

Addis Ababa University
Faculty of Technology
Mechanical Engineering Department
Graduate Program in *Thermal Engineering*

SIMULATION OF MODERN STEAM POWER PLANT

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A thesis submitted to the School of Graduate Studies of Addis Ababa University in partial fulfillment of the requirements of the Degree of Masters of Science in Mechanical Engineering (Thermal Engineering Stream)

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Symbols

m_s = mass flow rate of steam/water

m_g = mass flow rate of flue gases

m_a = mass flow rate of air

H = head of a pump

Q = volume flow rate

U = Overall heat transfer coefficient

U_s = Overall heat transfer coefficient of superheater

U_{ev} = Overall heat transfer coefficient of evaporator

U_{ec} = Overall heat transfer coefficient of economizer

U_{rh} = Overall heat transfer coefficient of reheater

U_{ah} = Overall heat transfer coefficient of air heater

cp_g = the specific heat capacity of flue gases

cp_a = the specific heat capacity of air

cp_s = the specific heat capacity of steam/water

T_{gos} = temperature of the flue gases leaving the superheater

T_{gis} = the temperature of flue gases entering the superheater

T_{goh} = the temperature of flue gases leaving the reheater

T_{goh} = the temperature of flue gases entering the reheater

T_{goec} = the temperature of flue gases leaving the economizer

T_{giec} = the temperature of flue gases entering the economizer

T_{goah} = the temperature of flue gases leaving the air-heater

T_{giah} = the temperature of flue gases entering the air-heater

T_{goev} = the temperature of the flue gases after the evaporator

T_{giev} = the temperature of the flue gases before the evaporator

t_{sis} =the temperature of the steam entering the superheater

t_{sos} =the temperature of the steam leaving the superheater

t_{sih} =the temperature of the steam entering the reheater

t_{soh} =the temperature of the steam leaving the reheater

t_{soec} =temperature of the steam leaving the economizer

t_{siec} =temperature of the steam entering the economizer

t_{oah} =temperature of the air leaving the air-heater

t_{amb} =the temperature of the ambient air

t_a = temperature of the air

A_{ev} =the heat transfer area of evaporator

A_s =the heat transfer area of the superheater

A_{ec} =the heat transfer area of the economizer

A_{rh} =the heat transfer area of the reheater

A_{ah} =the heat transfer area of air heater

hph =high pressure heater

lph = low pressure heater

$nheaters$ =number of heaters in the steam power plant cycle

$nhph$ = number of high pressure heater

$nlph$ = number of low pressure heater

$nhptext$ = number of high pressure turbine extraction

$niptext$ = number of intermediate pressure turbine extraction

$nlptext$ = number of low pressure turbine extraction

h_{in} = enthalpy of water entering the heater

h_{out} = enthalpy of water leaving the heater

h_{ext} = enthalpy of steam extracted from the turbine

h_{dr} = the enthalpy of water drained from the heater

m_{ext} = mass of the extraction from the turbine

m_{dr} =mass flow rate of the water drain from the heater

m_{in} =the mass flow rate of the water entering the heater

W_t =total power out put from the turbines

Abstract

The thesis describes a steam power plant simulator which has been developed using the MATLAB code. The simulation is important to study the behavior of the steam power plant system by means of a model, and the results obtained allow us to consider many different design possibilities as well as varieties operating conditions.

Therefore, different designs may be evaluated so that an acceptable design can be chosen and safe levels may be established for the operating conditions. These results are also used for optimization of the plant.

The basic considerations in the development of a mathematical model are first presented in the thesis, followed by a discussion of simulation of a steam power plant. Conservation equations and characteristic curves are used to model the component of the steam 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.

The *STEAM_POWER_PLANT_SIMULATION* program is used to simulate the power plant system. In the program there are two main solvers. These are: the turbine-heater solver and boiler solver. The turbine-heater solver consists of the turbine and feed water heaters including the pumps and the condenser. The boiler solver has its own residue generator function and Jacobean generator function and then Newton-Raphson technique is used to solve a set of nonlinear algebraic equations.

CHAPTER ONE

1.1 Introduction

By far the greater part of the mechanical and electrical energy generated for industrial purposes is furnished by steam power plant. Despite the rapid progress of the internal combustion engine and the intensive development of water power, the steam power plant still leads the field in power generation and will probably continue to do so for years to come.

The function of steam power plant is to convert the raw energy (chemical) in fossil fuels (coal, oil, gas) into mechanical or electrical energy through the expansion of steam from a high pressure to low pressure in a suitable prime mover.

Steam power plants employ either turbines or reciprocating engines, not only as prime movers but as drives also for auxiliary equipment, such as pumps, stokers, fans, etc. Steam is generated in boilers by burning fuel in furnaces and is conveyed to the prime movers through pipelines. A steam power plant can perform two purposes:

1. Production of electric power only
2. Production of electric power along with the production of steam for industrial purposes.

Cheap and abundant supply of electric power is the major factor in the development and progress of a country. Modern life is so much dependent upon electric power that the per capital consumption of electricity is often an index of the economic development, prosperity and standard of living of a nation. It is, therefore, quite clear that if the country is to become prosperous and economically sound, more and more emphasis should be laid on the continuous and increased rate of growth of electric power.

Therefore, optimization of a moderately complex steam power plant requires the prediction of the performance of the system, given certain

input conditions and performance characteristics of components .This is the process of system simulation. System simulation is not only a step in the optimization process but also permits examination of the operating conditions that may pinpoint possible operating and control problems at off-design conditions.

A system is a collection of components whose performance parameters are interrelated. The term system simulation means observing a synthetic system that imitates the performance of a real system. System simulation assumes knowledge of the performance characteristics of all components. Simulation is used when it is not possible or not economical to observe the real system. Another situation where the system simulation is employed is where the system is still in the design stage and no real system yet exists. The performance or control of the system at off-design may be of interest, so the planned system is “**run**” in advance of its construction. Most thermal systems are designed for some maximum load or demand but operate most of the time at loads below the design value. Economic and optimization analysis should be done throughout the range of operating conditions and not just at the design point.

This thesis mainly focuses on the development of a computer simulation program module of an ideal regenerative steam power plant and it is used to study the behavior of the steam power plant system by means of a model, and it also helps to investigate the sensitivity of the system to different variables, i.e. by varying the inlet steam temperature, pressure, mass flow rate to a turbine, and the number of heaters in the steam cycle.

1.2 Modeling and Simulation of Thermal System

Modeling is one of the most crucial elements in the analysis of thermal systems. Practical processes and systems are generally very complicated and must be simplified through idealization and approximations to make a problem solvable.

Modeling is needed for understanding and predicting the behavior and characteristics of thermal systems. Once a model is obtained, it is subjected to a variety of operating conditions and variations. If the model is a good representation of the actual system under consideration, the outputs obtained from the model characterize the behavior of the given system.

Mathematical modeling is at the very core of the modeling of the thermal systems because it brings out the basic considerations with respect to a given system. Focusing on the dominant mechanisms and neglecting less important aspects, it simplifies the problem by using approximations and idealizations. Conservation laws are used to drive the governing equations, which may be algebraic equations, integral equations, ordinary differential equations, partial differentials equations or combination of these.

Both equation fitting and mathematical modeling are processes of finding equations that represent properties of substances or performance characteristics of equipment. The distinction between the two is, generally that equation fitting starts with some data and uses only the numerical relationship among the data points to arrive at an equation, while mathematical modeling utilizes physical principles applicable to the substance, process, or component to arrive at an equation.

Equation fitting and mathematical modeling are not ends in themselves but are steps towards system simulation and optimization. In order to optimize a system, it is essential to predict the performance of that system to evaluate adjustments in component size and performance. System simulation becomes, then, a step in optimization as well as a useful activity in its own right.

To simulate a system, the performance of each component of that system must be known, and at the operating condition of the system all the individual performance characteristics of the components must be

satisfied. Steady-state simulation becomes, therefore, a simultaneous solution of the performance characteristics of all the components in the system.

1.2.1 Curve fitting

Curve fitting is an important and valuable technique that is used extensively to represent the characteristics and behavior of thermal systems. It is a means by which tabular or graphical information can be readily assimilated into a component model or a system simulation. Functional form of curve fit selection is perhaps the most crucial decision made in the curve fitting process since an inappropriate choice of functional form invariably results in a less than adequate curve fit. The simplest curve fitting form is the linear equation:

$$y = a + b * x \tag{1.1}$$

The general expression of the curve fitting form is given by the polynomial equation

$$y = a_0 + a_1 * x + a_2 * x^2 + \dots + a_m * x^m \tag{1.2}$$

Pump performance curves are well suited for representation by a quadratic equation.

1.2.2 Interpolation

Interpolation is a useful alternative to curve fitting when tabular information is available. Many methods of varying accuracy and complexity are available. Lagrange interpolating polynomial method is the most applicable, and is very convenient and compact for computer applications [1]. In this thesis, tabulated data, such as steam tables 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.

1.3 System Simulation Technique

The information and the techniques presented in the preceding section have barely scratched the surface of curve fitting and interpolation. They do, however, provide a convenient way to model hardware performance characteristics; and while much sophisticated techniques are available where higher accuracy is required, the results of the more sophisticated techniques are utilized in the same manner by the two simulation procedures.

The two general steady state simulation procedures used for energy systems are the multivariable Newton-Raphson and the Hardy-Cross methods.

1.3.1 Hardy-Cross Method

The Hardy-Cross method simulation method is the widest used method for steady-state fluid flow networks. It has the advantages of flexibility and ease of application. The Hardy-Cross method can be categorized as either loop or node. The generalized Hardy-Cross technique based on loop equation is fine for fluid systems in which continuity of mass and uniqueness of pressure must be enforced; however, such a methodology is not easily extendable to system simulation requiring matching of components with complex parametric representations [1].

1.3.2 Multivariable Newton-Raphson Simulation Method

A more generalized steady-state simulation method than the Hardy-Cross technique is available. This more general method uses information flow diagrams to represent each system component, links in an appropriate manner all the component information-flow

diagrams, and develops equations to represent the system. Such an approach results in a system of nonlinear algebraic equations that must be solved for steady-state values of the operating parameters. Obviously, the number of unknowns must be equal to the number of equations, and the equations must be independent.

The multivariable Newton-Raphson interactive approach is the main technique for solving systems nonlinear equations and solves the system simultaneously and, if well behaved, generally avoids convergence problems. It is developed by considering a Taylor's series expansion.

The following procedures are used for implementing the technique:

1. Rewrite all the equation in the form $R_i(x_1, x_2, x_3, \dots, x_n) = 0$;

where: $i=1, 2, 3, \dots, n$

2. Assume initial value of $x_{1t}, x_{2t}, x_{3t}, \dots, x_{nt}$

3. Calculate $R_i(x_{1t}, x_{2t}, x_{3t}, \dots, x_{nt})$ for $i=1, 2, 3, \dots, n$;

This is the step of determining the residue of the system.

4. Compute the Jacobin matrix of the system of equation, which is the partial differential form of the residue equations at the initial guess values.

5. Solve the set of linear equations

6. Update the solution for the next iteration

$$x_i = x_{it} - \delta x_i, \text{ for } i=1 \text{ to } n$$

7. Test for the convergence by substituting the values computed in step 6 in step 3. If convergence fails, repeat steps 4-7, otherwise stop iteration.

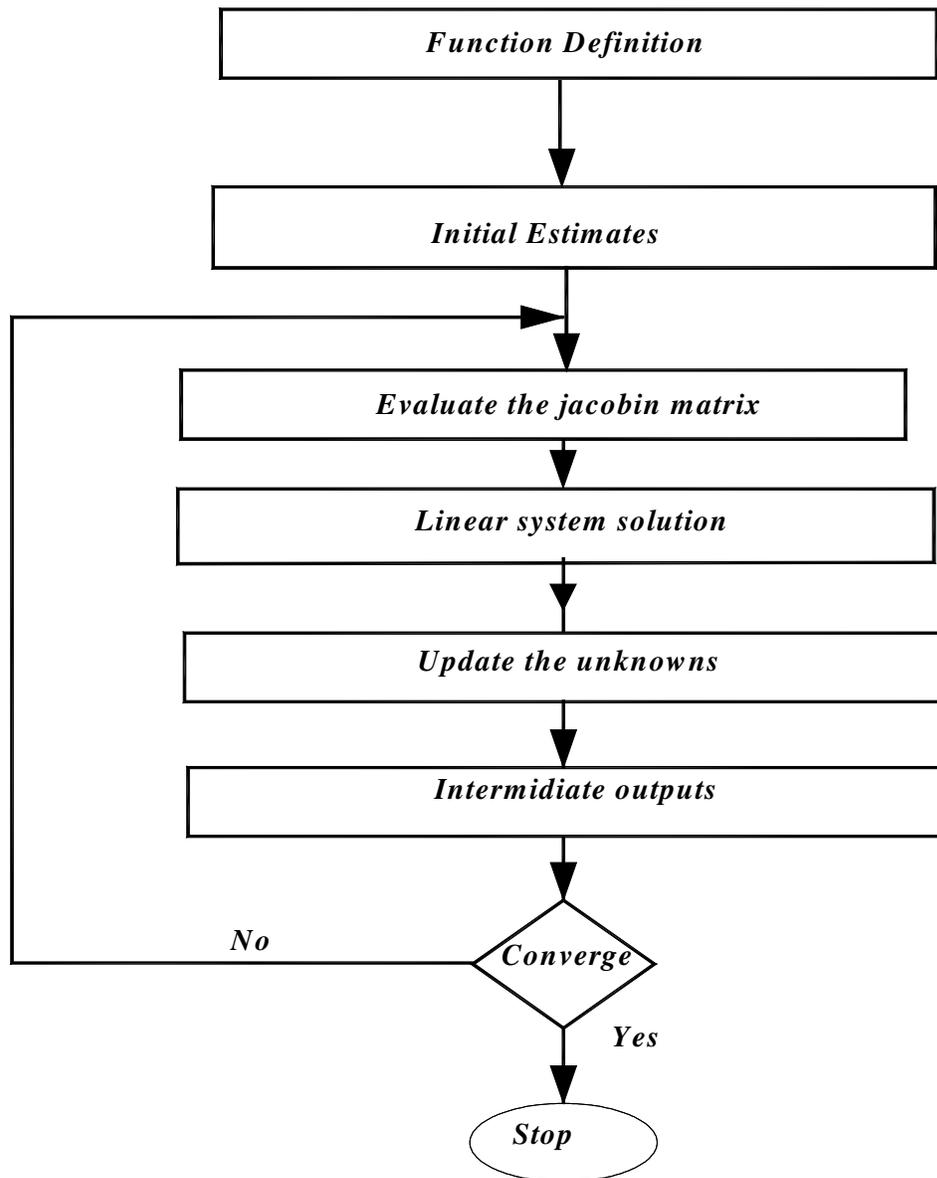


Fig. 1.1 Multivariable Newton-Raphson Simulation Method

CHAPTER TWO

Salient Features of a Modern Steam Power Plant

2.1 Boiler

In the boiler, water is converted into steam with the help of heat produced by the burning of coal. The boiler can be either fire tube or water tube boiler. Modern big boilers are all water tube boilers in which water flows through the tubes and the combustion gases across the tubes. A modern boiler may produce steam at the rate of 375 tones/hr at 14 MPa and 540°C and burning coal at the rate of 200 tones/hr. The temperature inside the furnace where fuel is burnt is of the order of 1500°C. The inside of boiler also contains separate set of tubes which constitutes heat exchanges in which heat in the flue gases is exchanged with other media. These are:

1. Superheater

The superheater is situated at the hottest part of the boiler. It is employed to raise the steam temperature above the saturation temperature by absorbing heat from fuel gases. The maximum temperature to which steam can be heated will depend upon the metallurgy and economy in initial cost and maintenance cost of the superheater. The present trend is to keep the steam temperature at 540°C. The superheating of steam makes it possible to recover more energy from steam which improves the cycle efficiency of the plant. It also eliminates the formation of water vapor during conveying of steam in pipelines and during its early flow through the turbine blades. From the superheater, the steam is led to high-pressure turbine.

The superheater is a control volume operating at constant pressure, but operates on steam only. Because the heat energy for the superheater generally comes from the same thermal reservoir as the

boiler, the energy inputs to both the boiler and the superheater are grouped as simply Q_b .

2. Reheater

The function of the reheater is to raise the temperature of steam after it has expanded in the high-pressure turbine. After being reheated, it passes through the intermediate and low-pressure turbines. In the reheater also, the temperature of steam is limited to 540°C. The higher a boiler pressure and the lower a condenser pressure, the lower the quality of the turbine exhaust would be.

A more practical option to achieving drier steam is the concept of reheat. In this system, as before, the steam is expanded in the turbine. Before complete expansion occurs, however, the steam is extracted and reheated to add enthalpy, and then finally taken to a low-pressure turbine for expansion to condenser pressure.

3. Economizer

The function of an economizer in a boiler is to absorb heat from the outgoing flue gases and raises the temperature of the feed water coming from the condenser before it enters the evaporative section of the boiler. It is usually located before the air heaters and following the superheater and reheater.

2.2 Steam Turbine

The function of a steam turbine is to convert the heat energy in the steam into rotational power of the shaft on which the turbine is supported. In a modern steam power plant, the steam turbine is usually a three cylinder tandem compound machine consisting of one single flow high pressure cylinder, one single flow intermediate pressure cylinder and one double flow low pressure cylinder. The rotational speed of the shaft is set by the frequency of the electricity supply and is 3000 rev/min (50 rps)

corresponding to an alternating electric supply at 50Hz (cps). The turbine is fitted with a precise oil operated speed governor.

2.3 Generator

The generator which is directly coupled to the turbine shaft, converts mechanical energy of turbine shaft into electrical energy. It consists of two electrical windings. One is mounted on the turbine shaft, rotating with it, and is called the rotor. The other is arranged as a shroud around the rotor, fixed to the floor, and is called the stator. The relative motion of rotor and stator generates the electricity.

2.4 Condenser

The function of the condenser is to condense the steam which has been discharged from low pressure turbine. The condenser is a large vessel containing a large number of tubes through which the cold water is circulated continuously for condensing the steam flowing outside the surface of the tubes. The use of condenser increases the output of the plant by lowering the exhaust pressure of steam and by providing hot feed water to the boiler.

2.5 Boiler Feed-Pump

A boiler feed pump is the heart of any steam power plant. It supplies feed water (coming from the condenser) to the boiler at a high pressure. This is one of the most sophisticated and the largest auxiliary of the power plant.

2.6 Feed Water Heaters

The function of feed water heaters is to raise the temperature of the feed water by means of bled steam before the feed water is supplied to the boiler from the hot well. There are two main types of feed water heaters: the open or contact type and closed or surface type. In the open heaters, feed water mixes with the heating steam and in closed heaters, the bled steam and feed water don't mix with each other.

2.6.1 Open or Contact Heaters

These heaters are usually constructed to remove non-condensable gases from water and steam along with raising the temperature of feed water and such a heater is termed as "**deaerator**". Oxygen and carbon dioxide dissolved in the feed water greatly corrode the inner surfaces of the boiler unit and the pipelines. Therefore, the feed water should be delivered to the boilers after it has undergone deaeration or degasification. The amount of gas dissolved in water depends upon its temperature. This decreases sharply with increasing temperatures and drops to almost zero at the boiling point. Hence thermal deaeration of the feed water is usually practiced. A thermal deaerator is nothing but a direct contact or open heater.

The direct contact feed water heaters (open heaters) have got the following advantages;

1. Complete conversion of steam to water is accomplished.
2. Non- condensable corrosive gases are removed from the feed water.
3. Removal of impurities in the water is possible
4. Water is brought to the temperature of the steam
5. Heater acts as a small reservoir

2.6.2 Closed or Surface Heaters

The closed feed water heater is a surface heat exchanger with the feed water inside the tubes, and the extraction steam condensing on the outside of the tubes. In contrast to the open feed water heater, which is a contact heater, the closed feed water heater can operate with different pressures for the extraction steam and the feed water. Because the two steams are at different pressures in the closed feed water heater, two principal alternatives exist for the condensed extraction steam from the heater drain:

1. Allow it to expand isenthalpically, and send it to a lower pressure feed water heater, or condenser.
2. Increase its pressure with a small pump and mix it into the feed water stream.

Both of these alternatives are frequently used.

In closed water heaters, the feed water can never be heated to the temperature of the steam. To maintain a high overall heat transfer for the heater, the water velocity should be high but pumping costs will limit the velocity.

In the ideal closed feed water heater, the extracted steam condenses and leaves the feed water as saturated liquid at the turbine extraction pressure. In the ideal case the feed water from the condenser is assumed to leave as compressed liquid at the same temperature as the condensed extracted steam. One advantage of a closed feed water heater is that the pressure of the extracted steam and the feed water can be significantly different. Owing to the type of construction, closed feed water heaters usually operate at higher pressures than the open type. However, since open feed water heater operates at fairly low pressures, they are less expensive. Another

advantage of the open heater is that it brings the feed water to the boiler up to its saturation temperature at the heater pressure.

2.7 Steam Traps

The water collected in the steam separator flows down into a steam trap. The function of the steam trap is to automatically drain water away or return to the boiler water, with out allowing the steam to escape. The steam traps are mainly of two types: **expansion** traps and **float** traps. In expansion traps, use is made of the expansion of the metals by heat. Also, the temperature of the water to be drained is quite lower than that of the steam. This difference in temperature produces a difference in the lengths of rods sufficient to open a valve which allows the water to escape and to close it when the water is discharged. Float type of traps use either a closed hollow float or a bucket. The principle of a closed hollow ball floats is similar to that in water tanks. The hallow ball floats in a closed vessel into which the condensed steam is discharged. As the water level rises, the float or ball rises up in the vessel and the discharge valve is opened which is attached to the float by means of a level arrangement. When most of the water goes out of the vessel, the float falls down and discharge valve is closed. In the bucket type of trap, a bucket floats in the vessel. As the level of water in the vessel goes on increasing, a stage will come when it is completely filled, it will sink down resulting in the opening of the discharge value and the water will start flowing out of the vessel. When most of the water has escaped, the bucket will rise up and the discharge value will be closed.

2.8 Disposal/ Drain Pumps

The two common methods to dispose off the heater drains are:

2.8.1 Cascade Method

In this method, the drain from one heater is led to the next lower heater. On its passage through this heater, the steam is condensed in heating the feed. As the drain is flowing from high pressure heater to low pressure heater, no pump is needed in this method.

2.8.2 Drain pump /Drip Pump

The drain from each heater is pumped directly into the feed line at a point immediately after the heater.

CHAPTER THREE

3.1 Working Principles of Modern Coal Fired Thermal Power Plants

The working of a modern coal fired thermal power plant can be studied conveniently with the help of the various cycles, such as flue gases cycle, ash cycle and water and steam cycle:

3.1.1 Steam and Water Cycle

Chemically treated water flowing through the water walls of the boiler furnace is evaporated into steam by absorption of heat. The steam is further heated in the superheaters. The dry, high pressure and high temperature steam is then fed to the steam turbine comprising of usually three cylinders: high pressure cylinder, intermediate pressure cylinder and low pressure cylinder. There, the steam is expanded through the three cylinders and thermal energy of steam is converted into mechanical energy of the turbine shaft, which is utilized to rotate a generator and produce electrical energy. The steam discharged from high pressure turbine is fed back to reheaters in the boiler. After it is reheated, the steam flows to intermediate pressure turbine and finally it expands in the low pressure turbine. After extraction of the useful energy in the steam turbine, the exhaust steam flows into a condenser where it is condensed to water. From the condenser, the condensed steam (condensate) is pumped to deaerator. The deaerator reduces the dissolved oxygen content in the condensate. From the deaerator, the feed pump forces the feed water under pressure to the economizer in the boiler, after the temperature of feed water is raised in high pressure feed water heaters with the help of steam taken from the high-pressure turbine. In the economizer, the feed water is further heated up by the hot flue gases leaving the boiler. From the economizer, the feed water enters the boiler drum to which water tube

walls and superheaters of the boiler are connected to generate superheated steam.

3.1.2 Flue Gas Cycle

The flue gases produced in the boiler furnace, by the combustion of fuel, evaporate water, running through the water walls of the boiler furnace, into steam. After losing much of its heat in converting water into steam, the flue gas flows upwards through the boiler, sucked by an induced draught fan. During its upward flow, the flue gas loses heat in the superheaters (to superheat the steam), reheaters (for reheating the steam), and economizer (to heat the boiler feed water) and in the air heater (for heating the air required for combustion of fuel in the boiler furnace). After passing through the ash precipitators, the flue gas leaves through the chimney.

3.1.3 Ash Cycle

Ash is the product of coal combustion in the furnace, along with flue gases. About 20% of the ash falls in the bottom ash hopper of the boiler and is periodically removed mechanically. The remaining ash (called fly ash) is carried by the flue gases. It is separated in ash precipitators (mechanical and electro static) and is mechanically removed and further disposed-off in accordance with ash disposal arrangements. The cleaner flue gases escape to the atmosphere through the chimney via induced draught fan.

3.2 Thermodynamic Properties of Steam

3.2.1 What Is Steam?

Steam is used in many forms of power generation, such as oil, coal, nuclear, and natural gas power plants as the final energy transfer

medium that spins the turbine. Steam is water vapor that is heated to the boiling point

3.2.2 How Is Steam Generated?

Energy in steam is held both in heat and pressure. When heat is applied to a water molecule, the electrons begin to move very rapidly around it. The change of state from a liquid to a gas takes place when the energized, moving molecule of water takes to the air because of its quickly moving electrons. The water molecule in air stays away from other water molecules. The distance between the molecules creates pressure. When countless billions of water molecules are heated, and all repelling each other, pressure builds. When this pressure is released, energy is generated.

3.2.3 What Does Steam Do?

When the pressure in steam is released, it transfers its energy to whatever it hits, usually a turbine or a piston. The transfer of energy from the steam to the object causes it to move. This movement can be used to generate electrical energy, or it can use the steam energy directly as in steam-powered trains, steam engines, and steam shovels. The transfer of energy is what is used in many power plants to transfer the heat energy released from the fuel to the turbine.

3.2.4 Phase-Change Processes

Two phases of a pure substance coexist in equilibrium in many practical situations. For example, water exists as a mixture of liquid and vapour in the boiler and the condenser of a steam power plant.

A liquid at a state where it is not about to vaporize is called a compressed liquid or sub cooled liquid. For example, water, say, at 1atm and 20°C is a super cooled liquid [9].

Compressed liquids are also called sub cooled liquids. There is not much data available on compressed liquids. However, since water is used as the fluid in fossil-fuel power plants, considerable data is available on it in the liquid region.

Compressed-liquid data in most cases can be approximated closely by using the property values of the saturated-liquid state at the given temperature. That is, the compressed-liquid data is more temperature dependent than it is pressure dependent. However, when the pressure difference between the saturated-liquid and the compressed-liquid is very large, the enthalpy value should be corrected for the pressure difference. In the absence of compressed-liquid data, the compressed liquid may be treated as saturated liquid at the given temperature [9].

A liquid which is about to vaporize is called a saturated liquid (e.g. water at 1 atm and 100°C). A vapour which is about to condense is called a saturated vapor. A substance at states between the saturated liquid and saturated vapour states is referred to as a saturated liquid-vapour mixture, since the liquid and vapour phases coexist in equilibrium at these states. A vapour which is not about to condense is called a superheated vapour.

3.3 Thermodynamics Steam Power Cycle

3.3.1 Rankine Cycle

The Rankine cycle is a vapor power cycle that forms the thermodynamics basis for most steam power plants. These plants may use coal, oil, gas, or nuclear power as fuel for a high temperature source, but the basic thermodynamic operation remains fairly constant. The thermal efficiency of a power cycle is maximized if all the energy supplied from a source occurs at the maximum possible temperature and all the energy rejected to a sink occurs at the lowest possible temperature.

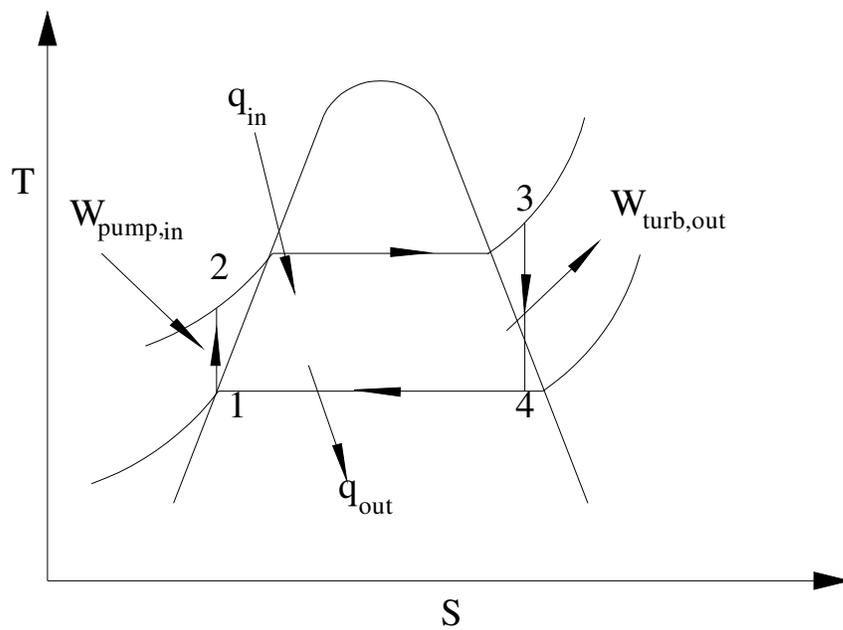
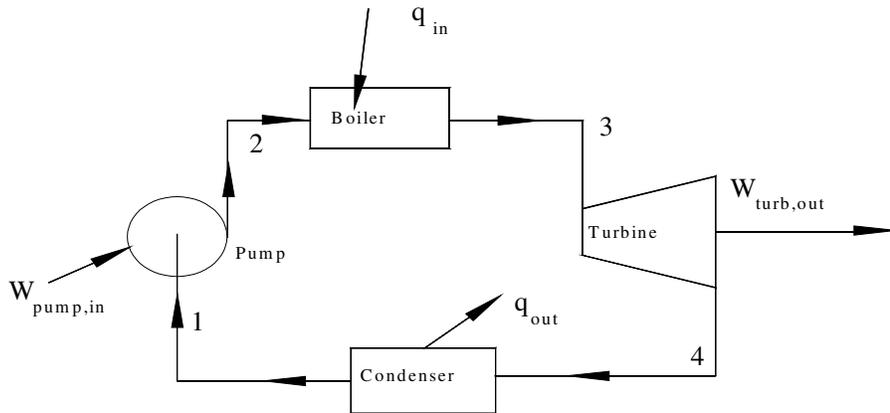


Fig.3.1 Rankine cycle.

The processes involved are isentropic compression in a pump (1-2), constant pressure heat addition in a boiler (2-3), isentropic expansion in a turbine (3-4), and constant pressure heat rejection in a condense (4-1). The cycle that results in these processes is the Rankine cycle.

The pump, boiler, and condenser associated with a Rankine cycle are steady-flow devices, and thus all the processes of this cycle can be analyzed as steady-flow processes. The Δke and Δpe of the steam are usually small compared with the work and heat transfer terms and are, therefore, neglected. Thus, the steady-flow energy equation per unit mass of steam is

$$q - w = h_e - h_i \quad (3.1)$$

Assuming the pump and turbine to be isentropic and noting that there is no work associated with the boiler and the condenser, the energy conservation relation for each device becomes:

$$W_{pump,in} = h_2 - h_1 = v^*(p_2 - p_1) \quad (3.2)$$

$$q_{boil,in} = h_3 - h_2 \quad (3.3)$$

$$W_{turb,out} = h_3 - h_4 \quad (3.4)$$

$$q_{cond,out} = h_4 - h_1 \quad (3.5)$$

The thermal efficiency of the Rankine cycle is given by

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \quad (3.6)$$

Where:

$$w_{net} = q_{in} - q_{out} = W_{turb,out} - W_{pump,in}$$

Like many other cycles, the Rankine cycle relies on the isentropic expansion of high pressure gas to produce work. The gas of choice for most Rankine cycles is steam. The system operates on the premise

that the easiest way to make high-pressure steam is to start with high pressure water and then heat that water at constant pressure. Since water is basically an incompressible liquid, it takes relatively little energy to compress it to high pressures, and operation within the vapor dome greatly simplifies control of steam properties.

3.3.2 Regenerative Cycle

In this cycle, the feed water heater is preheated by means of steam taken from some sections of the turbine, before it enters the boilers from the condenser. This process of draining steam from the turbine at certain points during its expansion and using this steam for heating the feed water supplied to the boiler, is known as "**bleeding**". The effect of this process is to supply the boiler with hotter water while a small amount of work is lost. There is a slight increase, in efficient but there is also a decrease in the power developed. The incidental advantages of improved thermal efficiency and reduced steam flow to the condenser are:

1. Smaller condenser and boiler;
2. The difficulty of passing large volumes of steam through the last stage in the low pressure turbine is lessened;
3. Improved turbine drainage, hence less trouble from erosion;
4. Increased blade height in the high pressure turbine to accommodate the initial increased steam consumption.

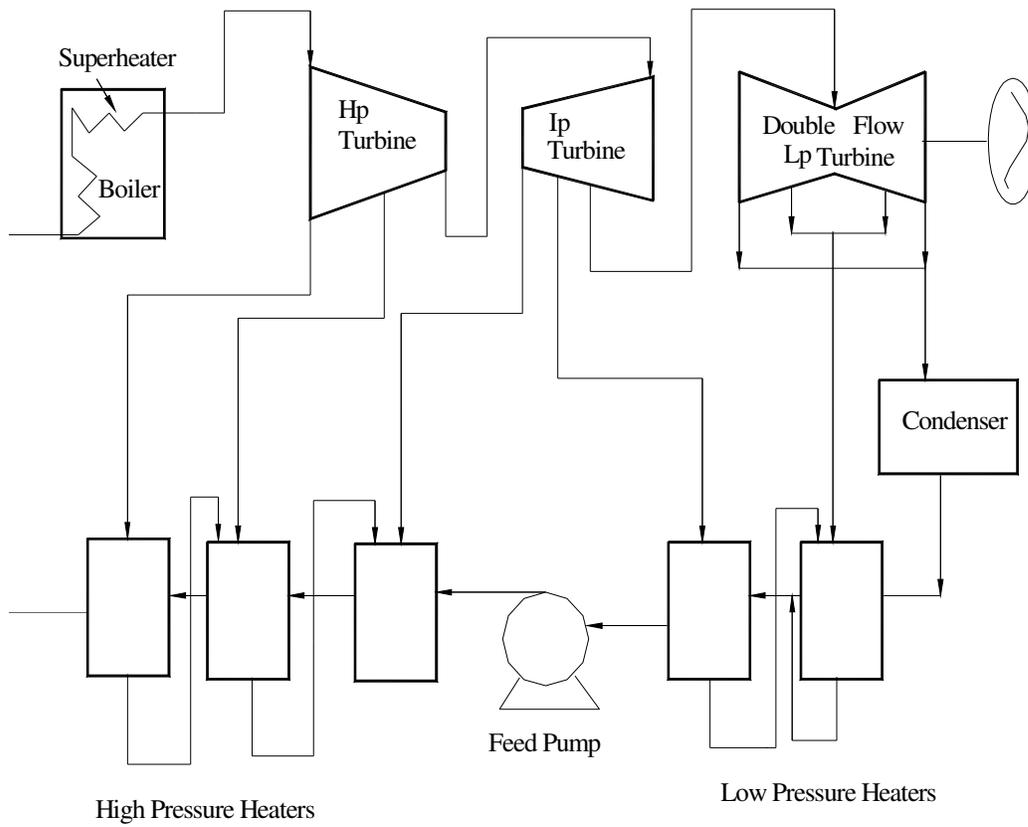


Fig.3.2 Regenerative cycle

CHAPTER FOUR

MATHEMATICAL MODEL OF STEAM POWER PLANT

COMPONENTS

4.1 Boiler Module

A steam generator or a boiler is a combination of systems and equipment in which the chemical energy of fossil fuels is converted into thermal energy which is then transferred to a working fluid (water) to convert it into steam at high pressure and temperature. This high pressure and temperature steam is then used for the development of power in a steam engine or turbine.

A boiler unit consists of the following elements: boiler evaporating section, superheater, reheater, economizer, air-preheater, furnace, refracting lining, framework, fittings, connecting piping and passages. The various heat transfer surfaces or elements can be grouped into two parts:

- 1. Primary heating elements:** include the evaporator, superheater and reheater surfaces.
- 2. Secondary heating elements:** include the preheater and economizer.

The air supply for combustion is forced through the furnace by the draught fan. The air is first heated, by passing it through air preheater, with the help of outgoing flue gases. The hot flue gases from the combustion chamber are drawn in by induced draught fan and are finally passed to the chimney. The heat is absorbed from the flue gases in the following order.

- 1.** The hot gases flow upwards through the combustion chamber which is surrounded with the water walls and evaporate the water to steam.

2. Then the hot flue gases pass over the tubes of the superheater, superheating the steam.
3. The hot gases may then pass over the steam reheater where the steam drawn from the turbine is re-superheated.
4. Then the hot gases pass through the economizer where heat is absorbed to raise the temperature of the feed water.
5. Finally heat is absorbed from the gases as they flow over the air preheater tubes.

The water circuits are as given below:

1. The feed water is pumped from the hot well by the feed pump.
2. Then it passes through the feed water heaters, where its temperature is raised by means of bled steam from the turbines.
3. It is further heated by outgoing flue gases in the economizer.
4. The hot feed water then passes through the water walls and water tube nests and is evaporated to saturated steam.
5. The saturated steam is next superheated by passing it through the superheaters.

In the above heat exchange devices, water or steam and flue gases are assumed to flow in opposite directions.

4.1.1 Superheater Module

The function of the superheater is to remove the last traces of moisture from the saturated steam leaving the boiler tubes and also to increase its temperature above the saturated temperature. The superheater is a control volume operating at constant pressure.

Most thermal systems including heat exchangers, expressing the performance of an existing heat exchanger is a frequent requirement. A typical situation encountered in system simulation is to calculate to outlet-fluid temperatures and the rate of heat transfer when the area,

overall heat transfer coefficient and inlet temperatures of the fluids are known.

The relation for a counter flow superheater with symbols as shown in Fig.4.1 will be presented first.

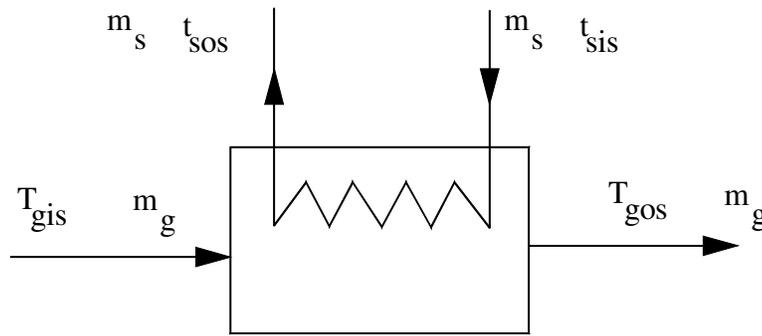


Fig.4.1. Counter flow superheater

Three equations for the rate of heat transfer q are:

$$q = U * A * \Delta Tlms \quad (4.1)$$

$$q = \dot{m}_s * cp_w * (t_{sos} - t_{sis}) \quad (4.2)$$

$$q = \dot{m}_g * cp_g * (T_{gis} - T_{gos}) \quad (4.3)$$

and the log mean temperature difference, $\Delta Tlms$

$$\Delta Tlms = \left[\frac{(T_{gis} - t_{sis}) - (T_{gos} - t_{sos})}{\ln \left(\frac{T_{gis} - t_{sis}}{T_{gos} - t_{sos}} \right)} \right] \quad (4.4)$$

Where: U = the overall heat transfer coefficient

A = the heat transfer area

\dot{m}_s =the steam mass flow rate

\dot{m}_g =mass flow rate of the flue gases

Here the modeling is important to find the steady-state outlet temperature of the water, t_{so} , and flue gases, T_{go} , and the heat transfer area A , given the inlet temperatures, flow rates and heat transfer rate and overall heat transfer coefficient.

4.1.2 Economizer module

When the combustion gases leave the boiler after giving most of their heat to water tubes, superheater tubes and reheater tubes, they still possess lot of heat which if not recovered by means of some devices, would go waste. Economizer is one of such heat recovering devices in which the temperature of the feed water is raised by means of outgoing flue gases, before it is delivered to boiler drums.

The same as superheater, the heat transfer equation for the economizer becomes

$$q = U * A * \Delta T_{lme} \tag{4.5}$$

$$q = \dot{m}_w * c_{p_w} * (t_{soec} - t_{siec}) \tag{4.6}$$

$$q = \dot{m}_g * c_{p_g} * (T_{giec} - T_{goec}) \tag{4.7}$$

and the log mean temperature difference, ΔT_{lme} for the economizer

$$\Delta T_{lme} = \frac{(T_{giec} - t_{soec}) - (T_{goec} - t_{siec})}{\ln \left(\frac{T_{giec} - t_{soec}}{T_{goec} - t_{siec}} \right)} \tag{4.8}$$

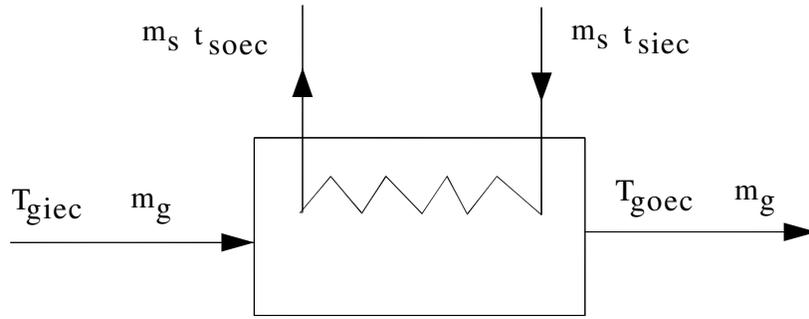


Fig.4.2 Economizer

Where: t_{siec} =the steam or water temperature entering to the economizer

t_{soec} =the steam or water temperature leaving from the economizer

T_{giec} =the temperature of the flue gases entering to the economizer

T_{goec} =temperature of the flue gases leaving from the economizer

4.1.3 Reheater Module

The function of the reheaters is to resuperheat the partly expanded steam from the turbine. This is done so that the steam remains dry as far as possible through the last stage of the turbine. Modern plants have reheater as well as superheaters in the same as passage of the boiler.

The same as the superheater heat transfer q_{rh} from the gases to the steam can be given by the following equations:

$$q_{rh} = \dot{m}_s * c_{p_s} * (t_{soh} - t_{sih}) \quad (4.9)$$

$$q_{rh} = \dot{m}_g * c_{p_g} * (T_{gih} - T_{goh}) \quad (4.10)$$

The heat load equation using the overall heat transfer coefficient U:

$$q_{rh} = U * A * delTlmrh \quad (4.11)$$

and the log mean temperature difference, delTlmrh

$$\text{delTlmrh} = \left[\frac{(T_{gih} - t_{sih}) - (T_{goh} - t_{soh})}{\ln \left(\frac{T_{gih} - t_{sih}}{T_{goh} - t_{soh}} \right)} \right] \quad (4.12)$$

Where: t_{soh} =the temperature of the steam leaving the reheater

t_{sih} =the temperature of the steam entering the reheater

A_{rh} =the heat transfer area for the reheater

4.1.4 Air Pre_heater Module

The air required in the boiler for combustion is preheated in the preheaters by means of flue gases leaving the economizer and thus the heat in the outgoing gases is recovered. The benefits of the air preheating are intensified and improved fuel combustion, increased thermal efficiency, and due to regenerative feed water heating, the inlet feed water temperature to economizer is quite high and it becomes impossible to recover large heat from flue gases.

The thermal analysis on the air pre_heater will give the following model equations:

The heat transfer q_{ah} from the gases to the air can be given by:

$$q_{ah} = \dot{m}_a * cp_a * (t_{oah} - t_{amb}) \quad (4.13)$$

$$q_{ah} = \dot{m}_g * cp_g * (T_{giah} - T_{goah}) \quad (4.14)$$

The log mean temperature difference, delTlmah

$$\text{delTlmah} = \left[\frac{(T_{giah} - t_{oah}) - (T_{goah} - t_{amb})}{\ln \left(\frac{T_{giah} - t_{oah}}{T_{goah} - t_{amb}} \right)} \right] \quad (4.15)$$

Using the overall heat transfer coefficient U_{ah} , the heat load will be:

$$q_{ah} = U_{ah} * A_{ah} * \Delta T_{lmah} \quad (4.16)$$

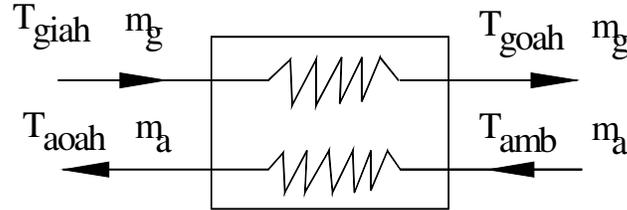


Fig.4.3 Air pre_heater

Where: t_{oah} = temperature of the outlet air

t_{amb} = temperature of the ambient air

T_{giah} = temperature of the inlet flue gases to the air pre-heater

T_{goah} = temperature of the outlet flue gases from the pre-heater

\dot{m}_a and \dot{m}_g = the mass flow rate of the air and the flue gases respectively.

4.2. Feed Water Heater Module

In the regenerative cycle, steam is extracted from the steam turbine at various stages and used to heat the feed water. This result in higher cycle efficiency by increasing the temperature of the feed water and by reducing the amount of energy lost in the condenser.

4.2.1 Open Feed Water Heater

Open feed water heater or deaerator is a contact device where extraction steam and sub cooled feed water are mixed directly. By necessity, both must be at the same pressure. The open feed water heater is a multiple stream control volume with continuity equation and steady-state, steady-flow energy equation as shown in the figure.4.3.No real law says that the output must be saturated liquid, but it generally is, and this is specified in

the heater description as “ $TTD=0$ ”. The terminal temperature difference, or TTD, is simply the difference between the saturation temperature corresponding to pressure (p) and the exit temperature of the liquid.

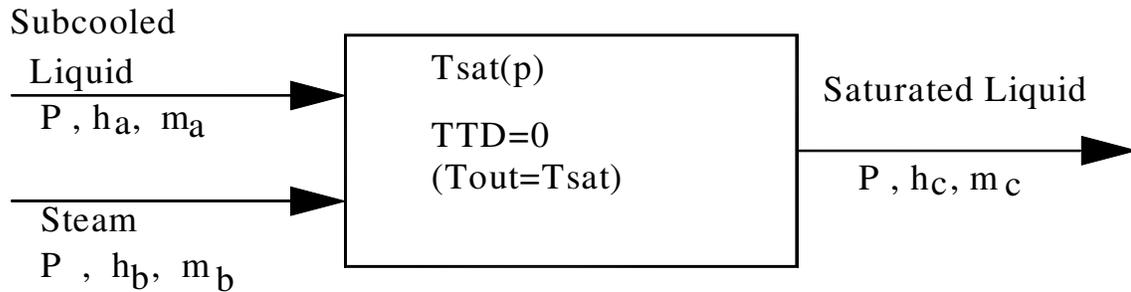


Fig.4.4 An open feed water heater

Energy and mass balance equation of the above control volume of the open feed water heater:

Continuity Equation:

$$\dot{m}_c = \dot{m}_a + \dot{m}_b \quad (4.17)$$

Energy Equation:

$$\dot{m}_c * h_c = \dot{m}_a * h_a + \dot{m}_b * h_b \quad (4.18)$$

Where:

\dot{m}_a and h_a = the mass flow rate and enthalpy of the sub cooled liquid

\dot{m}_b and h_b = the mass flow rate and enthalpy of the steam

\dot{m}_c and h_c = the mass flow rate and enthalpy of saturated liquid

4.2.2 Closed Feed Water Heater

Closed feed water heater is a surface heat exchanger with the feed water inside the tubes, and the extraction steam condensing on the outside of the tubes. The closed feed water heater can operate with different pressures for the extraction steam and the feed water. The heater is fully

specified by the three equations shown in the figure for TTD, DCA, and steady flow energy.

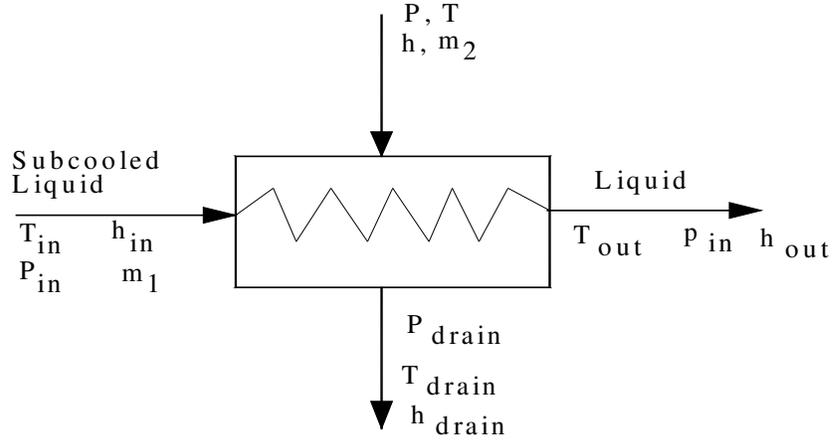


Fig.4.5 Closed feed water heater

Steady flow energy equation:

$$\dot{m}_1 * (h_{out} - h_{in}) = \dot{m}_2 * (h - h_{drain}) \tag{4.19}$$

Terminal temperature difference:

$$TTD = T_{sat} - T_{out} \tag{4.20}$$

Drain cooler energy equation:

$$DCA = T_{drain} - T_{in} \tag{4.21}$$

Where:

h_{out}, h_{in} = enthalpy of outlet and inlet of sub cooled liquid.

4.3. Pump Module

A pump is a machine that imparts energy into a liquid to lift the liquid to a higher level, to transport the liquid from one place to another, to pressurize the liquid for some useful purpose, or to circulate the liquid in a piping system by overcoming the frictional resistance of the piping system.

The most common type of kinetic pumps used in a modern central power plant is the centrifugal pump. Centrifugal pumps include radial, axial, and mixed flow types, with the radial flow volute type used for the bulk of power plant applications. Regenerative pumps are typically referred to as vortex, peripheral, or turbine pumps.

Pump performance curve are well suited for representation by a quadratic of the form [1]:

$$H = a_0 + a_1 * Q + a_2 * Q^2 \quad (4.22)$$

The head (H) developed by the centrifugal pump is a function of volume flow rate (Q). Three pair of information, (H_1, Q_1) , (H_2, Q_2) , and (H_3, Q_3) chosen in the range of interest can be substituted into equation (4.22) to yield the linear system;

$$\begin{aligned} H_1 &= a_0 + a_1 * Q_1 + a_2 * Q_1^2 \\ H_2 &= a_0 + a_1 * Q_2 + a_2 * Q_2^2 \\ H_3 &= a_0 + a_1 * Q_3 + a_2 * Q_3^2 \end{aligned} \quad (4.23)$$

Equation (4.23) can then be solved for coefficient a_0, a_1 , and a_2 by the matrix:

$$\begin{bmatrix} a_0 \\ a_1 \\ a_2 \end{bmatrix} = \begin{bmatrix} 1 & Q_1 & Q_1^2 \\ 1 & Q_2 & Q_2^2 \\ 1 & Q_3 & Q_3^2 \end{bmatrix} * \begin{bmatrix} H_1 \\ H_2 \\ H_3 \end{bmatrix} \quad (4.24)$$

The output of a pump for different flow rates and speeds (N, rpm) would have the form $H = f(Q, N)$. One satisfactory polynomial representation is

$$H = a_0(N) + a_1(N) * Q + a_2(N) * Q^2 \quad (4.25)$$

With the coefficient a_i ($i=0, 1, 2$)

$$a_i(N) = b_{i0} + b_{i1} * N + b_{i2} * N^2 \quad (4.26)$$

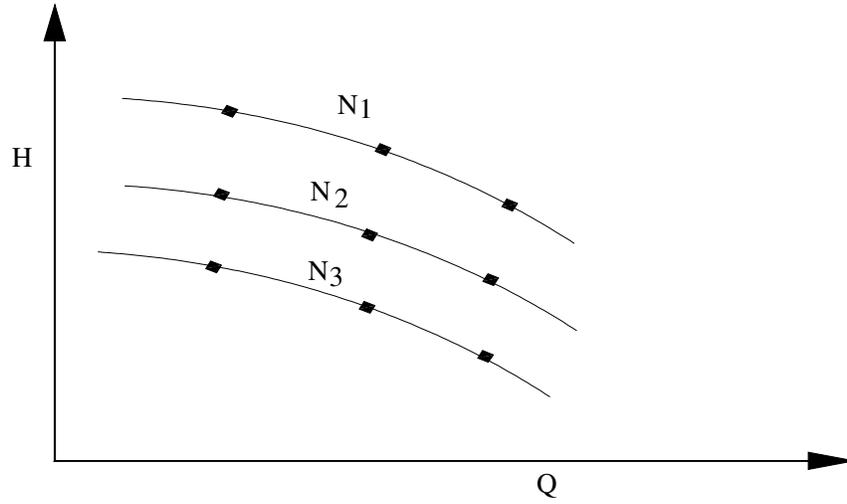


Fig.4.6 H as a function of two independent variables

The dependence on the speed in equation (4.25) is continued within the coefficients. Fig.4.5 illustrates this situation. The nine points shown on the sketch; three at each speed, are required to determine the coefficients in equation (4.25). At speed N_1 a quadratic curve fit using equation (4.25) yields

$$H_1 = a_{10} + a_{11} * Q_1 + a_{12} * Q_1^2 \quad (4.27a)$$

and at speed N_2 and N_3 the analogous expressions are

$$H_2 = a_{20} + a_{21} * Q_2 + a_{23} * Q_2^2 \quad (4.27b)$$

and

$$H_3 = a_{30} + a_{31} * Q_3 + a_{33} * Q_3^2 \quad (4.27c)$$

The same curve fitting procedure is used to evaluate the coefficients in the quadratic expression for $a_0(N)$:

$$a_0(N) = b_{00} + b_{01} * N + b_{02} * N^2 \quad (4.28)$$

The pairs (a_{10}, N_1) , (a_{20}, N_2) , and (a_{30}, N_3) are used to evaluate b_{00}, b_{01} , and b_{02} . The process is then repeated for $a_1(N)$ to obtain b_{10}, b_{11} , and b_{12} and for $a_2(N)$ to obtain b_{20}, b_{21} , and b_{22} . The final expression for H as a function of flow rate and speed then becomes

$$H = (b_{00} + b_{01} * N + b_{02} * N^2) + (b_{10} + b_{11} * N + b_{12} * N^2) * Q + (b_{20} + b_{21} * N + b_{22} * N^2) * Q^2 \quad (4.29)$$

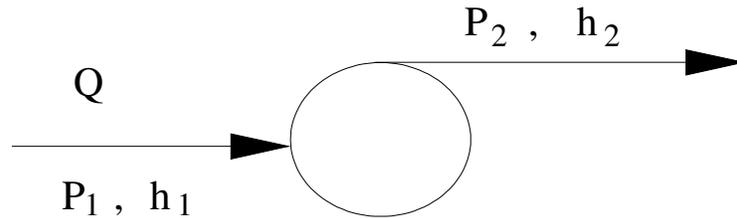


Fig.4.7 Pump in fluid flow diagram

The pump is basically a control volume that adiabatically increases the pressure of a liquid. The steady- state, steady –flow energy equation for this device reduces the following:

$$W_p = \dot{m} * (h_1 - h_2) \quad (4.30)$$

Where:

W_p = pump power, and

\dot{m} = mass flow rate of the fluid

If the process is considered as an incompressible liquid, the Gibbs equation, $Tds=dh-vdp$, yields the following:

$$h_2 - h_1 = v_f * (p_2 - p_1) \quad (4.31)$$

Where:

h_1 and h_2 = the enthalpy of the inlet and outlet fluid

p_2 and p_1 = the discharge and suction pressure of the pump

v_f = the specific volume of the fluid

The specific volume of water is very stable for constant temperature and is available from steam tables.

4.4 Condenser Module

The function of the condenser is to condensate the steam leaving the turbine, collect the condensate, and lower the turbine exhaust pressure.

The condensing of the steam requires the condenser to remove the heat of vaporization from the steam and reject it. Condensers are designed to reject this energy directly into cooling water or directly into the atmosphere. Condensers using cooling water are the norm, except in a few locations where water is very expensive, in that case air-cooled condenser may be used.

The condenser is a steam-to-water, tube and shell heat exchanger with the cooling water, normally called circulating water, passing through the tubes and the steam in the shell.

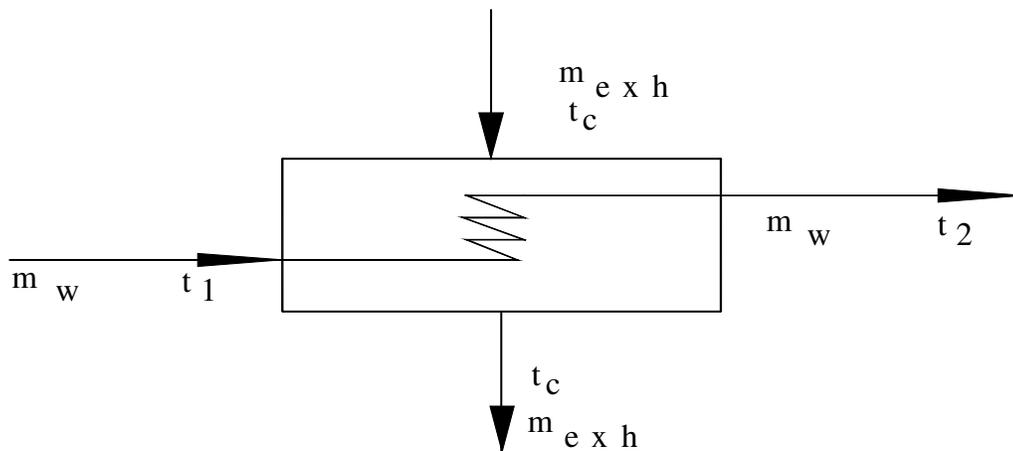


Fig.4.8 Condenser where one fluid remains at a constant temperature

In condenser, with no superheating or sub cooling of the fluid that changes phase, the condensing or evaporating fluid remains at a constant temperature, provided that its pressure does not change.

The heat transfer equation becomes

$$q = U * A * \Delta Tlms \quad (4.32)$$

$$q = \dot{m}_w * cp_w * (t_2 - t_1) \quad (4.33)$$

$$\Delta Tlms = \left(\frac{(t_c - t_1) - (t_c - t_2)}{\ln \left(\frac{t_c - t_1}{t_c - t_2} \right)} \right) \quad (4.34)$$

Applying the effectiveness concept to a condenser,

$$q = e * (\dot{m} * cp)_{\min} * (t_c - t_1) \quad (4.35)$$

$$q = (\dot{m} * cp)_{\min} * (t_2 - t_1) \quad (4.36)$$

and since $(\dot{m} * cp)_2$ is infinite, $(\dot{m} * cp)_1 = (\dot{m} * cp)_{\min}$

4.5. Steam Turbine Model

The steam turbine generator is the primary power conversion component of the power plant. The function of the steam turbine generator is to convert the thermal energy of the steam from the steam generator to electrical energy.

The thermal energy of the steam is converted to mechanical energy by expanding the steam through the turbine. The rotational mechanical energy is converted to electrical energy in the generator by the rotation of the rotor's magnetic field.

4.5.1. Steam Turbine Types

These are different ways of classifying steam turbines. The commonly used turbine types are described in the following sections:

1. Reheat turbines:

The reheat turbine returns its steam after partial expansion to the boiler for resuperheating and then expands it to back pressure.

2. Condensing or non-condensing turbines:

The condensing turbines exhausts to a condenser where the steam is condensed at sub atmospheric pressure(vacuum).The non-condensing turbine exhausts the steam above atmospheric pressure into a line for supply to the boiler, another turbine, or a process.

3. Bleeder or extraction turbines:

During the expansion of steam through a turbine, removing steam from an intermediate stage of the turbine is called an *extraction*. In most power plants, some steam is extracted from the turbine expansion process and supplied to heat exchangers for feed water heating.

4.5.2. Steam Turbine Module

Steam cycle performance is imparted by numerous designs and operating parameters .The operating parameters includes among others, main steam pressure and temperature, reheater system pressure drop, reheat temperature, and turbine back pressure.

Heat balances typically indicate the turbine expansion line end point and used energy end point of the steam as it exhaust to the condenser.

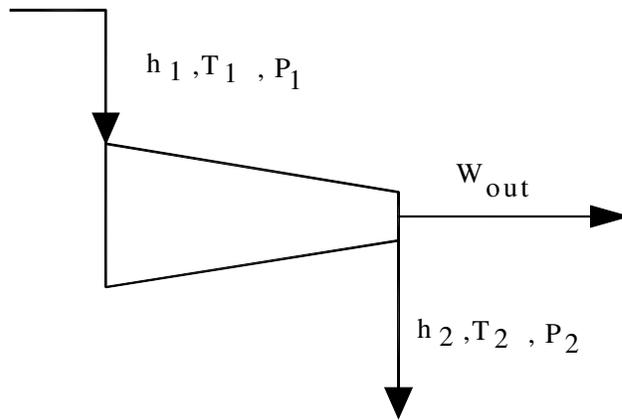


Fig.4.9 Steam Turbine

Energy balance for an adiabatic expansion in the steam turbine:

$$h_1 - h_2 = W_{out} \quad (4.37)$$

The efficiency of thermal cycle is the ratio of the work done to heat supplied.

$$\eta = \frac{\text{Net work done}}{\text{Heat supplied}} \quad (4.38)$$

$$\eta = \frac{h_1 - h_2}{h_1 - h_w}$$

Where: h_1 = enthalpy of entering steam

h_2 = enthalpy of exhaust steam

h_w = enthalpy of entering to the boiler

Superheated steam can be expanded in a high pressure turbine and exhaust reheated and then expanded in the low pressure turbine.

In this case the efficiency of the thermal cycle will be

$$\eta = \frac{\text{Net work done}}{\text{Total heat supplied}} \quad (4.39)$$

$$\eta = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_w) + (h_3 - h_2)}$$

Where: $h_1 - h_w$ = the heat supplied in the boiler

$h_3 - h_2$ = heat added in the reheater at constant pressure

h_3 = enthalpy leaving the reheater

h_2 = enthalpy exhaust from high pressure turbine

The **Turbine heat rate, HR** is defined as the heat energy needed by it to produce one unit of output. It is given as KJ per kw.hr. Assuming that the steam flow to turbine and the feed flow to the boiler are equal, the heat rate for a non-reheat turbine can be written as,

$$HR = \frac{\dot{m}_1 * (h_1 - h_w)}{\text{Output in Kw}}, (\text{kJ/kw.hr}) \quad (4.40)$$

Where: \dot{m}_1 = steam flow and feed flow, kg/hr

h_1 = total heat of steam entering to turbine, kJ/kg

h_w = enthalpy of final feed water entering the boiler, kJ/kg

In the case of **non-regenerative** turbine,

h_w = saturated feed water enthalpy at exhaust pressure and

For a **regenerative** turbine

h_w = saturated feed water enthalpy leaving the last feed water heater

For a re-heat unit,

$$HR = \frac{\dot{m}_1 * (h_1 - h_2) + \dot{m}_2 * (h_3 - h_2)}{\text{Output in Kw}}, (\text{kJ/kw.hr}) \quad (4.41)$$

Where: \dot{m}_2 = steam flow through reheater, kg/hr

h_3 = enthalpy at reheater, kJ/kg

h_2 = enthalpy of steam to reheater, kJ/kg

Energy balance on steam turbine having a number of bled steam or stage considered:

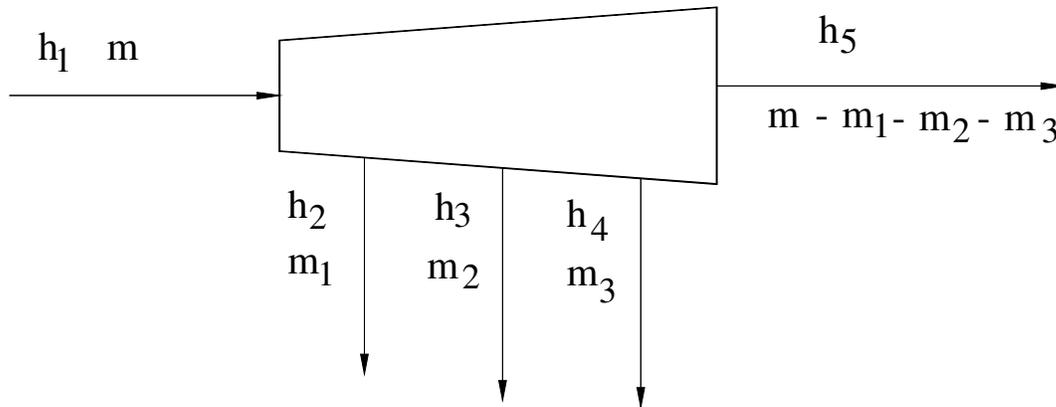


Fig.4.10 Turbine with different stages

Total work done=Work done between stages

$$w_{tt} = m*(h_1 - h_2) + (m - m_1)*(h_2 - h_3) + (m - m_1 - m_2)*(h_3 - h_4) + (m - m_1 - m_2 - m_3)*(h_4 - h_5) \quad (4.42)$$

Total heat supplied of feed water= $m*(h_1 - h_w)$ (4.43)

Therefore, the **efficiency** of the cycle will be

$$\eta = \frac{\text{Net work done}}{\text{Total heat supplied}} \quad (4.44)$$

CHAPTER FIVE

Simulation of steam power plant

5.1 Boiler Simulation

The boiler has a number of components, modeling of the individual components and assembling the component models into a system model is essentially the task of the boiler simulation process.

The mathematical formulation for much of our understanding of the thermal sciences rests on the concept of conservation of physically meaningful quantities as applied to control volumes. It should therefore be no surprise that the starting point for the development of many models involves control volume considerations.

The total heat Q_t supplied to the flue gases from the furnace can be calculated by the equation

$$Q_t = m_g * cp_g * (T_g - t_a) \quad (5.1)$$

Where: m_g =the mass flow rate of dry flue gases

cp_g =mean specific heat capacity of flue gases

T_g =temperature of flue gases leaving the furnace

t_a =temperature of the inlet air for the combustion

The equations to find the heat rate q absorbed by the individual components are:

1. Evaporator

$$q_{ev} = m_s * (h_{fg}) = m_s * (h_g - h_f) \quad (5.2)$$

2. Superheater

$$q_s = m_s * (h_{ss} - h_g) \quad (5.3)$$

3. Reheater

$$q_h = m_s * (h_{extrh} - h_{extrho}) \quad (5.4)$$

4. Economizer

$$q_{ec} = m_s * (h_{out(1)} - h_f) \quad (5.5)$$

5. Air Pre-heater

$$q_{ah} = Qt - (q_{ev} + q_s + q_{rh} + q_{ec}) \quad (5.6)$$

The heat transfer equation (5.6) for air_preheater is the assumption that the remain of the total heat is absorbed by the air, neglecting the loss of heat elsewhere.

Where: $q_{ev}, q_s, q_{rh}, q_{ec}$, and q_{ah} = the heat absorbed by evaporator, superheater, reheater, and economizer respectively.

m_s = mass flow rate of the steam from the boiler

h_f, h_g, h_{fg} = the enthalpy of the steam/water for the given pressure

$h_{out(1)}$ = the enthalpy water feed to the economizer outlet of the first heater

h_{ss} = the enthalpy of the steam out of the superheater

h_{extrh} = the enthalpy of the steam to the reheater

h_{extrho} = the enthalpy of the steam out of the reheater

Heat transfer equations:

1. Evaporator

$$q_{ev} = m_g * cp_g * (T_{giev} - T_{goev}) \quad (5.7)$$

$$delTlmev = \left[\frac{(T_{giev} - t_c) - (T_{goev} - t_c)}{\ln\left(\frac{T_{giev} - t_c}{T_{goev} - t_c}\right)} \right] \quad (5.8)$$

$$q_{ev} = U_{ev} * A_{ev} * delTlmev \quad (5.9)$$

2. Superheater

$$q_s = m_s * cp_s * (t_{sos} - t_{sis}) \quad (5.10)$$

$$q_s = m_g * cp_g * (T_{gis} - T_{gos}) \quad (5.11)$$

$$delTlms = \left[\frac{(T_{gis} - t_{sis}) - (T_{gos} - t_{sos})}{\ln\left(\frac{T_{gis} - t_{sis}}{T_{gos} - t_{sos}}\right)} \right] \quad (5.12)$$

$$q_s = U_s * A_s * delTlms \quad (5.13)$$

3. Reheater

$$q_{rh} = m_s * cp_s * (t_{sorh} - t_{sirh}) \quad (5.14)$$

$$q_{rh} = m_g * cp_g * (T_{girh} - T_{gorh}) \quad (5.15)$$

$$delTlmrh = \left[\frac{(T_{gih} - t_{sih}) - (T_{goh} - t_{soh})}{\ln\left(\frac{T_{gih} - t_{sih}}{T_{goh} - t_{soh}}\right)} \right] \quad (5.16)$$

$$q_{rh} = U_{rh} * A_{rh} * delTlmrh \quad (5.17)$$

4. Economizer

$$q_{ec} = m_s * cp_g * (t_{siec} - t_{soec}) \quad (5.18)$$

$$q_{ec} = m_g * cp_g * (T_{giec} - T_{goec}) \quad (5.19)$$

$$delTlmech = \left[\frac{(T_{giec} - t_{siec}) - (T_{goec} - t_{soec})}{\ln\left(\frac{T_{giec} - t_{siec}}{T_{goec} - t_{soec}}\right)} \right] \quad (5.20)$$

$$q_{ec} = U_{ec} * A_{ec} * delTlmech \quad (5.21)$$

5. Air Pre-heater

$$q_{ah} = m_a * cp_a * (t_{oah} - t_{amb}) \quad (5.22)$$

$$q_{ah} = m_g * cp_g * (T_{giah} - T_{goah}) \quad (5.23)$$

$$delTlmah = \left[\frac{(T_{giah} - t_{oah}) - (T_{goah} - t_{amb})}{\ln\left(\frac{T_{giah} - t_{oah}}{T_{goah} - t_{amb}}\right)} \right] \quad (5.24)$$

$$q_{ah} = U_{ah} * A_{ah} * delTlmah \quad (5.25)$$

Where:

m_s, m_a, m_g = the mass flow rate of the steam, air and flue gases respectively.

$T_{giev}, T_{gis}, T_{gih}, T_{giec},$ and T_{giah} = the temperature of the flue gases entering the evaporator, superheater, heater, economizer, and air heater respectively.

$T_{goev}, T_{gos}, T_{goh}, T_{goec}, \text{and}, T_{goah}$ = the temperature leaving the evaporator, superheater, reheater, economizer, and air heater.

$t_{sis}, t_{sih}, t_{siec}$ = the temperature of the steam entering the superheater, heater, and the economizer respectively.

$t_{sos}, t_{soh}, t_{soec}$ = the temperature of the steam/water leaving the superheater, heater, economizer resp.

t_{amb}, t_{oah} = the temperature of air inlet and outlet from the air heater

$delTlmev, delTlms, delTlmrh, delTlmeec, delTlmah$ = the log mean temperature difference of evaporator, superheater, reheater, and economizer and air heater respectively.

$A_{ev}, A_s, A_{rh}, A_{ec}, A_{ah}$ = heat transfer area for the evaporator, superheater, reheater, economizer & air_reheater respectively.

The first equations above for each component except for the air preheater the outlet steam or water temperature is known from steam table. Then reducing the above equations and writing them as a function of f or residue function R, 16 equations with 16 unknowns will be formed. The residue function {R} will be the right column matrix.

The Newton – Raphson method will now be used to find the values of the variables that reduce R to zero. First select trial values of the variables and compute the R value. Next, compute the partial derivatives of R with respect to the variables which is used to evaluate the matrix coefficient [pd], the Jacobean matrix. The partial derivatives become the coefficients and the residue subroutines {R} become the right column in the linear simultaneous equations.

$$\{pd\}\{\Delta V\} = \{R\} \quad (5.26)$$

Where: ΔV is the change of the variables.

The correction process consists of solving a set of 16 linear simultaneous equations, where the unknowns are the respective corrections to each

variable, the coefficients are the partial derivatives, and the numerical terms on the right side of the equations are the values of the functions computed with the current values of variables.

The partial derivatives are evaluated at the current values of the variables. Solution of the above simultaneous equations provides a value of $\{\Delta V\}$. The results of the successive refinements of the variables are continued until no function exceeds an absolute magnitude of 0.01.

5.2 Turbine – Heater Simulation

The modern steam power plant consists of high pressure turbine, intermediate pressure turbine and low pressure turbine. The turbines are extraction condensers type and the components in the turbine plant include condenser, feed water heaters and pumps. Generally, the relationships among thermodynamic properties are too complex to be expressed by simple equations. Therefore, thermodynamic properties are often represented in the form of tables. These tables are extremely useful for quantitative calculations. The intensive properties which are of primary interest in thermodynamic analysis of systems are pressure p , specific volume v , temperature T , specific internal energy u , specific enthalpy h and specific entropy s .

In modeling of steam turbine conservation laws, mass and energy conservation equations are used. The first step in cycle calculations is to determine heater shell pressure. While pressure of the extraction steam at the turbine is known, a pressure drop will occur in the extraction steam piping between the turbines. But since the pressure drop is small, here for our calculations the shell pressure is taken as the pressure of the extraction steam.

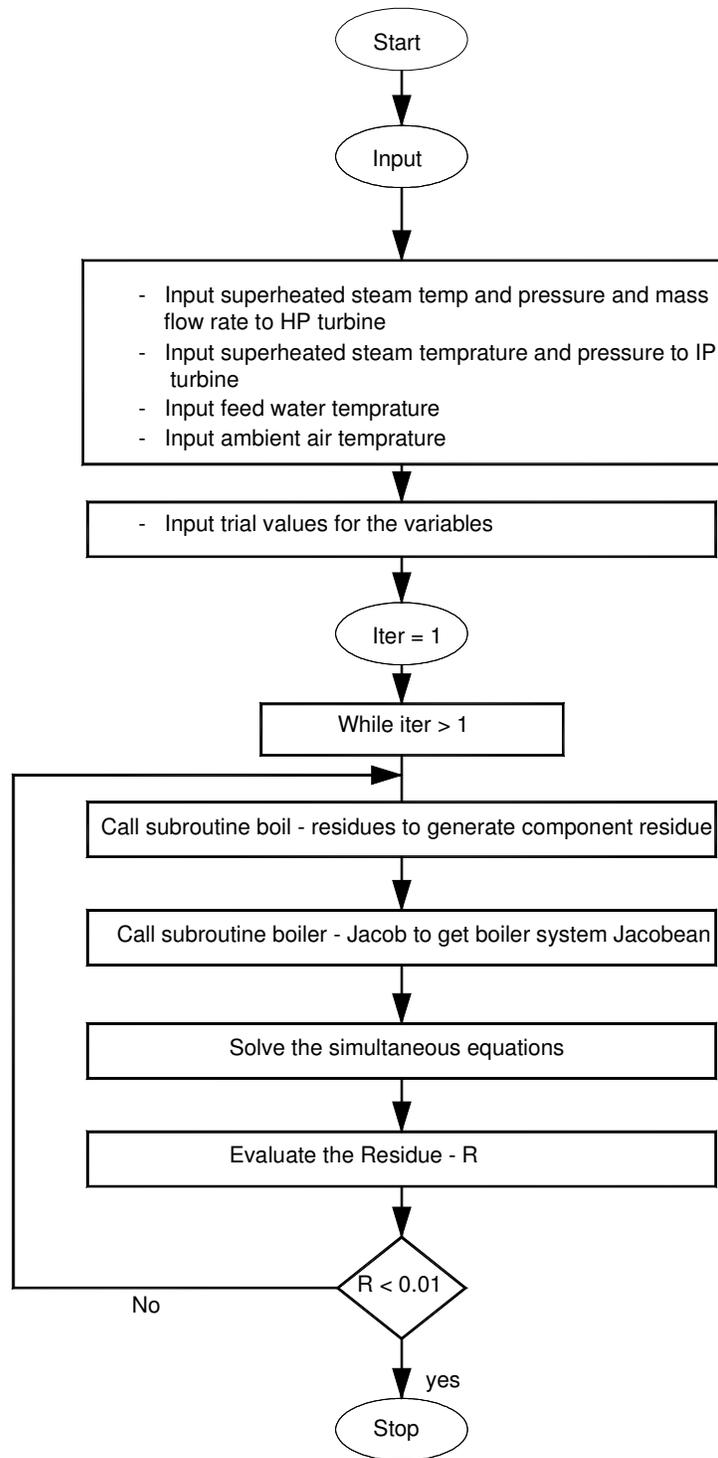


Fig.5.1 Flow chart for the simulation of boiler

A steam table can be used to determine the saturation temperature in the heater shells that corresponds to the shell pressure. By use of the established terminal temperature difference, the temperature and enthalpy of the feed water exiting each heater can be determined [8].

In the steam power cycle the temperature of the condensate or the feed water entering and leaving all the heaters is now known, except for that entering high pressure heater(hph) after the deaerator and low pressure heater(lph) after the condenser which condensate comes from the condenser, from that leaves at the saturation temperature corresponding to the condenser pressure. The condensate pump increases the condensate temperature, but this is minor and normally neglected [8].

The boiler feed pump also increases the boiler feed water temperature and enthalpy as it passes through the pump. This temperature rise is high enough not to be neglected, particularly on large high-pressure units. The rise in enthalpy through the pump can be estimated using the following formula:

$$\Delta h = V_f * (p_o - p_i) \quad (5.27)$$

Where: Δh = increase in enthalpy

V_f = specific volume of liquid at pump inlet conditions

p_o = pump outlet pressure

p_i = pump inlet pressure

The temperature of the feed water entering high pressure heater (hph) can be determined from its enthalpy using the compressed water portion of a standard steam table [8].

With the condensate or feed water temperature known for each of the heaters, the temperature of the drains from the drain coolers and their enthalpies can be established.

In the steam power plant cycle as shown in Fig.5.3, feed water heater 1 of the high pressure heaters heat balance is complete except for extraction flow and drain flow. As these two flows are equal, only one unknown remains.

Using the conservation of energy theory, a heat balance can be performed on this heater and the extraction flow determined as follows:

$$\text{Energy in} = \text{Energy out}$$

Feed water inlet flow × feed water inlet enthalpy + extraction flow × extraction enthalpy = feed water outlet flow × feed water outlet enthalpy + drain flow × drain enthalpy

$$(m_s * h_{in}) + (m_{ext} * h_{ext}) = (m_s * h_{out}) + (m_{ext} * h_{dr}) \quad (5.28)$$

Since drain flow is the same as the extraction flow, substituting extraction flow for drain flow, the equation can be solved for extraction flow:

$$\dot{m}_{ext}(1) = \left(\frac{\dot{m}_s * h_{out} - \dot{m}_s * h_{in}}{h_{ext} - h_{dr}} \right) \quad (5.29)$$

With the extraction and drain flows determined for heater 1, a similar heat balance can be performed for any number of high pressure heaters (nhph) until the deaerator.

In this case, drain flow from heater n equals extraction flow to heater n plus drains entering heater n from heater n-1.

$$\text{Energy in} = \text{Energy out}$$

$$\dot{m}_s * h_{in}(n) + \dot{m}_{ext}(n) * h_{ext}(n) + (\dot{m}_{ext}(2) + \dot{m}_{ext}(3) + \dots + \dot{m}_{ext}(n-1)) * h_{dr}(n-1) = \dot{m}_s * h_{out}(n) + (\dot{m}_{ext}(2) + \dot{m}_{ext}(3) + \dots + \dot{m}_{ext}(n-1) + \dot{m}_{ext}(n)) * h_{dr}(n)$$

Solving for the extraction flow for the high pressure heaters (hph) except heater 1, the following equation will be formed:

$$\dot{m}_{ext}(n) = \left[\frac{[\dot{m}_s * (h_{in}(n) - h_{out}(n)) + (\sum_{i=1}^{n-1} \dot{m}_{ext}(i)) * (h_{dr}(n-1) - h_{dr}(n))]}{(h_{dr}(n) - h_{ext}(n))} \right] \quad (5.30)$$

Proceeding to the deaerator, the extraction flow and the condensate flow from heater (nhph+2) after deaerator or the first low pressure heater (lph) are unknowns. However, by setting up an energy balance and a mass balance, two equations with two unknowns can be established that can be solved simultaneously giving both flows.

Mass in = Mass out

$$\dot{m}_{ext}(nhph+1) + \dot{m}_{in}(nhph+1) + \dot{m}_{dr}(nhph) = \dot{m}_s(\text{feed water out}) \quad (5.31)$$

Where: $\dot{m}_{dr}(nhph) = \dot{m}_{ext}(1) + \dot{m}_{ext}(2) + \dots + \dot{m}_{ext}(nhph)$ and

$$\dot{m}_{in}(nhph+1) = \dot{m}_{ext}(nhph+2) + \dot{m}_{ext}(nhph+3) + \dots + \dot{m}_{ext}(nhph+nlph+1) + \dot{m}_{exh}$$

The energy balance is as follows:

Energy in = Energy out

Condensate from heater (nhph+2) × condensate enthalpy + extraction steam × extraction steam enthalpy + drains from heater (nhph) × drain enthalpy = Feed water out × Feed water enthalpy or

$$\begin{aligned} & \dot{m}_{in}(nhph+2) * h_{in}(nhph+2) + \dot{m}_{ext}(nhph+1) * h_{ext}(nhph+1) + \dot{m}_{dr}(nhph) * h_{dr}(nhph) \\ & = \dot{m}_s * h_{dr}(nhph+1) \end{aligned}$$

Solving the above equation for the extraction flow to the deaerator, the next equation will be formed:

$$\dot{m}_{ext}(nhph+1) = \frac{(\dot{m}_s * (h_{dr}(nhph+1)) - \sum_{i=1}^{nhph} \dot{m}_{ext}(i) * h_{dr}(nhph) - (\dot{m}_s - \sum_{i=1}^{nhph} \dot{m}_{ext}(i)) * h_{out}(nhph+2))}{(h_{ext}(nhph+1) - h_{out}(nhph+2))} \quad (5.32)$$

Once the condensate flow is known, extraction flow can be calculated for low pressure heaters nhph+2, nhph+3 ... and nhph+nlph+1 using the following equation:

$$\dot{m}_{ext}(nhph+2) = \left[\frac{(\dot{m}_s - \sum_{i=1}^{nhph+1} \dot{m}_{ext}(i)) * (h_{out}(nhph+2) - h_{in}(nhph+2))}{h_{ext}(nhph+2) - h_{dr}(nhph+2)} \right] \quad (5.33)$$

$$\dot{m}_{ext}(i) = \left[\frac{(\dot{m}_s - \sum_{i=1}^{nhph+1} \dot{m}_{ext}(i)) * (h_{out}(n) - h_{in}(n)) + \sum_{i=nhph+2}^{n-1} \dot{m}_{ext}(i) * (h_{dr}(i) - h_{dr}(i-1))}{(h_{ext}(i) - h_{dr}(i))} \right] \quad (5.34)$$

Where: $i=nhph+3, nhph+4, \dots, n=nheater-1$

$$\dot{m}_{ext}(nheater) = \left[\frac{(\dot{m}_s - \sum_{i=1}^{n-1} \dot{m}_{ext}(i)) * (h_{out}(n) - h_{in}(n)) + \sum_{i=nhph+2}^{n-1} \dot{m}_{ext}(i) * (h_{dr}(n) - h_{dr}(n-1))}{(h_{ext}(n) - h_{in}(n) + h_{out}(n) - h_{dr}(n))} \right] \quad (5.35)$$

Where : $n=nheaters$

Based on extraction flows to each heater, the resulting steam flow through the turbine from extraction point to extraction point the turbine output can then be calculated.

Power output from **the high pressure turbine** \dot{W}_{t1} :

$$\begin{aligned} \dot{W}_{t1} = & \dot{m}_s * (h_{ss} - h_{ext}(1)) + (\dot{m}_s - \dot{m}_{ext}(1)) * (h_{ext}(1) - h_{ext}(2)) + \\ & +, ,, + (\dot{m}_s - \sum_{i=1}^{nhptext-1} \dot{m}_{ext}(i)) * (h_{ext}(nhptext-1) - h_{extrh}) \end{aligned} \quad (5.36)$$

Power output from **the intermediate turbine** \dot{W}_{t2} :

$$\begin{aligned} \dot{W}_{t2} = & \dot{m}_{extrh} * (h_{extrho} - h_{ext}(nhptext)) + (\dot{m}_{extrh} - \dot{m}_{ext}(nhptext)) * (h_{ext}(nhptext) - h_{ext}(nhptext+1)) \\ & +, ,, + (\dot{m}_{extrh} - \sum_{i=nhptext}^{nhptext+niptext-2} \dot{m}_{ext}(i)) * (h_{ext}(nhptext+niptext-2) - h_{ext}(nhptext+niptext-1)) \end{aligned} \quad (5.37)$$

Power output from **the Low pressure turbine** \dot{W}_{t3} :

$$\begin{aligned} \dot{W}_{t3} = & (\dot{m}_{extrho} - \sum_{i=nhptext}^{nhptext+niptext+1} \dot{m}_{ext}(i)) * (h_{ext}(nhptext+niptext-1) - h_{ext}(nhptext+niptext)) + \\ & \dot{m}_{exh} * (h_{ext}(nhptext+niptext) - h_{exh}) \end{aligned} \quad (5.38)$$

The **total power output** from the turbine will be \dot{W}_t :

$$\dot{W}_t = \dot{W}_{t1} + \dot{W}_{t2} + \dot{W}_{t3} \quad (5.39)$$

Where: $nhph$ = number of high pressure closed heater
 $nlph$ = number of low pressure closed heater
 $nhptext$ = number of extraction from high pressure turbine
 $niptext$ = number of extraction from intermediate turbine
 $nlptext$ = number of extraction from low pressure turbine
 $nheater$ = number of heaters in the cycle

The cycle heat rate **HR** is the thermal energy input to the cycle divided by the electric output. The heat rate then is as follows:

((Turbine inlet steam flow) × [Turbine inlet steam enthalpy-final feed water enthalpy] + [reheater steam flow] × [hot reheat enthalpy-cold reheat enthalpy]) ÷ (Turbine KW output).

Or

$$HR = \left[\frac{\dot{m}_s * (h_{ss} - h_w) + \dot{m}_{extrh} * (h_{extrh} - h_{extrho})}{\dot{W}_t} \right] \quad (5.40)$$

The procedure to simulate the Turbine_Heater is the same as the Boiler simulation but the number of equations will vary with the number of heaters in the steam power plant cycle.

Having known the number of heaters in the cycle, the extraction flow to each heater will be calculated by the above equations. By writing these equations as a function of f, or residue function R to be the right column matrix {R} and the partial derivatives become the matrix coefficient [pd], the Jacobean matrix and, then Newton-Raphson method will be used to find the value of the variable that reduce R to zero.

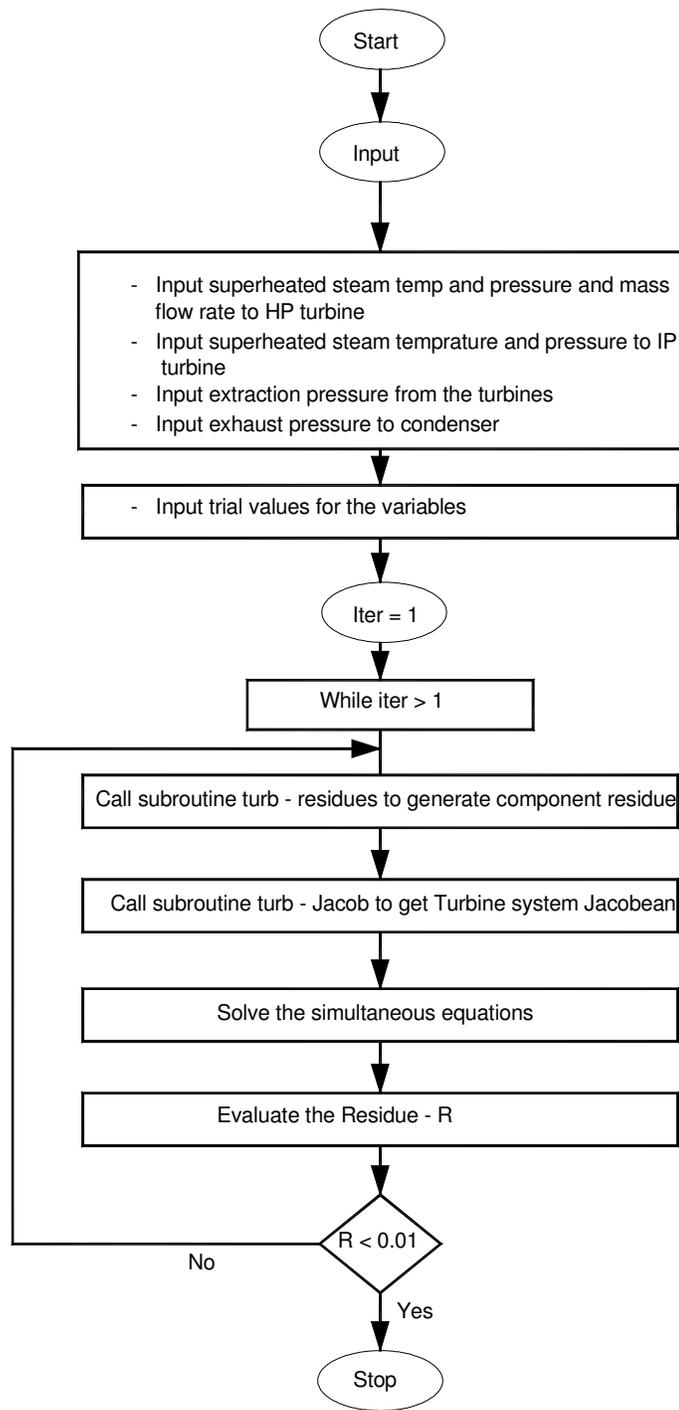


Fig.5.2 Flow chart for the simulation of Turbine-Heaters

5.3 Power Plant System Verification

Simulation programs can be verified with existing data to justify the model. In this work, an input data for the system is the steam entering into the high pressure turbine of mass flow rate $m_s = 0.083$ kg/s, temperature, $T = 550^\circ\text{C}$ and pressure, $p = 150$ bar, as shown in the steam power plant cycle Fig.5.3.

The main program contains many different subroutine 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 'input_SSPsim' program and then to main program steam power plant simulation.

The modern steam power plant is sketched below in the figure (Fig 5.3) and the Rankine cycle diagram is also shown in the Fig.5.4. The results of the simulation are displayed in the Table 5.1.

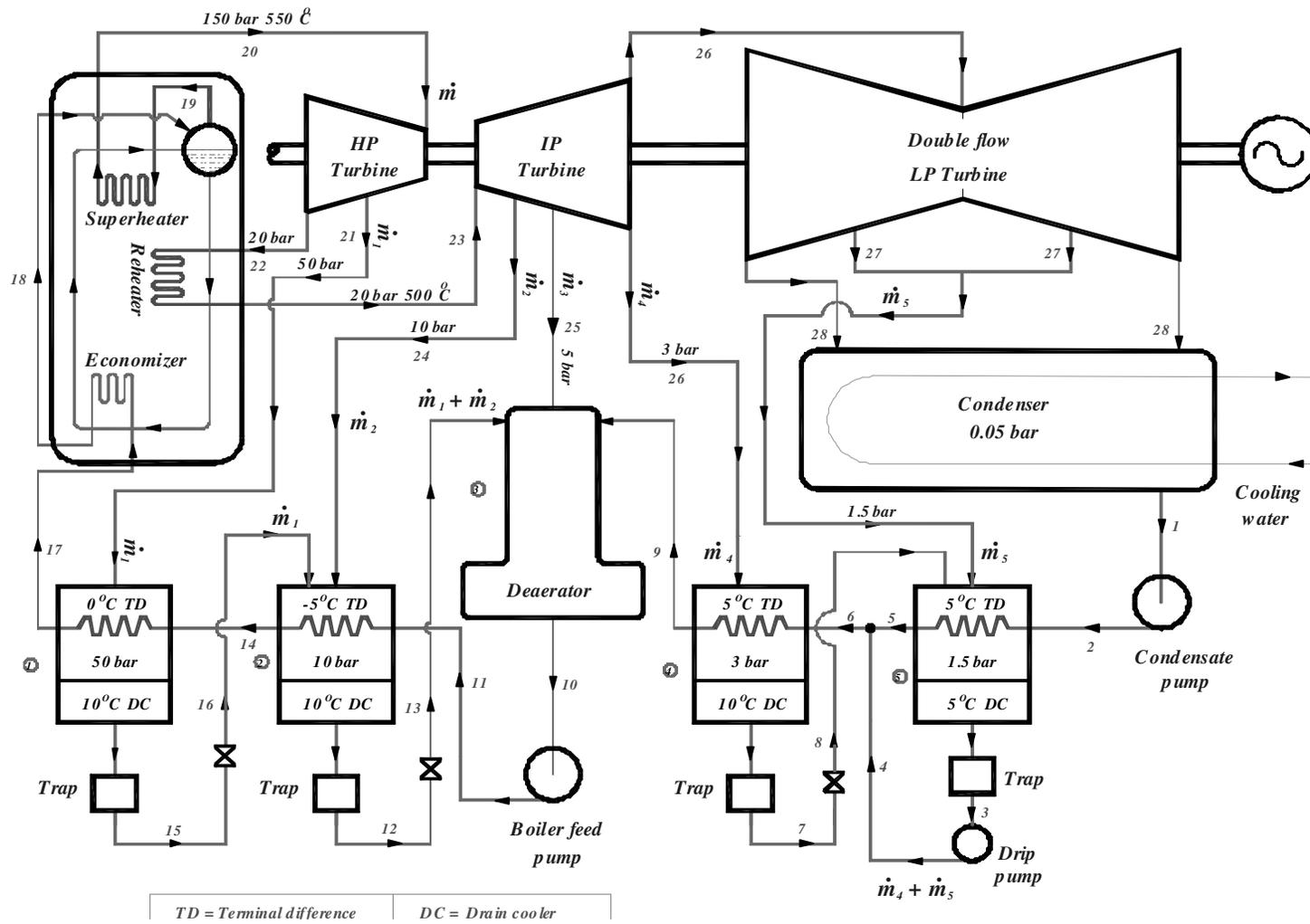


Fig.5.3 Steam Power Plant

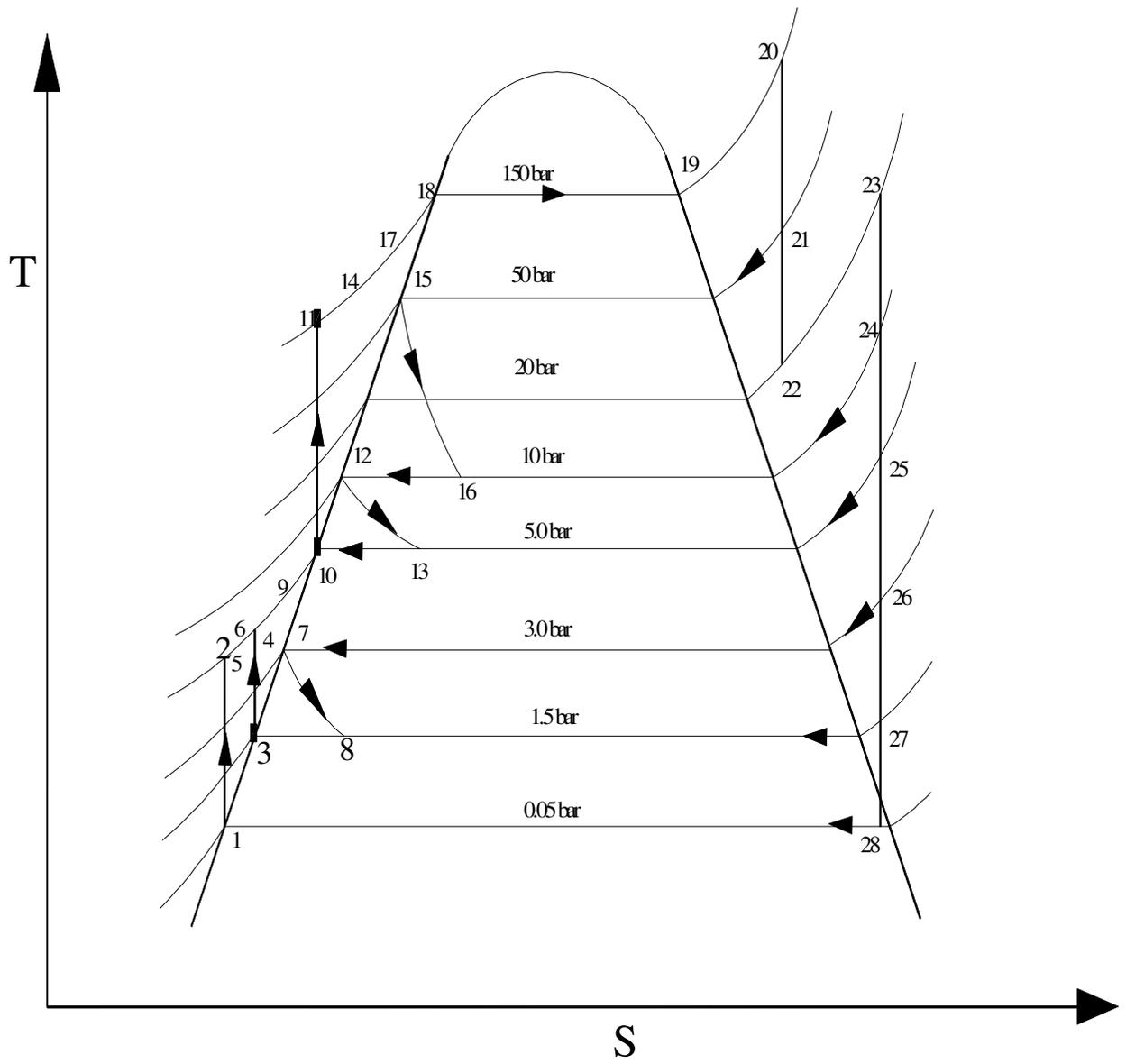


Fig.5.4 The T-S diagram of the Steam Power Cycle

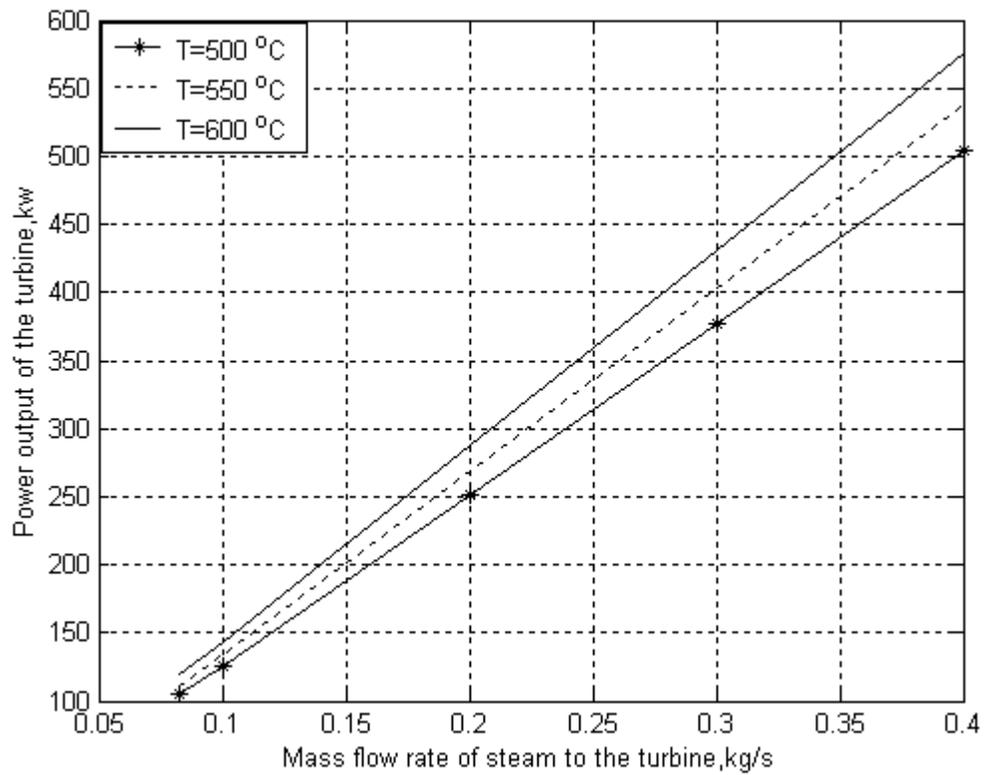


Fig.5.5 The characteristic of the Turbine Operating at the given Exhaust Pressure

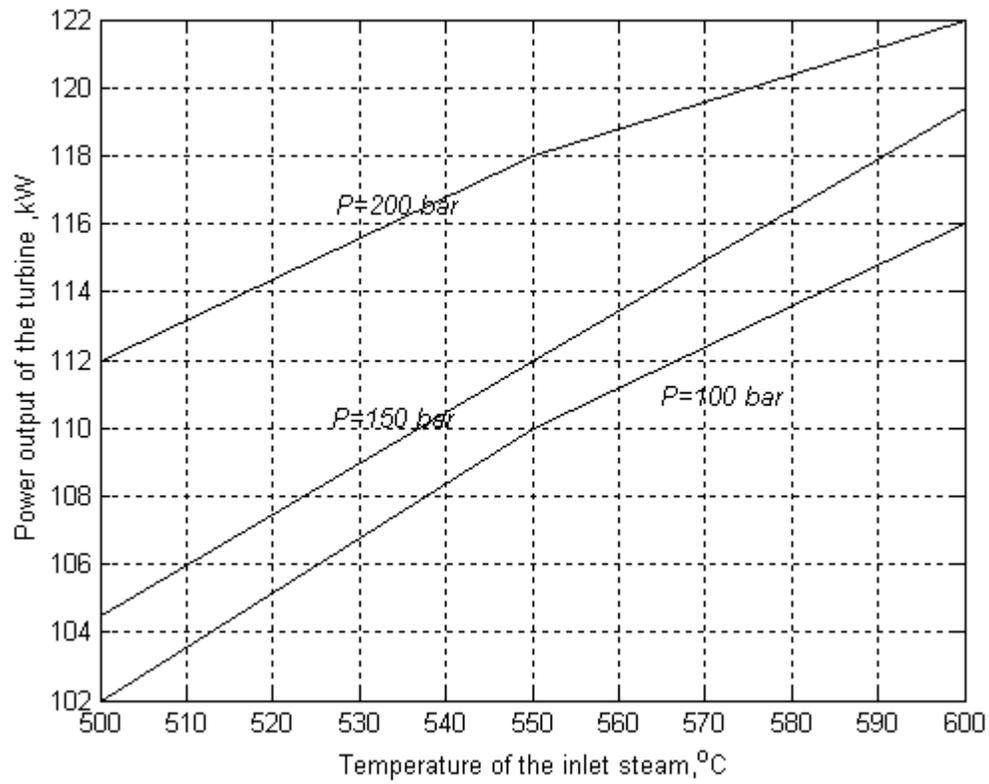


Fig.5.6 The Variation of the Power Output of the Turbine with the inlet Steam Temperature for the Given Inlet Pressure

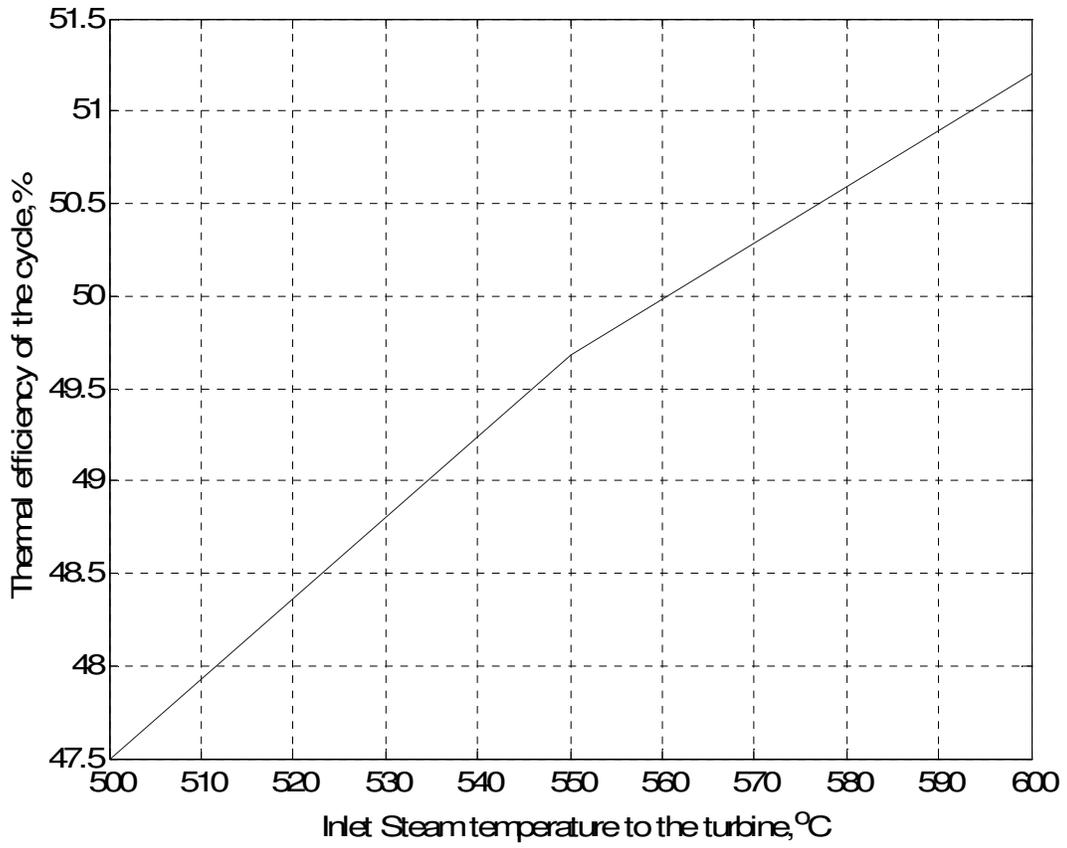


Fig.5.7 The Variation of the Thermal Efficiency with the inlet Steam Temperature

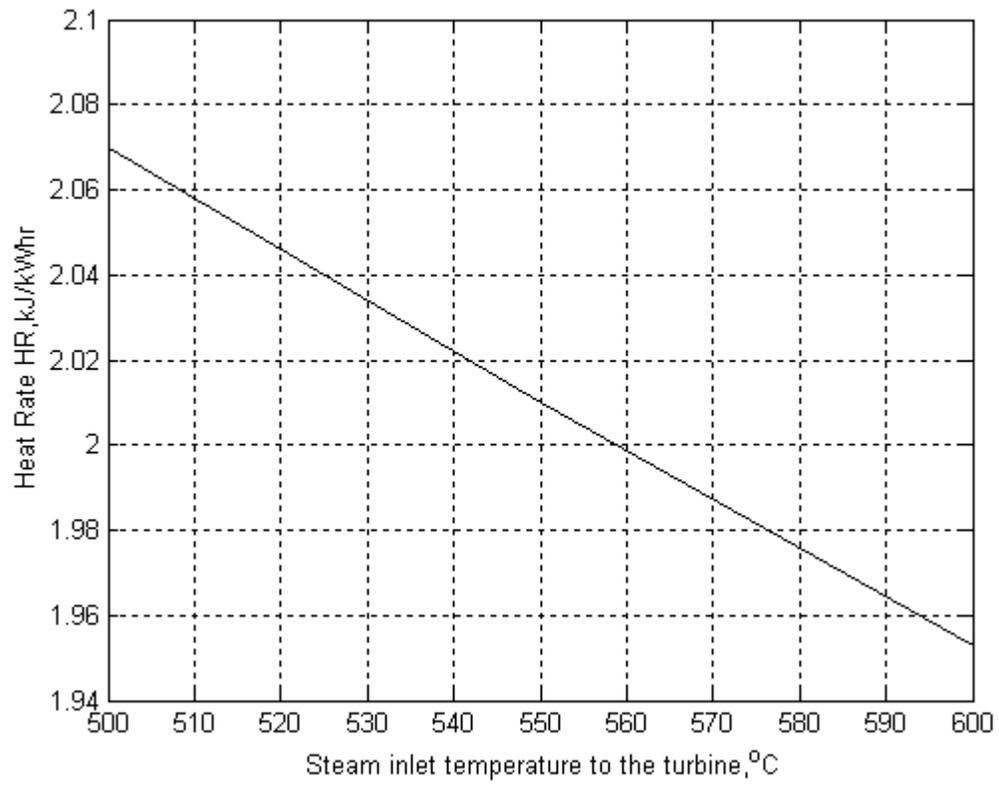


Fig.5.8 The Variation of Heat Rate HR with the Steam Inlet Temperature to the turbine

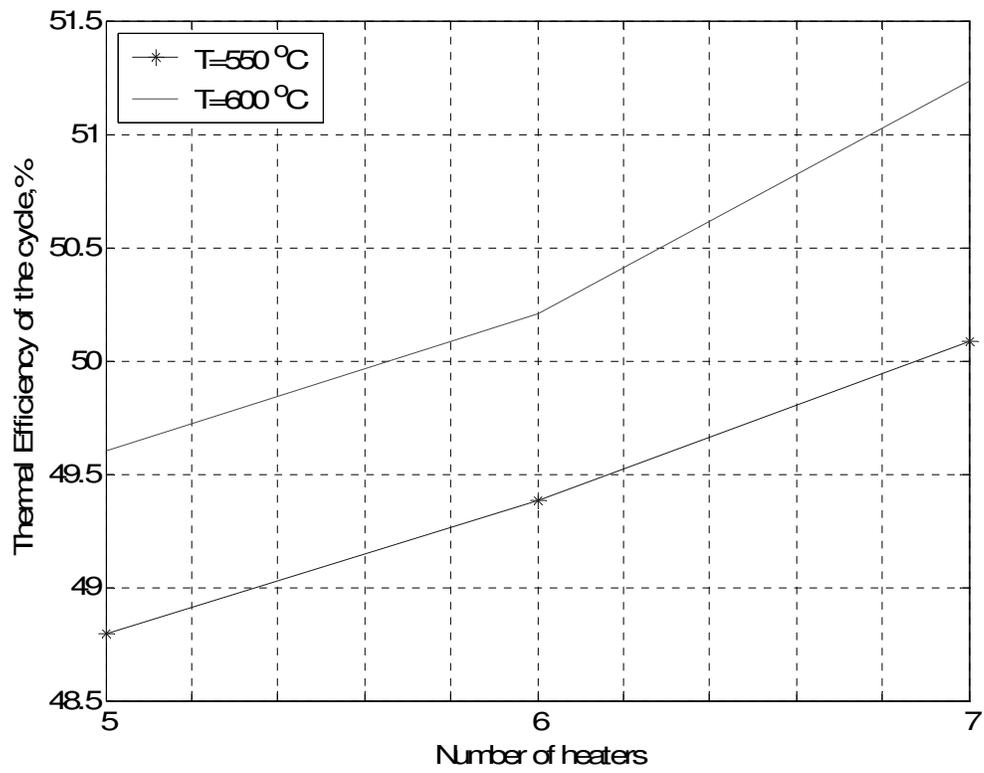


Fig.5.9 The Effect of the Thermal Efficiency with Increasing the number of Heaters

Stream Line	Mass flow rate, [kg/s]	Pressure, [Bar]	Temperature, [° C]	Enthalpy, [kJ/kg]	State of the steam
1	0.05439	0.05	32.88	137.82	saturated liquid
2	0.05439	5.00	32.88	138.32	>>
3	0.010175	1.50	111.7	467.11	>>
4	0.010175	5.00	111.7	467.48	liquid
5	0.05439	5.00	111.7	467.11	liquid
6	0.06458	5.00	111.7	467.17	liquid
7	0.002575	3.00	133.55	561.47	saturated liquid
8	0.002575	1.50	111.37	561.47	Liquid-vapour
9	0.06458	5.00	133.55	561.47	liquid
10	0.083	5.00	151.86	640.23	saturated liquid
11	0.083	150.00	153.53	656.46	>>
12	0.01755	10.00	179.91	762.81	>>
13	0.01755	5.00	151.86	762.81	Liquid-vapour
14	0.083	150.00	179.91	770.11	subcooled liq.
15	0.0163	50.00	263.99	1154.23	saturated liquid
16	0.0163	10.00	179.91	1154.23	Liquid-vapour
17	0.083	150.00	263.99	1153.10	subcooled liq.
18	0.083	150.00	342.24	1610.50	saturated liquid
19	0.083	150.00	342.24	2610.50	saturated vapor
20	0.083	150.00	550.00	3448.60	superheated va.
21	0.01630	50.00	367.96	3114.11	>>
22	0.067	20.00	245.13	2889.53	>>
23	0.067	20.00	500.00	3467.60	>>
24	0.00125	10.00	389.82	3242.27	>>
25	0.00122	5.00	292.54	3048.75	>>
26	0.002575	3.00	229.3	2926.56	>>
27	0.0076	1.50	148.73	2770.48	>>
28	0.05439	0.05	32.88	2266.00	Liquid-vapour

Table 5.1 The results of the simulation of the steam power Plant

5.4 CONCLUSION AND RECOMMENDATION

The simulation problem points the way toward using the simulation technique in general. One of the advantages of writing a computer program here used MATLAB, to perform the above calculation for the steam power plant is that once a program is available the operating conditions at many other input conditions can be determined simply.

As shown from the above figures that increasing the inlet steam mass flow rate and pressure for the given inlet steam temperature would result in:-increase in power output of the turbine and thermal efficiency. An increasing an initial steam temperature would always result in: increase in power output and thermal efficiency since heat will be added in the cycle at a higher temperature and decrease in moisture content of steam at turbine exhaust.

Parametric studies can be accomplished and the system behavior understood. The information available from the simulation program can be used in optimization studies; in fact, the entire system simulation program can be used as a subroutine for such studies. Also quick evaluations can be made of choosing a different steam power plant component, such as a turbine, a boiler or a pump with different performance characteristics.

The paper can be extended further by considering those neglected losses, minimizing the assumptions, avoiding limitations and for actual regenerative steam power plant cycle to make the program robust and predictive of the real case.

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COMPUTER PROGRAM

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%  
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%  
%THIS IS A MATLAB SIMULATION PROGRAM FOR MODERN STEAM POWER PLANT% %  
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%  
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
clear
```

```
clc
```

```
% Call the input values from the input data or input.m
```

```
load Inputs_SPPsim.m; inp=Inputs_SPPsim;
```

```
% Assign the values of the input values
```

```
ms=inp(1);
```

```
mg=inp(2);
```

```
ma=inp(3);
```

```
tss=inp(4);
```

```
tss2=inp(5);
```

```
pss=inp(6);
```

```
pss2=inp(7);
```

```
nheater=inp(8);
```

```
nhph=inp(9);
```

```
nlph=inp(10);
```

```
nhptext=inp(11);
```

```
niptext=inp(12);
```

```
nlptext=inp(13);
```

```
U(1)=inp(14);
```

```
U(2)=inp(15);
```

```
U(3)=inp(16);
```

```
U(4)=inp(17);
```

```
U(5)=inp(18);
```

```

Tgiev=inp(19);

pexh=inp(20);
for i=1:nheater
    pext(i)=inp(i+20);
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
gr=9.81;          % gravity

pextrh=pss2;      % the pressure of the steam out of the reheater

pdea=pext(nhph+1); % pressure of dearator

pcond=pexh;       % pressure of the cond. is the press. of the steam exhaust

% determination of saturated temperature in the shell of heaters

for i=1:nheater
    tsat(i)=sattable_givp_t(pext(i));
end

ss=superheat_entropy(tss+273.15,pss)*1e-03 ; %iscentropic expansion-entropy
calculation for hp turbine
% ENTHALPY CALCULATION
% Condenser
[tsc,hf,hfg,hg,sg,sf,sfg,vf,vg]=steam_p(pcond); % Read the steam property from the
table
tc=tsc;          % temperature of the exhaust steam
hfc=hf*1e-03;    % Enthalpy of the condensate liquid
sp.vol=vf;       % specific volume of the condensate
hin(nheater)=hfc + sp.vol*(pdea-pcond)*100 ; % condensate enthalpy leaving
the condensate pump in kJ/kg

```

```

% Dearator
[tsd,hfd,hfgd,hgd,sgd,sfd,sfgd,vfd,vgd]=steam_p(pdea); % read the steam property
from the table
hin(nhph)=hfd*1e-3+vfd*(pss-pdea)*100 ; % the enthalpy leaving the
feed pump or entering to heater nhph in KJ/kg
% Enthalpy calculation for the extraction of steam from hp turbine
hss=superheat_enthalpy(tss+273.15,pss)*1e-03; % enthalpy of steam inlet
to HP turbine
for i=1:nhptext-1
    hext(i)=superheat_p_s(pext(i),ss*1e+03)*1e-03;
end
hextrh=superheat_p_s(pextrh,ss*1e+03)*1e-03; % enthalpy of steam to
the reheater
% Enthalpy calculation for the extraction of steam from IP turbine
hextrho=superheat_enthalpy(tss2+273.15,pss2)*1e-03; % enthalpy of steam
leaving the reheater
ss2=superheat_entropy(tss2+273.15,pss2)*1e-03; % entropy for the
intermediate pressure turbine

for k=nhptext:niptext+nhptext-1
    hext(k)=superheat_p_s(pext(k),ss2*1e+03)*1e-03;
end

% Enthalpy calculation for the extraction of steam from lp turbine

hext(niptext+nhptext)=superheat_p_s(pext(niptext+2),ss2*1e+03)*1e-03;

% Enthalpy calculation for the exhaust steam stream
[Texh,hfex,hfgex,hgex,sgex,sfex,sfgex,vfex]=steam_p(pcond); % Read the steam
property for the given condensate pressure

xra=((ss2*1e+3)-sfex)/sfgex ; % the steam quality after the low pressure
turbine

```

```
hexh=hfex*1e-03 + xra*hfge*1e-03; % the exhaust steam enthalpy after the  
low pressure turbine
```

```
% Calculation of inlet and outlet water temperature of the heaters
```

```
for j=1:nhph  
    tdr(j)=tsat(j);  
    tout(j)=tdr(j);  
    hout(j)=comp_water_h1(pss,tout(j)-273.15);  
    hdr(j)=satsteam_hf_givp(pext(j))*1e-03;  
    if j==nhph  
        tin(j)=sattable_p_t(pdea);  
        hin(j)=hfd*1e-3+vfd*(pss-pdea)*100;  
    else  
        tin(j)=tsat(j+1);  
        hin(j)=comp_water_h1(pss,tin(j)-273.15);  
    end  
end  
for x=nheater:-1:nhph+1  
    tdr(x)=tsat(x);  
    tout(x)=tdr(x);  
    hdr(x)=satsteam_hf_givp(pext(x))*1e-03;  
    hout(x)=hdr(x);  
    if x==nheater  
        tin(x)=sattable_p_t(pcond);  
        hin(x)=hfc + sp.vol*(pdea-pcond)*100;  
    elseif x==nhph+1;  
        tin(x)=tout(nhph+2);  
        hin(x)=hout(nhph+2);  
    else  
        tin(x)=tsat(x+1);  
        hin(x)=hdr(x+1);  
    end  
end
```

```

end
% OUTPUT OF THE RESULTS
fprintf(1,'Heater Feed water inlet temperatures are %6.2f deg.cel\n',tin(1)-
273.15)
fprintf(1,'                %6.2f deg.cel\n',tin(2:nheater)-273.15)
disp('.....')
fprintf(1,'>>Heater Feed water outlet temperatures are %8.3f deg.cel\n',tout(1)-
273.15)
fprintf(1,'                %8.3f deg.cel\n',tout(2:nheater)-273.15)
disp('.....')
fprintf(1,'>>Extraction Enthalpy to heaters are %6.2f kJ/kg\n',hext(1))
fprintf(1,'                %6.2f kJ/kg\n',hext(2:nheater))
disp('.....')
fprintf(1,'>>Heater Feed water inlet enthalpy are %8.3f kJ/kg\n',hin(1))
fprintf(1,'                %8.3f kJ/kg\n',hin(2:nheater))
disp('.....')
fprintf(1,'>>Heater Feed water outlet enthalpy are %8.3f kJ/kg\n',hout(1))
fprintf(1,'                %8.3f kJ/kg\n',hout(2:nheater))
disp('.....')
fprintf(1,'>>Drain Enthalpy from heaters are %8.3f kJ/kg\n',hdr(1))
fprintf(1,'                %8.3f kJ/kg\n',hdr(2:nheater))
disp('.....')

% CALCULATION OF EXTRACTION STEAM MASS FLOW RATE TO THE
HEATERS
% FOR HIGH PRESSURE HEATERS
mext(1)=(ms*(hout(1)-hin(1)))/(hext(1)-hdr(1)); % extracted steam mass flow rate
to the first heater
for i=2:nhph
    summhph=0;
        for j=1:i-1
            mm(j)=mext(j)+summhph;
            summhph=mm(j);
        end
    end

```

```

    end
    mext(i)=(ms*(hout(i)-hin(i))-summhph*(hdr(i-1)-hdr(i)))/(hext(i)-hdr(i));
    mext(i);
end

mextrh=ms-summhph; % mass flow rate of extraction steam to reheater from
HP turbine
% DEARATOR
mml=0;
for j=1:nhph
    mml(j)=mext(j)+mml;
    mml=mml(j);
    summlp=mml;
end
mext(nhph+1)=(ms*hdr(nhph+1)-summlp*hdr(nhph)-(ms-
summlp)*hout(nhph+2))/(hext(nhph+1)-hout(nhph+2));
mext(nhph+1);
% FOR LOW PRESSURE HEATER
for i=nhph+2:nheater
    if i==nhph+2
        mext(nhph+2)=(ms-summlp-mext(nhph+1))*(hout(nhph+2)-
hin(nhph+2))/(hext(nhph+2)-hdr(nhph+2));
        mext(nhph+2);
    elseif i==nheater
        yy=ms-sum(mext);
        xx=yy*(hout(nheater)-hin(nheater))-mext(nheater-1)*(hdr(nheater-1)-
hdr(nheater));
        mext(nheater)=xx/(hext(nheater)-hdr(nheater)+hout(nheater)-hin(nheater));
        mext(nheater);
        mexh=yy-mext(nheater);
        mext;
    else

```

```

    mext(i)=((ms-summlp-mext(nhph+1))*(hout(i)-hin(i))-mext(i-1)*(hdr(i-1)-
hdr(i)))/(hext(i)-hdr(i));
    mext(i);
    end
end
disp('.....')
fprintf(1,'>>mass of extractions are %8.3f Kg/s\n',mext(1))
fprintf(1,'          %8.4f Kg/s\n',mext(2:nheater))
disp('.....')
fprintf(1,'>>mass of extraction to reheater is %8.3f Kg/s\n',mextrh)
disp('.....')
fprintf(1,'>>mass of extractions to exhaust is %8.3f Kg/s\n',mexh)
mexh;

% TURBINE POWER CALCULATION
%HIGH PRESSURE TURBINE POWER CALCULATION
whpo=0;
sumh=0;
for i=1:nhptext
    sumh(i)=sumh+mext(i);
    sumh=sumh(i);
    if i==1
        whp(i)=whpo+ms*(hss-hext(i));
        whpo=whp(i);
    elseif i==nhptext
        aa=ms-(sumh-mext(i));
        whp(i)=whpo+aa*(hext(i-1)-hextrh);
        whpt=whp(i);
    else
        whp(i)=whpo+(ms-(sumh-mext(i)))*(hext(i-1)-hext(i));
        whpo=whp(i);
    end
end

```

```

end
disp('.....')
msip=aa;
fprintf(1,'>>mass flow rate to intermediate pres. turbine is %8.3f Kg/s\n',msip)
disp('.....')
whpt=whp(nhptext);
%FOR INTERMEDIATE TURBINE POWER CALCULATION
hip=hextrh;
wipo=0;
sumi=0;
for i=nhptext:nhptext+niptext-1
    if i==nhptext
        wip(i)=wipo+(msip)*(hextrho-hext(nhptext));
        wipo=wip(i);
    else
        sumi(i)=sumi+mext(i-1);
        sumi=sumi(i);
        wip(i)=wipo+(msip-sumi)*(hext(i-1)-hext(i));
        wipo=wip(i);
    end
end
mlpt=msip-(sumi+mext(niptext+1));
fprintf(1,'>>mass flow rate to low pressure turbine is %8.3f Kg/s\n',mlpt)
disp('.....')
disp('.....')
fprintf(1,'>>power output from high pressure turbine is %8.3f kw\n',whpt)
disp('.....')
wipt=wip(nhptext+niptext-1);
fprintf(1,'>>power output from the intermediate pressure turbine is %8.3f
kw\n',wipt)

disp('.....')

```

```

%LOW PRESSURE TURBINE POWER CALCULATION
wlpo=0;
suml=0;
for i=nhptext- 1+niptext:nhptext- 1+niptext+nlptext- 1
    if i==nhptext- 1+niptext
        wlp(i)=wlpo+mlpt*(hext(i)-hext(i+ 1));
        wlpo=wlp(i);
    elseif i==nhptext- 1+niptext+nlptext- 1
        wlp(i)=wlpo+mexh*(hext(i)-hexh);
        wlpo=wlp(i);
        wlpt=wlpo;
    else
        suml(i)=suml+mext(i);
        suml=suml(i);
        wlp(i)=wlpo+(mlpt-suml)*(hext(i- 1)-hext(i));
        wlpo=wlp(i);
    end
end
fprintf(1,'>>power output from low pressure turbine is %8.3f kw\n',wlpt)
    %TOTAL POWER OUTPUT BY THE TURBINE
disp('.....')
    Wtt=whpt+wipt+wlpt;
    fprintf(1,'>> Total power output from the turbines will be %8.3f kw\n',Wtt)
    disp('.....')

% HEAT RATE CALCULATION
% Heat supplied from economizer,superheater,and evaporator are BO
BO=ms*(hss-hout(1)) ;    % Heat supplied by the boiler to the steam/water
RE=msip*(hextrho-hextrh); % Heat supplied from Reheater
HR=(BO+RE)*3600/Wtt;    % Heat Rate Calculation
Heat_Rate_kJ_Per_kwhr=HR;
fprintf(1,'>> Heat Rate is %8.3f kJ/kwhr\n',HR)
disp('.....')

```

```

% THERMAL EFFICIENCY CALCULATION
% E=(Wtt/(BO+RE))*100;      % Turbine Efficiency Calculation
% Efficiency_of_turbine=E;
% fprintf(1,'>> Efficiency of the turbine is %8.3f percent\n',E)
% disp('.....')
% Condenser
qc=mexh*(hexh-hfc);          % heat loss from the condenser
Heat_loss_from_condenser_kW=qc ;
fprintf(1,'>> Heat loss from the condenser is %8.3f kw\n',qc)
% condensate pump: Work done and Head
disp('.....')
Wp1=mexh*(hin(nheater)-hfc);
power_of_cond_pump_kw=Wp1;
H=(pdea-pcond)*100000*sp.vol/gr;
Head_of_condensate_pump_m=H;
fprintf(1,'>> Power for condensate pump is %8.3f kw\n',Wp1)
disp('.....')
fprintf(1,'>> Head of the condensate pump is %8.3f m\n',H)
disp('.....')
% feed pump : Head and work /pump power

[tsd,hfd,hfgd,hgd,sgd,sfd,sfgd,vfd,vgd]=steam_p(pdea);
hin(nhph)=hfd*1e-3+vfd*(pss-pdea)*100;      % in KJ/kg
Wp2=ms*(hin(nhph)-hdr(nhph+1));
Power_output_of_feed_pump_kw=Wp2;
H=(pss-pdea)*100000*vfd/gr;
Head_of_feed_pump_m=H;
fprintf(1,'>> Power for Feed pump is %8.3f kw\n',Wp2)
disp('.....')
fprintf(1,'>> Head of the Feed pump is %8.3f m\n',H)
disp('.....')
% Drip pump: work in put and Head

```

```

[tsp,hfp,hfgp,hgp,sgp,sfp,sfgp,vfp,vgp]=steam_p(pext(nheater));
hp=(hfp*1e-3)+vfp*(pdea-pext(nheater))*1e+5 ;           % in KJ/kg
  Wp3=(sum(mext(nhph+2:nheater)))*(hp-hdr(nheater));
power_output_of_drip_pump_kw=Wp3;
H=(pdea-pext(nheater))*1e+5*vfp/gr;
Head_of_drip_pump_m=H;
fprintf(1,'>> Power for drip pump is %8.3f kw\n',Wp3)
disp('.....')
fprintf(1,'>> Head of the drip pump is %8.3f m\n',H)
disp('.....')
%%%%%%%% BOILER SIMULATION SOLVER %%%%%%%%%%%%%%
cpg=1.005;      % specific heat capacity of flue gases
tamb=20;       % temperature of ambient air oC
cpa=1.05;      % spesfic heat capacity of air KJ/kg*K
Uev=U(1);     % Overall heat transfer coefficient for evaporator in KW/m^2*K
Us=U(2);      % Overall heat transfer coefficient for superheater >>
Urh=U(3);     % Overall heat transfer coefficient for economizer >>
Uec=U(4);     % Overall heat transfer coefficient for economizer >>
Uah=U(5);     % Overall heat transfer coefficient for air_heater >>
N=16;         % number of equation
% the heat of the flue gasses from the furnace
% temperature Tgiev=Temperature of the flue gases out of the furnace
Qt=mg*cpg*(Tgiev-200);
fprintf(1,'>> Total heat output from the furnace %8.3f kw\n',Qt)
disp('.....')
% Data found from steam table
hss2=hextrho;
hw=hout(1);
[tsat,hf,hfg,hg,sg,sf,sfg,vf,vg]=steam_p(pss);
tc=tsat-273 ;

% Calculation of heat absorbed by the boiler components

```

```

qev=ms*hfg*1e-03;           % heat transfer to the steam in the
evaporator in KW
qs=ms*(hss-hg*1e-03);      % heat transfer to the steam in the
superheater in KW
qec=ms*(hf*1e-03-hw);      % heat transfer to the steam in the
economizer in KW
qrh=mextrh*(hss2-hextrh);  % heat transfer to the steam in the
reheater in KW
qah=Qt-(qev+qs+qec+qrh);  % heat transfer to the air in the
air_heater in Kw
fprintf(1,'>> Heat absorbed by evaporator %8.3f kw\n',qev)
disp('.....')
fprintf(1,'>> Heat absorbed by superheater %8.3f kw\n',qs)
disp('.....')
fprintf(1,'>> Heat absorbed by economizer %8.3f kw\n',qec)
disp('.....')
fprintf(1,'>> Heat absorbed by reheater %8.3f kw\n',qrh)
disp('.....')
fprintf(1,'>> Heat absorbed by air_preheater %8.3f kw\n',qah)
disp('.....')
% temperature of the steam/water in the boiler

tw=tout(1)-273;           % temperature inlet to the economizer/outlet from the
first heater
tsih=superheat_temp_p_h1(pextrh,hextrh)-273.15;   % temperature inlet to
the reheater
tsoh=tss2;               % temperature outlet from the reheater
tsis=tc;                 % temperature inlet to the superheater
tsos=tss;                % temperature outlet from the superheater
tsoec=tc;                % temperature outlet from the economizer
tsiec=tw;                % temperature inlet to the economizer

% inter the initial conditions for the variables

```

```

Aev=0.15;   Tgoev=895;   delTlmev=504;
As=0.15;    Tgos=795;    delTlms=400;
Arh=0.11;   Tgoh=595;    delTlmrh=350;
Aec=0.09;   Tgoec=495;    delTlmeec=330;
Aah=0.03;   Tgoah=395 ;   delTlmah=570;   toah=45;

vt(1,1)=Tgoev;           % Outlet Temperature of the flue gases from the
evaporator ,oC
vt(2,1)=delTlmev;        % Log.mean temperature difference for the
evaporator,oC
vt(3,1)=Aev;             % Heat transfer Area for the evaporator,m^2

vt(4,1)=Tgos;           % Outlet temperature of the flue gases from the
superheater
vt(5,1)=delTlms;        % Log.mean temperature difference for the superheater
vt(6,1)=As;             % Heat transfer Area for the superheater

vt(7,1)=Tgoh;           % Outlet temperature of the flue gases from the reheater
vt(8,1)=delTlmrh;       % Log.mean temperature difference for the reheater
vt(9,1)=Arh;           % Heat transfer Area for the reheater

vt(10,1)=Tgoec;         % Outlet temperature of the flue gases from the
economizer
vt(11,1)=delTlmeec;     % Log.mean temperature difference for the economizer
vt(12,1)=Aec;          % Heat transfer Area for the economizer

vt(13,1)=Tgoah;         % Outlet temperature of the flue gases from the
Air_heater
vt(14,1)=toah;          % the outlet air temperature for the air_heater
vt(15,1)=delTlmah;      % Log.mean temperature difference for the Air_heater
vt(16,1)=Aah;          % Heat transfer Area for the Air_heater

% iteration sequence

```

```

iter=1;
while iter>0

% Intializing the variables

Aevo=Aev; Tgoevo=Tgoev; delTlmevo=delTlmev;
Aso=As; Tgoso=Tgos; delTlmsso=delTlms;
Arho=Arh; Tgoho=Tgoh; delTlmrho=delTlmrh;
Aeco=Aec; Tgoeco=Tgoec; deTlmeco=delTlmec;
Aaho=Aah; Tgoaho=Tgoah; delTlmaho=delTlmah; toaho=toah;

vt=[Tgoevo;delTlmevo;Aevo;Tgoso;delTlmsso;Aso;Tgoho;
delTlmrho;Arho;Tgoeco;deTlmeco;Aeco;Tgoaho;toaho;delTlmaho;Aaho];

Tgis=Tgoevo;
Tgih=Tgoso;
Tgiec=Tgoho;
Tgiah=Tgoeco;
tsiec=tw;
% Call the Residue function,R

R=boil_resid2(vt,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us,Urh,
Uec,Uah,tsis,tsos,tsih,tsoh,tsoec);
% Call the Jacobian matrix/the coefficient of the matrix,pd
pd=boil_jacob2(N,R,vt,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us
,Urh,Uec,Uah,tsis,tsos,tsih,tsoh,tsoec);
% solve the simultaneouse equation to get change of variables,V
V=inv(pd)*R;
% find the new variable values
vn=vt-V;
% Assign the new variable in the matrix
Tgoev=vn(1,1);
delTlmev=vn(2,1);

```

```

Aev=vn(3,1);

Tgos=vn(4,1);
delTlms=vn(5,1);
As=vn(6,1);

Tgoh=vn(7,1);
delTlmrh=vn(8,1);
Arh=vn(9,1);

Tgoec=vn(10,1);
delTlmev=vn(11,1);
Aec=vn(12,1);

Tgoah=vn(13,1);
toah=vn(14,1);
delTlmah=vn(15,1);
Aah=vn(16,1);
% Call the residue function
R=boil_resid2(vt,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us,Urh,
Uec,Uah,tsis,tsos,tsih,tsoh,tsoec);
%check on convergence
    if all(abs(R)<1)
        iter=0;
        v1=vn;
        disp('.....')
fprintf(1,'>> Tgoev is %8.3f deg.cel\n',vn(1,1))
disp('.....')
fprintf(1,'>> delTlmev is %8.3f deg.cel\n',vn(2,1))
disp('.....')
fprintf(1,'>> Aev is %8.3f m^2\n',vn(3,1))
disp('.....')
fprintf(1,'>> Tgos is %8.3f deg.cel\n',vn(4,1))

```

```

disp('.....')
fprintf(1,'>> delTlms is %8.3f deg.cel\n',vn(5,1))
disp('.....')
fprintf(1,'>> As is %8.3f m^2\n',vn(6,1))
disp('.....')
fprintf(1,'>> Tgoh is %8.3f deg.cel\n',vn(7,1))
disp('.....')
fprintf(1,'>> delTlmrh is %8.3f deg.cel\n',vn(8,1))
disp('.....')
fprintf(1,'>> Arh is %8.3f m^2\n',vn(9,1))
disp('.....')
fprintf(1,'>> Tgoec is %8.3f deg.cel\n',vn(10,1))
disp('.....')
fprintf(1,'>> delTlmech is %8.3f deg.cel\n',vn(11,1))
disp('.....')
fprintf(1,'>> Aec is %8.3f m^2\n',vn(12,1))
disp('.....')
fprintf(1,'>> Tgoah is %8.3f deg.cel\n',vn(13,1))
disp('.....')
fprintf(1,'>> toah is %8.3f deg.cel\n',vn(14,1))
disp('.....')
fprintf(1,'>> delTlmah is %8.3f deg.cel\n',vn(15,1))
disp('.....')
fprintf(1,'>> Aah is %8.3f m^2\n',vn(16,1))
disp('.....')

else
    iter=iter+1;
    vt=vn;
end
end
% Cycle efficiency calculation
disp('.....')

```

```

    eff_c=(qev+qrh+qec+qs-qc)/(qev+qrh+qec+qs);
    Cycle_efficiency=eff_c*100;
fprintf(1,'>> Cycle efficienct is %8.3f percent\n',Cycle_efficiency)
disp('_____')
disp('_____')

%%%%%%%%%%%%%BOILER_JACOBIN SOLVER%%%%%%%%%%%%%

function
pd=boil_jacob2(N,R,vt,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us
,Urh,Uec,Uah,tsis,tsos,tsih,tsoh,tsoec);

dvt=0.001;
N=16;%length(vt);
for k=1:N
    vd(k,1)=vt(k,1);
end
for j=1:N
    if abs(vt(j,1))<1e-30
        vd(j,1)=vt(j,1)+0.001;
    else
        vd(j,1)=(1+dvt)*vt(j,1);
    end
    %Rd=equation_heat_exchanger(vd,NTU,Cmin,Cmax,z,Cc,Ch,Thin,Tcin);
    %Rd=condenser_residue(vd,Twin,Twout,mexh,hexh,pcon,GD,mdc,hdc);

Rd=boil_resid2(vd,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us,Ur
h,Uec,Uah,tsis,tsos,tsih,tsoh,tsoec);

% [Rd,hfHX]=fheatertype1(Twin,mh,mw,Th,Tsat,pt,ps,GD,vd);
for i=1:N
    if abs(vt(j,1))<1e-30
        pd(i,j)=(Rd(i,1)-R(i,1))/0.001;
    else
        pd(i,j)=(Rd(i,1)-R(i,1))/(vt(j,1)*dvt);%sparse((Rd(i)-R(i))/(v(j)*dv))
    end
end
end
pd=pd;
vd(j,1)=vt(j,1);
end
format long
%%%%%%%%%%%%%BOILER_RESIDUE FUNCTION%%%%%%%%%%%%%

function
R=boil_resid2(vt,qev,qs,qrh,qec,qah,Tgiev,tc,tw,ma,mg,cpg,cpa,tamb,Uev,Us,Urh,
Uec,Uah,tsis,tsos,tsih,tsoh,tsoec)

```

```

% Define the variables in the column matrix
Tgoev=vt(1,1);
delTlmev=vt(2,1);
Aev=vt(3,1);

Tgos=vt(4,1);
delTlms=vt(5,1);
As=vt(6,1);

Tgoh=vt(7,1);
delTlmrh=vt(8,1);
Arh=vt(9,1);

Tgoec=vt(10,1);
delTlmec=vt(11,1);
Aec=vt(12,1);

Tgoah=vt(13,1);
toah=vt(14,1);
delTlmah=vt(15,1);
Aah=vt(16,1);

Tgis=Tgoev;
Tgih=Tgos;
Tgiec=Tgoh;
Tgiah=Tgoec;
tsiec=tw;

% residue functions
R(1,1)=qev-mg*cpg*(Tgiev-Tgoev);
R(2,1)=delTlmev-(((Tgiev-tc)-(Tgoev-tc))/(log((Tgiev-tc)/(Tgoev-tc))));
R(3,1)=qev-Uev*Aev*delTlmev;

R(4,1)=qs-mg*cpg*(Tgis-Tgos);
R(5,1)=delTlms-(((Tgis-tsos)-(Tgos-tsis))/(log((Tgis-tsos)/(Tgos-tsis))));
R(6,1)=qs-Us*As*delTlms;

R(7,1)=qrh-mg*cpg*(Tgih-Tgoh);
R(8,1)=delTlmrh-(((Tgih-tsoh)-(Tgoh-tsih))/(log((Tgih-tsoh)/(Tgoh-tsih))));
R(9,1)=qrh-Urh*Arh*delTlmrh;

R(10,1)=qec-mg*cpg*(Tgiec-Tgoec);
R(11,1)=delTlmec-(((Tgiec-tsoec)-(Tgoec-tsiec))/(log((Tgiec-tsoec)/(Tgoec-tsiec))));
R(12,1)=qec-Uec*Aec*delTlmec;

R(13,1)=qah-mg*cpg*(Tgiah-Tgoah);
R(14,1)=qah-ma*cpa*(toah-tamb);
R(15,1)=delTlmah-(((Tgiah-toah)-(Tgoah-tamb))/(log((Tgiah-toah)/(Tgoah-tamb))));
R(16,1)=qah-Uah*Aah*delTlmah;

```

%%%%%%%%%%TURBINE_JACOBI SOLVER%%%%%%%%%%

function
pd=turbine_jacob(N,R,vt,ms,mext,mexh,nhph,nlph,nheater,hext,hdr,hout,hin)

dvt=0.001;
N=6; %length(vt);
for k=1:N
 vd(k,1)=vt(k,1);
end
for j=1:N
 if abs(vt(j,1))<1e-30
 vd(j,1)=vt(j,1)+0.001;
 else
 vd(j,1)=(1+dvt)*vt(j,1);
 end

 %Rd=equation_heat_exchanger(vd,NTU,Cmin,Cmax,z,Cc,Ch,Thin,Tcin);
 %Rd=condenser_residue(vd,Twin,Twout,mexh,hexh,pcon,GD,mdc,hdc);

Rd=turbine_res(vd,ms,mext,mexh,nhph,nlph,nheater,hext,hdr,hout,hin);

% [Rd,hfHX]=fheatertype1(Twin,mh,mw,Th,Tsat,pt,ps,GD,vd);

 for i=1:N
 if abs(vt(j,1))<1e-30
 pd(i,j)=(Rd(i,1)-R(i,1))/0.001;
 else
 pd(i,j)=(Rd(i,1)-R(i,1))/(vt(j,1)*dvt); %sparse((Rd(i)-R(i))/(v(j)*dv))
 end
 end
 pd=pd;
 vd(j,1)=vt(j,1);
end
format long

%%%%%%%%%% TURBINE_RESIDUE FUNCTION%%%%%%%%%%

function R=turbine_res(vt,ms,mext,mexh,nhph,nlph,nheater,hext,hdr,hout,hin);

for i=1:nheater
 mext(i)=vt(i,1);
end

mexh=vt(6,1);

R(1,1)=mext(1)-(ms*(hout(1)-hin(1)))/(hext(1)-hdr(1)); % extracted steam mass
flow rate to the first heater

for i=2:nhph
R(i,1)=mext(i)-(ms*(hout(i)-hin(i))-sum(mext(1:i-1))*(hdr(i-1)-hdr(i)))/(hext(i)-hdr(i));

```

end

R(nhph+1,1)=mext(nhph+1)-(ms*hdr(nhph+1)-sum(mext(1:nhph))*hdr(nhph)-(ms-
sum(mext(1:nhph))*hout(nhph+2))/(hext(nhph+1)-hout(nhph+2)));

for i=nhph+2:nheater-1
R(i,1)=mext(i)-((ms-sum(mext(1:nhph))-mext(nhph+1))*(hout(i)-hin(i))-
mext(i)*(hdr(i-1)-hdr(i)))/(hext(i)-hdr(i)));
end

xx=ms-sum(mext(1:nheater-1));
yy=xx*(hout(nheater)-hin(nheater))-sum(mext(nhph+2:nheater-1))*(hdr(nheater-
1)-hdr(nheater));
R(nheater,1)=mext(nheater)-(yy/(hext(nheater)-hdr(nheater)+hout(nheater)-
hin(nheater)));

R(nheater+1,1)=mexh-(ms-sum(mext(1:5)));

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% THIS IS THE MATLAB PROGRAMME TO FIND THE EQUATION AND THE
% PERFORMANCE CURVE OF THE PUMP HEAD FOR ANY SPEED N,RPM %%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

clear
clc

% give three datas for heads Hd,and flow rate,Q for THE CONDENSATE PUMP

Q=[0 25 47];
Hd1=[243.576;242.327;228.314];
Hd2=[169.15;166.808;151.8338];
Hd3=[78.2146;74.12535;57.61215];
n=[3000 2500 1700];

% give datas ,Head Hd,and flow rate,Q for THE FEED PUMP

% Q=[0 36 74];
% Hd1=[1137.882;1135.3304;1066.1954];
% Hd2=[921.664;916.32;844.2283];
% Hd3=[365.3886;350.669;268.7];
% n=[3000 2700 1700];

g=9.81; % gravity

% coefficient of the performance equation

qq=[1 Q(1) Q(1)^2;1 Q(2) Q(2)^2;1 Q(3) Q(3)^2]; % the matrix form of the flow rate

a=inv(qq); % the inverse of the matrix

```

```

a0=a*Hd1;
a1=a*Hd2;
a2=a*Hd3;

nn=[1 n(1) n(1)^2;1 n(2) n(2)^2;1 n(3) n(3)^2]; % the matrix form of the rpm

b=inv(nn); % inverse of the matrix
b0=b*[a0(1);a1(1);a2(1)] % the 1st coefficient of the equation of the head
b1=b*[a0(2);a1(2);a2(2)] % the 2nd coefficient of the equation of the head
b2=b*[a0(3);a1(3);a2(3)] % the 3rd coefficient of the equation of the head

% Input datas for the inlet conditions

m=input('the mass flow rate m,in kg/s== ')
pi=input('the inlet presure of the fluid,bar== ')
n=input ('the rpm of the pump=== ')

% finding the data from the table

[Tdea,hfde,hfgde,hgde,sgde,sfde,sfgde,vfde]=steam_p(pi);
hi=hfde*1e-03;
Inlet_enthalpy=hi % KJ/Kg
specific_volume=vfde % m^3/kg
% vfde=The specifc volume

% the equation of the head will be,H

H=(b0(1,1)+b0(2,1)*n+b0(3,1)*n^2)+(b1(1,1)+b1(2,1)*n+b1(3,1)*n^2)*(m*vfde)+(b2(1,1)+b2(2,1)*n+b2(3,1)*n^2)*((m*vfde)^2); %calculated head
Head_of_the_pump=H

po=pi+H*g*(1/vfde)*1e-5 ; % pump out let pressure ,BAR
Outlet_pressure=po

wp=(m*vfde)*(po-pi)*100 ; % work done by the pump,KW
work_of_the_pump=wp

ho=hi+(wp/m) ; % Outlet fluid enthalpy ,KJ/KG
Outlet_enthalpy=ho

% Plot of the the characteristics of the pump

plot(Q,Hd1,',Q,Hd2,',Q,Hd3,',)
xlabel('the flow rate of the fluid ,Q')
ylabel('the head of the pump,H')
title('the performance curve of the pump')
legend('N=3000','N=2500','N=1700')
grid on

```