Abstract

The objective of the project is to evaluate the best way of adapting the thermal cycle of a Waste-to-Energy (WtE) plant to a higher performance.

Current situation

The current thermal cycle, the furnace/boiler allows a thermal load of 46,511 MWt from the combustion of the waste, generating steam at 61 bar (abs) and 380 ºC, which is sent to a steam turbine. The steam at the outlet of the turbine goes to the aero condenser, from there to the feed tank, and is finally returned to the furnace/boiler.

The impulsion of the feed water is done by an electric pump, although a turbo-pump is available to move the above-mentioned water flow.

A small fraction of steam is withdrawn from the steam drum so as to heat the air used for combustion in the furnace.

Expected situation

Due to different improvements in the furnace system, the admissible thermal load can be increased in 10%, reaching a total of 51,162 MWt. Moreover, a modification in the boiler economizer provides that the temperature of the exit gases (140.000 Nm3/h) can be reduced to 35ºC, which allows to generate even more steam in the boiler.

To deal with this excess of steam, and considering a global modification of the plant, the possibilities are the following:

To use 2.0 tons/h of live steam for a nitrogen oxide reduction facility, this steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank.

To use 2.4 tons/h of live steam in an external drying process, this steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank.

To permanently use the turbo- pump instead of the electric pump, and to send its exit steam to the degasser. This steam replaces, either partially or completely, the steam withdrawn from the main turbine so as to heat the feed tank.

Content

Preface

Origin

The current thermal cycle, the furnace/boiler allows a thermal load of 46,511 MWt from the combustion of the waste, generating steam at 61 bar (abs) and 380 ºC, which is sent to a steam turbine. The steam at the outlet of the turbine goes to the aero condenser, from there to the feed tank, and is finally returned to the furnace/boiler. The impulsion of the feed water is done by an electric pump, although a turbo-pump is available to move the above-mentioned water flow. A small fraction of steam is withdrawn from the steam drum so as to heat the air used for combustion in the furnace.

Motivation

This project work is to evaluate the best way of adapting the thermal cycle of a Waste to Energy (WtE) plant to a higher performance.

Detail Objectives

● Recalculate the new thermal balance of the circuit.

● Determine the electrical output generated by the turbine in the new situation.

● Calculate the net variation of electrical output (grid input) due to the increase of the turbine power output and to the current replacement of the electric pump by the turbine pump.

1. Introduction

Base on number prognostications by leading authorities, electric power will be the fastest growing source of end-use energy throughout the world over the next two decades. Worldwide electricity usage is projected to grow to 23 trillion kWh in 2020, nearly doubling present demand.

The cycle of a steam power plant is the group of interconnected major equipment components selected for optimum thermodynamic characteristics, including pressure, temperatures and capacities, and integrated into a practical arrangement to serve the electrical (and sometimes by-product steam) requirements of a particular project. Selection of the optimum cycle depends upon plant size, cost of money, fuel costs, non-fuel operating costs, and maintenance costs.

Maximum overall efficiency and economy of a steam power cycle are the principal design criteria for plant selection and design. In general, better efficiency, or lower heat rate, is accompanied by higher costs for initial investment, operation and maintenance. However, more efficient cycles are more complex and may be less reliable per unit of capacity or investment cost than simpler and less efficient cycles. Efficiency characteristics can be listed as follows: (1) Higher steam pressures and temperatures contribute to better, or lower, heat rates. (2) For condensing cycles, lower back pressures increase efficiency except that for each particular turbine unit there is a crossover point where lowering back pressure further will commence to decrease efficiency because the incremental exhaust loss effect is greater than the incremental increase in available energy. (3) The use of stage or regenerative feed water cycles improves heat rates, with greater improvement corresponding to larger numbers of such heaters(4) Larger turbine generator units are generally more efficient that smaller units. (5) Multi-stage and multi-valve turbines are more economical than single stage or single valve machines. (6) Steam generators of more elaborate design, or with heat saving accessory equipment are more efficient.

This project work is to evaluate the best way of adapting the thermal cycle of a Waste to Energy (WtE) plant to a higher performance.

The plant uses the steam which generating from the boiler. The boiler allows thermal loads from the combustion which produces the heat from the waste resources. The steam is sent to turbine to generate the power. The steam at the outlet the turbine goes to the air-condenser to condense to liquid, from here the water flows to the feed tank. The tank sends the water to the pump to increase the pressure, by then the water is sent back to the boiler.

Due to the development of the plant, we know the characteristics of the current situation system and the expected situation condition as following:

As the current situation, the current thermal the furnace/boiler allows a thermal load of 46.511 MWt from the combustion which generate the steam at 61 bars (abs) and 380 ˚ C, which is sent to the steam turbine. The steam at the turbine goes to the air-condenser, from there to the feed tank, and is finally returned to the boiler. The impulsion of the feed tank water is done by an electric pump, although a turbo-pump is available to move the above-mentioned water flow. A small fraction of steam is withdrawn from the steam drum so as heat the air used for combustion in the furnace.

The expected situation, due to different in the furnace system, the admissible thermal load can be increased in 10 %, reaching a total of 51.162 MWt. Moreover, a modification in the boiler economizer provides that the temperature of the exit gasses (140,000 Nm ^{3}/ h) can be reduced to 35˚ C, which allows generating even more steam boiler. To deal with this excess of steam, and considering a global modification of the plant, the possibilities are the following: (1) To use 2.0 tons/h of live steam for a nitrogen oxide reduction facility. This steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank. (2) To use 2.4 tons/h of live steam in an external drying process. This steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank. (3) To permanently use the turbo- pump instead of the electric pump, and to send its exit steam to the degasser. This steam replaces, either partially or completely, the steam withdrawn from the main turbine so as to heat the feed tank.

2. The Thermal Cycle of A Waste to Energy Plant

2.1. Power Plant Cycle

The thermal cycle of a Waste-to-Energy (WtE) plant uses the thermal load from the combustion of waste to produce the electricity. The steam mass flow is sent from the boiler. The boiler obtains the thermal energy from the combustion of the waste to generate the steam. This steam is sent to the turbine to generate electricity and other to operate various processes. The turbine uses the steam with high temperature and high pressure for generating electrical output. The exit steam of turbine with low pressure goes to the Air Condenser to condense the water. The water goes to the feed tank, and then the water is sent to the Motor Pump for compression the pressure. Finally the water is fed to the boiler.

The impulsion of the feed water is done by electric pump, although a turbo pump is available to move the water flow. A small fraction of steam is withdrawn from the steam drum to heat the air used for combustion in the furnace.

As we known that the improvement of the furnace/boiler system can be increased the thermal loads 10% of the combustion waste which allows generating more steam in the boiler.

The system works on the current situation. It will be improve to have a higher performance, improvement of the furnace/boiler in the new situation and the expected situation. The situation of the power plant systems are demonstrated as following:

2.2. Current Situation

The current thermal cycle (Fig 2.1), the furnace/boiler allows a thermal load of 46.511 MWt from the combustion of the waste, generating steam at 61 bar (abs) and 380 º C, which is sent to a steam turbine. The steam at the outlet of the turbine goes to the Air condenser, from there to the feed tank, and is finally returned to the furnace/boiler.

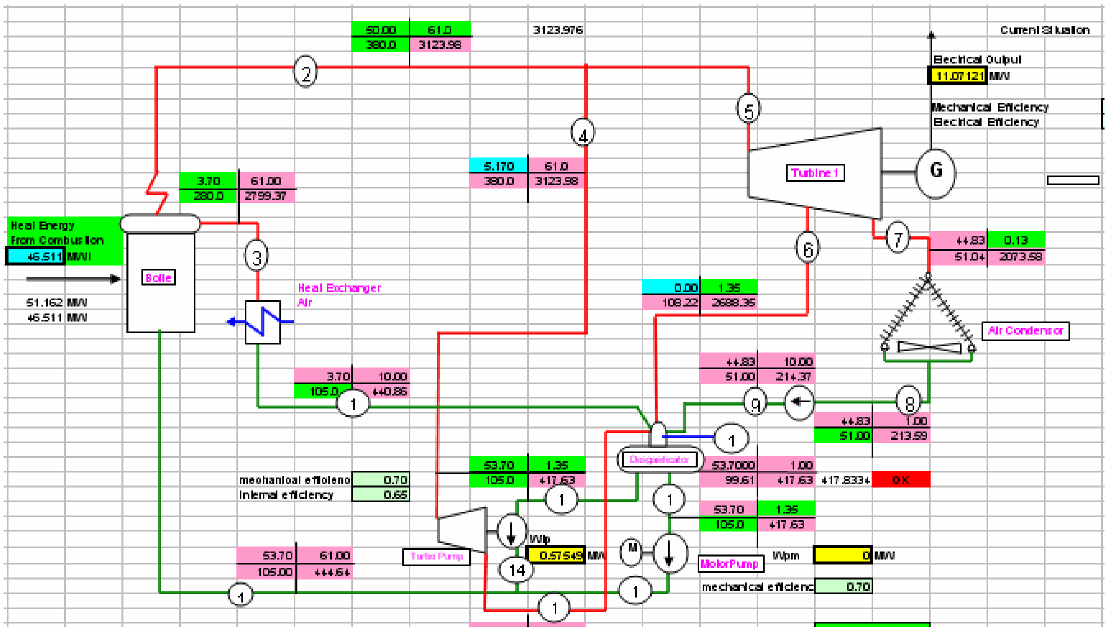

Figure 2.1 schematic of the current situation

The impulsion of feed water is done by an electric pump, although a turbo-pump is available to move the above mentioned water flow.

A small fraction of steam is withdrawn from the steam drum so as to heat the air used for combustion in the furnace.

2.3. Expected Situation

Due to different improvement in the furnace system, the admissible thermal load can be increased in 10%, reaching a total of 51.162 MWt. Moreover, a modification in the boiler economizer provides that temperature of the exit gases (140,000 Nm³/h) can be reduced to 35 º C, which allows to generate even more steam in the boiler.

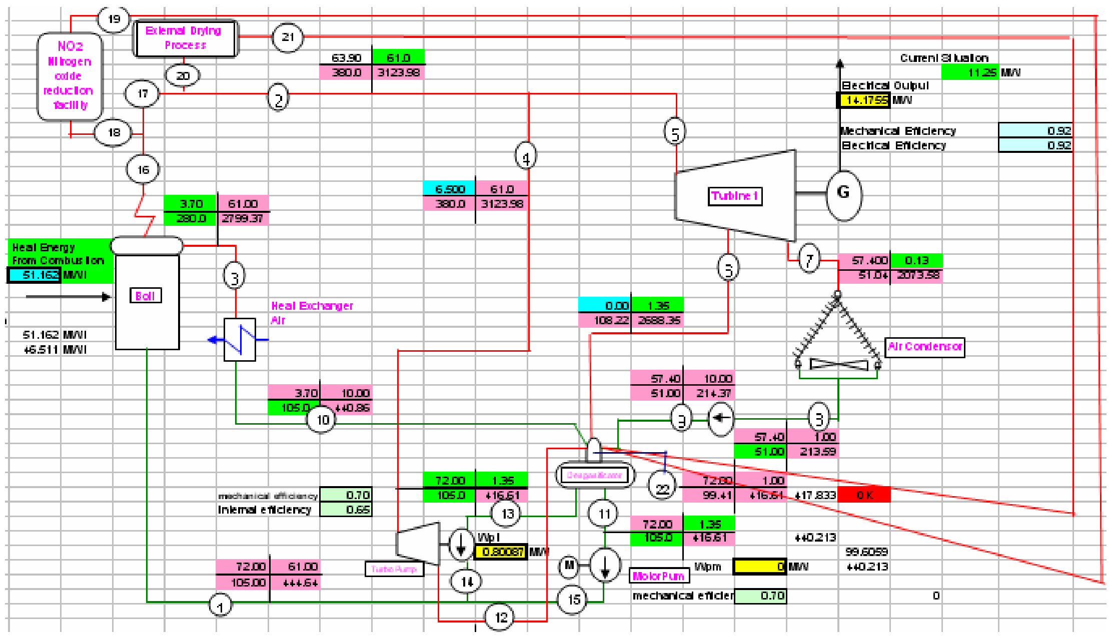

Figure 2.2 schematic of the expected situation

To deal with this excess of steam, and considering a global modification of the plant, the possibilities are following:

- To use 2 t/h of live steam for a nitrogen oxide reduction facility. This steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank

- To use 2.4 t/h of live steam in an external drying reduction process. This steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank.

- To permanently use the turbo pump instead of the electrical pump, and to send its exit to the deaerator. This steam replaces, either partially or completely, the steam withdrawn from the main turbine so as to the feed water tank.

2.4. Main Characteristic of the Devices

As we known, the power plant system works with different devices and generates various processes. The equipments are used in the system for instance: Boiler, Turbine, Air Condenser, Motor Pump, Deaerator, Turbo Pump, Heat Exchanger and Etc; The main characteristics of the equipments are explained as following:

2.4.1. Boiler

A boiler is a closed vessel in which water is heated under pressure. The fluid is then circulated out of the boiler in steam phase for using in the turbine and various processes. The source of heat for a boiler is combustion of Waste for example: the boiler allows the thermal load of 46.511 MWt of the current situation and the thermal load can be increased 10% of the expected situation due to improvement in the furnace. Heat recovery steam generators uses the heat rejected to other processes such as turbines.

2.4.2. Turbine

As we known in the thermal power plant, Steam turbine is used for the generation of electricity. The turbines have one moving part, a rotor assembly, which is a shaft with blades attached. Moving fluid acts on the blades, or the blades react to the flow, so that they rotate and impart energy to the rotor. The turbine is designed for a relatively narrow range of pressure and temperature for instance: in this plant temperature and pressure are kept constant T = 380 º C and P = 61 bar.

2.4.3. Turbo Pump

A turbo pump comprises two main components: a pump and a driving turbine, both mounted on the same shaft. As in the system, the impulsion of the feed water is generated by the motor pump, although the turbo pump is available to move the water flow.

2.4.4. Motor Pump

A motor pump is a device used to move liquids. A pump moves liquids from lower pressure to higher pressure. As in this plant system, the motor pump uses the electricity which is supplied from generating electricity of the turbine.

2.4.5. Air Condenser

An Air condenser, the turbine exhaust steam is piped directly to the air-cooled, finned tube, condenser. The steam trunk main has a large diameter and is as short as possible to reduce pressure losses, so that the cooling banks are usually as close as possible to the turbine.

2.4.6. Deaerator

A deaerator (“desgasificador”), the aim of this item is to bring the water very close to saturation point, to allow separation of non condensable; steam supply must be adjusted to be very close to saturation point but always in the liquid side. We have to balance all the water and the steam input in order to bring the water closely to saturation point.

2.4.7. Air Heat Exchanger

Two fluids: Gas and Steam, of different starting temperatures, flow through the heat exchanger. Steam flows through the tubes and the gas flows outside the tubes but inside the shell. Heat is transferred from steam to the gas through the tube walls in order to transfer heat efficiently. Besides, the flue gas heat exchanger allows us to recover a certain amount of heat, and this affects directly to boiler efficiency.

2.4.8. Nitrogen Oxide Facility

Due to the excess of steam and considering a global modification of the plant, we use 2 tons/h of live steam for a nitrogen oxide reduction facility. This steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank.

2.4.9. External Drying Process

Due to the excess of steam and considering a global modification of the plant, we use 2.4 tons/h of live steam in an external drying process. This steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank.

3. Calculation Theory

3.1. Energy Balance

Energy balance for any system undergoing any kind of process was expressed as:

Net Energy Transfer by heat, work and Mass	=	Change in internal, kinetic, potential, etc., energies

$$E_{in} - E_{out} = E_{system} \qquad \text{(eq 3.1)}$$

Or, in the rate form, as:

Rate of Net Energy Transfer by heat, work and Mass	=	Rate of Change in internal, kinetic, potential, etc., energies

$$\dot{E}_{in} - \dot{E}_{out} = dE_{system}/dt \qquad \text{(eq 3.2)}$$

For constant rates, the total quantities during a time interval Δt are related to the quantities per unit time as:

$$Q = \dot{Q}\Delta t,\ W = \dot{W}\Delta t \text{ and } \Delta E = (dE/dt)\,\Delta t \qquad \text{(eq 3.3)}$$

The energy balance can be expressed on a per unit mass basis as:

$$e_{in} - e_{out} = \Delta e_{system} \qquad \text{(eq 3.4)}$$

3.2. Flow work and the Energy of a Flowing Fluid

3.2.1. Total Energy of a Flowing Fluid

The total energy of a simple compressible system consists of three parts: internal, kinetic, and potential energies. On a unit-mass basis, it is expressed as:

$$e = u + ke + pe = u + \frac{V^2}{2} + gz \qquad \text{(eq 3.5)}$$

Where V is the velocity and z is the elevation of the system relative to some external reference point.

The fluid entering or leaving a control volume possesses an additional form of energy. The total energy of a flowing fluid on a unit-mass basis (denoted by ϴ) becomes:

$$\Theta = Pv + e = Pv + (u + ke + pe) \qquad \text{(eq 3.6)}$$

But the combination Pv + u have been previously defined as the enthalpy h.

So the relation in Eq. 2.5 reduces to:

$$\Theta = h + ke + pe = h + \frac{V^2}{2} + gz \qquad \text{(eq 3.7)}$$

By using the enthalpy instead of the internal energy to represent the energy of a flowing fluid, one does not need to be concerned about the flow work. The energy associated with pushing the fluid into or out of the control volume is automatically taken care of by enthalpy.

3.2.2. Energy Transport by Mass

Noting that ϴ is total energy per unit mass, the total energy of a flowing fluid of mass m is simply mϴ, provided that the properties of the mass m are uniform. Also, when a fluid stream with uniform properties is flowing at a mass flow rate of $\dot{m}$, the rate of energy flow with that stream is $\dot{m}$ ϴ (Fig. 3.1).That is,

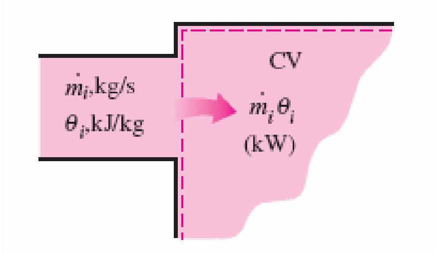

Figure 3.1 the product $\dot{m}$ i ϴi is the energy transported into control volume by mass per unit time.

Amount of energy transport: E mass = mΘ = m (h + $\frac{V^2}{2}$ + gz) (eq 3.8)

Rate of energy transport: $\dot{E}\,mass = \dot{m}\,\Theta = \dot{m}$ (h + $\frac{V^2}{2}$ + gz) (eq 3.9)

When the kinetic and potential energies of a fluid stream are negligible, as is often the case, these relations simplify to E mass = mh and $\dot{E}\,mass = \dot{m}$ h. In general, the total energy transported by mass into or out of the control volume is not easy to determine since the properties of the mass at each inlet or exit may be changing with time as well as over the cross section. Thus, the only way to determine the energy transport through an opening as a result of mass flow is to consider sufficiently small differential masses δm that have uniform properties and to add their total energies during flow.

3.3. Energy Analysis of Steady Flow System

A large number of engineering devices such as turbines, compressors, and nozzles operate for long periods of time under the same conditions once the transient start-up period is completed and steady operation is established, and they are classified as steady-flow devices. Processes involving such devices can be represented reasonably well by a somewhat idealized process, called the steady-flow process as a process during which a fluid flows through a control volume steadily. That is, the fluid properties can change from point to point within the control volume, but at any point, they remain constant during the entire process. (Remember, steady means no change with time). During a steady-flow process, no intensive or extensive properties within the control volume change with time. Thus, the volume V, the mass m, and the total energy content E of the control volume remain constant (Fig.3.2).

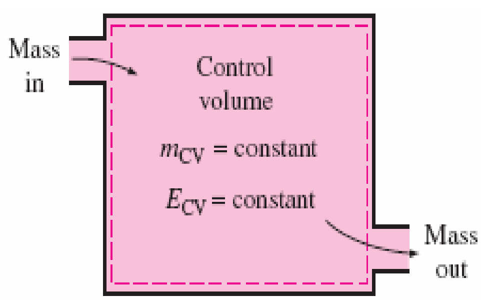

Figure 3.2 under steady-flow conditions, the mass and energy contents of a control volume remain constant.

As a result, the boundary work is zero for steady-flow systems (since Vcv = constant), and the total mass or energy entering the control volume must be equal to the total mass or energy leaving it (since m cv = constant and Ecv =constant). These observations greatly simplify the analysis.

The fluid properties at an inlet or exit remain constant during a steady flow process. The properties may, however, be different at different inlets and exits. They may even vary over the cross section of an inlet or an exit. However, all properties, including the velocity and elevation, must remain constant with time at a fixed point at an inlet or exit. It follows that the mass flow rate of the fluid at an opening must remain constant during a steady flow process (Fig. 3.3)

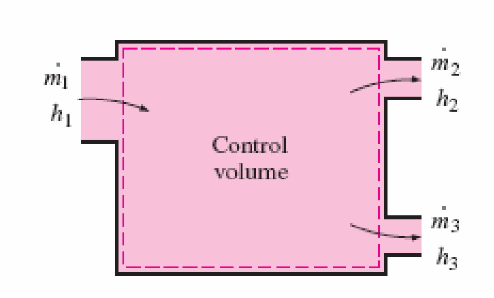

Figure 3.3 under steady-flow conditions, the fluid properties at an inlet or exit remain constant (do not change with time).

The mass balance for a general steady-flow system was given as:

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \qquad \text{(eq 3.10)}$$

The mass balance for a single-stream (one-inlet and one-outlet) steady-flow system was given as:

$$\dot{m}1 = \dot{m}2 \quad \text{or} \quad \rho1V1A1 = \rho2V2A2 \qquad \text{(eq 3.11)}$$

Where the subscripts 1 and 2 denote the inlet and the exit states, respectively, ρ is density, V is the average flow velocity in the flow direction, and A is the cross-sectional area normal to flow direction.

During a steady-flow process, the total energy content of a control volume remains constant (Ecv = constant), and thus the change in the total energy of the control volume is zero (ΔEcv = 0). Therefore, the amount of energy entering a control volume in all forms (by heat, work, and mass) must be equal to the amount of energy leaving it. Then the rate form of the general energy balance reduces for a steady-flow process to:

$$\dot{E}\text{ in} - \dot{E}\text{ out} = dE\text{ system}/dt = 0 \qquad \text{(eq 3.12)}$$

Or Energy Balance: $\dot{E}$ in - $\dot{E}$ out (eq 3.13)

Noting that energy can be transferred by heat, work, and mass only, the energy balance in Eq. 3.13 for a general steady-flow system can also be written more explicitly as:

$$\dot{Q}\text{ in} + \dot{W}\text{ in} + \sum_{in}\dot{m}\theta = \dot{Q}\text{ out} + \dot{W}\text{ out} + \sum_{out}\dot{m}\theta \qquad \text{(eq 3.14)}$$

Or

$$\dot{Q}\text{ in} + \dot{W}\text{ in} + \sum_{in}\dot{m}\ (h + \frac{V^2}{2} + gz) = \dot{Q}\text{ out} + \dot{W}\text{ out} + \sum_{out}\dot{m}\ (h + \frac{V^2}{2} + gz) \qquad \text{(eq 3.15)}$$

The energy balance relation just given is intuitive in nature and is easy to use when the magnitudes and directions of heat and work transfers are known. When performing a general analytical study or solving a problem that involves an unknown heat or work interaction, however, we need to assume a direction for the heat or work interactions. In such cases, it is common practice to assume heat to be transferred into the system (heat input) at a rate of heat, and work produced by the system (work output) at a rate of $\dot{W}$, and $\dot{Q}$ then solve the problem. The first-law or energy balance relation in that case for a general steady-flow system becomes:

$$\dot{Q} - \dot{W} = \sum_{out}\dot{m}\ (h + \frac{V^2}{2} + gz) - \sum_{in}\dot{m}\ (h + \frac{V^2}{2} + gz) \qquad \text{(eq 3.16)}$$

Obtaining a negative quantity for $\dot{Q}$ or $\dot{W}$ simply means that the assumed direction is wrong and should be reversed for single-stream devices, the steady-flow energy balance equation becomes:

$$\dot{Q} - \dot{W} = \dot{m}\ (h2 - h1 + \frac{V^2}{2}2 - \frac{V^2}{2}1 + g\ (z2 - z1) \qquad \text{(eq 3.17)}$$

Dividing Eq 3.17 by $\dot{m}$ gives the energy balance on a unit-mass basis as:

$$q - w = (h_2 - h_1 + \frac{V_2^2}{2} - \frac{V_1^2}{2} + g(z_2 - z_1) \qquad \text{(eq 3.18)}$$

When the fluid experiences negligible changes in its kinetic and potential energies (that is, $\Delta ke = 0$, $\Delta pe = 0$), the energy balance equation is reduced further to:

$$q - w = h_2 - h_1 \qquad \text{(eq 3.19)}$$

3.4. Some Steady-Flow Engineering Devices

Many engineering devices operate essentially under the same conditions for long periods of time. The components of a steam power plant (turbines, compressors, heat exchangers, and pumps), for example, operate nonstop for months before the system is shut down for maintenance. Therefore, these devices can be conveniently analyzed as steady-flow devices.

3.4.1. Turbines and Compressors

In steam, gas, or hydroelectric power plants, the device that drives the electric generator is the turbine. As the fluid passes through the turbine, work is done against the blades, which are attached to the shaft. As a result, the shaft rotates, and the turbine produces work. Compressors, as well as pumps and fans, are devices used to increase the pressure of a fluid. Work is supplied to these devices from an external source through a rotating shaft. Therefore, compressors involve work inputs.

Even though these three devices function similarly, they do differ in the tasks they perform. A fan increases the pressure of a gas slightly and is mainly used to mobilize a gas. A compressor is capable of compressing the gas to very high pressures. Pumps work very much like compressors except that they handle liquids instead of gases. Note that turbines produce power output whereas compressors, pumps, and fans require power input. Heat transfer from turbines is usually negligible ($\dot{Q} = 0$) since they are typically well insulated. Heat transfer is also negligible for compressors unless there is intentional cooling. Potential energy changes are negligible for all of these devices ($\Delta pe = 0$). The velocities involved in these devices, with the exception of turbines and fans, are usually too low to cause any significant change in the kinetic energy ($\Delta ke = 0$). The fluid velocities encountered in most turbines are very high, and the fluid experiences a significant change in its kinetic energy. However, this change is usually very small relative to the change in enthalpy, and thus it is often disregarded.

3.4.2. Heat Exchangers

As the name implies, heat exchangers are devices where two moving fluid streams exchange heat without mixing. Heat exchangers are widely used in various industries, and they come in various designs. The simplest form of a heat exchanger is a double-tube (also called tube and shell) heat exchanger, shown in Fig. 3.4. It is composed of two concentric pipes of different diameters. One fluid flows in the inner pipe, and the other in the annular space between the two pipes. Heat is transferred from the hot fluid to the cold one through the wall separating them. Sometimes the inner tube makes a couple of turns inside the shell to increase the heat transfer area, and thus the rate of heat transfer. The mixing chambers discussed earlier are sometimes classified as direct-contact heat exchangers.

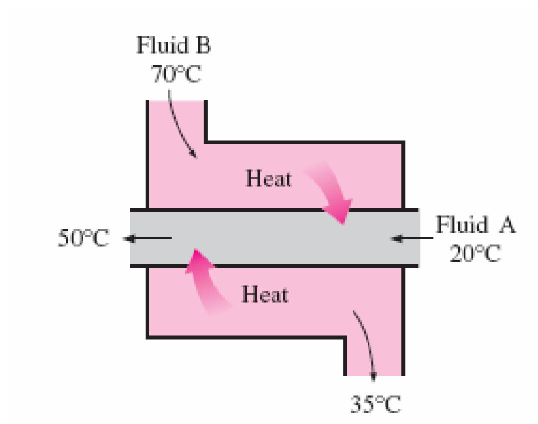

Figure 3.4 a heat exchanger can be as simple as two concentric pipes.

The conservation of mass principle for a heat exchanger in steady operation requires that the sum of the inbound mass flow rates equal the sum of the outbound mass flow rates. This principle can also be expressed as follows: Under steady operation, the mass flow rate of each fluid stream flowing through a heat exchanger remains constant. The conservation of mass principle for a heat exchanger in steady operation requires that the sum of the inbound mass flow rates equal the sum of the outbound mass flow rates. This principle can also be expressed as follows: Under steady operation, the mass flow rate of each fluid stream flowing through a heat exchanger remains constant.

$\dot{Q}$ .becomes zero, since the boundary for this case lies just beneath the insulation and little or no heat crosses the boundary (Fig. 3.5). If, however, only one of the fluids is selected as the control volume, then heat will cross this boundary as it flows from one fluid to the other and $\dot{Q}$ will not be zero. In fact, $\dot{Q}$ in this case will be the rate of heat transfer between the two fluids.

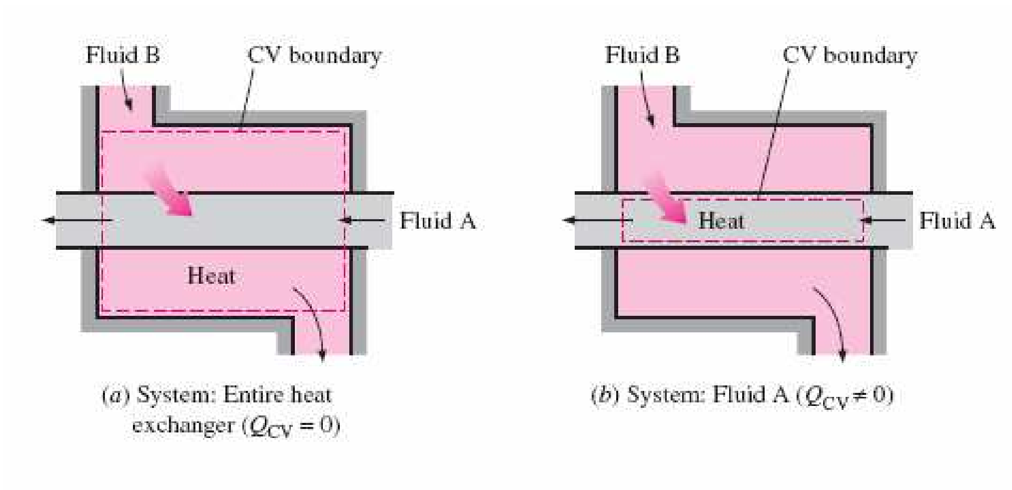

Figure 3.5 the heat transfer associated with a heat exchanger may be zero or nonzero depending on how the control volume is selected.

3.4.3. Pipe and Duct Flow

The transport of liquids or gases in pipes and ducts is of great importance in many engineering applications. Flow through a pipe or a duct usually satisfies the steady-flow conditions and thus can be analyzed as a steady-flow process. This, of course, excludes the transient start-up and shut-down periods. The control volume can be selected to coincide with the interior surfaces of the portion of the pipe or the duct that we are interested in analyzing.

Under normal operating conditions, the amount of heat gained or lost by the fluid may be very significant, particularly if the pipe or duct is long (Fig. 3.6). Sometimes heat transfer is desirable and is the sole purpose of the flow. Water flow through the pipes in the furnace of a power plant, the flow of refrigerant in a freezer, and the flow in heat exchangers are some examples of this case. At other times, heat transfer is undesirable, and the pipes or ducts are insulated to prevent any heat loss or gain, particularly when the temperature difference between the flowing fluid and the surroundings is large. Heat transfer in this case is negligible.

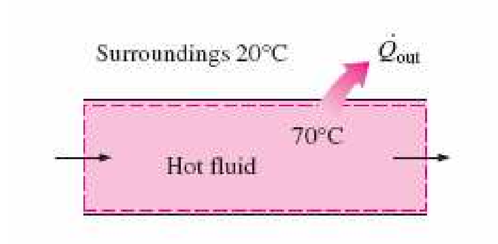

Figure 3.6 Heat losses from a hot fluid flowing through an
uninsulated pipe or duct to the cooler environment may be very significant.

3.5. Isentropic Process

We mentioned earlier that the entropy of a fixed mass can be changed by (1) heat transfer and (2) irreversibilities. Then it follows that the entropy of a fixed mass does not change during a process that is internally reversible and adiabatic (Fig. 3.7). A process during which the entropy remains constant is called an isentropic process. It is characterized by

Isentropic Process: $\Delta s = 0$ or $s2 = s1$ (eq 3.20)

That is, a substance will have the same entropy value at the end of the process as it does at the beginning if the process is carried out in an isentropic manner.

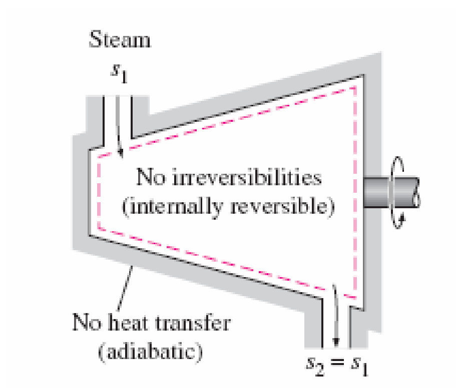

Figure 3.7 during an internally reversible, adiabatic (isentropic) process,
the entropy remains constant.

Many engineering systems or devices such as pumps, turbines, nozzles, and diffusers are essentially adiabatic in their operation, and they perform best when the irreversibilities, such as the friction associated with the process, are minimized. Therefore, an isentropic process can serve as an appropriate model for actual processes. Also, isentropic processes enable us to define efficiencies for processes to compare the actual performance of these devices to the performance under idealized conditions.It should be recognized that a reversible adiabatic process is necessarily isentropic (s2 = s1), but an isentropic process is not necessarily a reversible adiabatic process. (The entropy increase of a substance during a process as a result of irreversibilities may be offset by a decrease in entropy as a result of heat losses, for example.) However, the term isentropic process is customarily used in thermodynamics to imply an internally reversible, adiabatic process.

4. Main Characteristics of X Steam Function

In this chapter demonstrates the main characteristics of the X steam function used in calculation the thermodynamic properties of steam. The program X Stream is used for the main program which we use its code to find out the property of the steam with the different conditions, for instance we know temperature and pressure then we can find the enthalpy, entropy, density and etc.

In actually, many programs can be used to calculate the thermodynamics properties of the steam for instance: X Steam program and Win Steam program which we can use it to find out the value of enthalpy, entropy and other properties. We compared the value of the steam properties by using the several program functions and also comparing with the steam properties table of the reliable book for example: Van Wylen, Fundamentals of Classical Thermodynamics and Yunus A. Cengel and Micheal A. Boles Thermodynamics An Engineering Approach. The exit value of the X Steam program are given values closely the same the value in the steam table and other program.

We used the program X steam because the exit values which we getting are quite reliable, this program is used for the free software, the program is available at the internet and program X steam is prepared by Magnus Holmgren, www.x-eng.com

Main characteristics of the X Steam function are demonstrated as following:

4.1. Temperature Code

Tsat_p	Saturation temperature
T_ph	Temperature as a function of pressure and enthalpy
T_ps	Temperature as a function of pressure and entropy
T_hs	Temperature as a function of enthalpy and entropy

4.2. Pressure Code

Psat_T	Saturation pressure
P_hs	Pressure as a function of h and s.

P_hrho	Pressure as a function of h and rho (density).

4.3. Enthalpy Code

hV_p	Saturated vapor enthalpy
hL_p	Saturated liquid enthalpy
hV_T	Saturated vapor enthalpy
hL_T	Saturated liquid enthalpy
h_pT	Enthalpy as a function of pressure and temperature
h_ps	Enthalpy as a function of pressure and entropy
h_px	Enthalpy as a function of pressure and vapor fraction
h_Tx	Enthalpy as a function of temperature and vapor fraction
h_prho	Enthalpy as a function of pressure and density

4.4. Specific Volume Code

vV_p	Saturated vapor volume
vL_p	Saturated liquid volume
vV_T	Saturated vapor volume
vL_T	Saturated liquid volume
v_pT	Specific volume as a function of pressure and temperature
v_ph	Specific volume as a function of pressure and enthalpy
v_ps	Specific volume as a function of pressure and entropy

4.5. Density Code

rhoV_p	Saturated vapor density

rhoL_p	Saturated liquid density
rhoV_T	Saturated vapor density
rhoL_T	Saturated liquid density
rho_pT	Density as a function of pressure and temperature
rho_ph	Density as a function of pressure and enthalpy
rho_ps	Density as a function of pressure and entropy

4.6. Specific Entropy Code

sV_p	Saturated vapour entropy
sL_p	Saturated liquid entropy
sV_T	Saturated vapour entropy
sL_T	Saturated liquid entropy
s_pT	Specific entropy as a function of pressure and temperature
s_ph	Specific entropy as a function of pressure and enthalpy

4.7. Specific Internal Energy Code

uV_p	Saturated vapor internal energy
uL_p	Saturated liquid internal energy
uV_T	Saturated vapor internal energy
uL_T	Saturated liquid internal energy
u_pT	Specific internal energy as a function of pressure and temperature
u_ph	Specific internal energy as a function of pressure and enthalpy
u_ps	Specific internal energy as a function of pressure and entropy

4.8. Specific Isobaric Heat Capacity Code

CpV_p	Saturated vapor heat capacity
CpL_p	Saturated liquid heat capacity
CpV_T	Saturated vapor heat capacity
CpL_T	Saturated liquid heat capacity
Cp_pT	Specific isobaric heat capacity as a function of pressure and temperature
Cp_ph	Specific isobaric heat capacity as a function of pressure and enthalpy
Cp_ps	Specific isobaric heat capacity as a function of pressure and entropy

4.9. Specific Isochoric Heat Capacity Code

CvV_p	Saturated vapor isochoric heat capacity
CvL_p	Saturated liquid isochoric heat capacity
CvV_T	Saturated vapor isochoric heat capacity
CvL_T	Saturated liquid isochoric heat capacity
Cv_pT	Specific isochoric heat capacity as a function of pressure and temperature
Cv_ph	Specific isochoric heat capacity as a function of pressure and enthalpy
Cv_ps	Specific isochoric heat capacity as a function of pressure and entropy

4.10. Speed of Sound Code

wV_p	Saturated vapor speed of sound
wL_p	Saturated liquid speed of sound
wV_T	Saturated vapor speed of sound
wL_T	Saturated liquid speed of sound
w_pT	Speed of sound as a function of pressure and temperature
w_ph	Speed of sound as a function of pressure and enthalpy
w_ps	Speed of sound as a function of pressure and entropy

4.11. Synamic Viscosity Code

my_pT	Viscosity as a function of pressure and temperature
my_ph	Viscosity as a function of pressure and enthalpy
my_ps	Viscosity as a function of pressure and entropy

4.12. Thermal Conductivity Code

tcL_p	Saturated vapor thermal conductivity
tcV_p	Saturated liquid thermal conductivity
tcL_T	Saturated vapor thermal conductivity
tcV_T	Saturated liquid thermal conductivity
tc_pT	Thermal conductivity as a function of pressure and temperature
tc_ph	Thermal conductivity as a function of pressure and enthalpy
tc_hs	Thermal conductivity as a function of enthalpy and entropy

4.13. Surface Tension Code

st_T Surface tension for two phase water/steam as a function of T

st_p Surface tension for two phase water/steam as a function of T

4.14. Vapor Fraction Code

x_ph Vapor fraction as a function of pressure and enthalpy

x_ps Vapor fraction as a function of pressure and entropy

4.15. Vapor Volume Fraction Code

vx_ph Vapor volume fraction as a function of pressure and enthalpy

vx_ps Vapor volume fraction as a function of pressure and entropy

5. Resolution of the Problem

This chapter demonstrates the calculation of the maincharacteristic of the devices used of the power plant system. As we known that In the plant system has several equipment and different main characteristics. Due to the the different of the equipment, the condition of each devices are different, for instance we known that the turbines are designed for a relative narrow range of pressure and temperature so that we can find the enthalpy and entropy from these values. We can find the mass flow by balancing mass conservation. We can balance the power used of the motor pump, we can calculate the efficiency of the boiler when the boiler is increased the thermal load input and etc;.

5.1. Boiler

In the Current situation, Boiler allows a thermal load of 46.511MWt from the combustion of the waste which generating steam at 61 bar (abs) and 380 º C, which is sent to a steam turbine.

For the New situation, Due to different improvement in the furnace system, the admissible thermal load can be increased in 10%, reaching a total 51.162 MWt.

The expected situation, Moreover a modification in the boiler economizer provides the temperature of exit gases (140,000 Nm³/h) can be reduced to 35 º C which allows generating even more steam in the boiler.

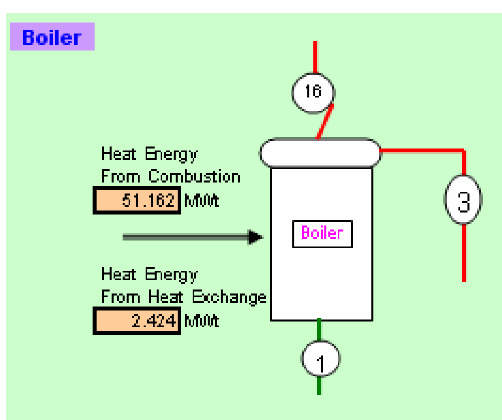

Figure 5.1 Schematic of Boiler

Point 1

The impulsion of the feed water is done by an electric pump although a turbo pump is available to move the water flows. The characteristics of this point demonstrate as following:

Mass flow: We can find the mass flow of this point by $m1 = m3+m16$

Pressure: We know the pressure $P1 = 61$ bar which is the maximum pressure of the system.

Temperature: We know the temperature $T1 = 105\ ^\circ C$ which is the same temperature T14, T15.

Enthalpy: We can calculate Enthalpy h1 by using the function h_pt (p, t)

$$h\,1 = h\_pt\,(P1, T1)$$

Point 3

A small fraction of steam is withdrawn from the steam drum so as to heat the air used for combustion in the furnace.

Mass flow: As we knows from the current situation mass flow of this point $m3 = 3.7$ t/h. In actually we can adjust mass flow m3 by control valve.

Pressure: We know the pressure $P3 = 61$ bar as the pressure of the boiler generating steam.

Temperature: The steam leaves out of the steam drum with temperature $T3 = 280\ ^\circ C$

Enthalpy: We can calculate Enthalpy h3 by using the function h_pt (p, t)

$$h\,3 = h\_pt\,(P3, T3)$$

Point 16

The steam is sent to the turbine and other useful parts. The characteristics are shown as following:

Mass flow: We can find the mass flow m16 by balancing mass flow, $m16 = m17 + m18$

Pressure: We know the pressure $P16 = 61$ bar as the pressure of the boiler generating steam.

Temperature: The steam leaves out of the boiler which the boiler controls the temperature T16 = 380 º C.

Enthalpy: We can calculate Enthalpy h16 by using the function h_pt (p, t)

h 16 = h_pt (P16, T16)

5.2. Turbine

Turbine receives the steam from the boiler with the high temperature and high pressure. As we known, turbine uses the steam to generate the electrical output. The characteristics of the turbine are designed for a relative narrow range of pressure and temperature.

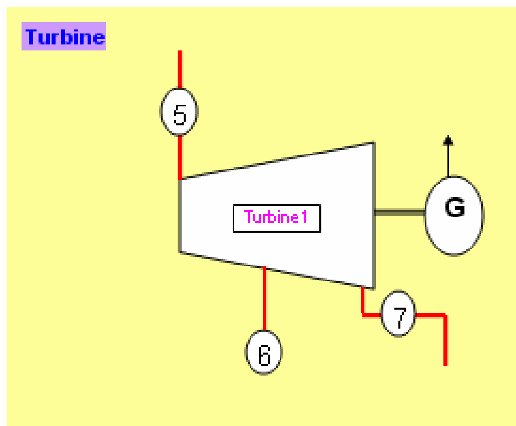

Figure 5.2 Schematic of Turbine

Point 5

The steam enters to the turbine under condition of high pressure and high temperature. The characteristics demonstrate as following:

Mass flow: We can find the mass flow m5 by balancing mass flow, m5 = m2-m4.

Pressure: Pressure of this point is designed for relative pressure range of the turbine. As we known pressure P5 = 61 bar.

Temperature: Temperature of this point is designed for relative range temperature of the turbine. As we known temperature T5 = 380 º C.

Enthalpy: We can calculate Enthalpy h5 by using the function h_pt (p, t)

h 5 = h_pt (P5, T5)

Entropy: We can calculate Entropy s16 by using the function s_pt (p, t)

s 5 = s_pt (P5, T5)

Point 6

The fraction of the steam is withdrawn from the turbine and sent to a deaerator. The aim of this pipe line sends the steam to the deaerator to adjust the water closely to saturation point.

Mass flow: The mass flow of this pipe line is adjusted by control valve to find the appropriated mass flow to send to deaerator.

Pressure: As we known pressure of this point, P6 = 1.35 bar.

Temperature: We can calculate temperature T6 by using the function Tsat_p (p)

T6 = Tsat_p (P6)

Enthalpy: We can calculate enthalpy h6 by using the function hV_p (p).

h 6 = hV_p(P6)

Point 7

The steam leaves out of the turbine in the condition low pressure (less than atmospheric pressure). The steam of this point is sent to an Air Condenser. The characteristics of this point are shown as following:

Mass flow: We can find the mass flow m7 by balancing mass flow, m7 = m5-m6.

Pressure: We know the pressure P7 = 0.13 bar below atmospheric pressure which the pressure is vacuum for condensing the steam phase to liquid phase.

Temperature: We can calculate the temperature T7 by using the function t_ps (p, s).

T7 = t_ps (P7, S7)

Enthalpy: We can calculate the enthalpy h7 by using the function h_ps (p, s).

h 7 = h_ps (P7, S7)

Entropy: We know the entropy s7 = s5. As we known, the entropy of leaved out turbine is equal entropy of entered to turbine.

We can calculate the Electrical Output as:

Electrical Output

= Mechanical Efficiency

x Electrical Efficiency

x ((m7 x (h5-h7) + m6 x (h5-h6))

5.3. Air Condenser

An air condenser receives the steam which it leaves out of turbine. The steam condenses to the phase of liquid. The liquid will be sent back to the system. The characteristics of the Air are shown as following:

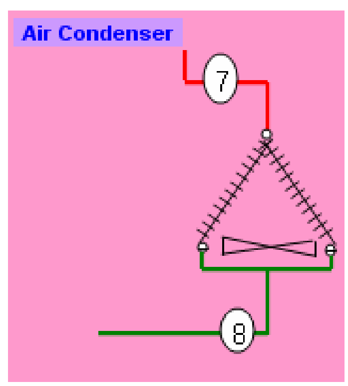

Figure 5.3 Schematic of Air Condenser

Point 8

The liquid in the Air Condenser tank is sent to the system. All of the water is in the phase of liquid under condition of low pressure. The characteristics of this point are present as below:

Mass flow: We can find the mass flow m8 by balancing mass flow, m8 = m7.

Pressure: As we known, the pressure leave out of the air condenser is atmospheric pressure as P8 = 1 bar.

Temperature: The temperature of this point T8 = 51 º C.

Enthalpy: We can calculate the enthalpy h8 by using the function h_pt (p, t).

$$h\,8 = h\_pt\,(P8, T8)$$

5.4. Pump (point 8 to point 9)

A pump moves liquids from lower pressure (point 8) to higher pressure (point 9), and overcomes this difference in pressure by adding energy to the system. The characteristics of the pump are shown as below:

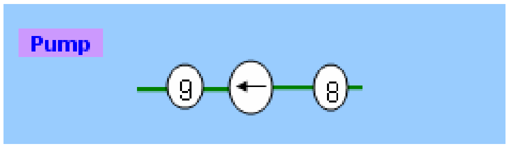

Figure 5.4 Schematic of the Pump (point 8 to point 9)

Point 9

The impulsion of the water is done by the pump; the water comes from low pressure (point 8) to the high pressure (point 9). The characteristics demonstrate as following:

Mass flow: We can find the mass flow m9 by balancing mass flow, m9 = m8 = m7.

Pressure: The water leaves out of the pump, the pressure P9 = 10 bar.

Temperature: The temperature of this point T9 = T8. Assumption: As we known, the temperature of the water goes to the pump is equal the temperature of water leaves out the pump.

Enthalpy: We can calculate the enthalpy h9 by using the function of h_pt (p, t).

h 9 = h_pt (P9, T9)

We can find the power of pump working from point 8 to point 9 as:

Wp 8-9 = m8 x (h9 – h8) / mechanical efficiency

5.5. Deaerator

Deaerator (desgasificador), the aim of this item is to bring the water very close to saturation point, to allow separation of non condensable; steam supply must be adjusted to be very close to saturation point, but always in the liquid side. The liquid several pipe lines come to the deaerator with different characteristics.

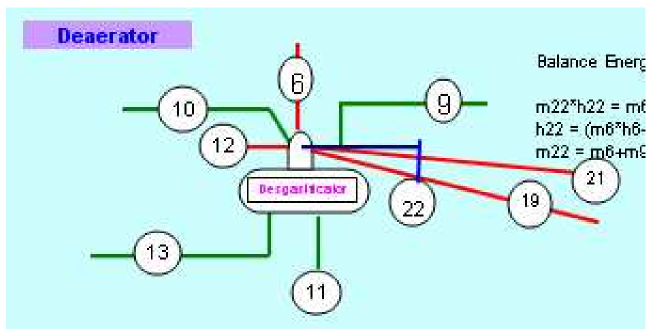

Figure 5.5 Schematic of the Deaerator

Point 10

The water leaves out of the air heat exchanger, and then this liquid are sent to the deaerator which the main characteristics of water show as below:

Mass flow: As we known, we can find the mass flow m10 by balancing mass flow, m10 = m3.

Pressure: As we known, the liquid goes out of air heat exchanger with control pressure P10 = 10 bar.

Temperature: The temperature of this point T10 = 105 º C.

Enthalpy: We can calculate the enthalpy h10 by using the function h_pt (p, t).

h 10 = h_pt (P10, T10)

Point 11

The water goes out from the deaerator to the motor pump. The water of this point is closely to the saturation point. The main characteristics of this point are shown as below:

Mass flow: We can calculate the mass flow m11 by balancing the mass flow, m11 = m6 + m9 + m10 + m12 + m13 + m19 + m21.

Pressure: The pressure of this point P11 = 1.35 bar.

Temperature: The temperature of this point T11 = 105 º C.

Enthalpy: We can calculate the enthalpy h11 by balancing energy,

(m11 x h11) = ((m6 x h6) + (m9 x h9) + (m10 x h10) + (m12 x h12) + (m13 x h13) + (m21 x h21) + (m19 x h19))

So then;

h11 = ((m6 x h6) + (m9 x h9) + (m10 x h10) + (m12 x h12) + (m13 x h13) + (m21 x h21) + (m19 x h19)) / m11

Point 12

The steam goes out of the turbo pump and returned hot to the deaerator. The steam characteristics of this point are demonstrated as below:

Mass flow: We can find the mass flow m12 by balancing mass flow, m12 = m4.

Remark: mass flow m4 is adjusted to the turbo pump to supply the power used of pump working (point13 to point14).

Pressure: The pressure of this point P12 = 1.35 bar.

Temperature: We can calculate the temperature T12 by using the function, t_ps (p, s).

T12 = t_ps (P12, S12)

Enthalpy: We can calculate the enthalpy h12 by using the function, h_ps (p, s).

h 12 = h_ps (P12, S12)

Point 13

This pipe line is worked to support the power of pump working when the turbo pump is available to impulse the water flow. The main characteristics of this point are shown as below:

Mass flow: We can calculate the mass flow m13 by balancing the mass flow, m13 = m6 + m9 + m10 + m12 + m11 + m19 + m21.

Noting: When the mass flow m13 is working and the mass flow m11 will be closed. Due to when the turbo pump is working and the motor pump will be stop.

Pressure: The pressure of this point P13 = 1.35 bar.

Temperature: The temperature of this point T13 = 105 ^{o} C.

Enthalpy: We can calculate the enthalpy h11 by balancing energy,

(m13 x h13) = ((m6 x h6) + (m9 x h9) + (m10 x h10) + (m12 x h12) + (m11 x h11) + (m21 x h21) + (m19 x h19))

So then;

h13 = ((m6 x h6) + (m9 x h9) + (m10 x h10) + (m12 x h12) + (m11 x h11) + (m21 x h21) + (m19 x h19)) / m13

Noting: When the mass flow m13 is working and the mass flow m11 will be closed. Due to when the turbo pump is working and the motor pump will be stop.

Entropy: We know the entropy s12 = s4. As we known, the entropy of leaved out turbine is equal entropy of entered to turbine.

Point 19

The small fraction of steam is withdrawn from the pipe line 16 to use 2 t/h of live steam for a nitrogen oxide reduction facility. This steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank. The main characteristics of point 19 are present as below:

Mass flow: As we known that mass flow m19 = m18 = 2 t/h to use for a nitrogen oxide reduction facility.

Pressure: The low pressure steam goes out of the Nitrogen oxide reduction facility .The pressure of this point P19 = 1.35 bar.

Temperature: We can calculate the temperature T19 by using the function, Tsat_p(p).

$$T19 = Tsat\_p\ (P19)$$

Enthalpy: We can calculate the enthalpy h19 by using the function, hL_p(p).

$$h\ 19 = hL\_p(P19)$$

Point 21

The small fraction of steam is withdrawn from the pipe line 17 to use 2.4 t/h of live steam in the external drying process. These steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank. The main characteristics of point 21 are present as below:

Mass flow: As we known that mass flow m21 = m20 = 2.4 t/h to use of live steam in the external drying process.

Pressure: The low pressure steam goes out of the external drying process .The pressure of this point P21 = 1.35 bar.

Temperature: We can calculate the temperature T21 by using the function, Tsat_p(p).

$$T21 = Tsat\_p\ (P21)$$

Enthalpy: We can calculate the enthalpy h19 by using the function, hL_p(p).

$$h\ 21 = hL\_p(P21)$$

5.6. Motor Pump

Motor pump impulses liquids from lower pressure (point 11) to higher pressure (point 15). The impulsion of feed water is sent to the system. The main characteristics of the motor pump are shown as below:

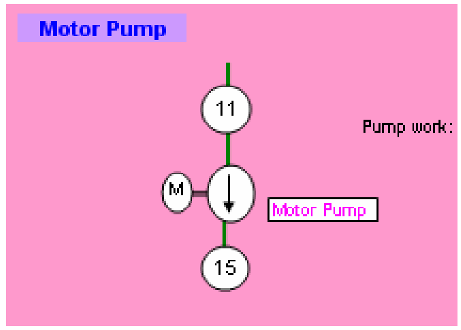

Figure 5.6 Schematic of Motor Pump

Point 15

The impulsion of feed water is sent out by the motor pump. The water of pipe line 15 is high pressure and this pressure is the highest pressure of the system. The main functions of point 15 are shown as below:

Mass flow: We can calculate the mass flow m15 by balancing mass flow m15 = m11.

Pressure: As we known that the pressure of this pipe line is the highest pressure. The pressure P15 = 61 bar.

Temperature: The temperature of this point T15 = 105 ^{o} C.

Remark: The temperature of T15 = T11

Enthalpy: We can calculate the enthalpy h15 by using the function, h_pt (p, t)

h 15 = h_pt (P15, T15)

We can calculate the power of pump working from point 11 to point 15 as below:

Wpm11-15 = m11 x (h15-h11) / Mechanical Efficiency

5.7. Turbo Pump

Turbo pump is used to impulse the feed water from lower pressure (point 13) to higher pressure (point 14). The impulsion of feed water is sent to the system. The main characteristics of the motor pump are shown as below:

Remark: The impulsion of the feed water is done by the turbo pump, although the motor pump is available to move the above mentioned water flow.

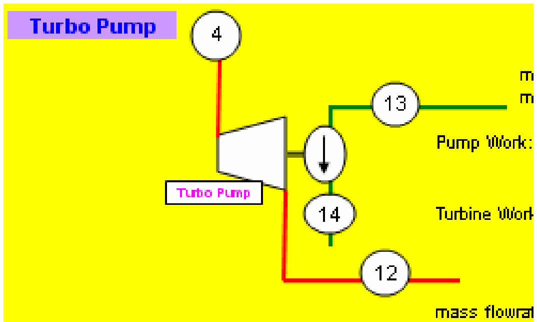

Figure 5.7 Schematic of the Turbo Pump

Point 14

The impulsion of feed water is sent out by the turbo pump. The water of pipe line 14 is high pressure and this pressure is the highest pressure of the system. The main functions of point 14 are shown as below:

Mass flow: We can find the mass flow by balancing mass flow m14 = m13.

Pressure: As we known that the pressure of this pipe line is the highest pressure. The pressure P14 = 61 bar.

Temperature: The temperature of this point T14 = 105 º C.

Remark: The temperature of T14 = T13

Enthalpy: We can calculate the enthalpy h14 by using the function, h_pt (p, t)

h 14 = h_pt (P14, T1)

We can calculate the power of pump working from point 13 to point 14 as below:

Wpt 13-14 = m13 x (h14-h13) / Mechanical Efficiency

Point 4

The steam is withdrawn from the pipe line 2 to send to the turbo pump. We can adjust the mass flow m4 to supply the power used of turbo pump working (point13 to point14).The main characteristics of point 4 are demonstrated as below:

Remark: When turbo pump is working, the other side the motor pump will be closed.

Mass flow: We can adjust the mass flow m4 to supply the power used of turbo pump working.

Pressure: As we known that the pressure P4 = P2 = 61 bar.

Temperature: As we known that the temperature T4 = T2 = 380 º C.

Enthalpy: We can calculate the enthalpy h4 by using the function, h_pt (p, t).

$$h\ 4 = h\_pt\ (P14, T14)$$

We can calculate the energy of turbo pump working from point 4 to point 12 as below:

$$Wt = m4 \times (h4-h12) \times \text{Internal Efficiency}$$

5.8. Air Heat Exchanger

Two fluids Gas and Steam, of different starting temperatures, flow through the heat exchanger. Steam flows through the tubes and the Gas flows outside the tubes but inside the shell. Heat is transferred from Steam to the Gas through the tube walls. In order to transfer heat efficiently.The flue gas heat exchanger allows us to recover a certain amount of heat, and this effect directly to boiler efficiency

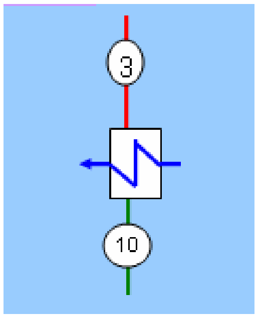

Figure 5.8 Schematic of Air Heat Exchanger

We can estimate the heat amount and directly add it to the transferred heat to the steam, that is

$$Qt, e = 1.1\ Qt, c + Qhe$$

Where:

Qt, c is the heat transferred to the steam in the current situation

Qhe is the heat recovered at the heat exchanger

Qt, e is the heat transferred to the steam in the expected situation

Where:

$$1.1 \times Qt,c = 1.1 \times 46.511\ MWt = 51.162\ MWt$$

$$Qhe = m3 \times (h3 - h10)$$

Which the value of Qt,e we can estimate the new steam flow as:

$$ms, e = Qt, e / (hs, o - hs, i)$$

Where:

ms, e is the steam mass flow in the expected situation.

hs, o and hs, i are the enthalpies at boiler outlet and inlet both in current and expected situation.

5.9 Nitrogen oxide reduction facility

The small fraction of steam is withdrawn from the pipe line 16 to use 2 t/h of live steam for a nitrogen oxide reduction facility. This steam is previously expanded up to 34 bar (abs) and returned hot and condensed to the feed tank.

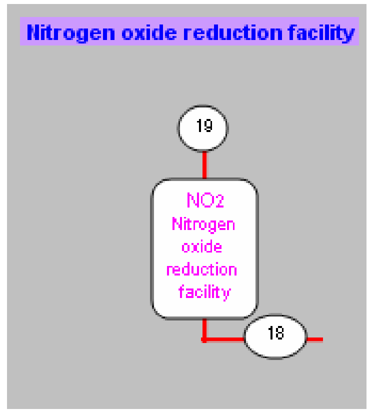

Figure 5.9 Nitrogen oxide reduction facilities

Point 18

The steam is sent to the Nitrogen oxide reduction facility. The main characteristics of point 18 are present as below:

Mass flow: As we known that mass flow m18 = 2 t/h to use for a nitrogen oxide reduction facility.

Pressure: The pressure of this point P18 = P16 = 61 bar.

Temperature: The temperature of this point T18 = T16 = 380 º C.

Enthalpy: We can calculate the enthalpy h18 by using the function, h_pt (p, t).

$$h\ 18 = h\_pt\ (P18,\ T18)$$

5.10 External Drying Process

The small fraction of steam is withdrawn from the pipe line 17 to use 2.4 t/h of live steam in the external drying process. These steam is previously expanded up to 28 bar (abs) and returned hot and condensed to the feed tank.

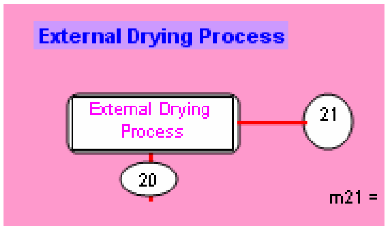

Figure 5.10 Schematic of External Drying Process

Point 20

The steam is sent to the external drying process. The main characteristics of point 21 are present as below:

Mass flow: As we known that mass flow m20 = 2.4 t/h to use of live steam in the external drying process.

Pressure: The pressure of this point P20 = P16 = 61 bar.

Temperature: The temperature of this point T20 = T16 = 380 º C.

Enthalpy: We can calculate the enthalpy h18 by using the function, h_pt (p, t).

$$h\ 20 = h\_pt\ (P20,\ T20)$$

6. Results

As we known that we have to work with three different situations, current situation, new situation and expected situation. Due to the different improvement in the boiler system, the admissible thermal load can be used in the system. So I will explain the result in each situation.

1. The efficiency of the boiler:

Expected Situation:

The boiler allows the thermal load of 46.511 MWt as shows in Fig 6.1

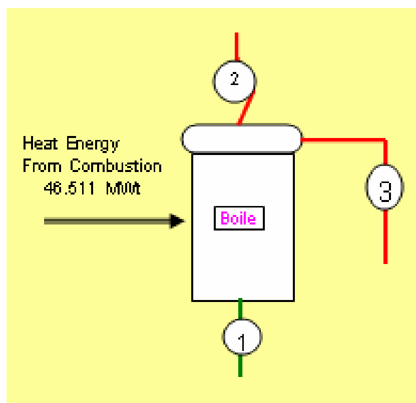

Figure 6.1 the boiler allows the thermal load of 46.511 MWt

The heat transferred to the steam in the current situation Qt, c:

$$Q_{t,c} = 46.511 \text{ MWt}$$

The heat energy is used to the system Euth, c:

$$E_{uth,c} = m_2 \times (h_2 - h_1) + m_3 \times (h_3 - h_1) = 39.633 \text{ MWt}$$

Boiler Efficiency ηb, c:

$$\eta_{b,c} = E_{uth,c} / Q_{t,c} = 0.8521$$

$$\eta_{b,c} = 85.21\%$$

Mass flow uses in the current situation = 53.7 t/h

New Situation:

The admissible thermal load can be increased in 10%, reaching a total 51.162 MWt as shown in Fig 6.2

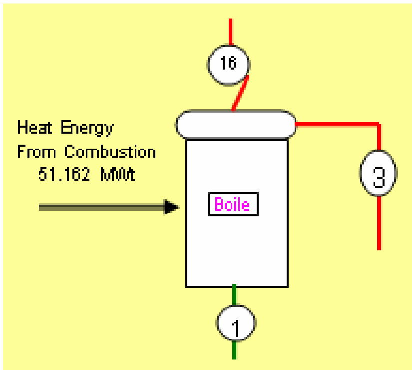

Figure 6.2 the boiler allows the thermal load of 51.162 MWt

The heat transferred to the steam in the new situation Qt, n:

$$Q_{t,n} = 51.162 \text{ MWt}$$

The heat energy is used to the system Euth, n:

$$E_{uth,n} = m_{16} \times (h_{16}-h_1) + m_3 \times (h_3-h_1) = 43.629 \text{ MWt}$$

Boiler Efficiency ηb, n:

$$\eta_{b,n} = E_{uth,n} / Q_{t,n} = 0.85277$$

$$\eta_{b,n} = 85.277\%$$

Assumption: As we known that the efficiency of the boiler is about 85.21 % and 85.277 % of current situation and new situation. So we can assume that the global efficiency of the boiler is constant (app 85 %).

Due to the thermal load can be increased 10% of the boiler and the efficiency of the boiler is constant (85%), then the new steam flow will be increased by about 10% too.

Mass flow uses in the new situation = 1.1 x 53.7 = 59.07 t/h

Expected Situation:

The modification in the boiler economizer provides the temperature of exit gasses (140,000 Nm³/h) can be reduced to 35 º C. The flue gas heat exchange allows recovering a certain amount of heat as shown in Fig 6.3.

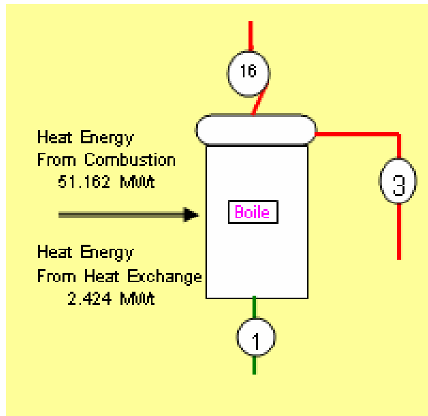

Figure 6.3 the heat energy form heat exchange

recover a certain amount of heat

Qt, e is the heat transferred to the steam in the expected situation:

$$Qt,e = 1.1\ Qt,c + Qhe$$

$$1.1 \times Qt,c = 1.1 \times 46.511 \text{ MWt} = 51.162 \text{ MWt}$$

Qhe is the heat recovered at the heat exchanger

$$Qhe = m3 \times (h3 - h10) = 2.424 \text{ MWt}$$

So we get

$$Qt,e = 51.162 + 2.424 = 53.586 \text{ MWt}$$

Which the value of Qt, e we can estimate the new steam flow as:

$$ms,e = Qt,e / (hs,o - hs,i) = 72 \text{ t/h}$$

2. Pressure Cycle of the system:

Length Point	Pressure Bars
1	61
16	61
17	61
2	61
5	61
7	0.13
8	1
9	10
11	10
15	61
1	61

Figure 6.4 Table of the pressure of

each point in the cycle system

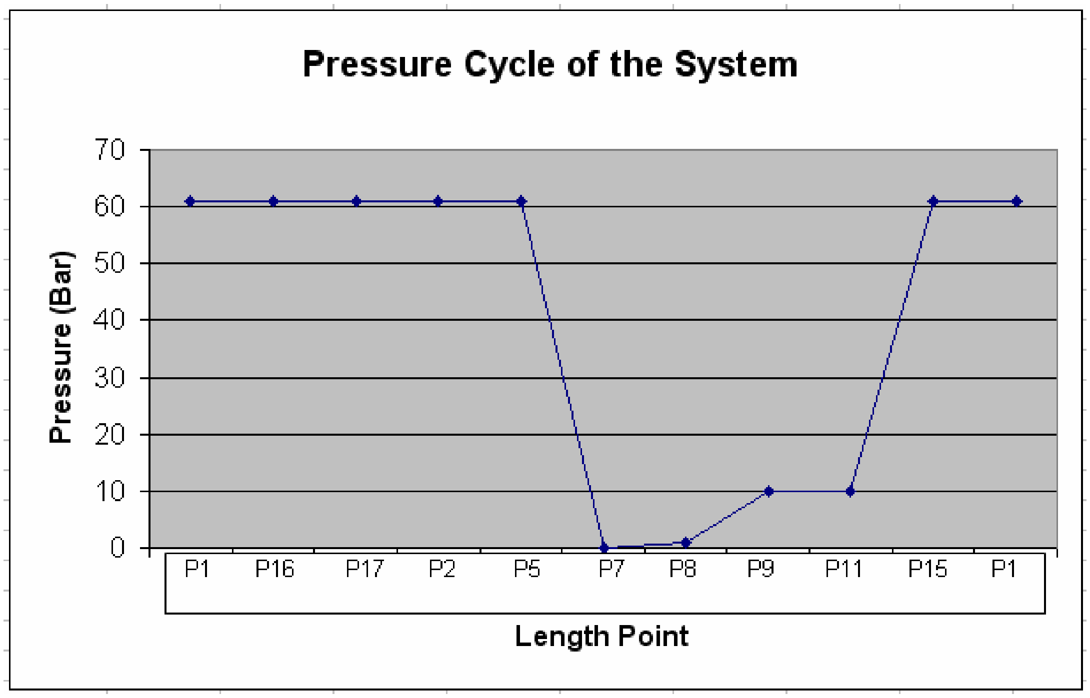

Figure 6.5 Graph demonstrates the pressure cycle of the system

3. The electrical output generated by the turbine in the new situation:

Consider at the system works with Motor Pump:

Elecrical Output, New Situation	
m 6 t/h	Elecrical Output MW
0	12.58
0.4	12.53
0.8	12.47
1.2	12.41
1.6	12.35
2	12.29
2.4	12.24
2.8	12.18
3.2	12.12
3.6	12.06
4	12
4.05	12.007
4.06	12
4.07	11.999
4.08	11.997
4.09	11.996
4.4	11.95

Figure 6.6 Electrical Output, New situation

Elecrical Output, Expected Situation	
m 6 t/h	Elecrical Output MW
0	15.78
0.4	15.72
0.8	15.66
1.2	15.607
1.6	15.549
2	15.49
2.4	15.43
2.8	15.37
3.2	15.31
3.6	15.26
4	15.2
4.4	15.144
4.8	15.087
5.1	15.043
5.15	15.036
5.16	15.034
5.17	15.033
5.2	15.029

Figure 6.7 Electrical Output, Expected situation

Consider at the system works with Turbo Pump:

Elecrical Output, New Situation	
m4 t/h	Elecrical Output MW
0	12.58
0.4	12.488
0.8	12.38
1.2	12.29
1.6	12.19
2	12.093
2.4	11.99
2.8	11.89
3.2	11.79
3.6	11.69
4	11.599
4.4	11.5
4.8	11.4
5.17	11.31
5.18	11.308
5.2	11.303
4.4	11.95

Figure 6.8 Electrical Output, New situation

Elecrical Output, Expected Situation	
m4 t/h	Elecrical Output MW
3	15.039
3.4	14.94
3.8	14.84
4.2	14.74
4.6	14.64
5	14.54
5.4	11.47
5.8	14.34
6.2	14.24
6.3	14.22
6.4	14.2
6.5	14.175
6.6	14.15

Figure 6.9 Electrical Output, Expected situation

Remark: The pipe line 6 will close the balance, since the steam flow coming from the turbo pump 12.

4. The net variation of electrical output (grid input) due to the increase of the turbine power output and to the current replacement of the electric pump by the turbine pump.

Consider at the system works with motor pump:

Vary the mass flow of pipe line 6 in order to bring the water in the Deaerator very close to saturation point. Assume the temperature 0.3 º C below saturation point, h = 417.8334 KJ/Kg

Varying the Mass Flow of Line 6, Current Situation				
m6 t/h	h15 Deaerator KJ/Kg	Power of Pump Working MW	Electrical Output MW	Net Varaition of E-O MW
0	229.97	4.57	12.337	7.767
0.4	248.4	4.18	12.29	8.11
0.8	266.83	3.78	12.23	8.45
1.2	285.26	3.396	12.17	8.774
1.6	303.68	3	12.11	9.11
2	322.11	2.161	12.05	9.889
2.4	340.54	2.218	12	9.782
2.8	358.97	1.839	11.94	10.101
3.2	377.4	1.43	11.86	10.43
3.6	395.83	1.04	11.827	10.787
4	414.25	0.647	11.77	11.123
4.05	416.56	0.598	11.762	11.164
4.06	417.02	0.588	11.761	11.173
4.07	417.48	0.578	11.759	11.181
4.4	432.68	0.2548	11.71	11.4552

Figure 6.10 Net Variation output, Current Situation

Varying the Mass Flow of Line 6, New Situation				
m6 t/h	h22 Deaerator KJ/Kg	Power of Pump Working MW	Electrical Output MW	Net Varaition of E-O MW
0	246.39	4.647	12.58	7.933
0.4	263.14	4.254	12.53	8.276
0.8	279.88	3.86	12.47	8.61
1.2	296.65	3.469	12.41	8.941
1.6	313.4	3.076	12.35	9.274
2	330.15	2.683	12.29	9.607
2.4	346.91	2.29	12.24	9.95
2.8	363.66	1.898	12.18	10.282
3.2	380.41	1.505	12.12	10.615
3.6	397.17	1.1128	12.06	10.9472
4	413.92	0.72	12	11.28
4.05	416.01	0.671	12.007	11.336
4.06	416.43	0.661	12	11.339
4.07	416.85	0.651	11.999	11.348
4.08	417.17	0.641	11.997	11.356
4.09	417.69	0.631	11.996	11.365
4.4	413.92	0.327	11.95	11.623

Figure 6.11 Net Variation output, New Situation

Varying the Mass Flow of Line 6, Expected Situation				
m6 t/h	h22 Deaerator KJ/Kg	Power of Pump Working MW	Electrical Output MW	Net Varaition of E-O MW
0	246.64	5.82	15.78	9.96
0.4	254.38	5.43	15.72	10.29
0.8	268.13	5.04	15.66	10.62
1.2	281.37	4.65	15.607	10.957
1.6	296.62	4.25	15.549	11.299
2	309.36	3.86	15.49	11.63
2.4	323.1	3.47	15.43	11.96
2.8	336.85	3.079	15.37	12.291
3.2	350.59	2.68	15.31	12.63
3.6	364.34	2.29	15.26	12.97
4	378.08	1.9	15.2	13.3
4.4	391.83	1.509	15.144	13.635
4.8	405.57	1.11	15.087	13.977
5.1	415.88	0.82	15.043	14.223
5.15	417.6	0.772	15.036	14.264
5.16	417.94	0.762	15.034	14.272
5.17	418.28	0.753	15.033	14.28
5.2	419.31	0.72	15.029	14.309

Figure 6.12 Net Variation output, Expected Situation

Consider at the system works of Turbo Pump

Varying the Mass Flow of Line 4, Current Situation				
m4 t/h	h16 Deaerator KJ/Kg	Pump Effective Power MW	Pump Gross Power MW	Electrical Output MW
0.8	259.01	3.95	0.138	12.15
1.2	273.53	3.64	0.208	12.05
1.6	288.05	3.33	0.277	11.95
2	302.57	3.02	0.346	11.85
2.4	317.09	2.71	0.416	11.75
2.8	331.61	2.4	0.48	11.65
3.2	346.13	2.099	0.55	11.55
3.6	360.65	1.789	0.624	11.45
4	375.17	1.48	0.69	11.36
4.4	389.68	1.17	0.76	11.26
4.8	404.2	0.861	0.83	11.16
5.17	417.63	0.57	0.89	11.076
5.2	418.72	0.55	0.9	11.063

Figure 6.13 Net Variation output, Current Situation

Varying the Mass Flow of Line 4, New Situation				
m4 t/h	h22 Deaerator KJ/Kg	Pump Effective Power MW	Pump Gross Power MW	Electrical Output MW
0	246.39	4.64	0	12.58
0.4	259.59	4.33	0.069	12.488
0.8	272.79	4.02	0.138	12.38
1.2	285.99	3.71	0.208	12.29
1.6	299.19	3.4	0.277	12.19
2	312.39	3.1	0.34	12.093
2.4	325.59	2.79	0.416	11.99
2.8	338.79	2.48	0.48	11.89
3.2	351.98	2.17	0.55	11.79
3.6	365.18	1.86	0.62	11.69
4	378.38	1.55	0.69	11.599
4.4	391.58	1.24	0.763	11.5
4.8	404.78	0.934	0.832	11.4
5.17	416.99	0.648	0.89	11.31
5.18	417.32	0.64	0.89	11.308
5.2	417.98	0.62	0.9	11.303

Figure 6.14 Net Variation output, New Situation

Varying the Mass Flow of Line 4, Expected Situation				
m4 t/h	h22 Deaerator KJ/Kg	Pump Effective Power MW	Pump Gross Power MW	Electrical Output MW
3	321.86	3.5	0.52	15.039
3.4	332.69	3.198	0.589	14.94
3.8	343.51	2.89	0.659	14.84
4.2	354.34	2.57	0.728	14.74
4.6	365.17	2.27	0.79	14.64
5	376	1.96	0.867	14.54
5.4	386.83	1.65	0.93	11.47
5.8	397.66	1.34	1.005	14.34
6.2	408.49	1.03	1.075	14.24
6.3	411.2	0.955	1.092	14.22
6.4	413.9	0.87	1.11	14.2
6.5	416.61	0.8	1.27	14.175
6.6	419.32	0.723	1.144	14.15

Figure 6.15 Net Variation output, Current Situation

Remark: We can adjust the mass flow m4 to supply the power used of turbo pump working (point13 to point14). Pump Gross Power have to have more power than Pump Effective Power, Because Pump Gross Power to have enough power to support the Pump Effective Power.

7. Conclusions

7.1 Conducting qualitative results

As we known that the objective of the project is to evaluate the best way of adapting the thermal cycle of a Waste to Energy (WtE) plant to a higher performance.

Due to the improvement in the furnace system, the admissible load can be increased and the modification in the boiler economizer provides the temperature of exit gasses, which allows generating even more steam in the boiler.

Dealing with the excess of steam and considering the global modification of the plant. The fraction of the steam are withdrawn to use of live steam for a nitrogen oxide reduction facility and to use of live steam in the external drying process.

Considering at the efficiency of the boiler, the current situation $\eta_{b,c}$ = 85.21% which the thermal load of Qt, c = 46.511 MWt and the thermal use of the system Euth, c = 43.62 MWt., the new situation $\eta_{b,n}$ = 85.277% which the thermal load of Qt, n = 51.162 MWt and the thermal use of the system Euth, n = 43.62 MWt according to modify the heat input (boiler furnace + Boiler) the load can be increased 10%.

As we known that the heat input can be increased 10%, the mass flow in the new situation is increased by 10% of the current situation 53.7 t/h to be 59.07 t/h. Consider the global efficiency constant approximated 85%. We use this mass flow $m_{s,n}$ = 59.07 t/h to calculate in the new system.

Besides, the flue gas heat exchanger allows us to recover the certain amount of heat. As we known that the heat recovered at the heat exchanger Qhe = 2.424 MWt which we add to the transferred heat to the system. So we will get the amount of the heat transferred to the steam in the expected situation Qt, e = 53.586 MWt. From this value Qt, e we can find the new steam mass flow used of the expected situation $m_{s,e}$ = 72 t/h. As we known we use 2 t/h of steam for NOx reduction facility and 2.4 t/h for external drying process.

Considering, the electrical output generated by the turbine in the new situation and the expected situation.

1. At the system works with the motor pump, we adjust the steam flow the pipe line 6 to bring the water in the deaerator very close to the saturation point.

The new situation, we get mass flow m6 = 4.09 t/h and electrical output = 11.996 MW (it's not the net variation electrical output).

The expected situation, we get mass flow m6 = 5.15 t/h and electrical output = 15.036 MW (it's not net variation electrical output).

2. At the system works with the turbo pump, we adjust the steam flow the pipe line 4 to bring the water in the deaerator very close to the saturation point.

The new situation, we get mass flow m4 = 5.17 t/h and electrical output = 11.31 MW .

The expected situation, we get mass flow m4 = 6.5 t/h and electrical output = 14.175 MW.

Considering, the net variation of electrical output (grid output) in the current situation, new situation and expected situation.

1. At the system works with the motor pump, we vary the mass flow of pipe line 6.

The current situation, we get:

m 6 = 4.07 t/h

h16 = 417.48 KJ/Kg (enthalpy in the deaerator)

Power of Pump Working = 0.578 MW

Electrical Output =11.759 MW

Net Variation of Electrical Output = 11.181 MW

The new situation, we get:

m6 = 4.09 t/h

h22 = 417.69 KJ/Kg (enthalpy in the deaerator)

Power of Pump Working = 0.631 MW

Electrical Output =11.996 MW

Net Variation of Electrical Output = 11.365 MW

The expected situation, we get:

m6= 5.15 t/h

h22 = 417.6 KJ/Kg (enthalpy in the deaerator)

Power of Pump Working = 0.772 MW

Electrical Output =15.036 MW

Net Value = 14.264 MW

2. At the system works with the turbo pump, we vary the mass flow of pipe line 4. The pipe line 6 will close the balance, since the steam flow coming from the turbo pump 12.

The current situation, we get:

m4 = 5.17 t/h

h16 = 417.63 KJ/Kg (enthalpy in the deaerator)

Power of the Pump working = 0.57 MW

Power of the Turbo Pump Working = 0.89 MW

Electrical Output = 11.076 MW

The new situation, we get::

m4 = 5.17 t/h

h22 = 416.99 KJ/Kg (enthalpy in the deaerator)

Power of the Pump working = 0.648 MW

Power of the Turbo Pump Working = 0.89 MW

Electrical Output =11.31 MW

The expected situation, we get:

m4= 6.5 t/h

h22 = 416.61 KJ/Kg (enthalpy in the deaerator)

Power of the Pump Working = 0.8 MW

Power of the Turbo Pump = 1.27 MW

Electrical Output = 14.175

7.2 Final Remarks

The different calculations performed show the feasibility of the proposed improvement.

The final performance of the plant is expected to be adapted to the reel needs of the facility with a higher performance and efficiency.

The development worksheet can be used in future following any improvement of the knowledge or the values of parameters assumed.

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