHAPTER 3

Basic Flow Models

3.1 Basic Equations of Fluid Flow in Pipes

Consider a fluid flowing from point1 to point 2 in a pipe of Fig. 3.1.

In the steady-state flow consideration, continuity equation will give:

$$m_1 = m_2 = m \tag{3-1}$$

If the flow is taken to be incompressible

$$V_1 A_1 = V_2 A_2 \tag{3-2}$$

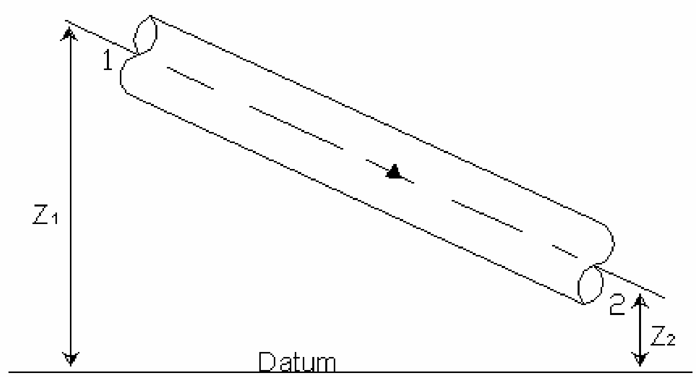

Fig.3.1 Fluid flow in pipe

where:

m = fluid mass flow rate, [kg/s]

V_1, V_2 = velocity of fluid in the pipe at the respective points, [m/s]

A_1, A_2 = pipe cross-sectional area at the respective points, [m^2]

The energy equation between the two points in the pipe can be written as:

$$\frac{p_1}{\gamma}+\frac{V_1^2}{2g}+Z_1=\frac{p_2}{\gamma}+\frac{V_2^2}{2g}+Z_2+h_f \qquad (3\text{-}3)$$

Assuming that the velocities at the two points are equal, Eqn.(3-3) can be reduced to:

$$h_f=\frac{\Delta p}{\gamma}+\Delta Z \qquad (3\text{-}4)$$

where: p_1, p_2 = Pressure at the respective points, [*Pa*]

Z_1, Z_2 = Elevation of the respective points from the datum, [*m*]

h_f = Total Head loss between the two points, [*m]*

$\gamma = \rho g$, [$kg/(m^2s^2)$]

ρ = Density of the incompressible fluid,[kg/m^3]

Eqn.(3-3) is a model equation which relates the pressure of a fluid flowing between two points in pipe taking into consideration only the skin friction loss and elevation change. It is also possible to consider devises, such as valves, heat exchangers and soon, between the two points and consider losses in these devices.

3.1.1 ***Total Head loss***

The total head loss, h_f , is composed of major and minor losses. Major losses are losses associated with the pipe wall skin friction over the length of the pipe, and minor loses are losses caused by the flow of fluids through fittings, valves, and process equipments.

3.1.1.1 Pipe friction Loss

The total loss due to the flow of a fluid at an average velocity through a length L of pipe with a diameter d is given by the *Darcy-Weisbach relation*ship as follows:

$$h_f = f \frac{L}{d_i} \frac{V^2}{2g} \tag{3-5}$$

where: h_f = head loss due to skin friction in a pipe

f = Darcy-Weisbach friction factor

$$f = f\left(\mathrm{Re}, \frac{\varepsilon}{d_i}\right) \tag{3-6}$$

Re = Reynolds number based up on the pipe diameter

$\frac{\varepsilon}{d_i}$ = relative roughness of the pipe.

Eqn.(3-6) is represented graphically in the *Moody's Diagram* [1].

Fluid flow in pipes can be categorized as turbulent or laminar based on its *Reynolds number*. The friction factor in the Moody diagram is also classified for turbulent and for laminar flow. This diagram is not convenient for computer programming. To make it convenient, either tabulated graphical values which can be interpolated or algebraic expressions which give friction factor as a function of Reynolds number and relative roughness of the pipe should be available. For laminar flow, the friction factor is not a function of the relative roughness of the pipe. In the Moody diagram, this can be seen as a straight line; i.e., only function of Reynolds number. This relation, for laminar flow can be algebraically expressed by the Darcy-Weisbach friction factor representation as follows:

$$f = \frac{64}{\mathrm{Re}} \tag{3-7}$$

There are many algebraic representations for the turbulent flow region proposed by different researchers. *Colebrook* friction factor representation has been accepted as the most accurate representation of the Moody diagram. It has been given as:

$$\frac{1}{\sqrt{f}} = \log\left(\frac{\varepsilon}{3.7d} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)^{-2} \tag{3-8}$$

But, the equation is implicit in the friction factor and so iteration is required to obtain the friction factor for a specified Re and ε / d

There are other alternative equations for friction factor identification in the turbulent region, which are explicit and simple, the Swamee-Jain equation and the Halaand equation.

The Swamee and Jain equation is given by:

$$f = \frac{0.25}{\left[\log\left(\frac{\varepsilon}{3.7d} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]^2} \tag{3-9}$$

and the Halaand equation is expressed as:

$$f = \frac{0.3086}{\left\{\log\left[\frac{6.9}{\mathrm{Re}} + \left(\frac{\varepsilon}{3.7d}\right)^{1.11}\right]\right\}^2} \tag{3-10}$$

Due to their small error and easiness, most literatures recommend use of the Halaand or the Swamee and Jain equation.

Churchill provides an expression which can be an alternative to Eqns. (3-7), (3-9) and (3-10) and can be solved for the friction factor at all regions of flow.

$$f = 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + \frac{1}{(A+B)^{1.5}}\right]^{1/12} \qquad \text{(3-11a)}$$

where: $$A = \left\{2.457\ln\left[\frac{1}{(7/\mathrm{Re})^{0.9} + (0.27\varepsilon/d)}\right]\right\}^{16} \qquad \text{(3-11b)}$$

$$B = \left(\frac{37530}{\mathrm{Re}}\right)^{16} \qquad \text{(3-11c)}$$

The usual approach in piping computation is to specify a Reynolds number limit for laminar flow and to declare the flow to be turbulent for all Reynolds numbers above that limit. The most referenced limit for laminar flow is 2300 [1].

The advantage of the Churchill's equation is that the friction factor for the laminar, transition and turbulent flow regimes is represented by a single expression, in which the Reynolds limit for laminar flow is implicit in the equation.

So, fluid flow in a pipe with only pipe skin friction loss consideration can have a model equation of:

$$(p_1 + \gamma Z_1) - (p_2 + \gamma Z_2) = f\frac{L}{d_i}\frac{\rho V^2}{2} \qquad \text{(3-12)}$$

3.1.1.2 Minor Losses

Minor losses are expressed in a form similar to that for the major head loss, except that, a loss coefficient, K, for the various fittings or valves can be taken instead of $f\frac{L}{d}$ in Eqn.(3-5). Hence, the head loss due to fittings can be given by:

$$h_f = K\frac{V^2}{2g} \qquad \text{(3-13)}$$

The loss coefficient K, can be identified from two approaches for different fittings. The first one is prescribing a single value of K for a given fitting which is invariant with respect to size and Reynolds number. These values of K for different fittings are given in **Table-A** in the appendix.

The other approach is to specify K for a given fitting in terms of the value of the complete turbulence friction factor f_T for the nominal pipe size. This is more preferable than the first approach if information is available. Detailed analysis of the minor loss coefficients has been made by Hodge [1].

Considering the two loss components in the flow of fluids in a pipe, the total head loss can be set as:

$$h_f = \left(f\frac{L}{d_i} + \sum_{i=1}^{n} K_i \right) \frac{V^2}{2g} \tag{3-14}$$

where: K_i is the minor loss coefficient of i^{th} devise.

Thus, for a fluid flow in a pipe between two points with major and minor (fittings and valves) losses considered, Eqn.(3-12) can be rewritten as:

$$(p_1 + \gamma Z_1) - (p_2 + \gamma Z_2) = \left(f\frac{L}{d_i} + \sum_{i=1}^{n} K_i \right) \frac{\rho V^2}{2} \tag{3-15}$$

3.1.2 Heat exchanger losses

There are two major sources of pressure drop on the tube-side of a heat exchanger: the friction loss in the tubes and the losses due to the sudden contraction and expansion and flow reversals that fluid experiences in flow through the tube arrangement.

The pressure losses due to contraction at the tube inlets, expansion at the exits, and flow reversal in the headers, can be taken to be significant of the total tube-side pressure drop. There is no entirely satisfactory method for estimating these losses. *Coulsen* [3] uses velocity heads to estimate these losses by counting the number of flow contractions, expansions and reversals, and using the factors for pipe fittings to estimate the number of velocity heads lost. For two tube passes, there will be two contractions, two expansions and one flow reversal. The head loss for each of these effects is [4]: contraction 0.5, expansion 1.0, 180^0 bends 1.5. Therefore, for two passes the maximum loss will be:

$$2 \times 0.5 + 2 \times 1.0 + 1.5 = 4.5 \text{ Velocity heads}$$

This can be taken as 2.25 per pass

Hence, the pressure drop in the tube-side of a heat exchanger can be estimated as:

$$\Delta p = N_p \left[4f \frac{L}{d_i} \left(\frac{\mu_b}{\mu_w} \right)^{-n} + 2.25 \right] \frac{\rho u_t^2}{2} \tag{3-16}$$

where: $n = 0.25$ for laminar flow, and

$= 0.14$ for turbulent flow

Δp = Tube-side pressure drop, [*Pa*]

N_p = Number of tube passes

u_t = Tube-side fluid velocity, [*m/s*]

L_t = Length of one tube, [*m*]

d_i = tube inside diameter, [m]

μ_b & μ_w = dynamic viscosity at bulk and wall temperature

Therefore, if there is a heat exchanger between two points of a flow, Eqn.(3-15) includes heat exchanger pressure loss. Thus:

$$(p_1 + \gamma Z_1) - (p_2 + \gamma Z_2) = \left(f \frac{L_p}{d_i} + \sum_{i=1}^{n} K_i \right) \frac{\rho V^2}{2} + N_p \left[4f \frac{L_t}{D_i} \left(\frac{\mu_b}{\mu_w} \right)^{-n} + 2.25 \right] \frac{\rho u_t^2}{2} \qquad (3\text{-}17)$$

Thus, Eqn.(3-17) is a general model equation which relates pressures between two points with many devices in-between. If there is a pump in the flow system, the model equation for the selected pump should be considered.

3.2 Pump Model

Modeling boiler-feed or condensate pumps would be started by the identification of the discharge pressure and flow rate from the pumps [25]. To determine this pressure, it is necessary to obtain the sum of the maximum boiler drum pressure and of all the frictional losses and control losses, in the case of the boiler feed pumps, and in the case of condensate pumps, the sum of all losses between the deaerator and the condensate pump, and the pressure of the deaerator.

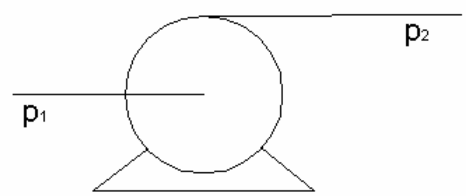

Fig.3.2 Pump schetch

The net pressure developed by the feed or condensate pump would be the difference between the required discharge pressure and the available suction pressure. Converting it into total head, the standard equation between points 1 and 2 in Fig.3.2 can be written as:

$$H = \frac{p_2}{\rho g} - \frac{p_1}{\rho g} \tag{3-18}$$

In a pump modeling task, the characteristic curve of the specific pump must be known. In most cases, boiler feed and condensate pumps would be variable speed pumps. And the pumps selected to be modeled here are variable speed pumps too. Hence, the total head developed by the pumps is a function of flow rate and speed.

$$H = f(Q, n) \tag{3-19}$$

Eqn.(3-19) can be expressed by one satisfactory polynomial representation as:

$$H = a_0(n) + a_1(n)Q + a_2(n)Q^2 \tag{3-20}$$

The coefficient $a's$ can expressed as:

$$a_i(N) = b_{i0} + b_{i1}n + b_{i2}n^2 \tag{3-21}$$

The dependence of head on speed in Eqn.(3-20) is contained with in the coefficients.

Each curve, in Fig.3.3, at constant speed (n) is curve fitted by a quadratic polynomial. Now, lets get the coefficients of Eqn.(3-20) which represents the pump head at three different speed, n_1, n_2 and n_3.

At n_1 the quadratic curve fit would be:

$$H_1 = a_{10} + a_{11}Q + a_{12}Q^2 \tag{3-22}$$

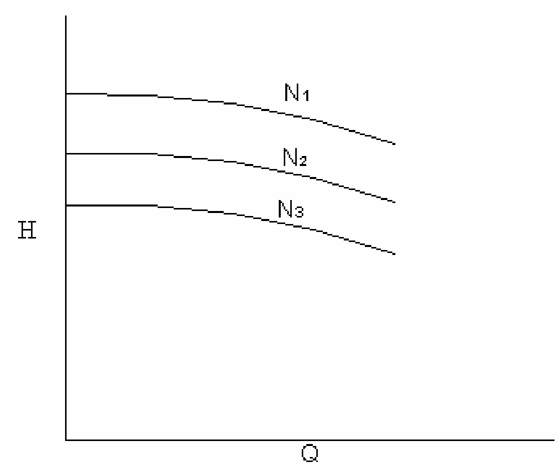

Fig. 3.3 Pump head as function of speed and flow rate

Similarly, at n_2 and n_3:

$$H_2 = a_{20} + a_{21}Q + a_{22}Q^2 \tag{3-23}$$

and,

$$H_3 = a_{30} + a_{31}Q + a_{32}Q^2 \tag{3-24}$$

The coefficients in Eqns.(3-22) to (3-24) can be identified from Eqn.(3-21).

Hence, for $i = 0$

$$a_0(N) = b_{00} + b_{01}n + b_{02}n^2 \tag{3-25}$$

For the three n value:

$$\begin{aligned} a_{10} &= b_{00} + b_{01}n_1 + b_{02}n_1^2 \\ a_{20} &= b_{00} + b_{01}n_2 + b_{02}n_2^2 \\ a_{30} &= b_{00} + b_{01}n_3 + b_{02}n_3^2 \end{aligned} \tag{3-26}$$

So, the coefficients will be:

$$\begin{Bmatrix} b_{00} \\ b_{01} \\ b_{02} \end{Bmatrix} = \begin{bmatrix} 1 & n_1 & n_1^2 \\ 1 & n_2 & n_2^2 \\ 1 & n_3 & n_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} a_{10} \\ a_{20} \\ a_{30} \end{Bmatrix} \tag{3-27}$$

Similarly, at $i = 1$

$$a_1(N) = b_{10} + b_{11}n + b_{12}n^2 \tag{3-28}$$

and the coefficient will be:

$$\begin{Bmatrix} b_{10} \\ b_{11} \\ b_{12} \end{Bmatrix} = \begin{bmatrix} 1 & n_1 & n_1^2 \\ 1 & n_2 & n_2^2 \\ 1 & n_3 & n_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} a_{11} \\ a_{21} \\ a_{31} \end{Bmatrix} \tag{3-29}$$

And for $i = 2$

$$a_2(N) = b_{20} + b_{21}n + b_{22}n^2 \tag{3-30}$$

So, the coefficients of Eqn.(3-30) can be set as:

$$\begin{Bmatrix} b_{20} \\ b_{21} \\ b_{22} \end{Bmatrix} = \begin{bmatrix} 1 & n_1 & n_1^2 \\ 1 & n_2 & n_2^2 \\ 1 & n_3 & n_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} a_{12} \\ a_{22} \\ a_{32} \end{Bmatrix} \tag{3-31}$$

The $a's$ vectors in the right hand side of Eqns.(3-27), (3-29) and (3-31) can be identified from Eqns.(3-22), (3-23) and (3-24) evaluating at different flow rates.

Hence, Eqn.(3-22) evaluated at Q_1, Q_2 and Q_3 will give:

$$\begin{aligned} H_{11} &= a_{10} + a_{11}Q_1 + a_{12}Q_1^2 \\ H_{12} &= a_{10} + a_{11}Q_2 + a_{12}Q_2^2 \\ H_{13} &= a_{10} + a_{11}Q_3 + a_{12}Q_3^2 \end{aligned} \tag{3-32}$$

So, the coefficients of Eqn.(3-32) will be:

$$\begin{Bmatrix} a_{10} \\ a_{11} \\ a_{12} \end{Bmatrix} = \begin{bmatrix} 1 & Q_1 & Q_1^2 \\ 1 & Q_2 & Q_2^2 \\ 1 & Q_3 & Q_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} H_{11} \\ H_{12} \\ H_{13} \end{Bmatrix} \tag{3-33}$$

Similarly, Eqn.(3-23) and (3-24) would give:

$$\begin{Bmatrix} a_{20} \\ a_{21} \\ a_{22} \end{Bmatrix} = \begin{bmatrix} 1 & Q_1 & Q_1^2 \\ 1 & Q_2 & Q_2^2 \\ 1 & Q_3 & Q_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} H_{21} \\ H_{22} \\ H_{23} \end{Bmatrix} \tag{3-34}$$

and,

$$\begin{Bmatrix} a_{30} \\ a_{31} \\ a_{32} \end{Bmatrix} = \begin{bmatrix} 1 & Q_1 & Q_1^2 \\ 1 & Q_2 & Q_2^2 \\ 1 & Q_3 & Q_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} H_{31} \\ H_{32} \\ H_{33} \end{Bmatrix} \tag{3-35}$$

Hence, the final expression for H as a function of speed and flow rate would be;

$$H = (b_{00} + b_{01}n + b_{02}n^2) + (b_{10} + b_{11}n + b_{12}n^2)Q + (b_{20} + b_{21}n + b_{22}n^2)Q^2 \tag{3-36}$$

The input data can be prepared in a matrix form from the characteristic curve as:

$$\begin{bmatrix} 0 & Q_1 & Q_2 & Q_3 \\ n_1 & H_{11} & H_{12} & H_{13} \\ n_2 & H_{21} & H_{22} & H_{23} \\ n_3 & H_{31} & H_{32} & H_{33} \end{bmatrix}$$

Thus, the two equations which characterize the pump model will be (Eqns.(3-18) and (3-36)):

$$H = \frac{p_2}{\rho g} - \frac{p_1}{\rho g}$$

$$H = \left(b_{00} + b_{01}n + b_{02}n^2\right) + \left(b_{10} + b_{11}n + b_{12}n^2\right)Q + \left(b_{20} + b_{21}n + b_{22}n^2\right)Q^2$$

p_1 is the pressure of deaerator for feed pumps, and the pressure of condenser in condensate pumps. p_2 is the pump discharge pressure in both types of pumps. So, the two equations can be solved simultaneously with Eqn(3-17) of the pipe equation. The simultaneous equations are solved using Newton-Raphson technique following the procedure indicated in Chapter-2.

3.3 General Flow Subroutines

The flow model in this paper takes the points as nodes, input and output nodes. Point 1 is an input node and point 2 is an output node as depicted on Fig.3.1. Between each pair of nodes there is one element, and Eqn.(3-17) should be written for each element which can relate the pressures at the nodes. Therefore, the number of equations will be as many as the number of elements selected in the flow system. If there is a pump in the system the model equations of the pump would be added and the number of equations will be raised. There are two subroutines which solve flow problems in this paper, *flow_press_steam* and *flow_press_water*. The first one is able to solve steam flow in the boiler from the drum to the inlet of the turbine. Since, the pressure and temperature at the inlet of the steam turbine are known from the design of the turbine; the main task of *flow_press_steam* subroutine is to identify the boiler drum pressure.

The second flow subroutine solves the pressures at the specified nodes on the condensate flow from the condenser to the deaerator, and the feed water flow from the deaerator to the boiler drum.

Here, the procedure is listed for the subroutine *flow_press_water* by considering only the flow condition between the deaerator and the boiler drum pressure.

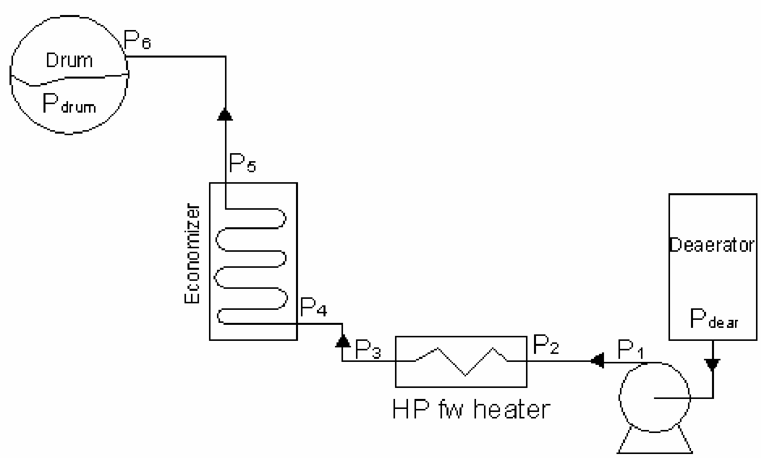

Fig.3.4 Feed pump-flow arrengement

Assuming a single high pressure heater, the boiler feed water is pumped from the deaerator pressure to the boiler drum pressure as shown in Fig.3.4. Nodal points would be taken at inlet and exit of pumps and heat exchangers, and elements between two consecutive nodes. Deaerator and drum pressures are boundary conditions of the two ends. The procedure for solving the pressures at each nodal point is discussed below, as indicated in Chapter 2.

1. Write Eqn.(3-17) for each element in the arrangement given in Fig. 3.4. Assuming $Z_1 = Z_2$ for all elements.

$$p_1 - p_2 = \left(f_1 \frac{L_1}{d_{i1}} + \sum_{i=1}^{n} K_{i1} \right) \frac{\rho V^2}{2}$$

$$p_2 - p_3 = N_p \left[4 f_{t1} \frac{L_{t1}}{D_{it1}} \left(\frac{\mu_b}{\mu_w} \right)^{-n} + 2.25 \right] \frac{\rho u_{t1}}{2}$$

$$p_3 - p_4 = \left(f_2 \frac{L_2}{d_{i2}} + \sum_{i=1}^{n} K_{i2} \right) \frac{\rho V_2^2}{2}$$

$$p_4 - p_5 = N_{p2} \left[4 f_{t2} \frac{L_{t2}}{D_{it2}} \left(\frac{\mu_b}{\mu_w} \right)^{-n} + 2.25 \right] \frac{\rho u_{t2}}{2}$$

$$p_5 - p_{drm} = \left(f_3 \frac{L_3}{d_{i3}} + \sum_{i=1}^{n} K_{i3} \right) \frac{\rho V_3^2}{2}$$

2. Since there is pump in the system the model equation for the pump is considered. So, write the pump model equations, Eqn.(3-18) and Eqn.(3-36), between the deaerator pressure and p_1.

$$H_p = \frac{p_2}{\rho g} - \frac{p_1}{\rho g}$$

$$H_p = \left(b_{00} + b_{01} n + b_{02} n^2\right) + \left(b_{10} + b_{11} N + b_{12} n^2\right) Q_{fw} + \left(b_{20} + b_{21} n + b_{22} n^2\right) Q_{fw}^2$$

3. Follow the procedure listed for solving non-linear system of equations using Newton-Raphson method in Chapter 2

The subroutine *flow-feed-water* able to solve the pressure at the different nodes, the total head of the pump and the speed of the pump with specified boundaries of deaerator and boiler drum pressure.

Similar approach is followed to identify pressures at specified nodes and condensate pump head and speed for similar arrangement of condensate flow from condenser to deaerator. In this arrangement the condenser and deaerator pressures are specified boundaries.

CHAPTER 4

Boiler Modeling and Simulation

Modeling and simulation of boiler need a thorough understanding of the physical phenomena involved in the process of the boiler. A basic description of the physics ranging from chemistry to thermodynamics is provided below. This description is based on literatures available on the subject of boiler modeling with emphasis on circulating fluidized bed combustion.

Circulating fluidized bed boilers are considered to be an improvement over the traditional methods associated with coal combustion. This technology has many confirmed advantages that include fuel flexibility, high combustion efficiency, low No_x emissions, and high sulfur capture efficiency. The circulating fluid bed combustors are more developed in chemical engineering fields than in power technology.

Mathematical modeling and simulation of CFB combustion furnace need the modeling of chemical reaction, heat and mass transfer, particle size reduction due to combustion, fragmentation and other mechanisms, and gas and solid flow structure.

Luhuilin and Bie Rushan [5] attempted to predict the performance of a CFB boiler with the steady-state mathematical model considering the detailed hydrodynamics, heat transfer, and combustion analysis in the dense zone and dilute region in the furnace. Their model of CFB predicts the distribution of the gas concentration, chemical species, temperature, and heat flux along the furnace in both the axial and radial locations.

Osman Bucak, Dogan and Uysal [6] did their paper on modeling heat transfer in CFB combustor by identifying the heat transfer coefficient between the gas-solid suspension and the water wall of the furnace. Their paper predicts only the heat transfer model of the combustor. Nganpradit and Piumsomboon [7] developed a simulator for CFB combustor using Aspen Plus. Their model was divided into two parts, reaction and hydrodynamic. The reaction model was represented by the Continues Stirred Tank Reactor (CSTR) model, where as the hydrodynamics was divided into two regions; lower region with one interval and upper region with three intervals, and they tried to calculate the height of the bed, void and volume in each region. There are a lot of papers done on CFB boilers and most of them have followed similar approaches.

As mentioned earlier modeling and simulation of combustion furnaces in a CFB boiler considers very complex phenomena. Here in this thesis it is tried to model the heat transfer in a CFB following the combined approach of many available literatures. This model is able to determine the furnace exit temperature of the combustion gases and the heat taken by the vapor-water mixture in the tube wall. And also the combustion reaction will be considered in a simplified form to avoid the reaction complexity in the combustion module.

The physical properties to be modeled in a boiler module are many. To make an overall model of the device, one should consider the combustion reaction in the furnace. By this process the energy required for the plant to produce steam is released. This released heat is transferred by all three modes, conduction, convection and radiation from the gas-solid suspension to the colder water walls constituting the sides of the combustion chamber. The gas leaving the furnace at high temperature passes through superheaters, economizer and air pre-heater, continuously releasing heat by all modes to the piping in the inner side, from where the water or steam or air present will remove the heat by convection.

4.1 Combustion Model

Combustion modeling needs the ultimate analysis of the dominant constituent of the fuel (in this case coal) to be known. These constituents are: carbon, hydrogen, nitrogen, moisture, sulfur and ash contents of the fuel. The percentage as-received base of the constituents can be given by: C^r for carbon, H^r for hydrogen, O^r for oxygen, N^r for nitrogen, S^r for sulfur, W^r for moisture content of the fuel, A^r for ash content of fuel. Now, let us do the combustion analysis procedurally.

I. Identify the stoichiometric moles of oxygen, say m_s, for each oxygen consuming constituents. The oxygen consuming constituents are, carbon, Hydrogen and sulfur.

Hence, for carbon:

$$C + O_2 \rightarrow CO_2 \tag{4-1}$$

To combust 1 mol of carbon completely to CO_2 1 mol of oxygen is required. So, m_s for carbon would be: $m_{sc} = 1$

Similarly, for Hydrogen and sulfur:

$$H_2 + 0.5O_2 \rightarrow H_2O \quad m_{sH_2} = 0.5 \tag{4-2}$$

$$S + O_2 \rightarrow SO_2 \qquad m_{sS} = 1 \tag{4-3}$$

II. Identify overall stoichiometric moles of oxygen per Kg of fuel, Ms, by multiplying the individual value of ms for each oxygen-consuming component by its molar/mass fraction, and summing them,

$$Ms = \frac{m_{sC}}{12}\frac{C^r}{100} + \frac{m_{sH2}}{2}\frac{H^r}{100} + \frac{m_{sS}}{32}\frac{S^r}{100} - \frac{1}{32}\frac{O^r}{100}$$
$$Ms = \frac{1\times C^r}{12\times 100} + \frac{0.5\times H^r}{2\times 100} + \frac{1\times S^r}{32\times 100} - \frac{O^r}{32\times 100} \qquad (4\text{-}4)$$

III. Identify the stoichiometric air mass ,Lo , per kg of fuel

Taking into account the weight fraction of oxygen in atmospheric air equals to 0.231,

$$L_0 = 4.33\times 32\times Ms \quad \text{kg of air/kg of fuel} \qquad (4\text{-}5)$$

The volume of dry air can be identified by dividing L_0 by density of dry air, 1.293 kg/m^3. Hence,

$$V^0 = \frac{L_0}{1.293} \qquad (4\text{-}6)$$

To ensure a possibly higher complete combustion of fuel in actual furnaces of boilers, it is necessary to introduce into a furnace more air than the theoretically required amount V^0. This can be introduced into the furnace as excess air, α

In a fluidized bed combustion furnaces α can be ranged between 1.25-1.5 [8].

Hence the actual combustion volume of air will be

$$V = \alpha V^0 \qquad (4\text{-}7)$$

4.1.1 Combustion Products

The combustion products can be identified by assuming that complete combustion is taking place in the furnace. The products of combustion consist, then, of carbon dioxide CO_2, sulfur dioxide SO_2, the unconsumed nitrogen N_2, excess oxygen O_2, and water vapor.

To obtain convenient formula for the calculation, it is expedient to introduce the concept of the theoretical volume of the combustion products, i.e. which is resulted from theoretical required air.

$$^{3}\ V_{cp}^{0} = V_{co_2} + V_{So_2} + V_{N_2}^{0} + V_{H_2o}^{0} \qquad (4\text{-}8)$$

Now, the volume of each gas can be identified as follows:

Carbon Dioxide

The density[4] of CO_2 is 1.977kg/m^3, hence the volume of one mole of CO_2 will be 22.26m^3. Therefore,

$$V_{Co_2} = \frac{22.26}{12} \times \frac{C^r}{100} = \frac{1.86C^r}{100}, \ [m^3/kg \text{ of fuel}] \qquad (4\text{-}9)$$

Sulfur dioxide

The density of SO_2 is 2.927kg/m^3, and the molar volume of SO_2 will be 21.89m^3. So,

$$V_{So_2} = \frac{0.699S^r}{100} \qquad (4\text{-}10a)$$

CFB combustor removes most of SO_2 gas from the combustion gases by capturing it with limestone. Most of this type of combustor is able to capture about 90% of this gas. Hence, Eqn.(4-10a) will be;

---

[3] *The superscript "0" indicates theoretical volume*
[4] *densities given at standard pressure and temperature*

$$V_{So_2} = \frac{0.0699S^r}{100} \tag{4-10b}$$

The volume of the tri-atomic gases would be given as the sum of Eqns. (4-9) & (4-10b)

$$V_{R0_2} = V_{Co_2} + V_{So_2}$$

$$V_{R0_2} = 1.866\left(\frac{C^r + 0.375S^r}{100}\right) \tag{4-11}$$

Nitrogen

Taking density of nitrogen as 1.25kg/m^3, the volume of it in the combustion product from the theoretical required, 0.79V^0, and from the fuel would be:

$$V_{N_2}^0 = 0.79V^0 + \frac{0.8N^r}{100} \tag{4-12}$$

Hence, the theoretical volume of dry exhaust gas will be the sum of Eqns.(4-11) & (4-12)

$$V_{dg}^0 = V_{R0_2} + V_{N_2}^0$$

$$V_{dg}^0 = 1.866\left[\frac{C^r + 0.375S^r}{100}\right] + \frac{0.8N^r}{100} + 0.79V^0 \tag{4-13}$$

Water Vapor

The water vapor in the combustion product would be from the combustion of hydrogen, the moisture content of the fuel and from the moisture content of the combustion air. Assuming moisture content of air is 10 gm/kg of dry air, the expression will have the form:

$$G^0_{H_20} = \frac{8.94H^r + W^r}{100} + 0.013V^0, \quad [kg/kg\ of\ fuel] \qquad (4\text{-}14)$$

Then, the theoretical volume of water vapor will be:

$$V^0_{H_20} = \frac{G^0_{H_20}}{\rho_{H_20}} \qquad (4\text{-}15)$$

Water vapor contained in the products of combustion is superheated, for its partial pressure is small and temperature is high. Assuming that water vapor obeys Avogadro law, the value ρ_{H_20} is found by dividing the molecular weight of H_20 by the molar volume under standard conditions.

$$\rho_{H_20} = \frac{18.016}{22.4} = 0.804 \ kg/m^3 \qquad (4\text{-}16)$$

Hence,

$$V^0_{H_20} = \frac{8.94H^r + W^r + 1.3V^0}{80.4} \qquad (4\text{-}17)$$

$$V^0_{H_20} = 0.111H^r + 0.0124W^r + 0.0161V^0 \quad m^3/kg\ of\ fuel \qquad (4\text{-}18)$$

Considering excess air ratio $(\alpha > 1)$, the volume of actual dry gas would be:

$$V_{dg} = V^0_{dg} + (\alpha - 1)V^0 \qquad (4\text{-}19)$$

where: $(\alpha - 1)$ is the amount of excess air.

Similarly,

$$V_{H_20} = V^0_{H_20} + 0.0161(\alpha - 1)V^0 \qquad (4\text{-}20)$$

Then, the total volume of gases per kg of fuel will be:

$$V_g = V_{R0_2} + V^0_{N_2} + V_{H_20} + (\alpha - 1)V^0 \qquad (4\text{-}21)$$

The volume fraction of tri-atomic gases and water vapor are respectively given by:

$$r_{R0_2} = \frac{V_{R0_2}}{V_g} \tag{4-22}$$

$$r_{H_20} = \frac{V_{H_20}}{V_g} \tag{4-23}$$

From the total mass balance, weight of flue gas will be:

$$G_g = 1 - \frac{A^r}{100} + 1.306 \times \alpha \times V^0 \tag{4-24}$$

4.1.2 Verifying Example

The ultimate analysis of lignite coal is given as follows, as-received base: $C^r = 30\%; H^r = 10\%; S^r = 2\%; N^r = 3\%; O^r = 10\%; W^r = 25\% \quad \& \ A^r = 20\%$ and consider a 35% excess air. The subroutine *combustion_module* gives the following output:

Quantity of air required = 8.8852kg of air/kg of fuel

Volume of dry flue gas = 6.3862 m^3/kgof fuel

Mass of water vapor in the gas = 1.2222 kg/kg of fuel

Mass of dry combustion gas = 8.4674kg/kg of fuel

Ratios of tri-atomic gases: $r_{RO_2} = 0.0708$

$r_{H_2O} = 0.1923$

Mass of flue gas = 9.6896 kg/kg of fuel

4.2 Furnace Heat Transfer Model

The heat transfer process in the furnace passes three phases. The first is that the heat transfer from the gas-solid suspension to the water wall tubes. This heat flow can be estimated from:

$$\dot{Q} = h_{sw} A_0 (T_g - T_{w0}) \qquad (4\text{-}25)$$

The second one is the conduction of heat through the wall of the riser tubes. Conduction through these pipes is governed by the Fourier relation for cylindrical bodies.

$$\dot{Q} = \frac{2\pi k_t l (T_{w0} - T_{wi})}{\ln \frac{r_0}{r_i}} \qquad (4\text{-}26)$$

Pressurized water flows inside the tubes and cools the tube material by convective heat transfer. This is the third mode of heat transfer in the boiler furnace which is governed by the Newton equation of cooling:

$$\dot{Q} = h_{TP} A_i (T_{wi} - T_{sat}) \qquad (4\text{-}27)$$

If overall heat transfer coefficient is designated by U_0, based on the outside surface area, it is expressed as;

$$U_0 = \frac{1}{\frac{d_o}{d_i}\frac{1}{h_{TP}} + \frac{r_0 \ln\left(d_0 / d_i \right)}{k_t} + \frac{1}{h_{sw}}} \qquad (4\text{-}28)$$

where: h_{sw} = Heat transfer coefficient between the gas-solid suspension and the wall, [*W/m²k*]

h_{TP} = Convective heat transfer coefficient of boiling water, [*W/m²k*]

$\dot{Q}$ = Heat absorbed by the furnace, [*W*]

T_g = Gas-solid suspension temperature (bed temperature), [*k*]

T_{wi}, T_{wo} = Furnace wall inner and outer temperatures, respectively, [*k*]

k_t = Tube Wall thermal conductivity, [*W/m-k*]

d_0, d_i = Tube inner and outer diameters, [*m*]

A = Heat transfer area, [m^2]

The heat transfer computation in a fluidized bed furnace is not different to other types of heat exchanger computations once the heat transfer coefficients have been identified. So, the next step in furnace heat transfer modeling would be the identification of the heat transfer coefficients.

4.2.1 Gas-solid Suspension Heat transfer Coefficients

In a circulating fluidized bed, the suspension-to-wall heat transfer comprises various modes including conduction due to particle clusters contacting the surface or particles sliding along the walls, gas convection to uncovered surface areas, and thermal radiation. The percent of surface area covered by particle clusters is an important parameter in the heat transfer identification. These three coefficients are discussed below.

a) Particle Convective Coefficient, h_{pc}

The particle convection is important in the overall bed-to-surface heat transfer. When particles or particle clusters contact the surface, relatively large local temperature gradients are developed. The rate of heat transfer can be enhanced with increased surface renewal rate or decreased cluster residence time in the convective flow of particles in contact with the surface.

Particle convective heat transfer coefficient is correlated well with the definition of the Nusselt number.

$$h_{pc} = \frac{Nu \times k_g}{d_p} \tag{4-29}$$

Nusselt number can be expressed by Archimedes number and the Prandtl number as follows [8].

$$Nu = 0.31 Ar^{0.27} \Pr \tag{4-30}$$

The Archimedes and the Prandtl numbers are given by:

$$Ar = \frac{d_p^3 \times g \times \rho_g \times (\rho_p - \rho_g)}{\mu_g^2} \tag{4-31}$$

$$\Pr = \frac{c_{p\,g} \times \mu_g}{k_g} \tag{4-32}$$

where: Nu = Nusselt number

Ar = Archimedes number

$\Pr$ = Prandtl number

d_p = mean particle diameter, [*m*]

ρ_g = gas density, $[kg/m^3]$

ρ_p = particle density, $[kg/m^3]$

μ_g = gas viscosity, [*kg/m-s*]

$c_{p\,g}$ = gas specific heat, [*J/kg-K*]

k_g = gas thermal conductivity, [*W/m-K*]

b) Gas Convective Heat Transfer Coefficient, h_{gc}

Gas convection coefficient can be computed considering that the up-flowing gases contain dispersed solids [6]

$$h_{gc} = \frac{k_g c_{pp}}{d_p c_{pg}} \left(\frac{\rho_b}{\rho_p} \right)^{0.3} \left(\frac{u_t^2}{g d_p} \right)^{0.21} P_r \qquad (4\text{-}33)$$

The terminal velocity, u_t, can be correlated with the Stokes law [10]:

$$u_t = \frac{g(\rho_p - \rho_g) d_p^2}{18 \mu_g} \qquad \text{for } \mathrm{Re}_p < 0.4 \qquad (4\text{-}34)$$

$$u_t = \left[\frac{4}{225} \frac{(\rho_p - \rho_g)^2 g^2}{\rho_g \mu_g} \right]^{1/3} d_p \qquad \text{for } 0.4 < \mathrm{Re}_p < 500 \qquad (4\text{-}35)$$

$$u_t = \left[\frac{3.1 g (\rho_p - \rho_g) d_p}{\rho_g} \right]^{1/2} \qquad \text{for } 500 < \mathrm{Re}_p < 200{,}000 \qquad (4\text{-}36)$$

where: $$\mathrm{Re}_p = \frac{d_p \rho_g u_t}{\mu_g} \qquad (4\text{-}37)$$

ρ_b = suspension density [*kg/m*^{3}]

c_{p_p} = solid specific heat [*J/kg-K*]

Re = Reynolds number

c) Radiation Heat Transfer Coefficient, h_r

The radiation heat transfer in a circulating fluidized bed can be treated by considering the radiation from the clusters, h_{cr} , and from the dispersed phase, that is, the remaining part of gas-solid suspension except clusters, h_{dr} . Hence:

$$h_r = \xi h_{cr} + (1-\xi)h_{dr} \tag{4-38}$$

where: ξ is the volume fraction of clusters in the bed. The two components of radiation coefficients can be defined by:

$$h_{cr} = \frac{\sigma_b\left(T_b^4 - T_w^4\right)}{\left(\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_w} - 1\right)(T_b - T_w)} \tag{4-39}$$

and,

$$h_{dr} = \frac{\sigma_b\left(T_b^4 - T_w^4\right)}{\left(\frac{1}{\varepsilon_d} + \frac{1}{\varepsilon_w} - 1\right)(T_b - T_w)} \tag{4-40}$$

The emissivity of the cluster ε_c can be determined from:

$$\varepsilon_c = 0.5(1 + \varepsilon_p) \tag{4-41}$$

And emissivity of the dispersed phase ε_d :

$$\varepsilon_d = \sqrt{\frac{\varepsilon_p}{(1-\varepsilon_p)B}\left(\frac{\varepsilon_p}{(1-\varepsilon_p)B} + 2\right)} - \frac{\varepsilon_p}{(1-\varepsilon_p)B} \tag{4-42}$$

where: *B* is taken as *0.5* for isentropic scattering and *0.667* for diffusely reflecting particles. Hence, convective heat transfer coefficient at tube outside surface h_{sw} can then be expressed as, [11]:

$$h_{sw} = fh_{pC} - (1-f)h_{gc} + h_r \tag{4-43}$$

where: f the wall surface area fraction with particles.

Bucak and Dogan [6] give f to be:

$$f = 3.5c^{0.37} \tag{4-44}$$

where c = solid fraction of the bed, (1-e), and e is bed void fraction.

Bed void fraction varies with the height of the bed. Natthapong and Plumsomboon [7] on their CFB combustor modeling identified the average mean voidage by dividing the bed into lower and upper beds, and further dividing the upper bed into three sections. Here, in this paper, the void fraction is taken as the average throughout the bed.

4.2.2 Boiling Heat Transfer Coefficient

The most well known correlation for heat transfer coefficient of boiling liquids is the Chen correlation. This correlation is described in detail by Collier [12]. This correlation covers both the saturated nucleate boiling region and two-phase forced convection region. The assumption is that both mechanisms occur to some degree over the entire range of the correlation and that the contribution made by the two mechanisms are additive. Hence:

$$h_{TP} = h_{NCB} + h_C \tag{4-45}$$

where: h_{TP} = the local heat transfer coefficient, [W/m^2-^oK]

h_{NCB} = the contribution due to nucleate boiling, [W/m^2-^oK]

h_C = the contribution due to convection, , [W/m^2-^oK]

a) Convective Boiling coefficient, h_C

This convective component, h_C**,** could be represented by Dittus-Boelter [12] equation.

$$h_C = 0.023\left[\frac{G(1-x)d_i}{\mu_f}\right]^{0.8}\left[\frac{\mu_f c_{pf}}{k_f}\right]\left[\frac{k_f}{d_i}\right]F \qquad (4\text{-}46)$$

Collier [12] suggested Chilton-Colburn analogy to identify the multiplying factor, F**,** in Eqn.(4-46)**.** This can be given as:

$$F = \left[\left(\frac{\mathrm{Pr}_f + 1}{2}\right)\phi_f^2\right]^{0.444} \qquad (4\text{-}47)$$

where:

ϕ_f^2 = a two-phase frictional multiplier which is tabulated in **Table-B** in the appendix as a function of pressure and steam quality for a vapor-water mixture.

G = Mass flow rate, $kg/s.m^2$

μ_f = Saturated liquid viscosity at boiler pressure, [*kg/m-s*]

Cp_f = Specific heat of saturated liquid, [*J/kg-^{o}K*]

k_f = Thermal conductivity of saturated liquid, [*W/m-^{o}K*]

F = Multiplying factor

x Vapor quality at the exit of the evaporator

d_i Inner water wall tube diameter, [*m*]

Pr_f Prandtl number of saturated liquid

b) Nucleate Boiling Coefficient, h_{NCB}

The nucleate boiling is most dominant at the lower region of the evaporator. The coefficient would be given by a single equation as follows:

$$h_{NCB} = 0.00122\left[\frac{\Delta T_{sat}^{0.24}\Delta p_{sat}^{0.75}{c_{p}}_{f}^{0.45}\rho_f^{0.49}k_f^{0.79}}{\sigma^{0.5}\mu_f^{0.29}h_{fg}^{0.24}\rho_g^{0.24}}\right]S \qquad (4\text{-}48)$$

where: $\Delta T_{sat} = T_w - T_{sat}$

Δp_{sat} = the difference saturation pressure corresponding to ΔT_{sat}

S = the suppression factor

T_w = wall temperature, [*K*]

T_{sat} = saturation temperature, [*K*]

h_{fg} = latent heat of vaporization, [*J/kg*]

S can be given as a function of the local two-phase Reynolds number, Re_{TP}. Its graphical value can be found in [12]. For programming convenience, it is tabulated in **Table-C** of the appendix. The Re_{TP} in the table can be identified from:

$$\mathrm{Re}_{TP} = \mathrm{Re}_f \times F^{1.25} \qquad (4\text{-}49)$$

In film boiling, or convective boiling, a layer of vapor forms on the heat transfer surface and the film conductance of the vapor is low. For a simplified analysis of the boiling phenomenon, the heat flux to the tube wall is assumed to be constant along the tube, with nucleate boiling established at the lower end and convective boiling near the upper end. In order to balance the heat flow rate to the fluid, the tube wall temperature near the top of the tube will exceed the corresponding temperature at the bottom of the tube. In effect, at the top section of the tube, a greater temperature difference between wall and fluid is required to counteract the low film conductance. This adjustment overheats and blisters the tube metal. Hence, for a simplified evaporator design, two important criteria may be examined. One is the maximum allowable percent by volume of the vapor in the two-phase mixture leaving a heat absorbing riser circuit. In other word, specifying the minimum amount of vapor-water mixture circulation ratio. Sorensen [13] provides this limited value of vapor volume as a function of operating pressure**.** And the second criterion is the minimum allowable velocity of the water entering the tube. For evaporating risers that are essentially vertical and absorb energy on one side or completely around the tube, this minimum allowable velocity will be in the range 0.3 to 1.5 m/s [13]. The above criteria are design and not be discussed here in detail, but important in verifying boilers with more practical design data.

Shvets and Tolubinsky [14] suggests that the water flow circulation, which is the reciprocal of mass fraction of vapor at the exit of the water walls, should not be less than 3 to avoid overheating of tube walls.

Sorensen [13] provides the approximate value of maximum vapor fraction by volume at discharge from the riser at the corresponding operating pressure of the boiler, as discussed before. From this the maximum mass fraction of the vapor, x_m, can be calculated as:

$$x_m = \frac{1}{1 + \frac{V_l}{V_v}\frac{v_g}{v_f}} \tag{4-50}$$

where: V_v = volume fraction of vapor

V_l = Volume fraction of liquid

v_g = Specific volume of gas phase, m^3 / kg

v_f = Specific volume of liquid phase, m^3 / kg

x_m = Maximum allowable mass fraction of vapor at riser exit

To be safe, x_m should be evaluated at the maximum working pressure of the boiler.

Therefore, water flow rate at the bottom of the evaporator (water-walls) would be:

$$\dot{m}_m = CR \times \dot{m}_s \tag{4-51}$$

where: CR - circulation ratio

$$CR = \frac{1}{x_m} \tag{4-52}$$

$\dot{m}_s$ = Steam load of the boiler

$\dot{m}_m$ = Flow of mixture at the bottom of the water wall tubes.

4.3 Convection Pass Components Modeling

Once the combustion mode is selected and modeled, thermal calculation of the convection elements of the boiler will be easier and similar approach is followed almost in all types of boilers. The convection heat transfer components will be modeled using the $NTU - \varepsilon$ heat exchangers analysis method.

Convection heating surfaces are usually made up of banks of tubes which are exposed to the flue gases in cross flow or counter flow. The banks of tubes may have in-line or staggered arrangement.

Using the $NTU - \varepsilon$ method, the effectiveness of the heater will be:

$$\varepsilon = \frac{\dot{Q}_{act}}{\dot{Q}_{\max}} \tag{4-53}$$

The actual heat transfer will be given by:

$$\dot{Q}_{act} = \dot{m}_g Cp_g (T_{gi} - T_{ge}) = \dot{m}_f Cp_f (T_{fe} - T_{fi})$$
$$\dot{Q}_{act} = C_g \left(T_{gi} - T_{ge}\right) = C_f \left(T_{fe} - T_{fi}\right) \quad (4\text{-}54)$$

The maximum possible heat transfer is that which would result if one fluid underwent a temperature change equal to the maximum temperature difference available, i.e. the temperature of the entering hot fluid or gas minus the temperature of the entering cold fluid (working fluid or combustion air). This maximum heat transfer occurs when the fluid of smaller C undergoes the maximum temperature difference available. Hence:

$$\dot{Q}_{\max} = C_{\min} \left(T_{gi} - T_{fi}\right) \quad (4\text{-}55)$$

Combining Eqns (4-53) & (4-55) will give

$$\dot{Q} = \varepsilon \times C_{\min} \times \left(T_{gi} - T_{fi}\right) \quad (4\text{-}56)$$

but, the effectiveness, ε, can be given as [14]:

$$\text{if } C_g < C_f : \qquad \varepsilon_g = \frac{T_{gi} - T_{ge}}{T_{gi} - T_{fi}} \quad (4\text{-}57)$$

$$\text{if } C_f < C_g : \qquad \varepsilon_f = \frac{T_{fe} - T_{fi}}{T_{gi} - T_{fi}} \quad (4\text{-}58)$$

Assuming counter flow mode ε can be expressed as a function of NTU and $\frac{C_{\min}}{C_{\max}}$ as follows:

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 - C\right)\right]}{1 - C\exp\left[-NTU\left(1 - C\right)\right]} \quad (4\text{-}59)$$

Air heaters are mostly cross flow exchangers that the waste gas flows in tubes and air flow, which is over the tube bundles, is mixed type. In this case effective ness will be:

if $C_g < C_a$:
$$\varepsilon = \frac{1}{C}\{1 - \exp[-C[1 - \exp(-NTU)]]\} \qquad (4\text{-}60)$$

if $C_a < C_g$:
$$\varepsilon = 1 - \exp\left\{-\frac{1}{C}[1 - \exp[-NTU * C]]\right\} \qquad (4\text{-}61)$$

where:
$$C = \frac{C_{\min}}{C_{\max}} \qquad (4\text{-}62)$$

$$NTU = \frac{U_0 A_0}{C_{\min}} \qquad (4\text{-}63)$$

S.Kakac [15] provides the overall heat transfer coefficient by considering the effect of slag deposit on the outside surface of the tubes as follows:

$$U_0 = \frac{\varphi}{\frac{r_0}{r_i}\frac{1}{h_i} + \frac{1}{h_0}} \qquad (4\text{-}64)$$

where: φ is the factor which can be used to take into consideration of the slag deposit on the outside of the tube wall. S.Kakac[15] also provides this value for the different types of convection heaters. Hence, for superheaters φ is between 0.6 and 0.65. For economizer its value can be taken between 0.65 and 0.7. And for tubular air heaters it can be taken in the range 0.8 to 0.85

where: U_o = overall heat transfer coefficient, [W/m^2- K]

T_{gi}, T_{ge} = Combustion gas temperature at the entrance and exit of convection component, [*K*].

T_{fi}, T_{fe} = Working fluid and/or combustion air entrance and exit temperatures, [*K*]

4.3.1 Heat Transfer Coefficients Inside Tube & Over Tube Banks

The gas side heat transfer coefficient, h_o, can be found from the convective and radiative coefficients as follows:

$$h_o = \omega \times h_c + h_r \tag{4-65}$$

where: ω = exposure factor, it is close to unity.

h_c = Convection heat transfer coefficient, [W/*m*²- *K*]

h_r = Radiation heat transfer coefficient, [W/*m*²- *K*]

Ganapathy [16.] puts the convective coefficient, h_c, simply as:

$$h_c = 0.33 \times G^{0.6} \times \frac{C_{pg}^{0.33} \times k_g^{\ 0.67}}{d_o^{0.4} \times \mu_g^{\ 0.27}} \tag{4-66}$$

where:

$$G = \frac{\dot{m}_g}{N_w \times L \times (S_T - d_o)}, \quad \text{in } kg/m^2.s \tag{4-67}$$

N_w = Number of tubes wide

S_T = Transverse pitch, [*m*]

L, d_o = Length and outer diameter of tube, respectively, [*m*]

c_{pg} = Specific capacity, [*kJ/kg-^oK*]

k_g = thermal conductivity, [*W/m-^oK*]

μ_g = viscosity of gas, [*kg/m-s*]

The coefficient of thermal radiation of the combustion products accounts for radiation of tri atomic gases (SO_2, H_2O, and CO_2) and the particles of fly ash.

S. Kakac [15] puts the radiation coefficient as:

$$h_r = 5.1\times10^{-8}\times a_g \times T_g^3 \times \left(\frac{1-\left(T_{aw}/T_g\right)^4}{1-\left(T_{aw}/T_g\right)}\right) \tag{4-68}$$

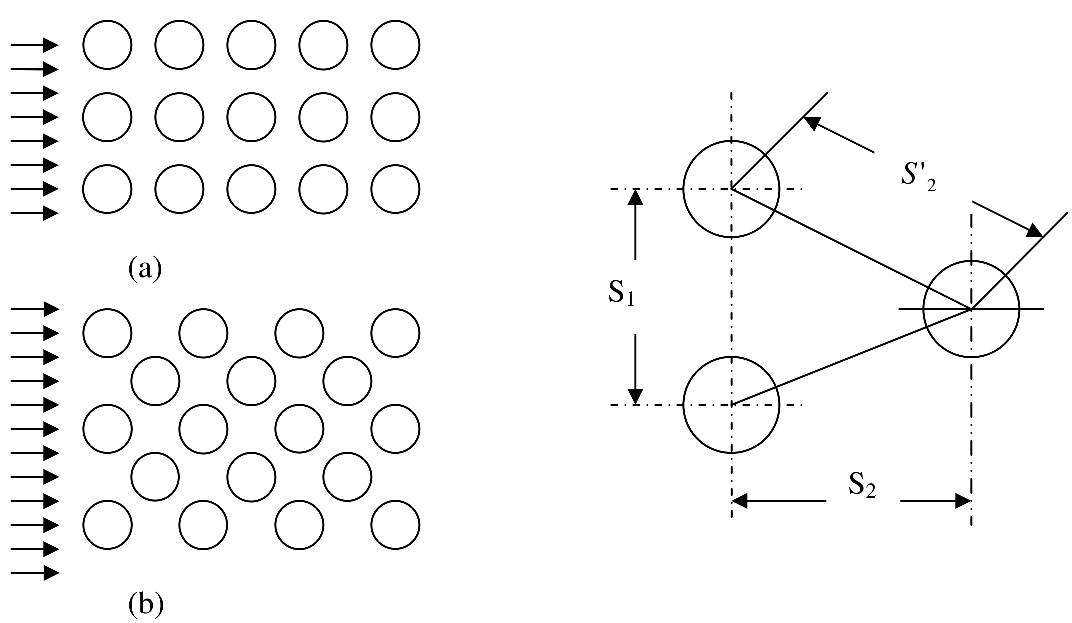

Fig.4-1 Flow across tube bundle: (a) in-line tube bundle, (b) staggered tube bundle

where: a_g, emissivity of gases :

$$a_g = 1-\exp(-kps) \tag{4-69}$$

k = effective coefficient of absorption

$$k = k_g r + k_a \mu_a \tag{4-70}$$

$$k_g = 10\left[\frac{0.78+1.6r_{H_2O}}{\left(10psr\right)^{0.5}} - 0.1\right]\left(1-0.37\frac{T_g}{1000}\right) \tag{4-71}$$

$$k_a = \frac{5990}{\left(T_g^2 d_o^2\right)^{1/3}} \tag{4-72}$$

s = Effective thickness of radiating layer of the tube bunddle

$$s = 0.9 d_o \left(\frac{4 S_1 S_2}{\pi d_o} - 1 \right) \tag{4-73}$$

$$\mu_a = \frac{A^w a_{fa}}{100 \dot{m}_g} \tag{4-74}$$

p =0.1MPa

where: a_{fa} = fly ash percentage in the flue gas

A^w = Ash mass fraction in fuel

T_g = Average gas temperature, [*K*]

T_{aw} = ashy wall average temperature, [*K*]

$$T_{aw} = \left(\frac{T_{f_i} + T_{fo}}{2} + \frac{T_{gi} + T_{go}}{2} \right) \tag{4-75}$$

$$r = r_{R0_2} + r_{H_20} \tag{4-76}$$

$$r_{R0_2} = \frac{V_{R0_2}}{V_g} \tag{4-77}$$

$$r_{H_20} = \frac{V_{H_20}}{V_g} \tag{4-78}$$

T_f = Average working fluid temperature, [*K*]

ξ = Fouling coefficient. For solid fuel burning, $\xi = 4.3 \left[m^2 \; K / kW \right]$

$\dot{Q}$ = Heat absorbed by the heat receiving surface, [*W*]

h_i =Tube side coefficient, [W/m^2-^oK]

At lower temperatures of flue gases, the radiation effect is neglected as in air pre-heaters.

For tube side coefficient, Holman [17] recommended for a fully developed turbulent flow in smooth tubes as;

$$Nu = 0.023 \times \mathrm{Re}^{0.8} \times \mathrm{Pr}^{0.4} \tag{4-79}$$

$$\mathrm{Re} = \frac{4\dot{m}}{\mu \pi d_i} \tag{4-80}$$

$$\mathrm{Pr} = \frac{\mu \times Cp}{k} \tag{4-81}$$

and,

$$h_i = 0.023 \times \left(\frac{k}{d_i} \right) \times \mathrm{Re}^{0.8} \times \mathrm{Pr}^{0.4} \tag{4-82}$$

where: $\dot{m}$ = tube side flow rate, [kg / s]

μ, Cp, k = tube fluid viscosity, specific heat and thermal conductivity, respectively. These values are estimated at the fluid bulk temperature.

4.3.2 Super Heater Module

The superheater sections are positioned after the furnace along the way of the gas flow in the boiler. The superheaters may be designed for radiative or convective heat transfer or combinations of both on the gas side. Radiative heat transfer will mainly be favored in the superheater sections positioned immediately after the furnace where the gasses are at high temperature.

The model of superheater in this paper is limited to tube buddle superheater in which heat transfer will be a combination of radiation and convection. This superheater is basically multi-pass counter-flow heat exchanger.

The saturated steam from the boiler drum will be divided into two streams in which the first stream passes through the superheater to raise the temperature of the steam and the other stream by-passes the heater and injected into the first stream at the exit of the superheater in order to control the superheated steam according to the design steam temperature at the turbine inlet. Therefore, the developed superheater module, in this paper, identifies the steam flow rate of the two streams, the exit temperature of the first stream from the super heater, the temperature of the gas exiting the super heater and the heater effectiveness.

The equations that govern the flow and heat transfer condition of the superheater can be set as follows.

The heat load of the superheater is given by Eqn.(4-56):

$$\dot{Q} = \varepsilon C_{\min}\left(T_{gi} - T_{si}\right)$$

in which the effectiveness of a counter flow heat exchanger can be given by Eqn.(4-59).

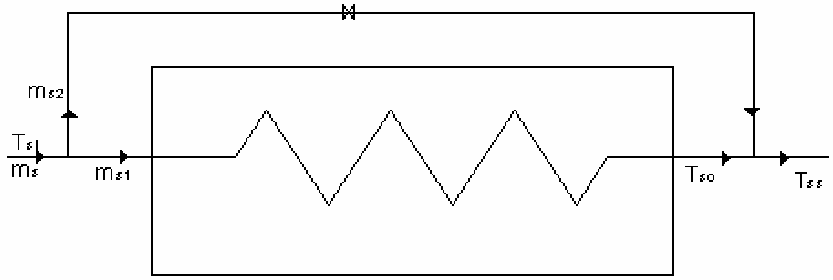

Fig.4.2 Superheater flow schematic

$$\varepsilon = \frac{1-\exp[-NTU(1-C)]}{1-C\exp[-NTU(1-C)]}$$

Conservation of energy between stream one and the gas flow gives:

$$q = \dot{m}_g c_{p_g}(T_{gi} - T_{go}) = \dot{m}_{s1} c_{p_s}(T_{so} - T_{si}) \quad (4\text{-}83)$$

And, conservation of mass & energy at the mixing junction of the two streams yield:

$$\dot{m}_s = \dot{m}_{s1} + \dot{m}_{s2} \quad (4\text{-}84a)$$

$$\dot{m}_{s1} c_{p_s}(T_{so} - T_{ss}) = \dot{m}_{s2} c_{p_s}(T_{ss} - T_{si}) \quad (4\text{-}84b)$$

$$\dot{m}_{s1} c_{ps} T_{so} + \dot{m}_{s2} c_{ps} T_{si} = (\dot{m}_{s1} + \dot{m}_{s2}) c_{ps} T_{ss} \quad (4\text{-}84c)$$

4.3.3 Economizer Module

The main purpose of economizer is to heat the feed water at the expense of the heat contained in the flue gases. However, in a number of cases water is not only heated in the economizer, but partly turns into steam. Economizers of this type are referred to as steaming economizers.

The feed water from the boiler feed pump passes through economizer to boiler drum. If attemperator is installed for super heated temperature control the feed water stream will be divided into two. One stream passes through economizer to boiler drum and the other to attemperator.

Similar to super heater model, the model of economizer is limited to tube buddle type in which heat transfer will be a combination of radiation and convection.

The thermal analysis on the economizer will give the following model equations.

The heat load on the economizer, q, using effectiveness would be given by Eqn(4-56):

$$\dot{Q} = \varepsilon \times C_{\min} \times \left(T_{gi} - T_{fw}\right)$$

Since, a counter flow arrangement is considered, effectiveness of the economizer will be given by Eqn.(4-59):

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1-C\right)\right]}{1 - C\exp\left[-NTU\left(1-C\right)\right]}$$

From energy conservation equation over the heater one can get:

$$\dot{Q} = c_{p_g}\dot{m}_g\left(T_{gi} - T_{go}\right) = \dot{m}_{fw}\left(h_{De} - h_{fw}\right)$$

where: $C = \frac{C_{\min}}{C_{\max}}$

$C_{\min}$ = the minimum of the capacities $c_{p_g}m_g$ and $c_{p_{fw}}m_{fw}$

$C_{\max}$ = the maximum of the capacities $c_{p_g}m_g$ and $c_{p_{fw}}m_{fw}$

The economizer is modeled perfectly by the above equations. The three unknowns, which are exit enthalpy of the feed water from the economizer, the gas exit temperature and the economizer effectiveness, can be identified by simultaneous solving of the above equations using Newton-Raphson technique.

There is a subroutine called *econo_module* that can solve the problem of economizer. *Economizer_residue* is also a subroutine that gives the residue of the conservation equations of this component. This subroutine also called by *boiler_solver* of the main boiler simulator program.

4.3.4 Air Pre-Heater Module

The need for air heaters in boiler units stems from the fact that with regenerative feed water heating copious cooling of the flue gases, flowing through economizer, is impossible due to a high inlet water temperature. At the same time it is of an advantage to preheat the air delivered into the furnace for improving fuel combustion, and increase available heat energy to the furnace.

In this paper the air heater to be considered for modeling is the recuperative type with a tubular design. In this type of air heater the combustion gases pass through the tubes and the combustion air passes over the tube banks in a cross flow mode. The heat transfer from the gases to the air can be given by effectiveness of the heater (Eqn.(4-56)):

$$\dot{Q} = \varepsilon C_{\min}\left(T_{gi} - T_{amb}\right)$$

And also conservation of energy over the heater yields,

$$\dot{m}_g c_{p_g}\left(T_{gi} - T_{go}\right) = \dot{m}_a c_{p_a}\left(T_{ca} - T_{amb}\right)$$

The effectiveness of a cross-flow type of this kind of heater depends on whether the flow streams are mixed or unmixed.

In boiler air heaters of this tubular type, the fluid flowing outside tubes, in this case combustion air is considered to be mixed flow; and the gases flowing inside tubes are unmixed. Hence:

If $C_{\min} = \dot{m}_g c_{p_g}$, (Eqn(4-60))

$$\varepsilon = \frac{1}{C}\{1 - \exp[-C[1 - \exp(-NTU)]]\}$$

Or, if $C_{\min} = \dot{m}_a c_{p_a}$, (Eqn(4-61))

$$\varepsilon = 1 - \exp\left\{-\frac{1}{C}[1 - \exp[-NTU(C)]]\right\}$$

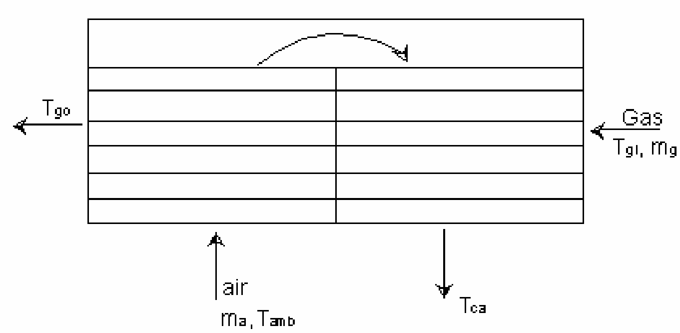

Fig.4.3 Tubular air heater schematic

Likewise, the problem of the air-heaters can be solved by the program; *air_heater_module*. The residue of the equations of this component can be evaluated by the function *air_heater_residue.*

4.4 Boiler Thermal Efficiency

Boiler efficiency is defined as the percentage of the heat input to the boiler, in the form of fuel, which is absorbed by the working fluid. This working fluid is the steam and/or water that flow in the boiler tubes.

To determine boiler efficiency, there are two methods recognized by ASME [8]. These are the Input-output method and the Heat-loss method. The first method, Input-output, is mostly used for oil and gas fueled boilers. It has great inaccuracy for coal fired boilers. The Heat-loss method is convenient for coal fueled boilers. It concentrates on determining the heat loss from the boiler envelop rather than measuring the actual heat absorbed by the working fluid.

The heat-loss method is derived as follows:

$$\eta_b = 100 \times \frac{Q_{abs}}{Q_{fuel}} \tag{4-85}$$

$$Q_{abs} = Q_{fuel} - Q_{loss} + Q_{credit} \tag{4-86}$$

$$\eta_b = \frac{100 \times (Q_{fuel} - Q_{loss} + Q_{credit})}{Q_{fuel}} \tag{4-87a}$$

$$\eta_b = 100 - 100 \times \frac{(Q_{loss} - Q_{credit})}{Q_{fuel}} \tag{4-87b}$$

Dividing Eqn.(4-87b) by the calorific value of the fuel:

$$\eta_b = 100 - 100 \times \frac{\left(\sum H_{loss} - \sum H_{credit}\right)}{H_{fuel}} \tag{4-88}$$

where: $\sum H_{loss}$ = sum of individual loss terms per unit flow of fuel.

$\sum H_{credit}$ = sum of individual heat credits per unit flow of fuel.

The heat losses and heat credits can be calculated as follows:

Heat losses

1) *Dry flue gas sensible loss*: this heat can be calculated as the heat loss which leaves the boiler envelop through boiler stack or heat loss of the waste gas.

$$H_{df} = m_{df} \times Cp_{df} \times \left(T_{df} - T_{amb}\right) \tag{4-89}$$

where: m_{df} = Dry flue gas flow per unit flow of fuel, [*kg/s*]

T_{df} = Temperature of dry flue gas, [*K*]

Cp_{df} = Specific heat of dry flue gas, [$J/kg\text{-}^{o}K$]

T_{amb} = ambient temperature, [*K*]

H_{df} = Waste gas heat per unit mass flow of fuel

2) *Loss due to water vapor in flue gas:* the change of water to vapor due to moister in the fuel, combustion air and fuel hydrogen constitute a heat loss. This heat loss is determined as follows:

$$H_{mf} = \dot{m}_w \times \left(h_{wv} - h_{wref}\right) \tag{4-90}$$

where:

$\dot{m}_w$ = mass of water in fuel, air & due to combustion of hydrogen per unit mass of fuel

h_{wv} = enthalpy of water vapor at T_{df}

h_{wref} = enthalpy of water at reference temperature (T_{ref} or T_{amb})

In the combustion module, it was assumed that the combustion would be complete and unburned carbon loss in the ash is neglected.

3) *Radiation and convection losses*: are loss terms that represent the losses from the boiler skinto the surrounding environment, due to each heat transfer mechanism. The ratio of this loss to the calorific value of the fuel becomes the smaller, when the higher the steaming capacity of the boiler. For small capacity boilers it can go up to 3 to 4% of the energy released in the boiler.

For a boiler unit operated under a load lower than its rated capacity, the radiant and convection losses can be given by:

$$RC = RC_{rated} \frac{\dot{m}_{rated}}{\dot{m}_s} \tag{4-91}$$

where: RC = radiation and convection loss percentage

$\dot{m}$ = steam rate of the boiler

Heat credits*:*

Heat credits can be taken as useful heat available to the boiler in a form other than fuel. These heats include heat in entering air, heat added by the auxiliary equipment, and sensible heat in the fuel.

1. *Heat in entering air (combustion air)*: by heating the combustion air it is possible to increase the boiler efficiency and decrease the exiting temperature of the waste gas.

$$H_{ca} = \dot{m}_{ca} \times c_{p_{ca}} \times (T_{ca} - T_{abm}) \tag{4-92}$$

2. *Sensible heat in the fuel:* it is the physical heat of the fuel if the fuel temperature is above the ambient temperature. This heat is neglected when feed coal is assumed to be not pre-heated.

Once the boiler thermal efficiency is identified the coal firing rate in the furnace can be found from:

$$Bc = \frac{\dot{m}_{s1}\left(h_{ss} - h_{fw}\right) + \dot{m}_{s2}\left(h_{sg} - h_{fw}\right) + \dot{m}_{bd}\left(h_{sw} - h_{fw}\right)}{Q_{fuel} \times \eta_b} \qquad (4\text{-}93)$$

where: Bc = coal firing rate, [kg/s]

h_{ss} = enthalpy of superheated steam, [J/kg]

h_{fw} = enthalpy of feed water heater at economizer inlet, [J/kg]

h_{sg} = enthalpy of saturated steam at drum pressure, [J/kg]

h_{sw} = enthalpy of saturated water at drum pressure, [J/kg]

$\dot{m}_{bd}$ = blow-down mass of steam, [kg/s]

$\dot{m}_{s1}$ & $\dot{m}_{s2}$ = the two stream of the steam flow rate, [kg/s]

4.5 Boiler Simulation Program Description

The boiler problem can be solved as a separate system or subsystem by a subroutine called *boiler_solver* in the main power plant program. Solving the problem of this subsystem requires the model equations of each component available in it. These components are: combustion furnace, superheaters, economizer and air pre-heater. The combustion module which gives the necessary data about the combustion gases will be called independently in the *boiler_solver*.

4.5.1 Components Summarized Model Equations

1) ***Combustion furnace***

The necessary equations which can define the heat transfer phenomena in the furnace will be (Eqns.(4-25) to (4-27)):

$$\dot{Q}_f = h_{sw} A_0 \left(T_g - T_{wo}\right)$$

$$\dot{Q}_f = h_{TP} A_i \left(T_{wi} - T_{sat}\right)$$

$$\dot{Q}_f = \frac{2\pi k_t l \left(T_{wo} - T_{wi}\right)}{\ln\left(\frac{r_o}{r_i}\right)}$$

The heat absorbed by the fluid, q_f, can be identified from the equation:

$$\dot{Q}_f = B_c H v_f - m_g C p_g \left(T_g - T_{amb}\right)$$

where: B_c = coal firing rate, [*kg/s*]

Hv_f = calorific value of fuel as-received base, [*J/kg*]

2) ***Super heaters***

The governing equations of superheater module with the by-pass stream for temperature control would be summarized from Eqns(4-56), (4-59), (4-83) & (4-84)

$$m_s C p_s (T_{so} - T_{si}) = \varepsilon_s C_{\min} (T_{gi} - T_{sat})$$

$$m_s C p_s (T_{so} - T_{si}) = m_g C p_g (T_{gi} - T_{go})$$

$$\varepsilon_s = \frac{1 - \exp[-NTU(1-C)]}{1 - C\exp[-NTU(1-C)]}$$

$$m_s = m_{s1} + m_{s2}$$

$$m_{s1} C p_s \left(T_{so} - T_{ss}\right) = m_{s2} C p_s \left(T_{ss} - T_{si}\right)$$

3) ***Economizer***

As defined before, the economizer can be a steaming one so it can be easily defined by enthalpy difference.

$$m_{fw}(h_e - h_{fw}) = \varepsilon_e C_{\min}(T_{gie} - T_{fw})$$

$$m_{fw}(h_e - h_{fw}) = m_g Cp_g (T_{gie} - T_{goe})$$

$$\varepsilon_e = \frac{1 - \exp[-NTU(1-C)]}{1 - C\exp[-NTU(1-C)]}$$

4) ***Air heater***

The tubular air heater also defined by the equations (Eqns.(4-56), (4-60), (4-61)):

$$m_g Cp_g (T_{gia} - T_{goa}) = m_a Cp_a (T_{ca} - T_{amb})$$

$$m_g Cp_g (T_{gia} - T_{goa}) = \varepsilon_a C_{\min} (T_{gia} - T_{amb})$$

Assuming that $C_{\min} = m_g Cp_g$

$$\varepsilon_a = \frac{1}{C}\{1 - \exp[-C[1 - \exp(-NTU)]]\}$$

5) ***Boiler drum***

Material balance at the drum will give:

$$m_s = x_r m_{cir} + x_{eco} m_{fw}$$

where:

$m_{cir} = CR * m_s$ taking the maximum recommended circulation ratio, CR

x_r = steam mass fraction at riser exit

x_{eco} = steam mass fraction of the feed water exiting economizer

6) ***Boiler efficiency & Coal firing rate***

The boiler efficiency from the losses can be given by Eqn.(4-88):

$$\eta_b = 100 - 100\left(\frac{H_{df} + H_{mf}}{H_{fuel}} + RC - \frac{H_{ca}}{H_{fuel}}\right)$$

And, the coal firing rate from Eqn.(4-93)

$$Bc = \frac{m_{s1}\left(h_{ss} - h_{fw}\right) + m_{s2}\left(h_{sg} - h_{fw}\right) + m_{bd}\left(h_{sw} - h_{fw}\right)}{Q_{fuel} \times \eta_b}$$

Now, all the components' governing equations are collected. Solving boiler problem means solving the above 17 equations simultaneously by Newton-Raphson simulation technique. Actually the residues of the model equations of each component are identified by their own residue generator functions and assembled in the main program *boiler_solver* by a subroutine *boiler_residu_ass* and stored in a vector *R*.

After the residue vector *R* for the boiler system is identified, the subroutine *jacob_boiler* is called by the program and the Jacobean of the system will be evaluated, and stored in matrix *pd*. Finally the system of equation will have the form:

$$[pd]\{\Delta v\} = \{R\} \qquad (4\text{-}94)$$

The Gaussian elimination subroutine *GAUSS* can solve the above form of equation (Eqn(4-94)) for the correcting vector Δv. This vector corrects the guessed initial values of the variables and the iteration is continued until convergence achieved.

4.5.2 Flow Chart of Boiler System Program

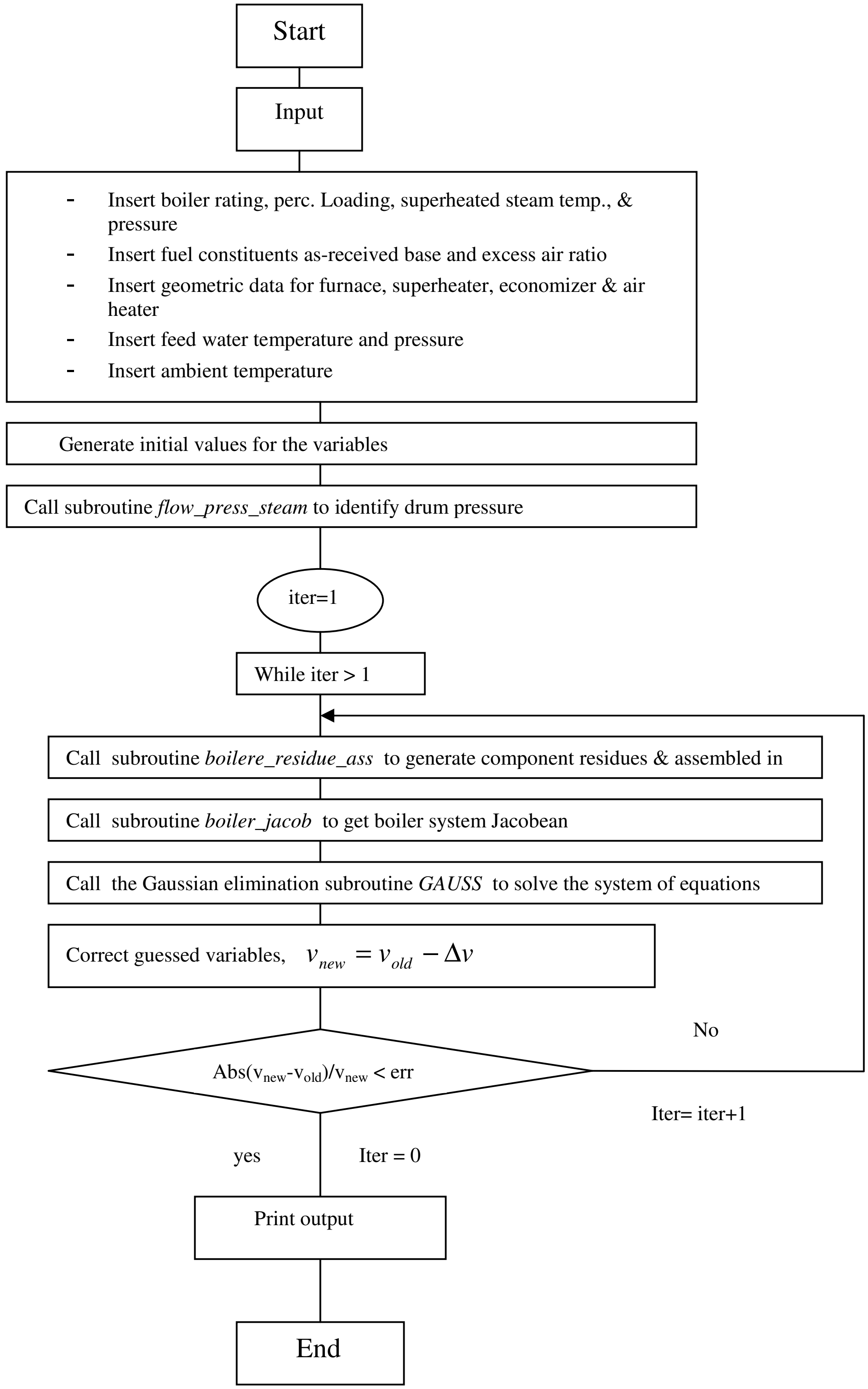

Fig.4.4 Flow chart for program *boiler_solver*

CHAPTER 5

Steam Turbine Modeling and Simulation

The components in the turbine plant, extraction condensing turbine, condenser feed water heaters and pumps, are modeled independently as the boiler components. These components have their own program to validate the model.

5.1 Extraction Condensing Turbine Model

The turbine which is considered here to be modeled is a non-reheat extraction condensing turbine; because it has low power generating capacity it is not economical to install a reheat unit.

Modeling of steam turbine needs different turbine characteristic curves which are provided by the manufacturer in the turbine thermal kit. These curves are used to determine the performance of the steam turbine for various steam cycle condition. The other important things in the modeling of steam turbine are the conservation laws, mass and energy conservation equations.

In the thermal kit the following are found:

- Extraction pressures versus flow to the following stages
- Over-all turbine internal efficiency
- Leakages and mechanical losses
- Turbine expansion line end points
- Correction to expansion line end points
- Generator loss curve, etc.

The turbine modeled in this paper assumes negligible steam leakages from glands and valves to avoid complexity of the model.

Turbine overall internal efficiency curve is available for various turbine inlet temperatures and throttled pressures. The efficiency curves are plotted by referencing LP exhaust pressure of 0.0508 bar [8].

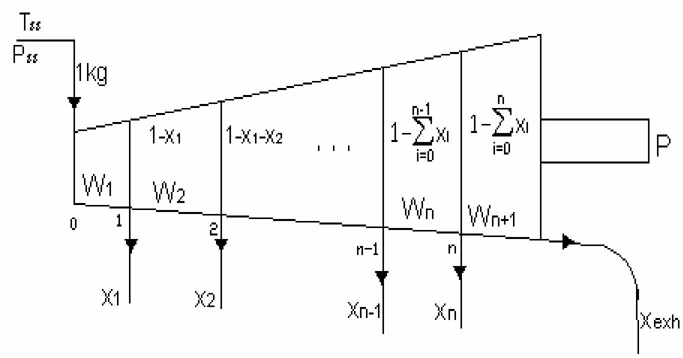

Fig.5.1 Schetch of extraction turbine

This exhaust pressure has been selected by the turbine manufacturer as a reference pressure convention. From this the expansion line end point (ELEP) of the turbine at a 0.0508 bar back pressure can then be identified.

The turbine efficiency from its characteristic curve can be adequately approximated by a polynomial fit as a function of inlet pressure and temperature with a reference exhaust pressure (0.0508 bar) as follows:

$$\eta = A_o(T) + A_1(T) \times p + A_2(T) \times p^2 \qquad (5\text{-}1)$$

with the coefficients $A_i\,(i = 0,1,2)$ expressed as

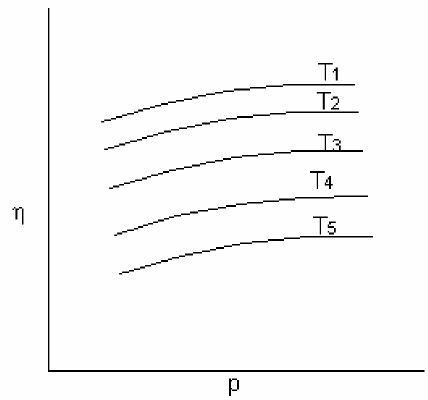

Fig. 5.2 Turbine isentropic efficiency at exhaust pressure of 0.0508 bars

$$A_i(T) = B_{i0} + B_{i1} \times T + B_{i2} \times T^2 \tag{5-2}$$

Taking nine points from the curve, three at each temperature, are required to determine the coefficients in Eqn.(5-1). At the different temperature Eqn.(5-1) can be written as:

At T_1,

$$\eta_1 = A_{10} + A_{11} \times p + A_{12} \times p^2 \tag{5-3a}$$

This forms the following equations at the three different pressure points.

$$\begin{aligned} \eta_{11} &= A_{10} + A_{11} \times p_1 + A_{12} \times p_1^2 \\ \eta_{12} &= A_{10} + A_{11} \times p_2 + A_{12} \times p_2^2 \\ \eta_{13} &= A_{10} + A_{11} \times p_3 + A_{12} \times p_3^2 \end{aligned} \tag{5-3b}$$

and then the coefficients of Eqn.(5-3) in a matrix form will be:

$$\begin{Bmatrix} A_{10} \\ A_{11} \\ A_{12} \end{Bmatrix} = \begin{bmatrix} 1 & p_1 & p_1^2 \\ 1 & p_2 & p_2^2 \\ 1 & p_3 & p_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} \eta_{11} \\ \eta_{12} \\ \eta_{13} \end{Bmatrix} \tag{5-4}$$

Similarly, for the other two temperatures the matrix would have the form:

$$\begin{Bmatrix} A_{20} \\ A_{21} \\ A_{22} \end{Bmatrix} = \begin{bmatrix} 1 & p_1 & p_1^2 \\ 1 & p_2 & p_2^2 \\ 1 & p_3 & p_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} \eta_{21} \\ \eta_{22} \\ \eta_{23} \end{Bmatrix} \quad (5\text{-}5)$$

$$\begin{Bmatrix} A_{30} \\ A_{31} \\ A_{32} \end{Bmatrix} = \begin{bmatrix} 1 & p_1 & p_1^2 \\ 1 & p_2 & p_2^2 \\ 1 & p_3 & p_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} \eta_{31} \\ \eta_{32} \\ \eta_{33} \end{Bmatrix} \quad (5\text{-}6)$$

The same curve-fitting procedure is used to evaluate coefficients in the quadratic expression Eqn.(5-2). Then for i=1

$$A_0(T) = B_{00} + B_{01} \times T + B_{02} \times T^2 \quad (5\text{-}7a)$$

the matrix derived from this equation will be:

$$\begin{Bmatrix} B_{00} \\ B_{01} \\ B_{02} \end{Bmatrix} = \begin{bmatrix} 1 & T_1 & T_1^2 \\ 1 & T_2 & T_2^2 \\ 1 & T_3 & T_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} A_{10} \\ A_{20} \\ A_{30} \end{Bmatrix} \quad (5\text{-}7b)$$

Similarly, for $i = 1 \,\&\, 2$ the coefficients in Eqn.(5-2) will be:

$$\begin{Bmatrix} B_{10} \\ B_{11} \\ B_{12} \end{Bmatrix} = \begin{bmatrix} 1 & T_1 & T_1^2 \\ 1 & T_2 & T_2^2 \\ 1 & T_3 & T_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} A_{11} \\ A_{21} \\ A_{31} \end{Bmatrix} \quad (5\text{-}8)$$

$$\begin{Bmatrix} B_{20} \\ B_{21} \\ B_{22} \end{Bmatrix} = \begin{bmatrix} 1 & T_1 & T_1^2 \\ 1 & T_2 & T_2^2 \\ 1 & T_3 & T_3^2 \end{bmatrix}^{-1} \begin{Bmatrix} A_{12} \\ A_{22} \\ A_{32} \end{Bmatrix} \quad (5\text{-}9)$$

The vector A's in the right-hand side of Eqns.(5-7b) – (5-9) can be identified from Eqns.(5-4) – (5-6). The final expression for η will then be:

$$\eta = (B_{00} + B_{01}T + B_{02}T^2) + (B_{10} + B_{11}T + B_{12}T^2)p + (B_{20} + B_{21}T + B_{22}T^2)p^2 \quad (5\text{-}10)$$

From the definition of isentropic line efficiency:

$$\eta = \frac{h_{in} - h_{exh}}{h_{in} - h_{s,exh}} \tag{5-11}$$

Now we are in a position to identify the exhaust enthalpy, i.e. ELEP, at the reference exhaust pressure. Hence,

$$h_{exh,r} = h_{in} - \eta \times \left(h_{in} - h_{s,exh}\right) \tag{5-12}$$

The enthalpy then corrected to the actual LP exhaust pressure. The correcting curve for the actual exhaust pressure is available in [8]. In this work, it has been tabulated in **Table-D** for computer programming convenience. The corrected exhaust enthalpy will then be:

$$h_{exh,a} = h_{exh,r} + \Delta ELEP \tag{5-13}$$

Hence, the actual expansion efficiency will be:

$$\eta_{act} = \frac{h_{in} - h_{exh,a}}{h_{in} - h_{s,exh,a}} \tag{5-14}$$

where: $\Delta ELEP$ = Enthalpy correction to actual exhaust pressure, (**Table-D**)

η_{act} = Internal efficiency of the turbine at actual exhaust pressure

T = Temperature of super heated steam at the turbine inlet, [*K*]

p = Pressure of throttled steam at turbine inlet, [*bar*]

h_{in} = Enthalpy of superheated steam at the turbine inlet, [*J/kg*]

$h_{exh,a}$ = Enthalpy of steam at turbine actual exhaust pressure, [*J/kg*]

$h_{s,exh,a}$ = Isentropic expansion enthalpy at actual exhaust pressure,[*J/kg*]

$h_{exh,r}$ = Enthalpy at the reference exhaust pressure of 0.0508 bar, [*J/kg*]

The inlet and exhaust conditions of the turbine are identified. Since the individual segment efficiency is difficult to identify, applying the straight line approach will simply join the inlet and exhaust conditions of the turbine on the *Mollier diagram* and assume all intermediate points lie on that line. This approach enables one to identify steam enthalpies at the extraction pressures of intermediate stages for feed water heating and for process heat demand. The *Gauss-Siedel* successive iteration is suitable to identify the enthalpy of steam at point of extraction. The procedure is listed as follows;

1. Guess a value of entropy such as the isentropic value
2. Solve the enthalpy at the desired pressure, which is the extraction pressure, and the current guess for entropy
3. Write a straight line equation for the expansion with enthalpy as the dependent variable. Since the two points' enthalpy and entropy are known,

$$s = s_{in} + (h - h_{in}) \times \left(\frac{s_{in} - s_{exh}}{h_{in} - h_{exh}} \right) \qquad (5\text{-}15)$$

where: s and h are entropy and enthalpy at the extraction pressure

4. Get h at the new entropy value found in Eqn.(5-15) and the extraction pressure using property program, a function named as *superheat_p_s*
5. Repeat the computation from step-2 until h does not vary significantly.

The above procedure can be used if it is necessary to identify enthalpies separately. The same enthalpies can also be solved in a turbine module simultaneously with other turbine variables.

5.1.1 *Turbine Model with Conservation Equations*

The conservation equations are important tools to identify the work of each expansion stage of the turbine and of the necessary steam flow rate at the turbine inlet for a given power output and extraction steam for regenerative heating and process heat demand

Taking *1kg* of steam entering into the turbine, the work of the turbine up-to the first extraction point would be (see Fig.5.1):

$$W_1 = h_{ss} - h_1 \tag{5-16}$$

and using turbine expansion efficiency relation:

$$W_1 = \eta_{act}\left(h_{ss} - h_{1s}\right) \tag{5-17}$$

where: η_{act} = Internal efficiency consist of non-isentropic expansion and mechanical loss.

W_1 = work done by 1kg of steam at the first expansion stage,[*J*]

h_{ss} = enthalpy of superheated steam at turbine inlet, [*J/kg*]

h_1 = Enthalpy of steam at the first extraction point, [*J/kg*]

h_{1s} = Isentropic expansion enthalpy at the first extraction point, [*J/kg*]

For the next expansion stage with x_1 kg of steam extracted at the first extraction point, the work produced by the turbine will be:

$$W_2 = \left(1 - x_1\right)\left(h_1 - h_2\right) \tag{5-18}$$

$$W_2 = \eta_{act}\left(1 - x_1\right)\left(h_1 - h_{2s}\right) \tag{5-19}$$

where:

W_2 = work done by $\left(1 - x_1\right)$kg of steam expanded from point1 to point2

h_2 = Actual enthalpy of steam at extraction point 2, [*J/kg*]

h_{2s} = Isentropic enthalpy of steam at extraction point 2, [*J/kg*]

Following similar procedure, for *n* number of extraction points of the turbine, as shown in Fig.5-1, the work done by the steam at the n^{th} expansion stage will be:

$$W_n = \left(1 - \sum_{i=1}^{n-1} x_i\right)\left(h_{n-1} - h_n\right) \tag{5-20}$$

$$W_n = \eta_{act}\left(1 - \sum_{i=1}^{n-1} x_i\right)\left(h_{n-1} - h_{ns}\right) \tag{5-21}$$

where: h_{n-1} = enthalpy at the (n-1)th extraction point, [*J/kg*]

h_n = enthalpy of steam at nth extraction point, [*J/kg*]

h_{ns} = isentropic enthalpy at the nth extraction point, [*J/kg*]

n = number of extraction points

x_i = steam extraction flow at i^{th} extraction point.

Finally, the last expansion stage work can be set as:

$$W_{n+1} = \left(1 - \sum_{i=1}^{n} X_i\right)\left(h_n - h_{exh}\right) \tag{5-22}$$

After identifying the work of each stage for a *1kg* of steam at turbine inlet, the necessary amount of steam flow rate at the inlet of the turbine can be given by:

$$\dot{m}_s = \frac{P}{W_t} \tag{5-23}$$

where: $\dot{m}_s$ = Mass flow rate of steam at the turbine inlet, [*kg/s*]

P = Power output at turbine shaft, [*W*]

W_t = Sum of work output of each stage with 1kg/s steam flow at turbine inlet. [*W*]

$$W_t = \sum_{i=1}^{n+1} Wi \tag{5-24}$$

Finally, conservation of mass around the turbine would give:

$$1 = \left(x_1 + x_2 + ... + x_n + x_{exh}\right) \tag{5-25}$$

where: $x_1, x_2, ..., x_n$ are extraction steam fraction at the respective extraction points

x_{exh} = fraction of steam exhausted from turbine to condenser

Solving the above set of linear equations is an easy task. Put the system of equations in a matrix and solve by Gaussian elimination method. But when turbine problem is going to be solved together with the rest of components such as feed water heaters, pumps, condensers and so on, which is the approach followed in this work, there are several non-linear equations and so a Newton-Raphson technique should be followed.

5.2 Feed Water Heaters Model

Steam is extracted from the intermediate stages of a turbine for regenerative feed water heating. This yields higher cycle efficiency by increasing the temperature of the feed water and by reducing the amount of energy lost in the condenser.

A turbine with a single-stage regenerative preheating of feed water, the lowest economic effect of regeneration will be obtained at a very high or low pressure of extracted steam, i.e. a pressure close to that of main steam or waste steam, and the highest effect will be obtained at a certain intermediate pressure. This is illustrated well in [14]. For more than one extraction points, the extraction points are arranged so that enthalpy of feed water increases in each water heater roughly by the same magnitude. That is, the heat drops of steam between adjacent extraction points are roughly the same.

The number of feed water heaters used in any specific cycle depends mainly on the size of the turbine, the inlet and exhaust steam conditions, and certainly the economic consideration.

The regenerative feed water heaters can be classified into open and closed heaters.

5.2.1 *Open Heaters Module (Mixing Type)*

These heaters are designed to deaerate the incoming condensate. This action liberates the dissolved, non-condensable gases consisting mainly of oxygen, nitrogen, ammonia, and carbon dioxide, from the condensate. This kind of heater consists of heater section, the vent condenser, and the storage section.

In heater section, the incoming condensate comes into contact with the extraction steam, heating the condensate, and condensing the steam. Since the condensate temperature is raised to the boiling point, all the dissolved gases are liberated.

The energy balance on the deaerator will give:

$$m_{hd}h_{hd} + m_c h_c + m_s h_s + m_{mw}h_{mw} = m_f h_f \qquad (5\text{-}26)$$

This equation is used to identify the quantity of steam bled from the turbine, m_s, which can make the drain of the deaerator to be saturated water at the deaerator pressure.

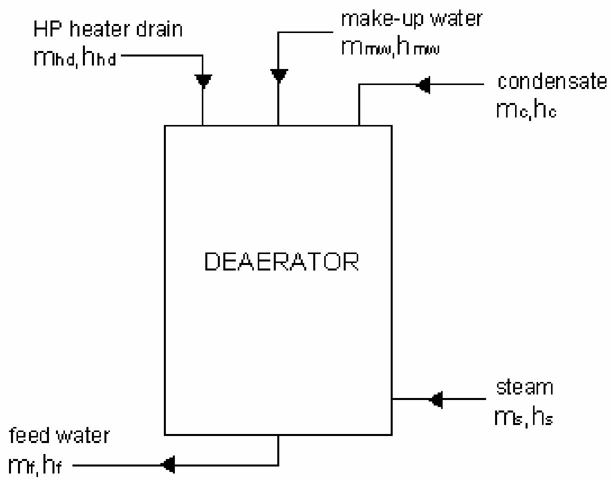

Fig.5.3 Deaerator flow schematic

5.2.2 Closed Heaters Module

The closed heaters are tube and shell design, with the condensate or feed water in the tube side and the extraction steam and resulting condensed steam (heater drains) in the shell side.

Modeling of closed feed water heaters considers only condensing zone except the last high pressure heater in which degree of superheat of the extracted steam may be high, hence, desuperheater is considered.

Since the steam condenses on the outside of the tubes, the outside tube temperature is the steam saturation temperature at the prevailing pressure. Therefore, the feed water temperature can only approach the saturation temperature even when the extraction steam entering is superheated. The drains from the heaters are taken to be saturated water at the shell pressure of the heaters.

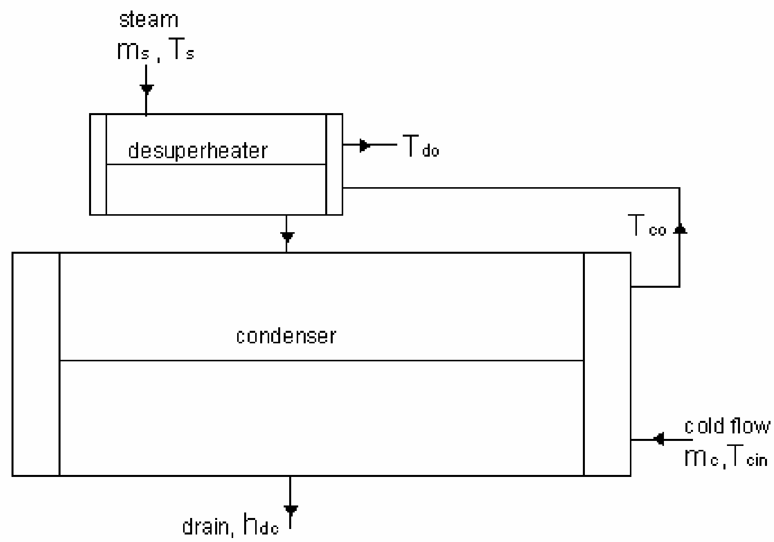

Fig. 5.4 Condenser & desuperheater components of feed-water heater

Similar to convection pass heaters in the boiler, the feed water heaters also modeled using $NTU - \varepsilon$ method.

A feed-water-heater can have three heat transfer zones; de-superheating zone, condensing zone and drain cooling zone. The last zone, drain cooling, is neglected in the modeling of the heaters in this paper and so the drain of all heaters is considered to be saturated liquid. The de-superheating zone is only accounted in the model of high pressure heaters in which the degree of superheat is expected to be significant. In the low pressure heaters the heat transfer is only assumed to be taking place by condensation of the extracted steam. Condensing zone, therefore, will be the main heat transfer zone of all feed water heaters.

Condensing Zone

Heat load of the condensing zone will have different expression depending on the condition of the extracted steam. Therefore, if the extracted steam is superheated, the heat load of the heater will be:

$$\dot{Q}_c = \dot{m}_s c_{p_s}(T_s - T_{sat}) + \dot{m}_s h_{fg} \tag{5-27}$$

While the extracted steam is wet, the heat load considers the steam quality. Hence:

$$\dot{Q}_c = \dot{m}_s x h_{fg} \tag{5-28}$$

where: $\dot{Q}_c$ = is heat transfer between the fluids, [*W*]

$\dot{m}_s$ = Extracted steam flow rate, [*kg/s*]

x = Extracted steam quality

h_{fg} = Heat of vaporization at the heater shell pressure, [*J/kg*]

T_s = Temperature of extracted steam, [*K*]

T_{sat} = Saturation temperature of steam at heater shell pressure, [*K*]

For the heaters model $NTU - \varepsilon$ approach is selected, so the heat transfer rate considering the effectiveness of the heater will be:

$$\dot{Q}_c = \varepsilon_c C_{\min,c}(T_s - T_{cin}) \tag{5-29}$$

where ε_c of the condenser is given by:

$$\varepsilon_c = 2\left[1 + \frac{1+\exp[-NTU]}{1-\exp[-NTU]}\right]^{-1} \tag{5-30}$$

$$NTU = \frac{U_0 A_0}{C_{\min}} \tag{5-31}$$

The overall heat transfer coefficient, U_0, will be identified later.

Desuperheating Zone

The desuperheater, which is usually included with high pressure heaters, can be in the same shell with the condenser or has its own shell. HP heater model in this paper considers the condenser and the desuperheater in a separate shell both as a two-pass heat exchanger. The feed water passes first through the condensing section and then through the desuperheater.

The model of the condenser of this high pressure heater is similar to the condenser modeled before. Hence, the heat load in the desuperheater will be;

$$\dot{Q}_d = \varepsilon_d C_{\min}\left(T_s - T_{cco}\right) \tag{5-32}$$

Taking temperature leaving the desuperheater as saturated steam, the heat balance equation between the two fluids will be:

$$\dot{m}_s c_{p_s}\left(T_s - T_{sat}\right) = \dot{m}_f c_{pf}\left(T_{dco} - T_{cco}\right) \tag{5-33}$$

For the two-pass shell and tube heat exchanger, the effectiveness of the desuperheater will have the form:

$$\varepsilon_d = 2\left[1 + C_d + \frac{1+\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}{1-\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}\left(1+C_d^2\right)^{1/2}\right]^{-1} \tag{5-34}$$

where: $NTU_d = \dfrac{U_d A_d}{C_{\min}}$

$$C_d = \frac{C_{\min,d}}{C_{\max,d}}$$

To solve the feed water heaters problem, first list out the known and unknown variables and second collect the necessary equations of the heat exchangers which can express characteristic of the heaters and the heat transfer between the fluids.

The unknown variables of a feed water heater composed of only a condenser are, $U_c, NTU_c, C_c, \varepsilon_c, T_{co}$ and m_s. The number of equations must equal the unknown variables of the equations to be solved. There are alternatives to reduce the number of equations to avoid programming complexity. One good alternative is to lag parameters that are nearly constant or secondary importance. That is, we can use values from the previous iteration for these parameters instead of specifying them as unknowns. Hence, NTU, C and U are lagged instead of attempting to solve them simultaneously with other variables. So, the unknown variables are reduced to three, and therefore three equations are enough to solve the system.

$$C_c(T_{cco} - T_{ccin}) = \varepsilon_c C_{\min,c}(T_s - T_{ccin}) \quad \text{(5-35a)}$$

$$\varepsilon_c = 2\left[1 + \frac{1+\exp[-NTU]}{1-\exp[-NTU]}\right]^{-1} \quad \text{(5-35b)}$$

The third equation depends on the steam condition at the inlet.

If the inlet steam is superheated:

$$C_c(T_{cco} - T_{ccin}) = \dot{m}_s c_{p_s}(T_s - T_{sat}) + \dot{m}_s h_{fg} \quad \text{(5-35c)}$$

Or, if it is wet steam:

$$C_c(T_{cco} - T_{ccin}) = m_s x h_{fg} \quad \text{(5-35d)}$$

When a desuperheater is included to the feed water heaters, the number of unknowns are raised by two, effectiveness of desuperheater, ε_d, and the temperature of the cold flow exiting desuperheater, T_{dco}. So, two equations should be added to the above three equations. These are:

$$\varepsilon_d = 2\left[1 + C_d + \frac{1+\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}{1-\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}\left(1+C_d^2\right)^{1/2}\right]^{-1} \quad \text{(5-35e)}$$

$$\varepsilon_d C_{\min}(T_s - T_{cco}) = C_c(T_{dco} - T_{cco}) \quad \text{(5-35f)}$$

5.2.3 ***Tube- and Shell-side Heat transfer Coefficients***

Tube side heat transfer coefficient for a non-condensing flow can be identified from Eqns.(4-79) to (4-82). Here only the shell side coefficients for condensing and non-condensing flow will be considered.

- ***Condensation Shell-side Heat Transfer Coefficient*** h_o

The laminar flow condensing shell-side heat transfer coefficient h_o, will be calculated by the Nusselt method with the Kern correction for condensate inundation.

$$h_o = 0.728 \times \left[\frac{\rho_f (\rho_f - \rho_g) \times g \times h_{fg} \times k_f^{\;3}}{\mu_f \times (T_{sat} - T_s) \times d_o} \right]^{1/4} \times \frac{1}{N_c^{1/6}} \qquad \text{(5-36a)}$$

For turbulent flow the relation is given by Butter Worth will be:

$$h_o = 0.416 \left(\frac{k_f}{d_o} \right) \left(1 + (1 + 9.47F)^{0.5} \right)^{0.5} \mathrm{Re}^{0.5} \qquad \text{(5-36b)}$$

where: $$F = \frac{g d_o \mu_f h_{fg}}{u_g^2 k_f \Delta T} \qquad \text{(5-36c)}$$

ρ = density, [*kg/m*^{3}]

μ = dynamic viscosity, [*kg/m-s*]

h_{fg} = heat of condensation, [*J/kg*]

k_f = thermal conductivity, [*W/m-^{o}K*]

N_c = number of tubes per column

subscripts: f = saturated liquid

s = surface

sat = saturation

g = saturated vapor

- ***Non-condensing Shell-side Heat Transfer Coefficients, h_o***

The complex flow pattern on the shell side, the great number of variables involved, make it difficult to predict the shell side heat transfer coefficient. Coulson [4] solves this coefficient procedurally as follows:

1. Calculate the area for cross-flow at the shell equator

$$A_s = \frac{(p_t - d_o) D_s l_b}{p_t} \tag{5-37}$$

where: p_t = tube pitch

d_o = tube outside diameter, [*m*]

D_s = shell inside diameter, [*m*]

l_b = baffle spacing, [*m*]

2. Calculate the shell-side mass velocity and the linear velocity:

$$G_s = \frac{m_s}{A_s} \tag{5-38a}$$

$$u_s = \frac{G_s}{\rho} \tag{5-38b}$$

where: m_s = fluid flow rate on the shell side, [*kg/s*]

ρ = shell-side fluid density, [*kg/m³*]

G_s = shell-side mass velocity, [*kg/m²-s*]

u_s = shell-side linear velocity, [*m/s*]

3. Calculate the shell-side hydraulic diameter.

   For a square pitch arrangement:

$$d_e = \frac{4\left(p_t^2 - \pi d_o^2 / 4\right)}{\pi d_o} \quad (5\text{-}39a)$$

   For triangular pitch arrangement:

$$d_e = \frac{1.1}{d_o}\left(p_t^2 - 0.917 d_o^2\right) \quad (5\text{-}39b)$$

where: d_e is equivalent diameter, [*m*]

4. Calculate the shell-side Reynolds number:

$$\mathrm{Re} = \frac{G_s d_e}{\mu} \quad (5\text{-}40a)$$

5. Identify the shell-side heat transfer coefficient from the Nusselt number definition.

$$Nu = \frac{h_o d_e}{k} = 0.36\,\mathrm{Re}^{0.55}\,\mathrm{Pr}^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad (5\text{-}40b)$$

$$h_o = 0.36\left(\frac{k}{d_e}\right)\mathrm{Re}^{0.55}\,\mathrm{Pr}^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad (5\text{-}40c)$$

Therefore, the overall heat transfer coefficient, U_o , for both zones can be given by:

$$U_o = \frac{1}{R_i + R_s + R_o} \quad (5\text{-}41a)$$

where: $R_i = \frac{d_o}{h_i d_i}$ (5-41b)

$$R_s = \frac{d_o \ln\left(\frac{d_o}{d_i} \right)}{2k_s} \tag{5-41c}$$

$$R_i = \frac{1}{h_o} \tag{5-41d}$$

5.3 *Condenser Module*

The power plant surface condenser will be attached to the low-pressure exhaust of a steam turbine. Its purposes are:

- To produce a vacuum or desired backpressure at the turbine exhaust for the improvement of plant heat rate.
- To condense turbine exhaust steam for reuse in the closed cycle.
- To deaerate the condensate, and
- To accept heater drains, steam line drains, and start-up and emergency drains

An economical turbine back pressure is from 0.034 to 0.118 bars. [8].

For a condenser to deaerate the condensate it must remove oxygen and other non-condensable gases to an acceptable level compatible with material selection and/or chemical treatment of the feed water (condensate).

Deaeration in a condenser is accomplished by Henry's law, which states that the concentration of the dissolved gas in a solution is directly proportional to the partial pressure of that gas in the free space above the condensate level in the hot well. In a condenser droplets of condensate are continually scrubbed with steam, liberating the O_2 and permitting it to flow to the low-pressure air-removal section, where it is discharged to the atmosphere by the air removal equipment.

The heat removed from the turbine exhaust steam in the condenser can be estimated by:

$$\dot{Q} = \dot{m}_s \left(h_s - h_f\right) \tag{5-42}$$

where: $\dot{m}_s$ = Steam mass flow to the condenser, [*kg/s*]

h_s = Steam enthalpy at condenser inlet, [*J/kg*]

h_f = Enthalpy of condensate at condenser exhaust, [*J/kg*]

In this paper, the enthalpy of condensate at the condenser outlet will be taken as enthalpy of saturated water at the condenser pressure.

The heat load of the condenser can also be estimated using the $NTU - \varepsilon$ method as before:

$$\dot{Q} = \varepsilon C_{\min} \left(T_{sat} - T_{win}\right) \tag{5-43}$$

If heat loss to the atmosphere is neglected, the energy balance between the fluids gives:

$$\dot{m}_s \left(h_s - h_f\right) = \dot{m}_w c_{p_w} \left(T_{wo} - T_{win}\right) \tag{5-44}$$

ε of the condenser can be identified from effectiveness definition of multi-pass shell and tube heat exchanger with $\frac{C_{\min}}{C_{\max}} = 0$. Because $C_{\max}$ goes to infinity.

$$\varepsilon = 2\left[1 + \frac{1 + \exp\left[-NTU\right]}{1 - \exp\left[-NTU\right]}\right]^{-1} \tag{5-45}$$

[18] gives recommended value for the terminal temperature difference for surface condensers to be in the range 5-10 ^{0}K. Hence, taking TTD =7^{0}K, the maximum rise in temperature of the cooling water:

$$T_{wo} = T_{sat} - 7 \tag{5-46}$$

In the above surface condenser model, the variables to be solved are the effectiveness of the condenser and the necessary amount of cooling water flow rate.

The shell side and tube side heat transfer coefficients can be identified from Eqns.(5-36) and (4-82); and the overall heat transfer coefficient can be found from Eqn.(5-41a).

5.4 Turbine System Program Description

The turbine subsystem, which consists of turbine and all heaters, can also be simulated independently by the program called *turbine_solver*. Similar to the boiler system, the components of the turbine system have their own residue generator function. The residues of each component assembled in a subroutine *turbine_residue_ass* and stored in vector *R*.

5.4.1 Components Summarized Model Equations

The model equation for the components can be collected as follows:

1) *Turbine*

The equations, which are of conservation equations, that determine the work done by each extraction stage are listed out in Eqns.(5-20) to (5-24). The number of equations and the number of variables depend on the number of steam extraction points from the turbine shell for regenerative heating or/and for process heat demand. For n number of extraction stages and taking $x_0 = 0$:

$$W_n = \eta_t \left(1 - \sum_{i=0}^{n-1} x_i\right)(h_{n-1} - h_{nise})$$

$$W_n = \left(1 - \sum_{i=0}^{n} x_i\right)(h_{n-1} - h_n)$$

$$m_s = P / W_t$$

$$W_t = \sum_{j=1}^{n} W_j$$

where: $P = \dfrac{P_{el}}{\eta_{gen}}$

The system of equations is solved by considering boundary conditions of the turbine ends. This makes the number of equations exceed the number of unknowns by one. So, one equation should be eliminated.

2) *Closed feed water heaters*

Residue for closed feed water heaters would be identified first for one heater and then assembled by a function called *resid_ass_fun* to the number of heaters given. This also finally assembled to the turbine system residue vector, R, with the turbine and condenser residues.

The three types of heaters, the low pressure (LP), the high pressure (HP) and open heater (deaerator), have defined by their own model equations in Eqn.(5-35). Assuming that the degree of super heat for LP heaters is very small and the drain for all types of heaters is saturated water at their respective shell pressure, the model equations can be collected as follows:

a) LP heaters

$$\dot{m}_s c_{p_s}(T_s - T_{sat}) + \dot{m}_s h_{fg} = \varepsilon_{lp} C_{\min}(T_s - T_{cin})$$

$$\dot{m}_s c_{p_s}(T_s - T_{sat}) + \dot{m}_s h_{fg} = \dot{m}_c c_{p_c}(T_{co} - T_{cin})$$

$$\varepsilon_{lp} = 2\left[1 + \frac{1+\exp[-NTU]}{1-\exp[-NTU]}\right]^{-1}$$

b) HP heaters

In these heaters the degree of superheat is expected to be significant and so de-super heater is considered. Hence, the equations which define the de-superheating and the condensing sections together must be considered.

$$\dot{m}_{fw} c_{p_{fw}}(T_{fw} - T_{cco}) = \varepsilon_d C_{\min d}(T_s - T_{cco})$$

$$\varepsilon_d = 2\left[1 + C_d + \frac{1+\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}{1-\exp\left[-NTU_d\left(1+C_d^2\right)^{1/2}\right]}\left(1+C_d^2\right)^{1/2}\right]^{-1}$$

$$\dot{m}_{fw} c_{p_{fw}}(T_{cco} - T_{cin}) = \dot{m}_s h_{fg}$$

$$\varepsilon_c C_{\min}(T_{sat} - T_{cin}) = \dot{m}_s h_{fg}$$

$$\varepsilon_{hp} = 2\left[1 + \frac{1+\exp[-NTU_c]}{1-\exp[-NTU_c]}\right]^{-1}$$

The number of equations of the total heaters equals the number of equations for a single heater multiplied by the number of heaters for both types of heaters.

3) Deaerator

It is an open heater in which the extracted steam, the condensate flow, the drain from HP heaters and make-up water are mixed together. The drain from this heater is taken to be saturated water too. The conservation equation around this heater gives Eqn.(5-29):

$$\dot{m}_{hd} h_{hd} + \dot{m}_c h_c + \dot{m}_s h_s + \dot{m}_{mw} h_{mw} = \dot{m}_f h_f$$

4) *Surface Condenser*

The variables to be identified in the condenser are the circulating water ratio, m_{cr} and the effectiveness of the condenser. The model equations are given in Eqns.(5-42) to (5-46). Since the number of unknowns are two, the number of equations which define the component should be two. Hence:

$$\frac{\dot{Q}_c}{\dot{m}_{cf}} = \varepsilon_c \dot{m}_{cr} c_{p\,w} (T_{sat} - T_{win})$$

$$\varepsilon_c = 2\left[1 + \frac{1 + \exp[-NTU]}{1 - \exp[-NTU]}\right]^{-1}$$

where: $\dot{m}_{cf} = \dot{m}_{exh} + \dot{m}_{lpd}$

$\dot{m}_{exh}$ = steam exhausted from the turbine

$\dot{m}_{lpd}$ = drain flow from LP heaters

The residue generator of the condenser, *condenser_residue*, identifies its residue vector and makes it ready for assembling in the main turbine system residue vector *R*.

Once system residue vector has been identified the next step is the identification of system Jacobean as Newton-Raphson simulation technique demands. So, *turbine_jacob* subroutine is called by the program and the Jacobean matrix, *[pd]*, would be found. Similar to boiler system, the system of equation finally will have the form:

$$[pd]\{\Delta v\} = \{R\} \quad (5\text{-}47)$$

Using Gaussian elimination function, *GAUSS*, the system of equation of the form Eqn.(5-47) is solved and the correcting vector, Δv, for the assumed initial values of the variables will be identified. Then the new values of the variables will be:

$$v_{new} = v_{old} - \Delta v \quad (5\text{-}48)$$

According to the procedure listed in chapter two the iteration is continued until convergence is achieved.

Verifying Example

A turbine generator plant with a capacity of 26.86MW is given with steam inlet temperature of 482 ^{o}C and a pressure of 59.62bar. The condenser back pressure is 0.0508 bars. The turbine has four regenerative heaters and produces electric power only **[8].**

The output of the turbine solver program and the reference value are tabulated in Table 5-1. & Table 5-2

Table 5-1 The total and extraction steam flow rate of the turbine system

	Reference value [kg/s]	Calculated value [kg/s]
Steam	28.95	23.90
Ext.1	1.71	1.40
Ext.2	1.52	2.00
Ext.3	1.62	0.84
Ext.4	1.83	0.90
Exh.	22.03	17.80

Table 5-2 performance data of the turbine system

	Reference value	Calculated value
Feed water temp. ^{o}C	185.15	187.85
Heat rate, [kJ/kWs]	2.8	2.3
Power gen. efficiency, [%]	36.0	43.0

5.4.2 Flow Chart of Turbine System Program

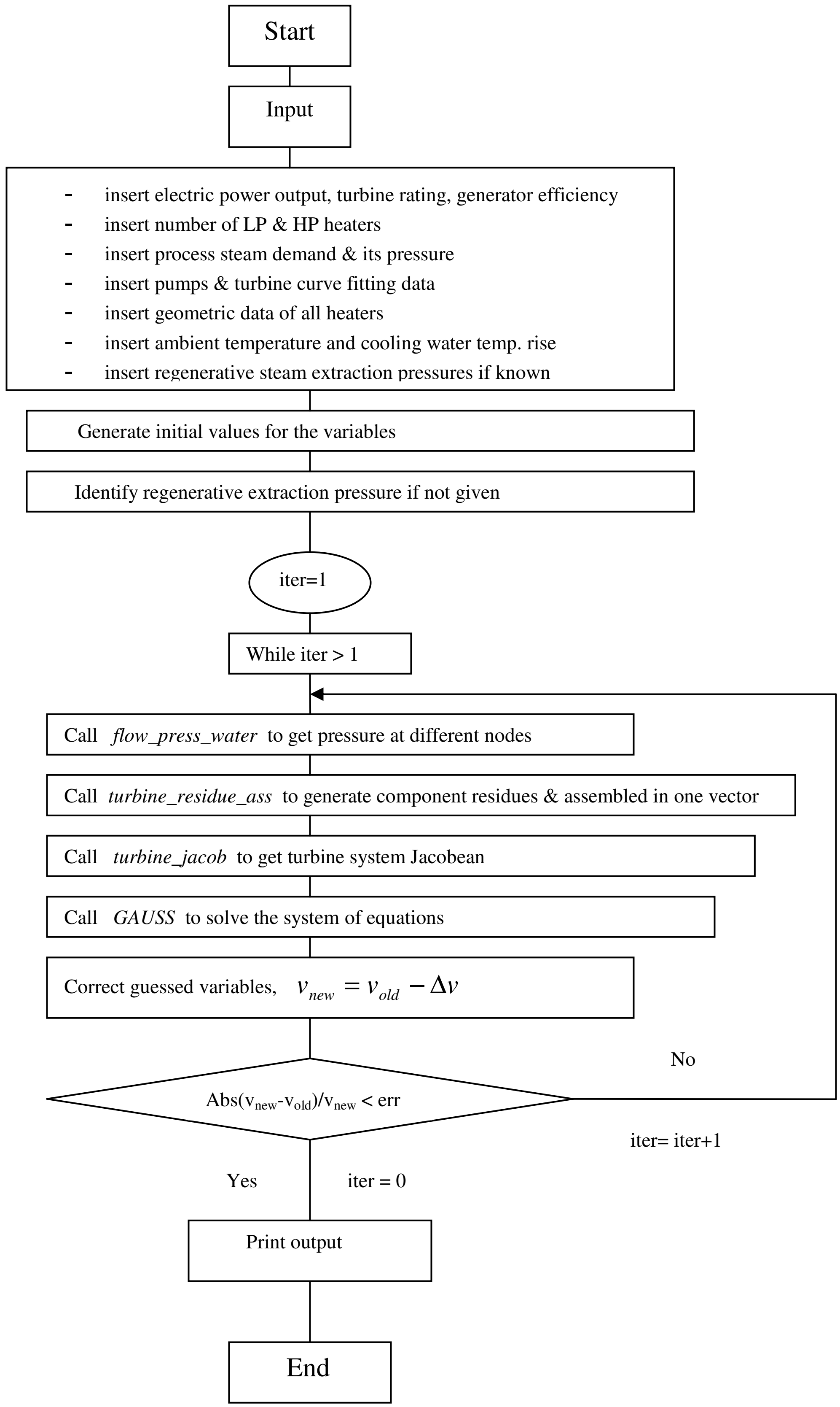

Fig.5.5 Flow chart for program *turbine_solver*

CHAPTER 6

System Verification and Discussion

6.1 Program Description

The overall simulation is composed of several simulation program subroutines. These are:

1. Turbine system solver
2. Boiler system solver
3. Fluid flow solver
4. Water property subroutine
5. Steam property subroutine
6. Combustion gas property subroutine
7. Combustion air property subroutine
8. Jacobean and Gaussian elimination subroutines, and so on

Each component of the power plant has been modeled independently and hence, has its own simulation subprogram. Since the simulation is based on the Newton-Raphson technique, the component simulators as well as the overall plant simulator consist of a residue generator functions, which are from the conservation equations, Jacobean generator function and the Gaussian elimination function to solve the set of linear equations.

The overall system simulator starts by assuming a guess value for steam extraction fraction from the condensing turbine, with given quantity and pressure of steam extracted for process heat demand. The rest of the guess values are generated by each solver.

The *turbine_solver* is able to identify all the necessary variables in the turbine system, turbine, regenerative heaters, condensate pump and condensers, when it is called by the main program. These variables are: the necessary amount of steam flow rate for the given electrical power output, feed water/condensate temperature at the exit of each heater, heaters effectiveness, the ratio of circulation cooling water at the condenser and so on.

The boiler simulator starts by taking steam flow rate and feed water temperature from the turbine simulator output as input. The output of the boiler simulator will be coal firing rate, thermal efficiency of the boiler, temperature of flue gas exiting the combustion furnace and at inlet and exit of convection heaters, quantity of gas produced, quantity of air required, and effectiveness of each heater.

From the above outputs of *boiler_solver* and *turbine_solver* programs the main power plant simulator, *powerplant_main*, is able to identify the performance of the plant as follows:

$$SHC = \frac{\lfloor m_s (h_{ss} - h_{fw}) \rfloor}{\left(P_{el} \times perlod \big/ 100\right)}, \quad [kJ/kW\text{-}s]$$

$$\eta_{powergen} = \frac{(P_{el} \times perlod / 100)}{B_c \times HHV}, \quad [\%]$$

$$SSC = \frac{3600}{ww \times 10^{-3}}, \quad [kg/kW\text{-}h]$$

where: ww = sum of power developed by each stage per kg of steam.

$$\eta_{cogeneration} = \frac{100(P_{el} \times (perlod/100)) + \dot{m}_{ph} h_{ph}}{B_c \times HHV}, [\%]$$

6.2 Power Plant System Verification

Simulation programs can be verified either with existing data or with hypothetical data to justify the fidelity of the model. In this work, a hypothetical data for a power plant of 20MW_e rated output is taken. Most of the data are taken from a similar power plant data and turbine manufacturers catalog

The main program contains many different subroutines or functions to be called by the program during simulation.

Before running the program all input data of the power plant should be given as input data file in the main program *powerplant_main.*

The total flow system of the power plant is sketched in **Fig.6.1** & **Fig.6.2** below and selected nodal points are numbered. These nodal points are representative points in the system to show the working fluid properties variations, especially at the inlet and outlet of components.

6.2.1 *Input Data File*

Combustion furnace

The combustion furnace is the fluidized bed type to which coal is crashed and fed without preheating.

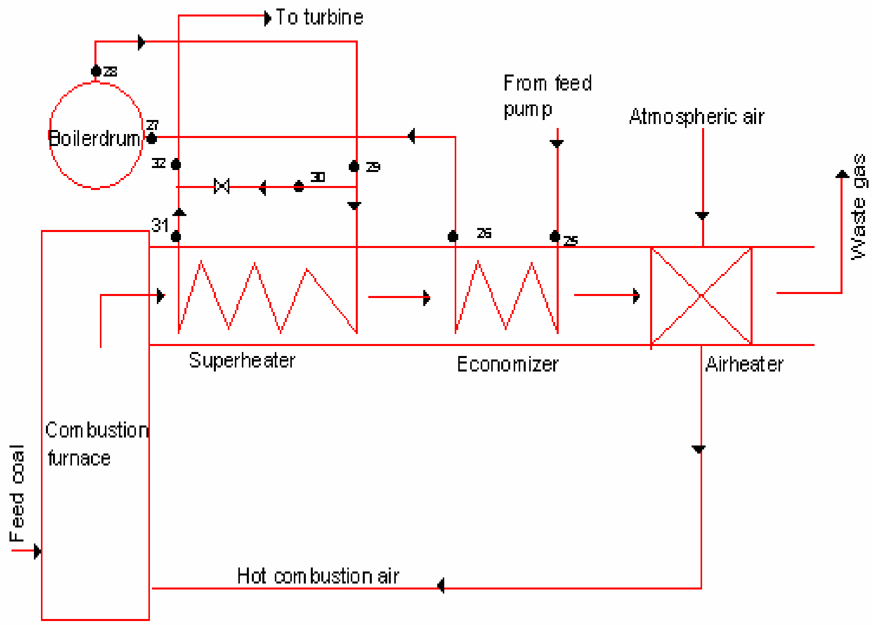

fig.6.1 Boiler system schematic diagram

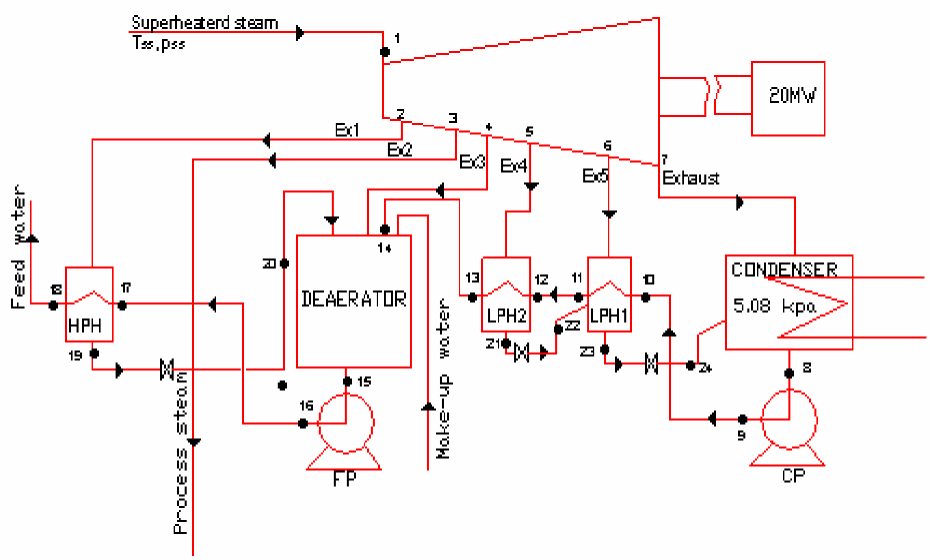

Fig.6.2 Schematic diagram of turbine system

The inside wall of the furnace is composed of N_t = 170 riser tubes with outside diameter d_o =0.0588m and wall thickness t_w = 0.003912m. The length of the tubes is taken to be L_t= 8m. Thermal conductivity of the tube wall is k_t = 50 W/m-k. These data of the furnace are stored in vector g_{vf}, $g_{vf} = [d_i, d_o, k_t, t_w, L_t, N_t]$.

Circulation water at the bottom of the riser is assumed to be saturated liquid at the working pressure and stability of the circulation is achieved at a circulation ratio of 6.

Convection Pass Heaters

Superheaters

This heater is composed of tube bundles with outside diameter d_{os} = 0.0267m and tube wall thickness t_{wt} = 0.00287m. The number of tubes in width N_{tc} = 36 and in the depth N_{tr} = 120. The length of one tube can be taken as 3m. The arrangement of the tubes are in-line and $S_1 = S_2 = S_{Ts}$ = 0.084.

The heat transfer area can be given by: $A_{hs} = N_{tr} \times N_{tc} \times \pi \times d_{os} \times L_t$

Gas flow area can be identified from: $A_{sg} = N_{tc} \times L_t \times (S_{Ts} - d_{os})$

Therefore, the above geometric data of the superheater are stored in vector g_{vs}:

$$g_{vs} = [d_{is}, d_{os}, A_{hs}, A_{sg}, N_{tr}, N_{tc}, L_t, S_1, S_2, S_{Ts}]$$

Economizer

The economizer is composed of tube bundles with outside diameter $d_{oe} = 0.02134m$ and wall thickness $t_w = 0.00277m$. The number of tubes in a row and column are 56 and 73, respectively. The tubes have in-line arrangement with $S_1 = S_2 = S_T = 0.075$. The length of one tube is about 3.4m. The heat transfer area and the gas flow area are identified as before in the superheater case. All the above data are stored in vector g_{ve}:

$$g_{ve} = [d_{ie}, d_{oe}, A_{he}, A_{eg}, N_{tr}, N_{tc}, L_t, S_1, S_2, S_T]$$

Air heater

Air heater consists of tubes of outside diameter 0.0334m and 0.0034m thick. The number of tubes in a row and column are given as 96 & 40, respectively. The combustion gas flows in the tubes side and combustion air outside the tubes in cross-flow mode with two-pass, crossing the flue gas twice. Overall length of one tube will have a size of 4.5m, and $S_T = 1.5d_o$ in an in-line arrangement. The heat transfer area is calculated as before, and the air flow area is given by: $A_a = N_{tr}(L_t/2)(S_T - d_o)$, and all data are stored in vector g_{va}:

$$g_{va} = [d_{ia}, d_{oa}, A_{ha}, A_a, N_{tr}, N_{tc}, L_t, S_T].$$

Turbine

The turbine considered here is of the extraction condensing type. The rated electrical power output is 20MW$_e$. Superheated steam temperature and pressure at the inlet of the turbine are 753K and 60bar, respectively. The internal efficiency of the turbine can be considered as 89% for the different loading. Exhaust pressure of the turbine is set as 0.0508bar. The pressure of automatic steam extraction for process heat demand is 10bar, and quantity of steam extracted depends on the demand of heat at the process.

The number of regenerative feed water heaters is four; two low-pressure heaters, one high-pressure heater and one open heater (deaerator). The extraction pressure is taken as $p_{ext} = [13.0, 5.0, 2.2, 0.635]$

Generator efficiency is assumed to be constant at different loading of the turbine and is taken as 95%.

Closed heaters

These heaters are of shell and tube heat exchangers with two tube-passes. Tube outside diameter is 0.0213m and 0.00277m thick. Tube length of LP heaters is 3m and of HP heater is 5m. The total number of tubes for the two heaters (LP & HP) is 80 & 100, respectively, and their shell diameter can be given by 0.393m for LP heaters & 0.426m for HP heaters.

The total designed closed heaters are three, as mentioned before. Between the heaters, there will be one gate-valve if it is necessary to make one of the heaters out of line and by-pass the flow, and at least two elbows. Minor losses of each element are taken to be the sum of all minor losses produced by the above devises that are available on the specific element, and so, the minor loss coefficients are inserted in a vector form for the system. Hence, for flow consideration of condensate with boundary conditions of condenser and deaerator pressure, the minor loss coefficients for the five available elements will be: K_{cond} = [1.67 0 1.67 0 1.67]. Similarly, feed water flow from deaerator to boiler drum would have minor loss coefficients of K_{feed} = [1.67 0 1.67 0 1.67]. Zeros in the vectors mean the element is a heat exchanger.

The condensate and feed pumps are taken as variable speed pumps to reduce losses due to throttling when the system is working at inferior loads. Hence, the total head developed by these pumps are functions of flow rate and speed. The characteristic curve of these pumps can be taken from manufacturers catalog and stored in matrixes as follows:

Condensate pump, $$T_{dc} = \begin{bmatrix} 0 & 0 & 25 & 47 \\ 3000 & 243.576 & 242.327 & 228.314 \\ 2500 & 169.15 & 166.808 & 151.8338 \\ 1700 & 78.2146 & 74.1254 & 57.6122 \end{bmatrix}$$

and, feed pump, $$T_{df} = \begin{bmatrix} 0 & 0 & 36 & 74 \\ 3000 & 1137.882 & 1135.3304 & 1066.1951 \\ 2700 & 921.664 & 916.32 & 844.2283 \\ 1700 & 365.3886 & 350.669 & 268.7 \end{bmatrix}$$

Condenser

The condenser consists of tubes of outside diameter 0.02134m with thickness 0.00277m. It is a shell- and tube design with two tube-passes. The length of one tube is 3m and its shell diameter is 2m. The baffle spacing and the tube pitch ratio are equal to 0.2m and 1.25, respectively. The thermal conductivity of the tube material is 54 *W/m-K*. The inlet temperature of the circulation water holds the atmospheric temperature, $T_{amb} = 20^0 C$, and has 7 ^{O}C TTD.

6.2.2 Simulation Output Discussion

The hypothetical power plant that has been taken for validation purpose is shown in **Fig.6.1** and **Fig.6.2**. The program *powerplant_main* simulates for rated load and for a process heat demand of 8715.9 kW_h. The outputs of the simulation at the different nodes and extraction points of the plant are tabulated below in **Table 6-1.** The Rankin cycle of the cogeneration is shown in Fig.6.3. On the cycle diagram, point 3 indicates the extraction of steam for process heat demand.

The power plant simulator is also tested by varying the quantity of extracted steam for process heat at a fixed pressure of 10 bars and also by varying the power loading of the turbine. The efficiency curves of boiler, power generation, and cogeneration of the plant are shown in **Fig.6.4** to **Fig.6.6.** The curves of **fig.6.4** show the change of the efficiency curves at different electric loading considering no steam extraction for process heat. In this case there is no cogeneration efficiency. The rest two efficiency curves are in relation to real power plant cases.

Fig.6.5 shows that cogeneration efficiency increases with the increase in power capacity of the plant at constant steam extraction for process heat. Increasing extraction heat for process, which means increasing heat to power ratio, shifting the cogeneration curve to have peak point at lower plant capacity. This is shown on

Table 6-1 ***Simulation results of a 20 MW_e and 8715.9 kW_h steam cogeneration power plant at rated load power output***.

Pos.	Steam/water Flow [t/h]	Temperature [K]	Pressure [bar]	Enthalpy kJ/kg	Steam quality
Node28	73.51	555.10	66.15	2778.1	1
Node29	73.51	552.50	63.59	2778.1	0.998
Node30	14.28	552.50	63.59	2778.1	0.998
Node31	59.23	800.66	63.00	3482.6	superheated
Node32	73.51	753.00	63.00	3370.0	superheated
Node1	73.51	751.0	60.00	3370.0	superheated
Node7	40.03	306.2	0.0508	2169.89	0.838
Node8	44.33	306.2	0.0508	138.41	-
Node9	44.33	306.2	5.27	138.41	-
Node10	44.33	306.2	5.25	138.41	-
Node11	44.33	358.28	5.01	356.47	-
Node12	44.33	358.28	4.99	356.47	-
Node13	44.33	393.16	4.76	503.76	-
Node14	44.33	423.07	4.75	631.81	-
Node15	77.18	423.07	4.75	631.81	-
Node16	77.18	423.07	66.86	636.12	-
Node17	77.18	423.07	66.84	636.12	-
Node18	77.18	462.67	66.27	807.52	-
Node25	77.18	460.75	66.26	807.52	-
Node26	77.18	555.1	66.16	1237.8	0
Node27	77.18	555.1	66.15	1237.8	0
Ext.1	4.88	582.71	13.00	3051.4	Superheated
Ext.2	10.80(proc. steam)	506.03	10.00	2905.3	Superheated
Ext.3	7.67	424.93	5.00	2747.4	0.9999
Ext.4	2.10	396.35	2.20	2583.0	0.9420
Ext.5	2.19	360.47	0.635	2384.0	0.881

Fig.6.6 when the plant is at 25% capacity and the extraction steam is 10.8 t/h. This maybe due to a significance reduction in heat loss at the condenser, because the steam that is expanding to the turbine exhaust will be reduced.

The model also tested by varying heat to power ratio at different power generating capacity. **Fig.6.7** shows the trend of the cogeneration efficiency at the different capacity. These curves help to identify optimum heat to power ratio of the cogeneration plant which gives higher efficiency, which means it helps to get the economic capacity of the cogeneration plant

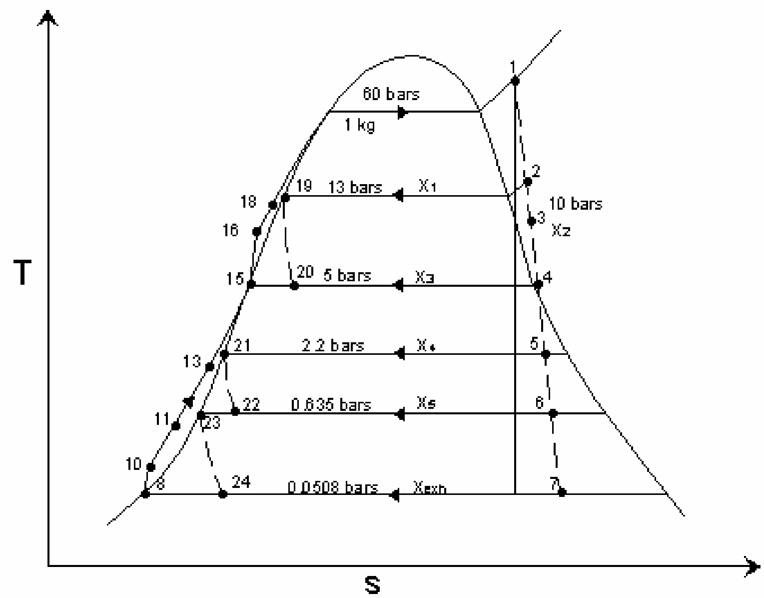

Fig. 6.3 The Rankin cycle of the cogeneration plant

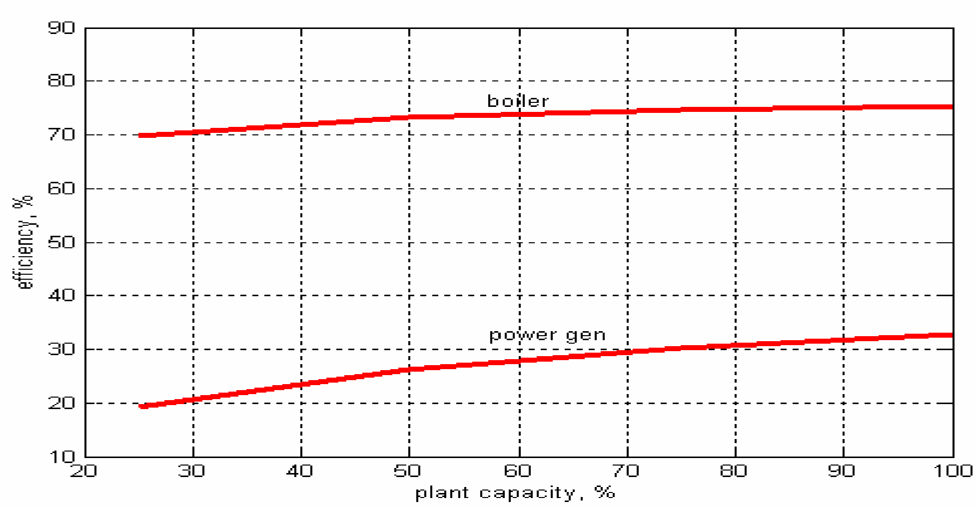

Fig.6.4 Efficiency curves without steam extraction for process heat

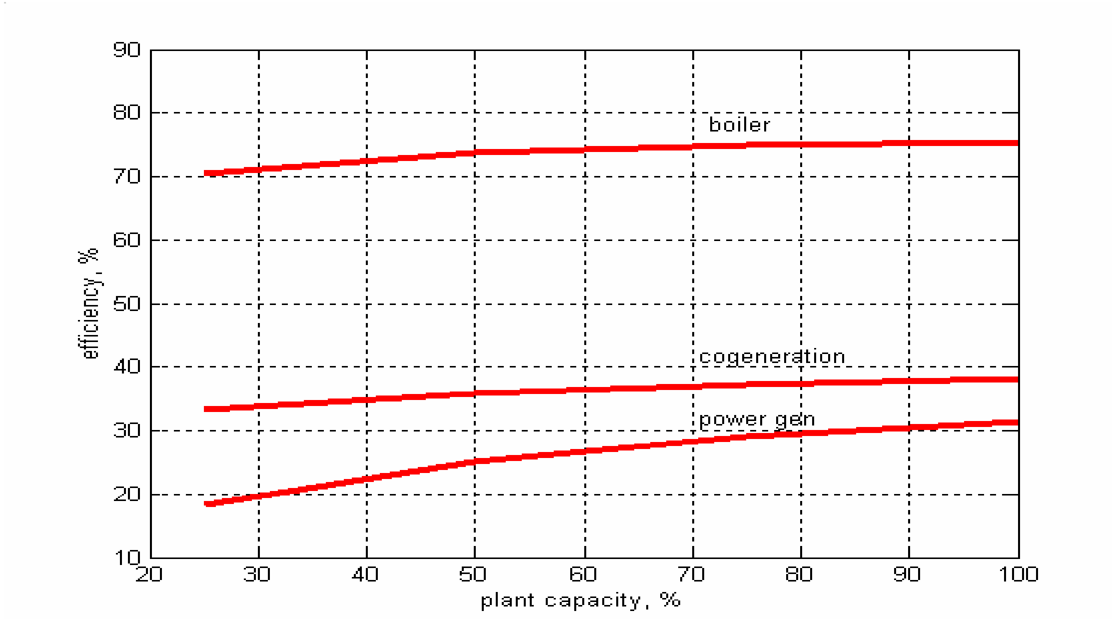

Fig.6.5 Efficiency curves of the plant with process steam extraction of 5.4t/h at 10 bars

Coal firing rate can have a linear relation with load or steam demand of the turbine. The consumption rate is graphed on **Fig.6.8** for different loading with out process heat extraction and with a process heat extraction of 10.8 t/h steam at 10 bars. The higher the coal consumption rate in curve-1 for a similar electric power output of curve-2 is that from the additional steam demand of the turbine due to the extraction of 10.8 t/h steam for process heat.

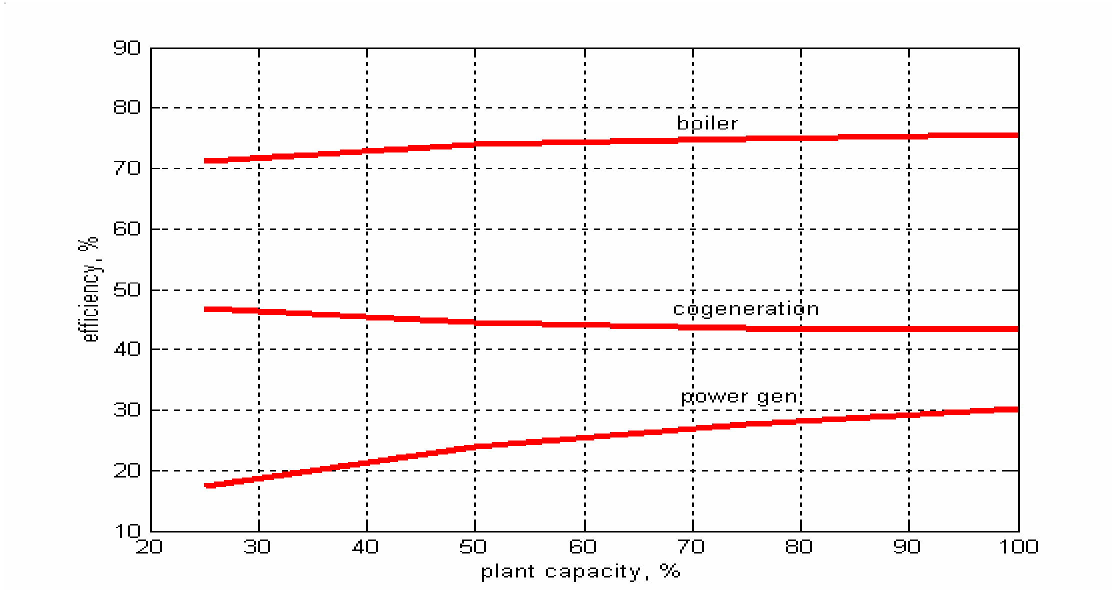

Fig.6.6   Efficiency curves of the plant with process steam extraction of 10.8t/h at 10 bars

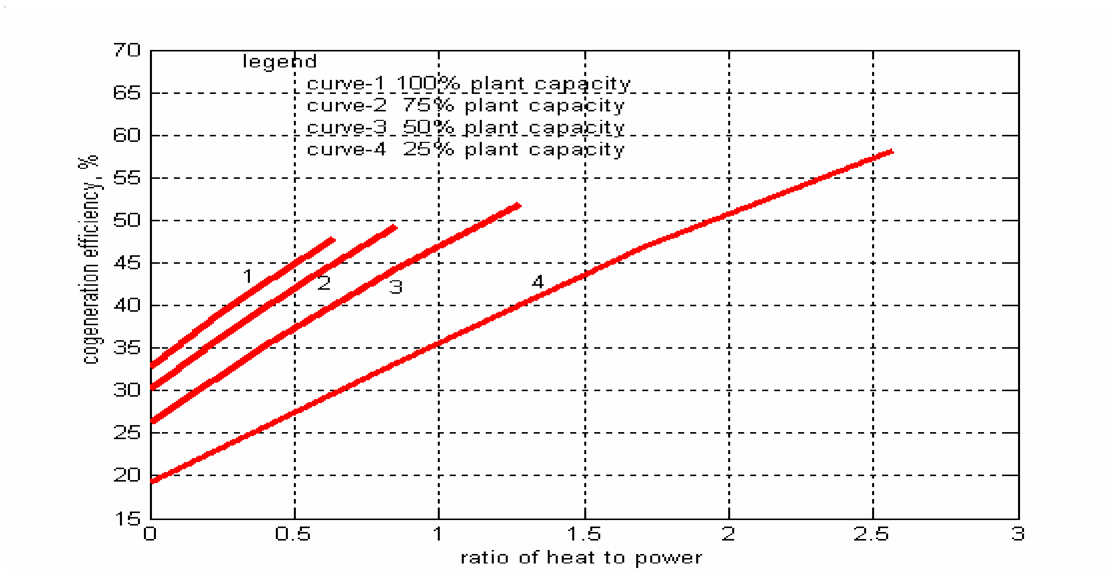

Fig.6-7 Cogeneration efficiency at different loading with varying process heat to power ratio

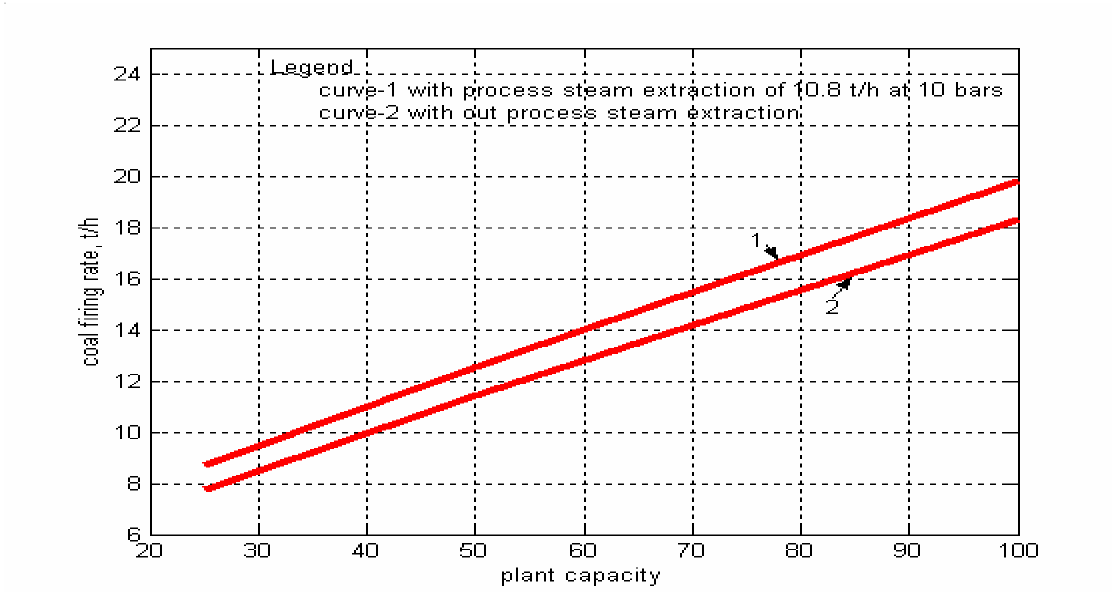

Fig.6-8 Plant capacity verses coal firing rate for different condition of the plant

The used Nusselt number correlations for various h values have their own error. The analysis has shown that the 1% change in Nusselt number applicable to economize, superheater and air preheater affects the boiler efficiency by 0.0066%, and so it is negligible

Conclusion and Recommendation

In this paper, computer program for a steam power plant simulation using MATLAB code has been developed. The program simulates the steady state operation of the plant. In the model there were lots of assumptions to avoid complexity. Some of them are:

- Coal combustion was taken to be complete, and there is no carbon monoxide in combustion gas and unburned carbon in the ash.
- The heat consumed for the decomposition of $CaCO_3$ to CaO was neglected
- Steam losses at turbine glands and valves were neglected
- Effects of steam throttling and exhaust losses at the turbine are not considered
- Calculation of plant efficiency was not consider power needed by the pumps
- Steam extraction pressure for process heat is limited to 10 bars
- Maximum quantity of steam extracted for process heat is 16.2 t/h

These are the limitations of the cogeneration plant modeled in this paper.

The model was intended to be verified by the power plant data of Coal-Phosphate Fertilizer Complex Project. Since the project is failed to be feasible due to financial problem, the simulation program of this work was verified using hypothetical data. The trends of the hypothetical plant output, as depicted in chapter 6, seem to be reasonable. But still more real power plant data are required to validate the model and the simulation program.

The paper can be extended further by considering those neglected losses, minimizing the assumptions and avoiding limitations to make the program robust and more predictive of the real case. It can also be used for power production cost analysis.

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Appendix-A

Table-A. Loss Coefficients for Fittings and Valves

Type of fitting or valve	*Loss coeff. K*
45^0 elbow, standard	0.350
90^0 elbow, standard	0.75
180^0 bend, close return	1.5
Tee, branching flow	1.0
Union	0.04
Gate valve, open	0.17
Diaphragm valve, open	2.3
Globe valve, bevel seat, open	6.4
Angle valve, open	3.0
Y or blow-off valve, open	3.0
Check valve, swing	2.0
Foot valve	15.0

Appendix-B

Table B. Values of the Two-Phase Frictional Multiplier ϕ^2 for the Martinelli-Nelson Model Steam-Water System

Steam Quality %by wt.	*Pressure, bar*								
	1.01	***6.89***	***34.4***	***68.9***	***103***	***138***	***172***	***207***	***221.2***
1	5.6	3.5	1.8	1.6	1.35	1.2	1.1	1.05	1.00
5	30	15	5.3	3.6	2.4	1.75	1.43	1.17	1.00
10	69	28	8.9	5.4	3.4	2.45	1.75	1.30	1.00
20	150	56	16.2	8.6	5.1	3.25	2.19	1.51	1.00
30	245	83	23.0	11.6	6.8	4.04	2.62	1.68	1.00
40	350	115	29.2	14.4	8.4	4.82	3.02	1.83	1.00
50	450	145	34.9	17.0	9.9	5.59	3.38	1.97	1.00
60	545	174	40.0	19.4	11.1	6.34	3.70	2.10	1.00
70	625	199	44.6	21.4	12.1	7.05	3.96	2.23	1.00
80	685	216	48.6	22.9	12.8	7.70	4.15	2.35	1.00
90	720	210	48.0	22.3	13.0	7.95	4.20	2.38	1.00
100	525	130	30.0	15.0	8.6	5.90	3.70	2.15	1.00

Appendix-C

Table-C Suppression Factor at Different Two-Phase Reynolds Number

Re_{TP}	S	Re_{TP}	S
$2*10^4$	0.8	$2*10^5$	0.25
$4*10^4$	0.7	$4*10^5$	0.15
$6*10^4$	0.6	$6*10^6$	0.11
$8*10^4$	0.5	$8*10^8$	0.1
$1*10^5$	0.4		

Appendix-D

Table-D Correction to Expansion Line End Point for Exhaust Pressure

P$_{exh}$ **[in.Hga]**	$\Delta ELEP_{Y=0}$ **[Btu/lb]**	**P**$_{exh}$ **[in.Hga]**	$\Delta ELEP_{Y=0}$ **[Btu/lb]**	**P**$_{exh}$ **[in.Hga]**	$\Delta ELEP_{Y=0}$ **[Btu/lb]**	**P**$_{exh}$ **[in.Hga]**	$\Delta ELEP_{Y=0}$ **[Btu/lb]**
0.5	-62.7	1.6	4.5	2.7	37.0	3.9	60.0
0.6	-52.3	1.7	8.0	2.8	39.4	4.0	62.0
0.7	-44.0	1.8	11.5	3.0	41.8	4.1	64.0
0.8	-36.5	1.9	15.0	3.1	44.0	4.2	65.9
0.9	-29.0	2.0	18.2	3.2	46.2	4.3	67.3
1.0	-22.8	2.1	21.2	3.3	48.5	4.4	69.0
1.1	-18.0	2.2	24.2	3.4	50.7	4.5	70.5
1.2	-13.0	2.3	27.0	3.5	52.9	4.6	72.0
1.3	-8.8	2.4	30.0	3.6	54.9	4.7	71.5
1.4	-4.0	2.5	32.0	3.7	56.7	4.8	75.0
1.5	0	2.6	34.2	3.8	58.4	4.9	76.5
						5.0	78.0

Appendix-E

Computer Programs

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                                                                 %
%        THIS IS MAIN SIMULATION PROGRAM, POWERPLANT_MAIN,        %
%          FOR COAL-FIRED COGENERATION POWER PLANT                %
%          DEPARTMENT OF MECHANICAL ENGINEERING                   %
%   BY YOHANNES TAMIRAT, DR-ING DEMISS ALEMU AND DR-ING ABEBAYEHU ASSEFA   %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

fid=fopen('powerplant11.text','w')
%
%
% INPUT DATA OF THE POWER PLANT
%   (1) TURBINE INPUT DATA
%
% THROTTLED STEAM CONDITION AND TURBINE EXHAUST PRESSURE. TEMPERATURE IN K, PRESSURE IN BAR
%
 pss=60;Tss=753;pexh=0.0508;
%
% PROCESS STEAM DEMAND, PRESSURE AND STEAM QUANTITY, IN BAR AND KG/S
%
   paut_ext=10;mstp=3;
%
% GENERATOR POWER OUT PUT, NUMBER OF EXTRACTION POINTS, NO. LOWPRESSURE HEATERS
% PERCENTAGE LOADING
%
 nlph=2;nhph=1;Pgen_rated=20e6;next=4;Gen_eff=95;perlod=100;
%
% CURVE FIT DATA OF TURBINE EFFICIENCY, STORED IN MATRIX CVD
%
   CVD=[0 150 350 550
       700 0.909 0.898 0.892
       850 0.924 0.915 0.908
       1000 0.934 0.926 0.923];
%
% CURVE FIT DATA OF PUMPS PERFORMANCE, STORED IN VECTOR Tdc AND Tdf,FOR CONDENSATE AND FEED
PUMPS
%
   Tdc=[0 0 25 47
       3000 243.576 242.327 228.314
       2500 169.15 166.808 151.8338
       1700 78.2146 74.12535 57.61215];
   Tdf=[0 0 36 74
       3000 1137.882 1135.3304 1066.1951
       2700 921.664 916.32 844.2283
       1700 365.3886 350.6695 268.7];
%
% HEATERS GEOMETRIC DATA STORD IN VECTORs
%
  do=0.02134;thw=0.00277; di=do-2*thw;Dslph=0.393;Dshph=0.426;B=0.2;ks=54;PR=1.25;CL=0.87;
  NP=2;Ltlph=3;Ntlph=80;Lthph=5;Nthph=100;
  gvlph=[di do Dslph B ks PR CL NP Ltlph Ntlph];
  gvhph=[di do Dshph B ks PR CL NP Lthph Nthph];
  GDlph=[di do NP Ltlph Ntlph];
  GDhph=[di do NP Lthph Nthph];
%
%
% CONDENSER DATA
%
  Twi=20+273;
  do=0.02134;tcw=0.00277;di=do-2*tcw;Ds=2;B=0.2;ks=54;PR=1.25;CL=1;NP=2;Lt=3;
  GDc=[di do Ds B ks PR CL NP Lt];
%
% LENGTH, DIAMETER OF PIPES B/N HEATERS,AND MINOR LOSS COEFFICIENTS,
```

```
%           CONDENSER TO DEAEARATOR
   Dic=[0.1 0.1 0.1];Lc=[1.5 1.5 1.5];Kc=[1.67 1.67 1.67];
%           DEAEARATOR TO BOILER DRUM
   Dif=[0.14 0.14 0.14];Lf=[1.5 5 3];Kf=[1.67 1.67 1.67];
%
%   (2) BOILER INPUT DATA; FURNACE, SUPER HEATERS AND ECONOMIZER
%  FUEL DATA
%  FUEL CONSTITUENR AS-RECIEVED BASE X=[C H S N O W A]
 X=100*[0.3 0.1 0.02 0.03 0.1 0.25 0.2];
 HHV=12e6;
% EXCESS AIR RATIO
 alpha=1.35;

  Tamb=20+273.15;
%
% FURNACE DATA
%
 dof=0.0588;tfw=0.003912;dif=dof-2*tfw;ktf=50;L=8;Nt=170;

 gvf=[dif dof ktf tfw L Nt];

%
% GEOMETRIC DATA OF SUPER HEATERS
%
 dos=0.0267;tsw=0.00287;dis=dos-2*tsw;Ntrs=120;Ntcs=36;Lts=3;STs=0.084;
 S1s=0.084;S2s=0.084;Npst=6;Ntohs=420;
 Asg=Ntcs*Lts*(STs-dos);
 Ahs=Ntrs*Ntcs*pi*dos*Lts;
 gvs=[dis dos Ahs Asg Ntrs Ntcs Lts S1s S2s STs];
 GDs=[dis dos Npst Lts Ntohs];
%
% PIPING DATA BETWEEN BOILER DRUM & SUPERHEATER, AND SUPERHEATER & TURBINE INLET
%
 Ks=[0.75 0.75];
 Ls=[2 5];
 Dis=[0.23 0.23];
% GEOMETRIC DATA OF ECONOMIZER

 doe=0.02134;tew=0.00277;die=doe-2*tew;Ntce=110;Ntre=73;Le=3.4;STe=0.075;
 S1e=0.084;S2e=0.084;Npet=13;Ntohe=616;
 Aeg=Ntce*Le*(STe-doe);
 Ahe=Ntre*Ntce*pi*doe*Le;
 gve=[die doe Ahe Aeg Ntre Ntce Le S1e S2e STe];
 Aeg=Ntce*Le*(STe-doe);
 GDe=[die doe Npet Le Ntohe];

% GEOMETRIC DATA OF AIR-HEATER

 doa=0.0334;taw=0.00338;dia=doa-2*taw;Ntra=96;Ntca=40;La=3;STa=1.5*doa;Npet=1;
 Aha=2200;
 Aa=Ntra*(La/2)*(STa-doa);
 gva=[dia doa Aha Aa Ntra Ntca La STa];

 gv=[gvf gvs gve gva];
%
% IF EXTRACTION PRESSURE FOR REGENERATIVE HEATING ARE GIVEN ind = 2, IF NOT ind = 1
%
 ind=2;
% CALL TURBINE SOLVER
%

[mext,mexh,ms,mcolw,Tfw,Hcp,Ncp,p_dea,vtout,vfc,Th]=turbine_solver(pss,Tss,pexh,nlph,Pgen_rated,CVD,Tdc,perlod,gvlph,gvhph,T
wi,GDc,Dic,Lc,Kc,paut_ext,Gen_eff,mstp,ind);
%
% CALL STEAM FLOW SOLVER TO GET DRUM PRESSURE
%
 vss=superheat_specific_volume(Tss,pss);
 gama_b2=9.81/vss;
 v_stm=flow_press_steam(pss*1e5,GDs,gama_b2,Dis,Ls,ms,Ks,1);
```

```
%
  pdrm=v_stm(1);pv_stm=v_stm*1e-5;
%
% CALL FEED WATER FLOW SOLVER
%
  GDff=[GDhph GDe];
%
% MAKE INITIAL PRESSURE GUESS BETWEEN DEAERATOR PRESSURE AND DRUM PRESSURE
%
  pdiffer=(pdrm*1.2-pdrm)/5;
  pguess=[pdrm*1.2 pdrm+4*pdiffer pdrm+3*pdiffer pdrm+2*pdiffer pdrm+pdiffer];
  vtp=[pguess 200 3000];
  nhtrs=2;

  vfedflow=flow_press_water(vtp,ms,pdrm,p_dea*1e5,GDff,Tdf,Dif,Lf,Kf,nhtrs);
%   vfedflow=flow_solver_feed(ms,p_dea,pdrm,Tdf,GD);
 pv_fw=1e-5*vfedflow(1:5);Hfp=vfedflow(6);Nfp=vfedflow(7);
 pecin=pv_fw(4);peco=pv_fw(5);
%
% CALL BOILER SOLVER
%
% RATED CAPACITY OF BOILER 75 t/hr
%
  mr=75/3.6;
 [eff,Bc,Twg,vbout]=boiler_solver(HHV,Tamb,gv,pv_stm,Tfw,pecin,peco,pss,Tss,ms,mr,X,alpha);
%
%===================== OUTPUTS==========================================
%
% TURBINE STEAM RATE, KG/KWh
%
mextf=mext./ms;
msf1=1-mextf(1);msf2=msf1-mextf(2);msf3=msf2-mextf(3);msf4=msf3-mextf(4);msf5=msf4-mextf(5);
ww=vtout(14)+(vtout(15)/(msf1))+(vtout(16)/(msf2))+(vtout(17)/(msf3))+(vtout(18)/(msf4))+(vtout(19)/(msf5));
steam_rate=1*3600/(ww*1e-3)
%
%  HEAT RATE, KJ/KWs
%
hss=superheat_enthalpy(Tss,pss);
[p,hffw,hfg,hg,sf,sfg,sg,vg,vf]=steam_t(Tfw);
heat_rate=(ms*(hss-hffw)-vtout(21)*mstp)*1e-3/(Pgen_rated*1e-3*perlod/100);
%
% STEAM FLOW RATE
msteam=3.6*ms;
%
% COAL COMBUSTION RATE (T/HR)
%
Ccomb=Bc*3.6;
%
% BOILER THERMAL EFFICIENCY
%
boiler_ther_eff=eff;
%
% TURBINE THERMAL EFFICIENCY
%
Tur_therm_eff=100/heat_rate;
%
% POWER PLANT CYCLE EFFICIENCY
%
over_all_eff=100*(Pgen_rated*(perlod/100)/(HHV*Bc-vtout(21)*mstp));
%
% COOLING WATER CONSUPTION RATIO, WITH A MAX. TEMP. RISE OF 15 DEG.CEL,
%
cool_water=mcolw;
% PROCESS HEAT-TO-POWER RATIO
h_p=(vtout(21)*mstp)/(Pgen_rated*(perlod/100));

fprintf(fid,'-----------------------------------------------------------------------------------\n')
fprintf(fid,'msteam[t/h]    Ccomb[t/h]    wat_cir_ratio   steam_rate[kg/kWh]   heat_rate[kJ/kWs]\n')
fprintf(fid,'-----------------------------------------------------------------------------------\n')
fprintf(fid,'%8.4f %14.4f %16.4f %14.4f %19.4f\n',msteam,Ccomb,mcolw,steam_rate,heat_rate)
```

```
fprintf(fid,'\n')
fprintf(fid,'\n')
fprintf(fid,'-----------------------------------------------------------------------------------\n')
fprintf(fid,'boiler_eff    Tur_therm_eff      over_all_eff      heat-to-power ratio\n')
fprintf(fid,'-----------------------------------------------------------------------------------\n')
fprintf(fid,'%8.4f %14.4f %20.4f %20.4f\n',eff,Tur_therm_eff,over_all_eff,h_p)
status=fclose(fid)



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%       THIS PROGRAM IS USED TO SIMULATE AN EXTRACTION CONDENSING TURBINE.           %
%       IT CONSISTS OF TURBINE MODULE, FEEDWATER HEATERS MODULE,                      %
%      CONDENSER MODULE AND CONDENSATE FLOW MODULE                                    %
%      DEPARTMENT OF MECHANICAL ENGINEERING                                           %
%      BY YOHANNES TAMIRAT, DR-ING DEMISS ALEMU AND DR-ING ABEBAYEHU ASSEFA           %
%                                                                                     %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

function [mext,mexh,ms,mcolw,Tfw,Hcp,Ncp,p_dea,v,vfc,Th]=turbine_solver(pss,Tss,pexh,nlph,Pgen_rated,CVD,Tdc,
perlod,GDlph,GDhph,Twi,GDc,Di,L,K,paut_ext,Gen_eff,mstp,x);

%
%         INITIALLY ASSUMED VALUES

% ASSUMED EXTRACTION STEAM FRACTION AT THE DIFFERENT POINTS
    mst=15; mprf=mstp/mst;mmw=mstp+0.05*mst;
    mext=[0.1 mprf 0.1 0.1 0.1];mexh=1-sum(mext);
    next=length(mext)-1;
%
% GENERATE INITIAL VALUE OF TURBINE WORK AT THE DIFFERENT STAGE AND
% EXTRACTED STEAM ENTHALPY AND STORE IN VECTOR vtt
%
    nhph=next-(nlph+1);
    nhph=next-(nlph+1);
    hss=superheat_enthalpy(Tss,pss);
    [Texh,hfex,hfgex,hgex,sgex,sfex,sfgex]=steam_p(pexh);
    hdiff=(hss-(hfex+0.7*hfgex))/(next+1);
    htril=[hss-hdiff,hss-2*hdiff,hss-3*hdiff,hss-4*hdiff, hss-5*hdiff];
    w1t=hss-htril(1);w2t=htril(1)-htril(2);w3t=htril(2)-htril(3);
    w4t=htril(3)-htril(4);
    w5t=htril(4)-htril(5);
    w6t=htril(5)-(hfex+0.6*hfgex);
    wtt=w1t+w2t+w3t+w4t+w5t+w6t;
    wttril=[wtt,w1t,w2t,w3t,w4t,w5t,w6t];
    Pgen=Pgen_rated*(perlod/100);
    vtt=[wttril htril mst mst*mexh];
%
% GENERATE INITIAL VALUE VECTOR FOR THE HEATERS
%
    vth1=[mst*mext(5) 363 0.5 mst*mext(4) 383 0.5];
    vth2=[mst*mext(1) 410 0.5 0.5 433 mst*mext(3)];
    vht=[vth1 vth2];;
    vct=[80 0.5];
%
% GENERATE INITIAL VALUE FOR THE CONDENSATE FLOW AND THE PUMP HEAD AND SPEED
%
      vp=1e5*[15.6 14.5 14 13.6 13.3];
      vtf= [vp 200 3000];

   GDf1=[GDlph(1) GDlph(2) GDlph(8) GDlph(9) GDlph(10)];
   GDf2=[GDlph(1) GDlph(2) GDlph(8) GDlph(9) GDlph(10)];
   GDf=[GDf1 GDf2];
%
% STORE TURBINE, LP AND HP HEATERS AND CONDENSER GUESSED VALUE IN VECTOR v_as
%
    v_as=[vht vtt vct];
%
% ADENTIFY THE EXTRACTION PRESSURE
```

```
%
switch x
case 1
Tbsat=sattable_givp_t(pss);
 Tc=sattable_givp_t(pexh);
 Tcoff=(Tbsat-Tc)/(next+1);
 Ttt=Tbsat;
 for ii=1:next
     Thtrsat(ii)=Ttt-Tcoff;
     pheatrs(ii)=sattable_givt_p(Thtrsat(ii));
     Ttt=Thtrsat(ii);
end
   pext=pheatrs/0.95;
case 2
  pext=[13 5 2.2 0.635];
  pheatrs=pext*0.95;
end
  p_dea=pheatrs(next-nlph);
  hext=htril;
  hvar=hext;
  nheaters=next;
  nvar=length(v_as);
%
%    START ITERATION OF THE TURBINE SYSTEM
%
iter=1;
while iter>0
%
% CALL THE CONDENSATE FLOW SOLVER
%
   mc=mst*mexh+mst*sum(mext(next:next+1));
%
  vfc=flow_press_water(vtf,mc,p_dea*1e5,pexh*1e5,GDf,Tdc,Di,L,K,nlph);
%
%
% CALL THE RESIDUE VECTOR GENERATOR FUNCTION OF THE TURBINE-HEATRS SYSTEM
%

[R,Th]=turbine_residue_ass(v_as,nlph,nhph,mst,mext,mexh,hext,pext,pexh,GDlph,GDhph,GDc,Twi,pss,Tss,CVD,Pgen,vfc,paut_ext,Ge
n_eff,mmw);
%
%
% CALL THE JACOBIAN MATRIX GENERATOR FUNCTION OF THE TURBINE-HEATRS SYSTEM
%

pd=turbine_jacob(nvar,R,v_as,nlph,nhph,mst,mext,mexh,hext,pext,pexh,GDlph,GDhph,GDc,Twi,pss,Tss,CVD,Pgen,vfc,paut_ext,Gen_e
ff,mmw);
%
% CALL THE GAUSSIAN ELEMINATION FUNCTION TO SOLVE THE SYSTEM OF EQUETIONS
%
   vcorr=GAUSS(nvar,pd,R);
%
% CORRECT THE GUESSED VALUES OF THE VERIABLES
%
   vn=v_as-(vcorr/2);
%
% TEST FOR CONVERGENCE
%

if all((abs(vn-v_as)/v_as )<0.01)
    iter=0;
    v=v_as;
    Tfw=v(11);
    mext=[v_as(7) mstp v_as(12) v_as(4) v_as(1)];
    mexh=[v_as(26)];
    mcolw=v_as(27);
    ms=v_as(25);
    Hcp=vfc(6);
    Ncp=vfc(7);
    hautext=v_as(21);
```

```
else
    v_as=vn;

    mst=v_as(25);
    mext=(1/mst)*[v_as(7) mstp v_as(12) v_as(4) v_as(1)];
    mexh=[v_as(26)/mst];

    iter=iter+1;
end
end
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%     THIS PROGRAM SIMULATES THE FLUIDIZED BED BOILER PROBLEM WITH ALL HEATERS              %
%                                                                                                                                     %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function [eff,Bc,Twg,v]=boiler_solver(HHV,Tamb,gv,pv_stm,Tfw,pecin,peco,pss,Tss,ms,mr,X,alpha);
%
vtf=[1100 700 600 ];vts1=[753 950 0.5 ms-2 2];vte=[2000e3 700 0.5];vta=[500 600 0.5];
vtdrm=[0.12];vteff=[76 6];
vt=[vtf vts1 vte vta vtdrm vteff];
%
[mac,mgc,mwc,mdgc,r_Ro2,r_H2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
%
nvar=length(vt);
iter =1;
while iter > 0
%
  [RR]=boiler_residu_ass(vt,ms,mr,Tfw,Tss,HHV,Tamb,gv,pv_stm,pecin,peco,pss,X,alpha);
%
  pd=jacob_boiler(nvar,RR,vt,ms,mr,Tfw,Tss,HHV,Tamb,gv,pv_stm,pecin,peco,pss,X,alpha);
%
%   CALL THE GAUSSIAN ELEMINATION SUBROUTINE
%
  vcorr=GAUSS(nvar,pd,RR);
  vn=vt-(vcorr/2);
%
 if vn(4)<Tss
     vn(8)=0;
     vn(7)=ms;
     vn(4)=Tss;
  end
%
if all((abs(vn-vt)/vt )<0.001)
    iter=0;
    v=vn;
    eff=v(16);Bc=v(17);Twg=v(13);
else
    iter=iter+1;
    vt=vn;
end
end
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   THIS SUBROUTINE IS USED TO CALL COMPONENT RESIDUE GENERATOR FUNCTIONS OF INDIVIDUAL %
%    TURBINE SYSTEM COMPONENTS AND MAKE THE ASSEMBLY OF THE SYSTEM RESIDUES                    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function
[ff,Th]=turbine_residue_ass(v,nlph,nhph,mst,mext,mexh,hext,pext,pcond,GDlph,GDhph,GDc,Twi,pss,Tss,CVD,Pgen,vfc,paut_ext,Gen_
eff,mmw);
%
 nvar=length(v);
```

```
 next=length(mext)-1;
 vh=v(1:12);vt=v(13:nvar-2);vc=v(nvar-1:nvar);
 nvart=length(vt);
 nvarh=length(vh);
 nvarc=length(vc);
 Tbsat=sattable_givp_t(pss);
 Tc=sattable_givp_t(pcond);
 Tcoff=(Tbsat-Tc)/(next+1);
 Ttt=Tbsat;
 mprf=mext(2);
 Tamb=Twi;
%
  pheatrs=pext*.95;
%
% CALL THE TURBINE RESIDUE GENERATORE FUNCTION
%
  [Rt,hexh]=turbine_new(vt,next,mext,Tss,pss,pcond,CVD,pext,nlph,Pgen,paut_ext,Gen_eff);
  hext=[hext(1),hext(3),hext(4),hext(5)];
  hexh=hexh;

 nvarh=length(vh);
 vhh=vh(1:nvarh-1);
 vde=vh(nvarh);
 nvarhh=length(vhh);
 g=9.81;
 hvar=hext;
 nheaters=next;
 mc=mst*mexh+mst*sum(mext(nhph+3:next+1));
 %
 % SATURATION TEMPERATURE AT THE HEATERS' SHELL PRESSURE
 %
for i=1:nheaters
   j=(nheaters+1)-i;
   ps(i)=pext(j);

   hext(i)=hvar(j);
end
mexthtr=[mext(1),mext(3),mext(4),mext(5)];
for i=1:nheaters
   j=(nheaters+1)-i;
   Ts_htrs(i)=sattable_p_t(ps(i)*0.95);
   ms(i)=mst*mexthtr(j);
   hdc(i)=satsteam_hf_givp(ps(i)*0.95);
   Th(i)=steam_temp_p_h(ps(i),hext(i));
end

mdc_hph=mst*sum(mext(1:nhph));
hh=hext(nlph+1);
[Texh,hfex,hfgex,hgex,sgex,sfex,sfgex,vfex]=steam_p(pcond);
Tsat_dea=sattable_givp_t(pheatrs(next-nlph));
Tdc_h=sattable_givp_t(pheatrs(next-(nlph+1)));
rho=1/vfex;
rho_g=(1/vfex)*g;
p_dea=pheatrs(next-nlph)*1e5;
mfw=mst+0.05*mst;
[Ts,hf,hfg,hg,sg,sf,sfg,vfw]=steam_p(p_dea*1e-5);
Qf=mfw*3600/(1/vfw);
Q=mc*3600/(1/vfex);
p1=pcond*1e5;
p2=p_dea;
Tcin=vh(2);
p=vfc(1:5);
ptubin=[p(2) p(4) 1.25*pss*1e5];
Tcin=Tsat_dea;
p1=p_dea;
Tc1=vh(nlph*3-1);
hfw=sattable_t_hf(Tsat_dea);
hc1=sattable_t_hf(Tc1);
hdc_hph=sattable_t_hf(Tdc_h);
p_cond=pcond*1e5;
```

```
nlph=nlph;
nhph=nhph;
mc=mc;
flowrate=mc*3600/(1/vfex);
%
% MASS BALANCE TO GET THE DRAIN FROM EACH HEATERS
%
mpdchp=0.0;
for i=nheaters:-1:nlph+2
   mdc(i)=mpdchp+ms(i);
   mpdchp=mdc(i);
end
mpdclp=0.0;
for i=nlph:-1:1
   mdc(i)=mpdclp+ms(i);
   mpdclp=mdc(i);
end
mdc(nlph+1)=mfw;
mdc=mdc;

mdcr=0.0;
for jj=nheaters:-1:nlph+2
   mdcthph(jj)=mdcr;
   mdcr=mdc(jj);
end
mdcr=0.0;
for kk=nlph:-1:1
   mdctlph(kk)=mdcr;
   mdcr=mdc(kk);
end
%
% CALL THE RESIDUE VECTOR OF CONDENSER
%
 Rc=condenser_residue(vc,Twi,mexh*mst,hexh,pcond,GDc,mdc(1),hdc(1));
%
% GENERATE RESIDUES FROM THE HEATERS AND ASSEMBLED INTO A SINGLE VECTOR
%

nvph=3;          % number of varibles per heater
ff=zeros(nvar,1);         % initialization of system residue vector
ind1=zeros(nlph,1);  % initialization of index vector
ind2=zeros(nhph,1);
Tcin=Tc;
pt=ptubin*1e-5;

for lph=1:nlph
   Tcout=vhh(2+(lph-1)*3);
   Ttb=(Tcin+Tcout)/2;
   lphs=(nheaters+1)-lph;
   ind1=ind_fun(lph,nvph,1);
   vth=vhh(ind1);
   R3=fheatertype1(vth,Tcin,mc,Th(lph),hext(lph),ps(lph),pt(lph),GDlph,mdctlph(lph),hdc(lph),1);
   ff=resid_ass_fun(ff,R3,ind1);
   Tcin=Tcout;

end

 Rdea=deaerator(vde,Tcin,Ts_htrs(nlph+1),Ts_htrs(nlph+2),mdc(nlph+2),hext(nlph+1),mfw,mst,next,mext,nhph,mexh,mmw,Tamb);
 ptin=1.25*pss;
 Tcin=Ts_htrs(nlph+1);

for hph=nlph+1:1:nheaters-1
   hphi=hph+2;
   Tcout=vhh(5+(hph-2)*3);
   Ttb=(Tcin+Tcout)/2;
   pthp=ptin*1e-5;

     if hph==nheaters-1
        factor=2;
        ind2=ind_fun(hph,nvph,2);
```

```
        vth=vhh(ind2);
      else
        factor=1;
        ind2=ind_fun(hph,nvph,1);
        vth=vhh(ind2);
      end

     R3=fheatertype1(vth,Tcin,mfw,Th(hph+1),hext(hph+1),ps(hph+1),pt(hph),GDhph,mdcthph(hph+1),hdc(hph+1),factor);

    ff=resid_ass_fun(ff,R3,ind2);
    Tcin=Tcout;
 end

ff(3*(nlph+nhph)+3)=Rdea;
ff(3*(nlph+nhph)+4:nvar-2)=Rt;
ff(nvar-1:nvar)=Rc;




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%         THIS FUNCTION GENERATES THE RESIDUE VECTOR FOR A TURBINE                        %
%         THE TURBINE MODELED HERE IS AN EXTRACTION CONDENSING ONE                        %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

function [R1,hexhac]=turbine_new(v,next,mext,Tss,pss,p_lpexh,CVD,pext,nlph,Pgen,paut_ext,Gen_eff);

g=9.81;
Pmech=0.4e6;
nvar=length(v);
nhph=next-(nlph+1);
hss=superheat_enthalpy(Tss,pss);
sss=superheat_enthropy(Tss,pss);
%
% GET TURBINE EXPANSION EFFICIENCY FROM CURVE FITTING
%


pp=inv([1 CVD(1,2) CVD(1,2)^2
       1 CVD(1,3) CVD(1,3)^2
       1 CVD(1,4) CVD(1,4)^2]);
TT=inv([1 CVD(2,1) CVD(2,1)^2
       1 CVD(3,1) CVD(3,1)^2
       1 CVD(4,1) CVD(4,1)^2]);
A1=pp*CVD(2,2:4)';
A2=pp*CVD(3,2:4)';
A3=pp*CVD(4,2:4)';

B0=TT*[A1(1);A2(1);A3(1)];
B1=TT*[A1(2);A2(2);A3(2)];
B2=TT*[A1(3);A2(3);A3(3)];
%
T=(9/5)*(Tss-273.15)+32;

P=pss/(6894.8*1e-5);
etha=(B0(1)+B0(2)*T+B0(3)*T^2)+(B1(1)+B1(2)*T+B1(3)*T^2)*P+(B2(1)+B2(2)*T+B2(3)*T^2)*P^2;
%
%BY ASSUMING THE LP TURBINE EXHAUST PRESSURE TO BE 1.5 IN.Hga ONE CAN IDENTIFY THE ELEP ENTHALPY
%
[Texh,hfex,hfgex,hgex,sgex,sfex,sfgex,vfex]=steam_p(0.0508);
%
%FOR AN ISCENTROPIC EXPASION THE ENTHALPY AT THE EXHAUST OF THE LP TURBINE WILL BE
%
xr=(sss-sfex)/sfgex;
hexis=hfex+xr*hfgex;
hexhre=hss-etha*(hss-hexis);
X=(hexhre-hfex)/hfgex ;
Y=(1-X)*100;
hend=end_line_exp_point_corr(p_lpexh);
delELEP_y=hend*0.87*(1-0.01*Y)*(1-0.0065*Y);
```

```
%
% ELEP AT ACTUAL EXHAUST PRESSURE
%
hexhac=hexhre+delELEP_y;
hc=hexhac;
%
[Texh,hfc,hfgex,hgex,sgex,sfex,sfgex,vfex]=steam_p(p_lpexh);
xa=(hexhac-hfex)/hfgex;
%
sexh=sfex+xa*sfgex;
%
%TO GET THE EXPANSION LINE EFFICIENCY WITH THE ACTUAL EXHAUST PRESSURE
xis=(sss-sfex)/sfgex;
hexis=hfex+xis*hfgex;
hext=v(next+4:nvar-2);
%
pext1=[pext(1) paut_ext pext(2:next)];
hvt=[hss hext(1:next)];
pp=[pss pext1(1:next)];
%
% IDENTIFY THE ISCENTROPIC EXPANSION ENTHALPY AT EACH EXTRACTION POINT
%
sx_as=sss;
for i=1:next+1
   hextis(i)=superheat_p_s(pext1(i),sx_as);
end

 hisc=hextis;
%
% DEFINE THE VARIABLES IN THE VECTOR v
%
wt=v(1);w=v(2:next+3);h=v(next+4:nvar-2);mst=v(nvar-1);mexh=v(nvar);
%
% IDENTIFY STEAM FLOW FRACTION AFTER EACH EXTRACTION
%
mxy=0.0;
for i=1:next+1
   mtf1(i)=mext(i)+mxy;
   mxy=mtf1(i);
   mtf2(i)=1-mtf1(i);
end
mtf=[1 mtf2];
%
% GENERATING THE RESIDUAL VECTOR, R
%
hin=hss;
for ii=1:next+2
   if ii==next+2
      R1(ii)=w(ii)-mtf(ii)*(hin-hc);
   else
      R1(ii)=w(ii)-mtf(ii)*(hin-h(ii));
      hin=h(ii);
   end
end

hin=hss;
for jj=next+3:nvar
   k=jj-(next+2);
   if jj==nvar-2
      R1(jj)=wt-sum(w);
   elseif jj==nvar-1
      R1(jj)=mst-(Pgen/(Gen_eff/100))/wt;
   elseif jj==nvar
      R1(jj)=mexh-mst*(1-sum(mext));
   else
     R1(jj)=w(k)-(etha_rh_ac*mtf(k)*(hin-hisc(k)));
     hin=h(k);
  end
end
R1=R1;
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                  FEED WATER HEATER RESIDUE GENERATOR FUNCTION                  %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function R=fheatertype1(v,Tcin,mc,Th,h,ps,pt,GD,mdc,hdc,factor);
%
[Tsat,hf,hfg,hg,sg,sf,sfg]=steam_p(ps);
mh=v(1);Tcout=v(2);eps=v(3);

di=GD(1);do=GD(2); Ds=GD(3);B=GD(4);ks=GD(5);PR=GD(6);CL=GD(7);NP=GD(8);
Lt=GD(9);Nt=GD(10);
Ao=NP*pi*do*Lt*Nt;
Ai=NP*pi*di*Lt*Nt;
PT=do*PR;
Tcb=(Tcout+Tcin)/2;
%
%IDENTIFING CONVECTIVE HEATRANSFER COEFFICIENT FOR BOTH FLOW
%
hi=hi_tube_side_fwheater(Tcb,mc,di);
ho=ho_condenser(hi,Tsat,Tcout,do,Nt,Tcb,Th);

Rs=do*log(do/di)/(2*ks);
Ri=do/(hi*di);
Ro=1/ho;
%
%THE OVERALL HEAT TRANSFER COEFFICIENT, Uo, W/m^2.C
%
Uo_Ao=Ao/(Ri+Rs+Ro);
%
% GET SPECIFIC HEAT OF BOTH FLUID AT MEAN TEMPERATURE
%
Cpc=cp_steam_and_water(Tcb,pt);

if Th>Tsat
  Thb=(Th+Tsat)/2;
  Cph=cp_steam_and_water(Thb,ps);
end

Cc=mc*Cpc;
Cmin=Cc;
Ao=NP*pi*do*Nt*Lt;
NTU=Uo_Ao/Cmin;

%
% IF THE HEATER HAS DESUPERHEATER
%
if factor==2
   epsd=v(4);Tfw=v(5);
   Thb=(Th+Tsat)/2;
  Cphd=cp_steam_and_water(Thb,ps);
  Ccd=mc*Cpc;
  Chd=mh*Cphd;
  Cmind=min([Ccd Chd]);
  Cmaxd=max([Ccd Chd]);
  Cd=Cmind/Cmaxd;
  Aod=(1/4)*Ao;
  Aid=(1/4)*Ai;

  Pt=do*1.25;Ds=0.5*Ds;Lt=0.25*Lt;
  HS= [Ds B CL Pt];CTP=0.93;
  Ntd=0.875*(CTP/CL)*(Ds^2)/(PR^2*do^2);
  hod=ho_desuperheater(HS,do,Tcout,Tfw,Th,Tsat,ps,mh);
  Rs=log(do/di)/(2*pi*ks*Lt*Ntd);
  Ri=1/(hi*pi*di*Ntd*Lt);
  Ro=1/(ho*pi*do*Ntd*Lt);
  Uo_Aod=1/(Ri+Rs+Ro);
```

```
   NTUd=Uo_Aod/Cmind;

end

switch factor
case 1
if Th > Tsat
   R(1)=eps*Cmin*(Tsat-Tcin)-(mh*Cph*(Th-Tsat)+mh*hfg+mdc*hdc);
else
    xvap=(h-hf)/hfg;
   R(1)=eps*Cmin*(Tsat-Tcin)-(mh*xvap*hfg+mdc*hdc);
end

   R(2)=eps-2*(1+((1+exp(-NTU))/(1-exp(-NTU))))^(-1);
   R(3)=eps*Cmin*(Tsat-Tcin)-Cc*(Tcout-Tcin);
otherwise

   R(1)=epsd*Cmind*(Th-Tcout)-Ccd*(Tfw-Tcout);
   R(2)=epsd-((1-exp(-NTUd*(1-Cd)))/(1-Cd*exp(-NTUd*(1-Cd))));
   R(3)=eps*mc*Cpc*(Tsat-Tcin)-mh*hfg;
   R(4)=mc*Cpc*(Tcout-Tcin)-mh*hfg;
   R(5)=eps-2*(1+((1+exp(-NTU))/(1-exp(-NTU))))^(-1);
end




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                    CONDENSER RESIDUE GENERATOR SUBROUTINE                    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function R=condenser_residue(v,Twin,mexh,hexh,pcon,GD,mdc,hdc);
%
[Tsat,hf,hfg,hg,sg,sf,sfg]=steam_p(pcon);
Two=Tsat-5;
mw=v(1);eps=v(2);
%
%GIVEN HEAT EXCHANGER GEOMETRICAL DATA
%
di=GD(1);do=GD(2); Ds=GD(3);B=GD(4);ks=GD(5);PR=GD(6);CL=GD(7);NP=GD(8);
Lt=GD(9);
Nc=24;
if NP==1
    CTP=0.93;
elseif NP==2
    CTP=0.9;
end
Nt=0.875*(CTP/CL)*(Ds^2)/(PR^2*do^2);

Aco=pi*do*Lt*Nt;
Aci=pi*di*Lt*Nt;PT=do*PR;

 Ttb=(Two+Twin)/2;
%
%IDENTIFING CONVECTIVE HEATRANSFER COEFFICIENT FOR BOTH FLOW
%
[Cp,meu,k,rho,Pr]=water_properties(Ttb);
%
hi=0.02785*(mw^0.8)*(Cp^0.4)*(k^0.6)/(meu^0.4*di);
Cpw=Cp;
Th=Tsat;
%
ho=ho_condenser(hi,Tsat,Two,do,Nt,Ttb,Th);
%
[p,hf,hfg,hg,sf,sfg,sg,vg]=steam_t(Tsat);
[Cp,muf,kf,rhof,Pr]=water_properties(Tsat);
rhog=1/vg;
rhof=rhof;
g=9.81;
%FOR UNIT LENGTH OF THE PIPE, THE THERMAL RESISTANCE OF THE PIPE MATERIAL, Rs
```

```
%THERMAL RESISTANCE OF THE INSIDE FLUID,Ri, AND THERMAL RESISTANCE OF THE OUT SIDE FLUID,Ro
Rs=log(do/di)/(2*pi*ks*Lt*Nt);
Ri=1/(hi*pi*di*Nt*Lt);
Ro=1/(ho*pi*do*Nt*Lt);
%
%THE OVERALL HEAT TRANSFER COEFFICIENT, Uo, W/m^2.C
%
Uo_Ao=1/(Ri+Rs+Ro);
Cmin=Cpw*mw*(mdc+mexh);
Ao=pi*do*Nt*Lt;
NTU=Uo_Ao/Cmin;
qc=mdc*(hdc-hf)+mexh*(hexh-hf);
%
% THE RESIDUE WILL BE GIVEN BY
%
  R(1)=(qc/(mdc+mexh))-eps*mw*Cpw*(Tsat-Twin);
  R(2)=eps-2*(1+((1+exp(-NTU))/(1-exp(-NTU))))^(-1);




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                    RESIDUE GENERATOR FUNCTION FOR DEAERATOR                    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function f=deaerator(v,Tc,Tsat,Tdc,mdc,hs,mf,mst,next,mext,nhph,mexh,mmw,Tamb);
%
hf=sattable_t_hf(Tsat);
hmw=sattable_t_hf(Tamb);
hc=sattable_t_hf(Tc);
hdc=sattable_t_hf(Tdc);
mext1=mext(1);mext2=mext(2);mext3=mext(3);mext4=mext(4);mext5=mext(5);
%
ms=v(1);

f(1)=mf*hf-(ms*hs+mdc*hdc+((mext4+mext5)*mst+mst*mexh)*hc+mmw*hmw);



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%  THIS SUBROUTINE IS USED TO IDENTIFY THE JACOBEAN OF THE TURBINE SYSTEM                %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

function   pd=turbine_jacob(nvar,R,v,nlph,nhph,mst,mext,mexh,hext,pext,pexh,GDlph,GDhph,GDc,Twi,pss,Tss,CVD,Pgen,vfc,
paut_ext,Gen_eff,mmw);
%
dv=0.01;
for k=1:nvar
   vd(k)=v(k);
end

for j=1:nvar
   if abs(v(j))<1e-30
      vd(j)=v(j)+0.001;
   else
      vd(j)=(1+dv)*v(j);
   end

[Rd,Th]=turbine_residue_ass(vd,nlph,nhph,mst,mext,mexh,hext,pext,pexh,GDlph,GDhph,GDc,Twi,pss,Tss,CVD,Pgen,vfc,paut_ext,Gen
_eff,mmw);
   for i=1:nvar
      if abs(v(j))<1e-30
         pd(i,j)=(Rd(i)-R(i))/0.001;
      else
         pd(i,j)=(Rd(i)-R(i))/(v(j)*dv);
      end

   end
   vd(j)=v(j);
end
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%        THIS IS A SUBROUTINE PROGRAM WHICH CALLS INDIVIDUAL BOILER COMPONENTS          %
%         RESIDUE GENERATOR FUNCTION AND MAKE THE ASSEMBLY OF THE RESIDUES              %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function [RR]=boiler_residu_ass(vt,mst,mr,Tfw,Tss,HHV,Tamb,gv,pv_stm,pecin,peco,pss,X,alpha);

gvf=gv(1:6);
gvs=gv(7:16);
gve=gv(17:26);
gva=gv(27:34);
pbd=pv_stm(1);ps1in=pv_stm(2);ps1ex=pv_stm(3);
dif=gvf(1);dof=gvf(2);Afw=gvf(3);kt=gvf(4);tw=gvf(5);Ntf=200;
vtf=vt(1:3);
vts1=vt(4:8);
%
vte=vt(9:11);vta=vt(12:14);vtdrm=vt(15);v_eff=vt(16:17);xf=vtdrm;
nvar=length(vt);nvarf=length(vtf);nvarsh1=length(vts1);nvare=length(vte);nvara=length(vta);
[Tsat,hsw,hfg,hg,sg,sf,sfg]=steam_p(pbd);
%
Tgis1=vtf(1);Tgis2=vts1(2);Tgie=vts1(2);Tgia=vte(2);
Tin1=Tsat;
Tca=vta(1);Twg=vta(2);Bc=v_eff(2);
[mac,mgc,mwc,mdgc,r_Ro2,r_H2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
%
mg=Bc*mgc;
hss=superheat_enthalpy(Tss,pss);
hfw=comp_water_h(pecin,Tfw);
[Tsat,hsw,hfg,hg,sg,sf,sfg]=steam_p(pbd);
%
mbd=0.05*mst;
mwe=mst+mbd;
Tgb=(Tgia+Twg)/2;
Tab=(Tamb+Tca)/2;
Tf=(3*Tgb+Tab)/4;
[Cpa,ka,meua]=comb_air_properties(Tf);
LHV=HHV-25e3*(9*X(2)+X(6));
Hav=LHV+mac*Cpa*(Tca-Tamb);
%
Reff=boiler_eff(v_eff,Twg,Tca,HHV,vts1(4),vts1(5),mr,X,alpha,hss,hfw,hg,mbd,hsw);
%
ma=Bc*mac;
%
RR=zeros(1,nvar);
Rf=furnace_exit_temp(vtf,Bc,mg,pbd,Tca,mst,HHV,Tamb,dif,dof,gvf,kt,tw,Cpa,Tsat,Ntf,xf,X,alpha);
%
Rs1=super_heater_module1(vts1,mst,Tgis1,Tin1,Tss,pss,ps1in,ps1ex,X,alpha,gvs,Bc,pbd);
%
[Re,xeco]=economizer_module(vte,mwe,mg,Tgie,Tfw,pecin,peco,X,alpha,gve);
%
Ra=air_heater_module(vta,ma,mg,Tgia,Tamb,X,alpha,gva);
%
% MATERIAL BALANCE OVER THE DRUM
%
mcirc=6*mst;
Rdrm(1)=mst-(mcirc*xf+mwe*xeco);
%
% STORE THE RESIDUE IN A VECTOR RR
%
RR(1:nvarf)=Rf;
RR(nvarf+1:nvarf+nvarsh1)=Rs1;
RR(nvarf+nvarsh1+1:nvarf+nvarsh1+nvare)=Re;
RR(nvarf+nvarsh1+nvare+1:nvarf+nvarsh1+nvare+nvara)=Ra;
RR(nvarf+nvarsh1+nvare+nvara+1)=Rdrm;
RR(nvarf+nvarsh1+nvare+nvara+2:nvar)=Reff;
RR=RR;
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%    THIS SUBROUTINE IDENTIFY THE RESIDUES OF OVERALL BOILER EQUATION,        %
%    EFFICIENCY AND COAL FIRING RATE                                          %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function R=boiler_eff(v_eff,Twg,Tca,H,ms1,ms2,m_rated,X,alpha,hss,hfw,hg,mbd,hsw);
%
ms=ms1+ms2;
eff=v_eff(1);Bc=v_eff(2);
%
%%%%%%FIRST IDENTIFY ALL LOSSES
%
%ENERGY IN THE DRY STACK GAS
%
Ta=20+273.15;patm=1;
%
% CALL COMBUSTION MODULE
%
[ma,mg,mw,mdg,r_Ro2,r_H2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
x_dCo2=m_Co2/mdg;  x_dN2=m_N2/mdg;
x_dSo2=m_So2/mdg; x_dO2=m_O2/mdg;
Cpdg=comb_dry_gas_Cp(Twg,x_dCo2,x_dN2,x_dSo2,x_dO2);%(t,n,x,mdg)
hwg=mdg*Cpdg*(Twg-Ta);
%
%ENERGY IN THE STACK GAS ATTRIBUTED TO THE MOISTER IN THE FUEL AND HYDROGEN
%IN THE FUEL. FIRST IDENTIFY THE PARTIAL PRESSURE
%
molfw=(mw/18)/((mw/18)+(m_N2/28)+(m_O2/32)+(m_So2/64)+(m_Co2/44));
pm=molfw*patm;
if Twg > 374
  hv=superheat_enthalpy(Twg,pm);
else
  hv=suturated_liquid(Twg);
end
 [Tsd,hf,hfgd,hgd,sgd,sfd,sfgd,vfd,vgd]=steam_p(1);
hm=mw*(hv-hf);
hloss=hwg+hm;
%
% RADIATION LOSS (EXTERNAL COOLING)FOR SMALLER BOILERS IT CAN BE IN THE RANGE 3 TO 4%
%
qr_loss=3*m_rated/(ms);
%
%NEXT IDENTIFYING THE HEAT CREDIT. IN THIS, WE ONLY SEE THE HEAT IN THE ENTERING
%COMBUSTION AIR, BY ASSUMING COMBUSTION AIR TEMP. WHICH AFTERWARD SHOULD BE CORRECTED
%
[Cpa,k_air,meu_air]=comb_air_Cp((Ta+Tca)/2);
hca=ma*Cpa*(Tca-Ta);
%
% THEN, THE RESIDUES OF THE EQUATIONS
%
R(1)=eff-(100*(1-((hloss/H)+(qr_loss/100)-(hca/H))));
R(2)=Bc-((ms1*(hss-hfw)+ms2*(hg-hfw)+mbd*(hsw-hfw))/(H*eff/100));

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%      FURNACE RESUDUE VECTOR GENERATOR SUBROUTINE                          %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function [R,q]=furnace_exit_temp(vt,Bc,mg,pbd,Tca,mr,HHV,Tamb,di,do,gvf,kt,tw,Cpa,Tsat,Nt,xv,X,alpha);
%
psat=pbd;
di=gvf(1);do=gvf(2);kt=gvf(3);tw=gvf(4);L=gvf(5);Nt=gvf(6);
Ao=Nt*L*pi*do;
Ai=Nt*L*pi*di;
%
Tg=vt(1);Tw1=vt(2);Tw2=vt(3);
B=0.667;g=9.81;
d=3.265;w=2.8;h=6;

Afw=Ao;

por=0.9;
```

```
[ma,mgc,mwc,mdg,rRo2,rH2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);

[px,hfx,hfgx,hgx,sfx,sfgx,sgx,vgx,vfx]=steam_t(Tsat);
hfg=hfgx;
Cpp=1.6876e3;
Cpcao=(10+0.00484*Tg-108000/(Tg^2))/(56.08*0.2388);
Cpi=Cpcao*1e3;
sigma=5.669e-8;
rhob=54;
dp=1000*1e-6;
rhop=400.45;
rhocg=0.5;
ep=0.8;
x_Co2=m_Co2/mgc;  x_N2=m_N2/mgc;
x_So2=m_So2/mgc; x_O2=m_O2/mgc;x_H20=mwc/mgc;
Cpg=comb_gas_Cp(Tg,x_Co2,x_N2,x_So2,x_O2,x_H20);
meug=comb_gas_viscosity(Tg,x_Co2,x_N2,x_So2,x_O2,x_H20);
kg=comb_gas_therm_conductivity(Tg,x_Co2,x_N2,x_So2,x_O2,x_H20);
Ar=dp^3*g*rhocg*(rhop-rhocg)/(meug^2);
Pr=Cpg*meug/kg;
Nu=0.31*Ar^0.27*Pr;
hpc=Nu*kg/dp;
%   Gas convective coeff, hgc
   ut=6;
%
   hgc=(kg*Cpp/(dp*Cpg))*(rhob/rhop)^0.3*(ut^2/(g*dp))^0.21*Pr;
%   identify radiation coefficient, hr
   ep3=ep/((1-por)*B);
   ed=sqrt(ep3*(ep3+2))-ep3;
   ec=0.5*(1+ep);
   ew=0.72;
%
   hcr=sigma*(Tg^4-Tw1^4)/(((1/ec)+(1/ew)-1)*(Tg-Tw1));
   hdr=sigma*(Tg^4-Tw1^4)/(((1/ed)+(1/ew)-1)*(Tg-Tw1));
   alph=0.4;
%
    hr=hcr+hdr;
%
   T=Tsat;
   fb=3.5*(1-por)^0.37;
   hsw=(fb*hpc+(fb-1)*hgc+hr);
[pamb,hfamb,hfgamb,hgamb,sfamb,sfgamb,sgamb,vgamb,vfamb]=steam_t(Tamb);
   hfgw=hfgamb;
%     Boiling heat transfer coefficient
   sigm=0.058;
   kf=thermal_conductivity(T,psat);
   Cpf=cp_steam_and_water(T,psat);
   muf=viscosity_of_steam(T,psat);
   rhof=1/vfx;rhog=1/vgx;
   Prf=Cpf*muf/kf;
   Acc=Nt*pi*do^2/4;
   ms=mr;
   phisqr=two_phase_multiplier(xv,psat);
   F1=((((Prf+1)/2)*phisqr)^0.444);
   mw=8*ms;
   G=4*mw/Acc;
   Ref=di*G*(1-xv)/muf;
   Retp=Ref*F1^1.25;
   S=supresion_factor(Retp);
   hc=0.023*Ref^0.8*Prf^0.4*(kf/di)*F1;
   delTsat=abs(Tw2-Tsat);
   [pb,hfb,hfgb,hgb,sfb,sfgb,sgb,vgb,vfb]=steam_t(delTsat+Tsat);
   pw=pb;
   delpsat=abs(psat-pw)*1e5;
   hncb1=delTsat^0.24*delpsat^0.75*Cpf^0.45*rhof^0.49*kf^0.79;
   hncb2=sigm^0.5*muf^0.29*hfg^0.24*rhog^0.24;
   hncb=0.00122*S*hncb1/(hncb2);
%
   htp=(hc+hncb);
```

```
    ro=do/2;
    Uo=1/((di/(do*htp))+(ro*log(do/di)/kt)+(1/hsw));
%   Identifing energy release rate, Er
    Er=HHV+ma*Cpa*(Tca-Tamb);
    LHV=HHV-mwc*hfgw;
  q=Er*Bc-mg*Cpg*(Tg-Tamb);

 R(1)=q-Afw*Uo*(Tg-Tsat);
 R(2)=q-Ao*hsw*(Tg-Tw1);
 R(3)=(Tw1-Tw2)*Ao*(kt/(3.5e-3))-q;



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%    THIS SUBROUTINE IS USED TO IDENTIFY THE RESIDUES OF SUPER HEATER MODDEL EQUATIONS    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function R=super_heater_module1(vt,ms,Tgi,Tin,Tss,pss,pin,pex,X,alpha,gv,Bc,pbd);
%
% CALL THE COMBUSTION MODULE
%
[mac,mgc,mwc,mdg,rRo2,rH2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
say=0.63;
Tso=vt(1);Tgo=vt(2);eps=vt(3);ms1=vt(4);ms2=vt(5);
mg=Bc*mgc;
di=gv(1);do=gv(2);A=gv(3);Ag=gv(4);Nw=gv(5);Ntc=gv(6);L=gv(7);S1=gv(8);S2=gv(9);ST=gv(10);
Nt=Nw*Ntc;
hss=superheat_enthalpy(Tss,pss);
hso=superheat_enthalpy(Tso,pss);
[Tsat3,hsw3,hfg3,hg3,sg3,sf3,sfg3]=steam_p(pbd);
hs2=hg3;
Ac=X(7)/100;
af=10;
%
% ASH PARTICLE DIAMETER, M
%
da=16e-6;
Tfb=(Tss+Tin)/2;
Tgb=(Tgo+Tgi)/2;
%
pe=(pex+pin)/2;
%
% EVALUTE SPECIFIC HEAT, VISCOSITY AND THERMAL CONDUCTIVITY OF FLUIDS
%
Cps=cp_steam_and_water(Tfb,pe);
Cps2=cp_steam_and_water((Tin+Tss)/2,pss);
%
x_Co2=m_Co2/mgc;  x_N2=m_N2/mgc;
x_So2=m_So2/mgc; x_O2=m_O2/mgc;x_H20=mwc/mgc;
Cpg=comb_gas_Cp(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
meug=comb_gas_viscosity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
kg=comb_gas_therm_conductivity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);

Cg=mg*Cpg ;
Cs=ms1*Cps;
Cmin=min([Cs Cg]);
Cmax=max([Cs Cg]);
C=Cmin/Cmax;
%
% EVALUTE HEAT TRANSFER COEFFICIENTS
%
hi=hi_tube_side_conv_pass(di,Tfb,pe,ms1,1);
hr=radition_coefficient(Tgb,Tfb,Tgo,S1,S2,do,da,rH2o,rRo2,Ac,af,mgc);
Tin=Tin;
Gg=mg/Ag;
Reg=Gg*do/meug;
Prg=meug*Cpg/kg;
hc=0.33*Gg^0.6*Cpg^0.33*kg^0.67/(do^0.4*meug^0.27);
omega=1;
ho=omega*hc+hr;
```

```
Uo=1/((1/ho)+(1/hi));
NTU=A*Uo/Cmin;
%
% GET RESIDUES OF THE COMPONENT MODEL
%
R(1)=eps*Cmin*(Tgi-Tin)-Cpg*mg*(Tgi-Tgo);
R(2)=Cpg*mg*(Tgi-Tgo)-Cps*ms1*(Tso-Tin);
R(3)=eps-((1-exp(-NTU*(1-C)))/(1-C*exp(-NTU*(1-C))));
R(4)=ms-ms1-ms2;
R(5)=ms1*(Tso-Tss)-ms2*(Tss-Tin);




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%          ECONOMIZER CONSERVATION EQUATIONS RESIDUAL FUNCTION                %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function [R,xeco]=economizer_module(vt,mw,mg,Tgi,Tfw,pein,peco,X,alpha,gv);
%
di=gv(1);do=gv(2);A=gv(3);Ag=gv(4);Nw=gv(5);Ntc=gv(6);L=gv(7);S1=gv(8);S2=gv(9);ST=gv(10);
Nt=Nw*Ntc;
hwo=vt(1);Tgo=vt(2);eps=vt(3);
say=0.68;
fc=1;% THE SELECTION FLUID IS WATER
[Cp,meu,k,rho,Pr]=water_properties(Tfw);
G=mw/(Nt*0.25*pi*di^2);
ut=G/rho;
[pein,hffw,hfgfw,hgfw,sffw,sfgfw,sgfw,vgfw,vffw]=steam_t(Tfw);

da=16e-6;
Ac=X(7);
af=10;
pe=(pein+peco)/2;
%call saturation property at the exit of economizer
[Tsat,hf,hfgex,hgex,sg,sf,sfg,vf,vg]=steam_p(peco);
if hwo>hgex
   xeco=(hwo-hf)/hfgex;
else
   xeco=0;
end
Tee=Tsat;
% Get inside and outside tube coefficients
Tgb=(Tgi+Tgo)/2;
Twb=(Tfw+Tee)/2;
[mac,mgc,mwc,mdg,rRo2,rH2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);

x_Co2=m_Co2/mgc;  x_N2=m_N2/mgc;
x_So2=m_So2/mgc; x_O2=m_O2/mgc;x_H20=mwc/mgc;

meug=comb_gas_viscosity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);%,nc,x,mgn)
Cpg=comb_gas_Cp(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
kg=comb_gas_therm_conductivity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);

Cpw=cp_steam_and_water(Twb,pe);
Cg=mg*Cpg;
Cw=mw*Cpw;
Cmin=min([Cw Cg]);
Cmax=max([Cw Cg]);
C=Cmin/Cmax;
Cec=C;
C2=Cmin/Cg;

hi=hi_tube_side_conv_pass(di,Twb,pe,G,2);
hr=radition_coefficient(Tgb,Twb,Tgo,S1,S2,do,da,rH2o,rRo2,Ac,af,mgc);

omega=1;
Vg=(mg/Ag)*(1+Tgb/273);

Gg=mg/Ag;
```

```
Reg=Gg*do/meug;
Prg=meug*Cpg/kg;

hc=0.33*Gg^0.6*Cpg^0.33*kg^0.67/(do^0.4*meug^0.27);
ho=omega*hc+hr;
Uo=say/((1/ho)+(do/(hi*di)));
NTU=A*Uo/Cmin;

q=eps*Cmin*(Tgi-Tfw);

hfw=hffw;

R(1)=Cpg*mg*(Tgi-Tgo)-(mw*(hwo-hfw));
R(2)=eps*Cmin*(Tgi-Tfw)-(mw*(hwo-hfw));
R(3)=eps-((1-exp(-NTU*(1-C)))/(1-C*exp(-NTU*(1-C))));



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   THIS SUBROUTINE IDENTIFIES THE RESIDUES OF AIR HEATER COMPONENT                %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function R=air_heater_module(vt,ma,mg,Tgi,Tamb,X,alpha,gv);
%
Tao=vt(1);Tgo=vt(2);eps=vt(3);
di=gv(1);do=gv(2);Ao=gv(3);Aa=gv(4);Nw=gv(5);Ntc=gv(6);L=gv(7);ST=gv(8);
Nt=Nw*Ntc;
say=0.83;
% Get inside and outside heat transfer coefficients
Tgb=(Tgi+Tgo)/2;
Tab=(Tamb+Tao)/2;
Tw=(Tgb+Tab)/2;
Tf=(3*Tgb+Tab)/4;
[mac,mgc,mwc,mdg,rRo2,rH2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
Gg=mg/(Nt*0.25*pi*di^2);
x_Co2=m_Co2/mgc;  x_N2=m_N2/mgc;
x_So2=m_So2/mgc; x_O2=m_O2/mgc;x_H20=mwc/mgc;

kg=comb_gas_therm_conductivity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
Cpg=comb_gas_Cp(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
meugb=comb_gas_viscosity(Tgb,x_Co2,x_N2,x_So2,x_O2,x_H20);
meugw=comb_gas_viscosity(Tw,x_Co2,x_N2,x_So2,x_O2,x_H20);

Re=Gg*di/(meugb);
Prg=meugb*Cpg/kg;
Nu=0.021*Re^0.8*Prg^0.33*(meugb/meugw)^0.14;
hi=Nu*kg/di;

[Cpa,ka,meua]=comb_air_properties(Tf);
Cg=mg*Cpg;
Ca=ma*Cpa;
Cmin=min([Ca Cg]);
Cmax=max([Ca Cg]);
C=Cmin/Cmax;
Ga=ma/(Aa);
Rea=Ga*do/meua;
Pra=meua*Cpa/ka;

ho=0.33*Ga^0.6*Cpa^0.33*ka^0.67/(do^0.4*meua^0.27);

Uo=say/((do/(hi*di))+(1/ho));
NTU=Ao*Uo/Cmin;
q=eps*Cmin*(Tgi-Tamb);

R(1)=Cpg*mg*(Tgi-Tgo)-Cpa*ma*(Tao-Tamb);
R(2)=Ca*(Tao-Tamb)-eps*Cmin*(Tgi-Tamb);
R(3)=eps-(1-exp(-(Cmax/Cmin)*(1-exp(-NTU*C))));
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%            THIS FUNCTION COMPUTES THE MOLES OF COMBUSTION GAS CONSTITUENTS          %
%                                                                                      %
%                   X - VECTOR OF FRACTION OF FUEL CONSTITUENTS                        %
%        WHERE C FOR CARBON, H FOR HYDROGEN, N FOR NITROGEN, S FOR SULFUR              %
%     O FOR OXYGEN, W FOR MOISTUR CONTENT OF COAL, A FOR ASH CONTENT OF COAL           %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function [ma,mg,mwv,mdg,r_Ro2,r_H2o,m_N2,m_O2,m_So2,m_Co2]=combustion_module_exact(X,alpha);
%
C=X(1);H=X(2);S=X(3);N=X(4);O=X(5);W=X(6);A=X(7);
Ms=(C/(12*100))+(0.5*H/(2*100))+(S/(32*100))-(O/(32*100));
% MASS OF DRY AIR
Lo=4.33*32*Ms;
ma=Lo*alpha;
% VOLUME OF DRY AIR
Vo=Lo/1.293;
% VOLUME OF CRBON DIOXID
V_Co2=1.86*C/100;
m_Co2=V_Co2*1.977;
V_So2t=0.699*S/100;
% ASSUMING 90% OF SULFUR DIOXIDE IS CAPTURED BY CALCIUM CARBONATE
V_So2=V_So2t*0.1;
m_So2=V_So2*2.927;
V_Ro2=V_Co2+V_So2;
Vo_N2=0.79*Vo+0.8*N/100;
% THEORETICAL VOLUME OF DRY GAS
Vo_dg=V_Ro2+Vo_N2;
% WATER VAPOR
Vo_H2o=0.111*H+0.012*W+0.016*Vo;
% ACTUAL VOLUME OF DRY GAS
V_dg=Vo_dg+(alpha-1)*Vo;
V_H2o=Vo_H2o+0.0161*(alpha-1)*Vo;
mwv=V_H2o*0.804;
% TOTAL VOLUME OF GAS
Vg=V_Ro2+Vo_N2+V_H2o+(alpha-1)*Vo;
m_N2=(Vo_N2+(alpha-1)*Vo*0.79)*1.25;
m_O2=(alpha-1)*Vo*0.21*1.281;
mdg=V_Co2*1.977+V_So2*2.927+Vo_N2*1.25+(alpha-1)*Lo;
r_Ro2=V_Ro2/Vg;
r_H2o=V_H2o/Vg;
mg=1-(A/100)+1.306*alpha*Vo;
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%          SUBROUTINE USED TO IDENTIFY TUBE SIDE HEAT TRANSFER COEFFICIENT            %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function hi=hi_tube_side_conv_pass(di,T,p,G,a);
%
 if a==1 % a=1 indicates the equipment is super heater
    a=1;
    Tc=T;
    Pc=p;
   meu=viscosity_of_steam(T,p);
   k=thermal_conductivity(T,p);
   Cp=cp_steam_and_water(T,p);
   Pr=meu*Cp/k;
elseif a==2 % a=2 indicates the equipment is economizer
   [Cp,meu,k,rho,Pr]=water_properties(T);
elseif a==3 % a=3 indicates the equipment is air_heater

   [Cp,k,meu]=comb_air_properties(T);
   Pr=meu*Cp/k;
end
%
Re=G*di/(meu);
%
Nu=0.023*Re^0.8*Pr^0.4;
```

```
hi=Nu*k/di;



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%            SUBROUTINE USED TO IDENTIFY RADIATION HEAT TRANSFER COEFFICIENT            %
%             OF BOILER CONVECTION PASS CONPONENTS                                      %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function hr=radition_coefficient(Tgb,Tfb,Tge,S1,S2,do,da,rh2o,rRo2,A,af,mg);
%
p=0.1;% pressure in MP
zeta=4.3;% m^2.k/kw fouling coefficient, for solid fuel burning
r=rh2o+rRo2;
S=0.9*do*(((4*S1*S2)/(pi*do^2))-1);
kg=10*(((0.78+1.6*rh2o)/(10*p*S*r)^0.5)-0.1)*(1-0.37*Tge/1000);
%
ka=5990/((Tgb^2*da^2)^(1/3));
muea=A*af/(100*mg);
k=kg*r+ka*muea;
ag=1-exp(-k*p*S);
Taw=(Tgb+Tfb)/2;
hr=5.1e-8*ag*Tgb^3*(1-(Taw/Tgb)^3.6)/(1-(Taw/Tgb));



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%      SHELL SIDE HEAT TRANSFER COEFFICIENT IDENTIFING SUBROUTINE                       %
%       IF THERE IS NO PHASE CHANGE SUCH AS DESUPER HEATER                              %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function ho=ho_desuperheater(HS,Do,Tc1,Tc2,Th1,Th2,ps,m);
%
Ds=HS(1);B=HS(2);C=HS(3);Pt=HS(4);
x=1;%PITCH SELECTION, 1 FOR SQUARE OR 2 FOR TRIANGULAR
switch x
case 1
   De=4*(Pt-pi*Do^2/4)/(pi*Do) ; %square pitch
otherwise
   De=4*((Pt^2*3^0.5/4)-(pi*Do^2/8))/(pi*Do/2);  %triangular pitch
end
As=Ds*C*B/Pt;
Gs=m/As;
%call steam property table at the Tb=(Th1+Th2)/2
Tb=(Th1+Th2)/2;
%
Tw=0.5*((Tc1+Tc2)/2+(Th1+Th2)/2);
mub=viscosity_of_steam(Tb,ps);
muw=viscosity_of_steam(Tw,ps) ;
kb=thermal_conductivity(Tb,ps);
Cpb=cp_steam_and_water(Tb,ps);
ho=(kb/De)*0.36*(De*Gs/mub)^0.55*(Cpb*mub/kb)^(1/3)*(mub/muw)^0.14;



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%             SUBROUTINE USED TO IDENTIFY CONVECTIVE HEAT TRANSFER COEFFICIENT         %
%             WITH PHASE CHANGE SUCH AS CONDENSATION                                   %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function hoc=ho_condenser(hi,Tsat,Tcout,do,Nt,Ttb,Th);
%
[p,hf,hfg,hg,sf,sfg,sg,vg]=steam_t(Tsat);
[Cp,muf,kf,rhof,Pr]=water_properties(Tsat);
```

```
rhog=1/vg;
g=9.81;

Tcout=Tcout;
Tsb=(Tsat+Th)/2;

Tst=(Ttb+Tsb)/2;
Ts=Tst;
Tdiff=abs(Tsat-Ts);
% k=1.0;

Nc=Nt/2;
hoct=0.728*((rhof*(rhof-rhog)*g*hfg*kf^3)/(muf*Tdiff*do))^0.25*Nc^(-1/4);
hoc=hoct;



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                    THIS FUNCTION SOLVES WATER FLOW PROBLEM IN PIPES                    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function v=flow_press_water(vt,m,pb2,pb1,GD,Td,Di,L,K,nhtrs)

 g=9.81;

 [Tsd,hf,hfg,hg,sg,sf,sfg,vfd,vg]=steam_p(pb2*1e-5);
 [Tsc,hf,hfg,hg,sg,sf,sfg,vfc,vg]=steam_p(pb1*1e-5);
 gama_dea=g*(1/vfd);
 gama_cond=g*(1/vfc);
 Tdiff=(Tsd-Tsc)/2;
 Tv=[Tsc,Tsc,Tsc+Tdiff,Tsc+Tdiff,Tsc+2*Tdiff];
nvar=length(vt);
%
% START ITERATION
%
iter=1;
while iter > 0
R=flowresgen_cond(vt,Td,pb2,pb1,GD,Tv,gama_dea,gama_cond,Di,L,m,nhtrs,K);
pd=jacob_flowp_cond(nvar,R,vt,Td,pb2,pb1,GD,Tv,gama_dea,gama_cond,Di,L,m,nhtrs,K);

ran=rank(pd);
siz=size(pd);
vcorr=GAUSS(nvar,pd,R);
vn=vt-(vcorr/2);
%
% CHECK CONVERGENCE
%
if all((abs(vn-vt)/vn )<0.0001)
   iter=0;
   v=vn;
else
   iter=iter+1;
   vt=vn;
end
end



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%           RESIDUE GENERATOR SUBROUTINE FOR WATER FLOW PROBLEM                     %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function R=flowresgen_cond(v,Td,pb2,pb1,GDf,T,gama_b2,gama_b1,Di,L,m,nhtrs,K);
%
nnds=2*nhtrs+1;
p=v(1:nnds);
Hp_c=v(nnds+1);N=v(nnds+2);
g=9.81;
[c0,c1,c2]=pump_module(Td);
GD1=GDf(1:5);GD2=GDf(6:10);
```

```
%
% GAMMA CALCULATION
%
for i=1:nnds
  [Tsd,hf,hfg,hg,sg,sf,sfg,vfd(i),vg]=steam_p(p(i)*1e-5);
   rho(i)=1/vfd(i);
   gama(i)=rho(i)*g;
end
ut=[1.25,1.25,1.25];
Qc=3600*m/rho(1);
a=1;
for ii=1:nhtrs+1
    jj=a+1;
if ii==nhtrs+1
    f(ii)=pipe_friction_coeff(m,T(a),p(a),Di(ii),ut(ii),1);
    hf(ii)=((f(ii)*L(ii)/Di(ii))+K(ii))*ut(1)^2/(2*g);
    R(ii)=p(a)-gama(a)*((pb2/gama_b2)+hf(ii));
else
    f(ii)=pipe_friction_coeff(m,T(a),p(a),Di(ii),ut(ii),1);
    hf(ii)=((f(ii)*L(ii)/Di(ii))+K(ii))*ut(1)^2/(2*g);
    R(ii)=p(a)-gama(a)*((p(jj)/gama(jj))+hf(ii));
     a=a+2;
 end
end
a=2;
for ii=nhtrs+2:2*nhtrs+1
    jj=a+1;
    zz=ii+1-4;
    if zz==1
        GD=GD1;
    else
        GD=GD2;
    end
    Tmb=(T(a)+T(jj))/2;
    delp(zz)=tube_side_pressure_loss(m,Tmb,p(a),GD,1);
    Hx(zz)=delp(zz);
     R(ii)=p(a)-gama(a)*((p(jj)/gama(jj)))-Hx(zz);
     a=a+2;
end

R(2*nhtrs+2)=Hp_c-((p(1)/gama(1))-(pb1/gama_b1));
R(2*nhtrs+3)=Hp_c-((c0(1)+c0(2)*N+c0(3)*N^2)+(c1(1)+c1(2)*N+c1(3)*N^2)*Qc+(c2(1)+c2(2)*N+c2(3)*N^2)*Qc^2);



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%     JACOBEAN GENERATOR SUBROUTINE FOR WATER PIPE FLOW PROBLEM                %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function pd=jacob_flowp_cond(nvar,R,v,Tdc,pdea,pcond,GD,Tv,gama_dea,gama_cond,Di,L,mst,nhtrs,K);

dv=0.001;
for k=1:nvar
    vd(k)=v(k);
end
for j=1:nvar
    if abs(v(j))<1e-30
        vd(j)=v(j)+0.001;
    else
        vd(j)=(1+dv)*v(j);
    end

    Rd=flowresgen_cond(vd,Tdc,pdea,pcond,GD,Tv,gama_dea,gama_cond,Di,L,mst,nhtrs,K);
    for i=1:nvar
        if abs(v(j))<1e-30
            pd(i,j)=(Rd(i)-R(i))/0.001;
        else
            pd(i,j)=(Rd(i)-R(i))/(v(j)*dv);
        end
    end
```

```
    vd(j)=v(j);
end
format short




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   THIS SUBROUTINE IS USED TO IDENTIFY PRESSURES AT DIFFERENT NODES OF A STEAM FLOW      %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function pnds=flow_press_steam(pb2,GD,gama_b2,Di,L,m,K,nhtrs);
%
nnds=2*nhtrs+1;
g=9.81;

Tav=643.15;pav=65e5;
vsp=superheat_specific_volume(Tav,pav*1e-5);
rho=1/vsp;
ut=[16,16,16];
%
gama=[rho*g rho*g rho*g rho*g rho*g];
Hx=zeros(nnds,1);
hf=zeros(nnds,1);
%
% GENERATE TOTAL HEAD LOSS VECTOR, hf
%
for i=1:nhtrs+1
    jj= 2*i-1;
f(jj)=pipe_friction_coeff(m,Tav,pav,Di(i),ut(i),2);
hf(jj)=((f(jj)*L(i)/Di(i))+K(i))*ut(i)^2/2*g;
end
%
% GENERATE HEAT EXCHANGERS' LOSS VECTOR, Hx
%
for j=1:nhtrs
    Hx(2*j)=tube_side_pressure_loss(m,Tav,pav,GD,2);
end
%
% CALCULATE NODAL PRESSURES SEQUISIALLY STARTING FROM THE GIVEN BOUNDARY PRESSURE
%
pb=pb2;
gamapb=gama_b2;
for i=nnds:-1:1
    pnds(i)=pb+gama(i)*hf(i)+Hx(i);
    pb=pnds(i);
    gamapb=gama(i);
end




%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%    INDEX FUNCTION FOR ASSEMBLING COMPONENT RESIDUES                                      %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function index=ind_fun(iel,nnel,fact);
%
 edof = nnel;

switch fact
case 1
    start = (iel-1)*(nnel);

   for i=1:edof
      index(i)=start+i;
   end
case 2
    start = (iel-1)*(nnel);

   for i=1:edof+2
```

```
     index(i)=start+i;
   end
end



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                    FEED WATER HEATERS RESIDUE VECTOR ASSEMBLLY                              %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function ff=resid_ass_fun(ff,f,index);
 edof = length(index);
 for i=1:edof
   ii=index(i);
     ff(ii)=ff(ii)+f(i);

 end



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   GAUSSIAN ELEMINATION SUBROUTINE
%     DEFINITION OF VARIABLES                    %
%      N = NUMBER OF EQUATIONS                   %
%      A = MATRIX OF COEFFICIENTS                %
%      C = RIGHT-HAND-SIDE VECTOR                %
%      X = UNKNOWNS                          %
%      O = ORDER VECTOR                          %
%      S = SCALE VECTOR                         %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function X=GAUSS(N,A,C);

for i=1:N
    O(i)=i;
    S(i)=abs(A(i,1));
    for j=2:N
        if abs(A(i,j))>S(i);
            S(i)=abs(A(i,j));
        end
    end
end
%elimination
for k=1:N-1
%put pivoting
    pivot=k;
    big=abs(A(O(k),k)/S(O(k)));
    for ii=k+1:N
        dummy=abs(A(O(ii),k)/S(O(ii)));
        if dummy>big
            big=dummy;
            pivot=ii;
        end
    end
    idum=O(pivot);
    O(pivot)=O(k);
    O(k)=idum;
    for jj=k+1:N
        factor=A(O(jj),k)/A(O(k),k);
        for aa=k+1:N
            A(O(jj),aa)=A(O(jj),aa)-factor*A(O(k),aa);
        end
        C(O(jj))=C(O(jj))-factor*C(O(k));
    end
end
% substitution
X(N)=C(O(N))/A(O(N),N);
for i=N-1:-1:1
    sum=0.0;
    for j=i+1:N
        sum=sum+A(O(i),j)*X(j);
    end
```

```
   X(i)=(C(O(i))-sum)/A(O(i),i);
   format long;
end



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% THIS FUNCTION MANUPILATE THE TUBE SIDE PRESSURE DROP OF HEAT EXCHANGERS              %
%                                                                                       %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function delpt=tube_side_pressure_loss(m,T,p,GD,fc);
di=GD(1);do=GD(2);Np=GD(3);L=GD(4);Nt=GD(5);
%
Ac=Nt*pi*di^2/4;
% FACT IDENTIFYIES THE TYPE OF FLUID IN THE PIPE
switch fc
case 1    % IF THE FLUID IS WATER
   [Cp,mu,k,rho,Pr]=water_properties(T);
   rho=rho;
case 2     % IF THE FLUID IS STEAM
   v=superheat_specific_volume(T,p*1e-5);
   rho=1/v;

end
ut=(m/(rho*Ac));
f=pipe_friction_coeff(m,T,p,di,ut,fc);
delpt=Np*(8*f*(L/di)+2.5)*rho*ut^2/2;



%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% THIS FUNCTION USED TO IDENTIFY FRICTION COEFFICIENTS OF FLUID FLOW IN PIPES              %
% WITH GIVEN VALUES: m - MASS FLOW RATE OF FLUID IN THE PIPE, m/s                          %
%                  T - MEAN BULK TEMPERATURE OF THE FLUID IN THE PIPE, K                   %
%                  p - PRESSURE, pa                                                        %
%                  Di- INSIDE DIAMETER OF THE PIPE, m                                      %
%                  fc- FLIUD CODE; 1 FOR LIQUID WATER,                                     %
%                              2 FOR SUPER-HEATED STEAM                                    %
%                                                                                          %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
function f=pipe_friction_coeff(m,T,p,Di,ut,fc);

switch fc
case 1
   [Cp,mu,k,rho,Pr]=water_properties(T);
case 2

   v=superheat_specific_volume(T,p*1e-5);
   rho=1/v;
   mu=viscosity_of_steam(T,p*1e-5);
end
eps=1e-5;
Re=ut*rho*Di/mu;
if Re>2300
   fdw=0.3086/(log10((6.9/Re)+(eps/(3.7*Di))^1.11))^2;
else
   fdw=64.0/Re;
end
f=fdw;
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%This function used to perform the curve fit polynomial approximation of a pump           %
%given performance characteristic, that is, H=f(Q,N)                                       %
%The input Tdc is a matrix of data from the performance characteristic curve               %
%                                                                                           %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function [bc0,bc1,bc2]=pump_module(Tdc);
%
format long;
QTdc=[1 Tdc(1,2) Tdc(1,2)^2
     1 Tdc(1,3) Tdc(1,3)^2
     1 Tdc(1,4) Tdc(1,4)^2];
NTdc=[1 Tdc(2,1) Tdc(2,1)^2
     1 Tdc(3,1) Tdc(3,1)^2
     1 Tdc(4,1) Tdc(4,1)^2];
ac1=inv(QTdc)*Tdc(2,2:4)';
ac2=inv(QTdc)*Tdc(3,2:4)';
ac3=inv(QTdc)*Tdc(4,2:4)';


bc0=inv(NTdc)*[ac1(1);ac2(1);ac3(1)] ;
bc1=inv(NTdc)*[ac1(2);ac2(2);ac3(2)]  ;
bc2=inv(NTdc)*[ac1(3);ac2(3);ac3(3)] ;
```