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# Performance Evaluation of Bagasse Fuelled Cogeneration System –The Case of Wonji-Shoa Sugar Mill

By: Asaye Mebratu

ID: GSR/2964/07

Advisor: Dr. Tesfaye Dama

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PERFORMANCE EVALUATION OF BAGASSE FUELLED COGENERTION SYSTEM OF WONJI-SHOA SUGAR MILL

By: Asaye Mebratu Adem

**Approved by Board of Examiners**

| | | |
|---|---|---|
| Dr.Yilma | ______________________ | ______________ |
| Chairman, | | |
| Thermal Engineering stream | Signature | Date |
| Tesfaye Dama (Dr.) | ______________________ | ______________ |
| Advisor | Signature | Date |
| Dr. -Ing. Demiss Alemu | ______________________ | ______________ |
| Internal Examiner | Signature | Date |
| Dr.-Ing. Edessa Dribsa | ______________________ | ______________ |
| Internal Examiner | Signature | Date |

# DECLARATION

I, the undersigned, declare that this thesis is my original work and has not been presented for a degree in any University, and that all the source of materials used for the thesis has been duly acknowledged.

Declared by:

Name: Asaye Mebratu Adem

Signature: ___________________________

Date: _______________________________

Confirmed by:

Name: Tesfaye Dama (Dr.)

Signature: ______________________________

Date: _________________________________

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List of symbols

| | |
|---|---|
| A | Total heat transfer area of the heat exchanger in ($m^2$) |
| $C_P$ | Specific heat |
| $d_o$ | tube outside diameter (m) |
| $\dot{E}_I$ | Energy input(kW) |
| $\dot{E}_o$ | Energy output(kW) |
| h | Enthalpy |
| $h_f$ | Enthalpy of saturated liquid |
| $h_g$ | Enthalpy of saturated vapor |
| $h_o$ | Enthalpy at reference state |
| HR | Heat rate of turbine(kJ/kWhr) |
| K | Thermal conductivity (W/m.K) |
| M | Mass (kg) |
| $\dot{m}$ | Mass flow rate(kg/s) |
| n | Number of tubes per pass |
| p | Pressure(bar) |
| $p_o$ | Pressure at reference state(bar) |
| P | Power(kW) |
| q | Quantity of fuel used per hour(kg/hr) |
| Q | Quantity of steam generated per hour(kg/hr) |
| s | Entropy |
| $s_f$ | Entropy of saturated liquid |
| $s_g$ | Entropy of saturated vapor |
| $s_o$ | Entropy at reference state |
| t | Tube thickness(m) |

| | |
|---|---|
| $T$ | Temprature(K) |
| $T_a$ | Ambient temperature($^oC$) |
| $T_{1-cw,I}$ | Inlet temperature of process heater cooling water(K) |
| $T_{1-cw,o}$ | Exit temperature of process heater cooling water(K) |
| $T_{2-cw,I}$ | Inlet temperature of condenser cooling water(K) |
| $T_{2-cw,o}$ | Exit temperature of condenser cooling water(K) |
| $T_f$ | Flue gas temperature($^oC$) |
| $T_o$ | Temperature of reference state(K) |
| U | Overall heat transfer coefficient($W/m^2.K$) |
| v | Specific volume($m^3/kg$) |
| $v_f$ | Specific volume of saturated liquid($m^3/kg$) |
| $v_g$ | Specific volume of saturated vapor($m^3/kg$) |
| $\dot{W}_{CEST}$ | Turbine generated power(kW) |
| W.D | Work done (kW) |
| $\dot{W}_{net}$ | Net power available(kW) |
| $\dot{W}_{PUMP}$ | Pump input power(kW) |
| x | Wetness fraction(%) |
| $\boldsymbol{\eta}_{ex}$ | Exergetic efficiency(%) |
| $\boldsymbol{\eta}_{gearbox}$ | Gearbox efficiency(%) |
| $\boldsymbol{\eta}_{generator}$ | Generator efficiency(%) |
| $\boldsymbol{\eta}_{pump}$ | Isentropic efficiency of pump(%) |
| $\boldsymbol{\eta}_{turbine}$ | Isentropic efficiency of turbine(%) |
| $\boldsymbol{\eta}_{th,PP}$ | Thermal efficiency of the power plant(%) |
| $\Psi_{in}$ | Exergy input |
| $\Psi_{out}$ | Exergy output |

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List of abbreviations and acronyms

| | |
|---|---|
| AAS | Actual mass of Air Supplied |
| AFC | Alkaline Fuel cell |
| | |
| BFB | Bubbling Fludized Bed |
| BPST | Back Pressure Steam Turbine |
| BS | British Standard |
| | |
| CEST | Condensing Extraction Steam turbine |
| CFB | Circulating Fludized Bed |
| CHP | Combined Heat and Power |
| CI | Compression Ignition |
| | |
| EA | Excess Air |
| | |
| GCV | Gross Calorific Value |
| GEM | Global Environmental Protection |
| | |
| HHV | Higher Heating Value |
| HP | High Pressure |
| HRSG | Heat Recovery Steam Generator |
| HVA | Hangler Vonder Amsterdam |
| | |
| LHV | Lower Heating Value |
| LMTD | Log Mean Temperature Difference |

| | |
|---|---|
| LP | Low Pressure |
| LPG | Liquefied Petroleum Gas |
| | |
| MCFC | Molten Carbonate Fuel Cell |
| MCR | Maximum Continuous Rating |
| | |
| NCV | Net Calorific Value |
| | |
| PAFC | Phosphoric Acid Fuel Cell |
| PEM | Proton Exchange Membrane |
| | |
| SI | Spark Ignition |
| SJAE | Steam Jet Air Ejector |
| SOFC | Solid Oxide Fuel Cell |
| | |
| TCD | Tons of cane Crushed per Day |
| TPH | Tons Per Hour |
| | |
| WSSF | Wonji-Shoa Sugar Factory |

## Abstract

*The performance of sugar cane bagasse boilers is commonly analyzed through the first law of thermodynamics, using the energy balance method and the fuel gross calorific value as a calculation base. This work presents a first law analysis by comparing from the two different methods: the input/output method and the energy balance method. Among the different energy losses in the boiler the contribution of the loss due to sensible heat carried away by dry flue gas is highest i.e. 10.23%. The combustion efficiency of the bagasse boiler as calculated using the energy balance method is 67.85% and it is lower than the rated efficiency of the boiler which is 72%.Energy and exergy analysis of the condensing-extraction steam turbine show that the first and second law efficiencies are 86.37% and 75.98% respectively. Moreover the influence of increasing boiler output temperature on energy and exergy efficiencies of the turbine by combusting fixed amount of bagasse at constant pressure is investigated. The result of the investigation shows that both the energy and exergy efficiencies improve at higher temperatures.*

KEY WORDS: *Bagasse, Combustion efficiency, Exergy, Performance.*

# CHAPTER ONE

## 1-**Introduction**

The development of sugar industry in Ethiopia has been remarkable since 1954E.C. when the Dutch Company, HVA established Wonji Sugar Factory with a crushing rate of 1400 tons of cane per day (TCD). Wonji Sugar Factory is the pioneer sugar factory in Ethiopia. Eight years after the establishment of Wonji Sugar Factory, in 1962 Shoa Sugar Factory was established with a crushing rate of 1600 TCD. They are under one management and collectively called Wonji-Shoa Sugar Factory (WSSF). However, presently the two oldest factories are not operational and as an extension a new factory is established at Dodota some 21Km from Wonji. The rated crushing capacity of the factory is 6250 TCD. The Factory is located in the rift valley, Oromia region, in East Central Ethiopia, some 149 Km South East of Addis Ababa at an elevation of 1540 above sea level.

The present system of sugar manufacturing process is essentially a combination of juice extraction from sugar cane by milling or diffusion or a combination of both, clarification of juice, concentration of juice by evaporation to syrup, crystallization of sucrose by vacuum pan boiling, centrifugal separation of sugar and molasses from massacuites, drying and cooling of sugar and sugar grading and packing . All these processes need relatively large amount of energy in different forms such as power for prime movers. This power is needed mainly in the form of electricity for the electric motors and medium to high-pressure steams for the processes and steam turbine. The processing of raw cane also requires a large amount of heat provided by medium-pressure steam for the processing of juice into commercial sugar.

Sugar cane industry is one of the few industries which are able to supply their own fuel (in the form of bagasse). The fiber in the cane is generally sufficient to supply all the steam necessary for power and sugar production when utilized as a fuel in the boiler furnaces. With normal fiber content in cane (12-14%) a well-balanced factory will be left with a surplus bagasse. The present objective of cane sugar industry is to extract the maximum power from the available fuel by efficiently utilizing the latent heat of steam and reducing the losses to a minimum. The losses are inevitably sustained in the flue gases of steam boiler, steam of condensing turbine and in the cooling water of condensers.

Significant improvements can be achieved by improving boiler house, and other factory departments leading to maximum process heat economy, bagasse drying to improve its calorific value as fuel and generation of extra electricity from excess bagasse for export to the grid. Shoa sugar factory is the factory that was established to supply its own fuel from the byproduct bagasse and remains with some excess of this bagasse, which will be used during stoppage of mill so that no additional fuel would be used whatever the case may be.

Cogeneration is the production of electricity and process heat from a single thermodynamic machine. Biomass, such as bagasse is used to fuel the thermodynamic machine that heats up steam, which then drives a turbine to produce electricity. Cogeneration strictly refers to an energy recovery system, which sequentially generates electrical power and thermal energy. Bagasse leaving the milling department would be burnt in boilers to generate steam, which is used to drive prime movers and the excess energy is exported in the form of electricity to the grid.

Bagasse being a by-product of sugarcane industry has almost zero fuel cost. Historically, it is used in sugar industries for generating power to fulfill steam and electricity requirement of the mills. Mass and energy balance computation showed that modern sugar factories with efficient cogeneration plant crushing cane of about 14 % fiber require about 60 to 70 % of bagasse produced for steam and electricity requirement of the mills. The remaining quantity can be used as raw material for paper production, other fibrous products or for production of extra electric energy in condensing mode turbines for sale to the grid.

In Ethiopia, the government is planning to expand the existing three sugar mills and to establish five new sugar factories with a total cane crushing capacity of 65,500 tons of cane per day (TCD) each having cogeneration and an ethanol plant. These sugar and ethanol production processes are energy intensive, requiring steam and electricity. While, the current scenario of the existing three sugar factories revealed that cogeneration in these factories is inefficient, operating with low pressure boilers and backpressure turbines generating electric power mostly for captive power use and partial requirements of irrigation and residential houses of the factory workers.

Bagasse is a useful source of energy with a gross calorific value of about 9314 kJ/kg but it contains a very high level of moisture (around 50% by weight) and needs specially designed handling, feeding and combustion systems. Many stages – such as diffusion, evaporation, crystallization and distillation – need heat in the form of steam. Electrical energy is consumed by material handling equipment, shredders, mill drives, clarifiers/centrifuges and other process pumps. Using a by-product – bagasse – as an energy source and producing combined heat and power (CHP) to meet their sugar mills' demand makes immense sense for customers. In addition, surplus electrical power from the cogeneration process can be exported to the state grid, providing the customer with an extra revenue stream.

## 1.1 **Statement of the problem**

Wonji-Shoa sugar factory is the factory that was established to supply its own fuel from the byproduct bagasse and remains with some excess of this bagasse, which will be used during stoppage of mill so that no additional fuel would be used whatever the case may be. At the present condition the reality reveals that the factory is using additional fuel(fuel oil) during mill stoppage.

No Significant improvements are made to extract maximum power by improving boiler house, and other factory departments leading to maximum process heat economy, bagasse drying to improve its calorific value as fuel and generation of extra electricity from excess bagasse for export to the grid.

The current status of Wonji-Shoa sugar factory revealed that performance of cogeneration in this factory is inefficient, operating with low pressure boiler and steam turbine. Therefore, this research is about performance evaluation of cogeneration in the sugar industry of Ethiopia, the case of Wonji-Shoa sugar factory with the objective of obtaining the sugar mill's maximum power extraction amount from the available fuel when efficiently utilizing the latent heat of steam and reducing the losses to a minimum. The losses are inevitably sustained in the flue gases of steam boiler, steam of condensing turbines and in the cooling water of condenser and process heater.

## 1.2-Objectives

### 1.2.1-General objective

The general objective of this research is to conduct performance evaluation of the bagasse fueled cogeneration system of the Wonji-Shoa sugar mill by analyzing the thermodynamic steam power cycle of the cogeneration unit( i.e.- the Rankine cycle) with the objectives of predicting the working value of the parameter of steam for each sub components and comparing them with the actual working value as obtained from the process control board and comparing the efficiency of cogeneration system obtained by calculation to the rated efficiency.

### 1.2.2-Specific objectives

The specific objective of this research is

- Assessing the performance of cogeneration of the Wonji-Shoa sugar mill based on the first and second laws of thermodynamics,
- Finding the actual power output and cogeneration efficiency,
- Comparing the predicted value of operating parameters with the actual daily values and
- Plot graphs using EES , EXCEL, and MATLAB

# CHAPTER TWO

## 2-Literature review

There seems to have been less number of studies or reports on the energy and exergy analysis of biomass based cogeneration systems in the open literatures, because of using biomass for producing power and process heat is relatively new technology compared to the conventional coal fired power generation. The literatures and relevant studies are taken from two categories; such as background review and energy and exergy analyses review. In the recent years, cogeneration systems have become popular technology to implement into many industrial or residential applications for producing power and process heat.

## 2.1-Background Review

A comprehensive study in the cogeneration applications were conducted by John and Martin [8].They introduced the classification of different types of steam and gas based cogeneration systems with various combination of steam withdrawal that helped to identify the combinations and arrangements of the component located in the cogeneration plant. In addition, the study also included the practical methods such as net heat to process and fuel chargeable to power to estimate the usability in implementing the cogeneration systems to the industry applications. In the present time, cogeneration system approves its potential to compensate the incredible energy consumption rate in the current sharing of primary energy supply.

Smouse *et al*. [6] gave credit to the biomass cogeneration system in sugar industry in India. According to their expectation the energy demand will reach to 300,000 MW for the next 25 years. Biomass based cogeneration system not only provided a cost effective to produce electricity and hot steam compared to the conventional power plant, but also presented a solution to the pollutions problems as the Global Environmental Protection(GEM) pointed out the consequence in terms of reducing of greenhouse emissions.

The feasibility and cost effectiveness are also described by Branch *et al*. [12]. They undertook a comprehensive analysis in implementing biomass cogeneration system to a small or medium scale industry application and included the economic analysis as well. This report specifically compared the conventional oil based captive power generation to the proposed biomass fuelled cogeneration. As the result, the feedback from the new system was very impressive due to its fuel saving and environmental friendly, even though the initial cost on installation was higher.

Jayen et al [13] proved that using biomass with cogeneration system resulted in diminishing the emission problems efficiently and therefore there is high potential to increase its capacity in the near future.

**2.2-Exergy Analysis review**

Derya Burcu *et al.* [15] – Cogeneration may be defined as a system which contains electricity production and regain of the thermal energy of flue gases simultaneously. The examination is based on the data's of an active cogeneration plant. In this study, it is aimed to determine which component of the system should be revised first to raise the efficiency and decrease the loss of exergy. For this purpose, thermodynamic analysis of different components was made to consider the effects of environmental conditions and take the quality of energy into consideration as well as the quantity of it. The exergy balance equations are produced and exergy loss is calculated for each component. 44.44 % loss of exergy in heat exchanger, 29.59% in combustion chamber, 18.68 % in steam boiler, 5.25 % in gas turbine and 2.03 % in compressor is calculated.

Exergetic and thermo economic analyses of the biomass based Power Plant in Thailand is conducted by Prachuab et al.[16] The purpose of this study is to analyze a biomass based power plant, and consider the efficiency of the power plant through both the first law and the second law of thermodynamics. The energetic and exergetic performance criteria such as thermal efficiency, exergy efficiency, and exergy loss have been found to be useful methods in design, evaluation, optimization and improvement of thermal power plants. The exergetic efficiency of a power plant shows that the boiler is the major component contributing to total loss, with its low exergetic efficiency of 52%, while the steam turbine has the exergetic efficiency of 76%. The study also evaluates the economic consideration of a biomass fired power plant for heat and power production. The capacity of the plant is 576 tons paddy/day. Results of the study also show that rice husk-fueled cogeneration is beneficial to power generation.

Rosen and Cornelia [17], conducted the study on differentiating energy and exergy, and recommended using exergy analysis to understand and improve the efficiency of electrical power technologies. This paper essentially explained and provided details in differentiating energy and exergy analyses. According to the result, the exergy analysis was concluded to have significant role in evaluating and improving the efficiencies of electrical power technologies and provided a useful tool for engineers and scientists as well as decision and policy makers. In some situations, the use of energy analysis could be misleading and confusing while doing the energy analysis, therefore by cooperating exergy analysis it could solve that issue easily and provided more details on the energy transferring through each component.

Relevant Studies In Industrial Applications In the performance evaluation aspect, the study of improving more efficient cogeneration plants had been conducted by Premalatha *et al*. [18]. This paper -Efficient Cogeneration Scheme for Sugar Industry, was prepared to be specific on considering the result of obtaining higher efficient power generation by applying the high pressure boilers and condensing extraction turbines for processing hot steam. Overall, the results proved the replacement of low pressure boiler with high pressure boiler could provide more efficient power generation, and because of that, the estimation of different operating conditions were carried out in the present work by way of considering to have better performance in the cogeneration systems.

Mujeebu *et al.* [19] presented a case study on the biomass fueled steam turbine cogeneration for a sugar mill with surplus power and studied the technical and economic feasibility of implementing a biomass based cogeneration and finally concluded the additional cogeneration plant in the sugar mills provided option for surplus power and suggested the replacement of bottoming cycle with a steam turbine with topping cycle to operate in the sugar mills. According to the results of the additional cogeneration system, the excess electricity production 200KW was obtained in its full time operation. In the selection of fuels, natural gas becomes a popular option in the current energy resources.

In the present time, the biomass fueled cogeneration with condensing extraction steam turbine start attracting attention to consider as the combustible since the exhausted gas is cleaner and less emissions compared to the conventional coal fired facility. The study was conducted by Reddy and Butcher [20]. They examined the influences of the different operating conditions such as reheat, inter-cooling, ambient temperature and pressure ratio to the system performance.

# CHAPTER THREE

## 3-Cogeneration system

### 3.1-Cogeneration power plant

Cogeneration is well known as producing both power and process heat simultaneously at very high efficiency, and operates at optimum efficiency from the single fuel source. Nowadays, renewability and sustainability make a vital meaning in the future energy generation, because the results of over exploiting current energy resources are becoming a disaster. Therefore, the present work is to evaluate the biomass based cogeneration systems, because biomass can balance the carbon dioxide from its own growth and does not add additional $CO_2$ into the atmosphere. The biomass can continuously be used as energy resource and yield very small pollution compared to the conventional power plant, and thus the necessity of increasing the capacity of cogeneration in the near future is the priority decision.

Producing electricity with smaller devices located near the user can dramatically increase the overall efficiency of energy use. Distributed generation is a term used for small power generators located near the point of electricity use as opposed to central power plants using transmissions grids[3][5][6]. However as smaller scale power generation is usually less efficient in the production of electricity, the full benefits of distributed generation are only achieved with cogeneration applications.

### 3.2-Bioenergy from sugar cane

In a sugar factory, raw sugar is extracted from sugarcane through a series of processes, which also deliver bagasse and molasses as by-products. In order to discard bagasse and simultaneously save external sources of energy for the manufacture of raw sugar, bagasse-fired power plants were set up and coupled to sugar factories to produce process steam required in raw sugar extraction (Baguant, 1984). In addition to steam, bagasse-fired power plants could also generate electricity to satisfy the internal needs of both the sugar factory and the power plant. Any excess electricity is then sold to the national grid and is referred to as exportable electricity (Baguant, 1984). Since the power plant generates both heat, in the form of process steam, and electricity from a single fuel, namely bagasse, it is commonly called a cogeneration plant. On the other hand, molasses could be processed in a distillery to produce bioethanol which is then used as transportation fuel. Thus, a sugarcane bioenergy system typically consists of a sugar factory with a cogeneration plant and/or a distillery coupled to it.

### 3.3-Some of the advantages of cogeneration

- ✓ Improved Efficiency of Fuel Energy use

The production of both electricity and thermal energy from a cogeneration system leads to energy use efficiencies of 80 to 85%. A typical cogeneration system can convert 30 to 35% of the energy in the fuel to electricity with another 50 to 55% produced as steam and/or hot water. Using absorption based refrigeration systems, the thermal energy could also be used for cooling and refrigeration. By locating the cogeneration plant near the end user, electricity transmission losses are also reduced further improving overall energy efficiency (Anderzej, 2002).

- ✓ Reduction of Load on Existing Transmission Systems

By locating energy production near the user, loads on existing transmission systems can be reduced negating the need for expensive upgrades or new transmission lines. Reducing the load on existing transmission systems also improves operating stability of the system. Cogeneration plants sized to supply the thermal load to a facility may produce excess electricity that can be sold back to the grid (Anderzej, 2002).

- ✓ Improved security of power

With the increase in use of computer equipment for plant operation, security and quality of power is increasing in importance. The conventional transmission system represents a weak link for the electricity user as the control and operation of the transmission system is with the utility. Cogeneration systems are often installed and operated by the electricity user and can be maintained to provide more dependable power. Existing transmission systems may also act as a backup to the cogeneration system (Anderzej, 2002).

- ✓ Reduced Emissions of Pollutants and Greenhouse gases

Improving the overall efficiency of fuel energy use also leads to reduced emissions of pollutants and greenhouses gases into the atmosphere. The cogeneration system supplies both electricity and thermal energy, in place of purchasing electricity from the grid and firing natural gas in a separate boiler to generate thermal energy for plant use. By eliminating the need for a separate steam boiler fired with natural gas or other fuel, the emissions of nitrogen oxides, sulfur oxides, particulates are also reduced. The improvement of fuel use efficiency of cogeneration over central power plants, more than halve the emissions of carbon dioxide-a greenhouse gas, can be as high as 3 to 6% of the electricity used (Anderzej, 2002).

- ✓ Development of Competition within the Electricity

Supply Industry Cogeneration power plants are usually small plants sized to either the users electricity demand or thermal demand. Often they are installed and operated by small independent power producers rather than the utilities controlling the transmission grid. The availability of a number of cogeneration plants operated by independent companies leads to increased competition in the production and sale of electricity that may lead to improved quality and reduced cost to the user (Anderzej, 2002).

**3.4-CHP Technology Descriptions**

CHP systems involve the simultaneous generation of electricity and heat, and thus are often referred to as cogeneration systems. Depending on the application, electricity or heat can be the primary on-site need, with the other energy source being used on-site to replace either: 1) electricity from the grid, or 2) an additional piece of equipment that provides steam, space heating, water heating, space cooling, or dehumidification. Because one technology, or a group of technologies, such as a boiler/steam turbine system, can provide both electricity and heat, the systems operate more efficiently than a separate electric and heat source.

Depending on the CHP technology and on the type of businesses using CHP, the heat recovered from the CHP system can be used for different applications. Typically, heat is used for one of the following five applications:

1) Steam for an industrial process,

2) Space heating,

3) Water heating,

4) Space cooling (through an absorption or adsorption chiller), and

5) Dehumidification (with the use of a desiccant).

Among the main engine cycles the most important and that are related to CHP technology are the following.

- Reciprocating engines

Reciprocating engines convert pressure into linear motion, via pistons, that is then converted into rotating motion. Most stationary reciprocating engines have four strokes to each cycle: the intake stroke, compression stroke, power stroke, and exhaust stroke. In the intake stroke, the piston moves down, allowing air or a fuel/air mixture into the combustion chamber. The compression stroke follows with the piston moving up and compressing the gases. The gases are either ignited by a source, as in the spark ignition type, or the fuel is added to them and they are ignited by the increased temperature caused by the compression, as in the compression ignition type. The combustion of the gases pushes the piston downward, which is called the power stroke. Finally, during the exhaust stroke, gases are released from the chamber. The rotational motion is converted into the power output of the machine. Reciprocating engines are widely used technology.

Table **3.1**-Characteristic descriptions of reciprocating engine.

| **Characteristics** | **Description** |
|---|---|
| Engine type | *Reciprocating engine* |
| Size | A few kW to several MW |
| Electrical efficiency | 30-50% |
| Overall efficiency | 70-80% |
| Type of technologies | Spark ignition (SI) and compression ignition (CI) designs are more applicable to CHP. |
| Fuel | Diesel and residual oil (compression ignition);natural gas, biogas, liquefied petroleum gas (LPG),and propane(spark ignition) |
| Thermodynamic cycle | Otto cycle for the SI designs and diesel cycle for the CI designs |
| Heat recovery | Heat can be recovered from the engine exhaust, engine coolant, lube oil cooler, and turbochargers intercooler and aftercooler. Steam is generated. |

- Gas Turbines

Gas turbine generators, also known as combustion turbines (CTs), are a mature technology. These systems consist of a compressor, a combustion chamber, and a turbine. Air is compressed in the compressor, and fuel is mixed with the air and ignited in the combustion chamber. The gas then enters the turbine's blades at high velocity through a nozzle, causing pressure on the blades. This pressure energy is converted into rotational energy and converted into the power output of the generator.

Table **3.2**-Characteristic descriptions of gas turbine.

| **Characteristics** | **Description** |
|---|---|
| Engine type | Gas turbine |
| Size | 500kW to 250MW, with larger sizes for central power generation. |
| Electrical efficiency | Simple cycle (25-40%) and combined cycle(40-60%). |
| Overall efficiency | 70-75% |
| Type of technologies | Not applicable |
| Fuel | Natural-gas,biogas,propane, distillate oil |
| Thermodynamic cycle | Brayton cycle |
| Heat recovery | The exhaust gas can be used directly for process energy or heat recovery steam generator (HRSG) can be added to the system for steam generation. |

- Steam Turbines

A steam turbine converts the thermal energy in steam into rotational energy. The turbine consists of both stationary blades, called nozzles, and rotating blades. The steam enters the blades through the nozzles at a high velocity, thus inducing pressure on the buckets and causing them to rotate. The rotational motion is converted into the power output. In contrast with other CHP technologies, steam turbines usually generate electricity as a byproduct of steam generation. While steam turbines are competitively priced compared to other technologies, the costs of the boiler, fuel handling, and overall steam systems and the custom nature of most installations tend to drive up equipment costs.

Table **3.3**-Characteristic descriptions of steam turbine.

| **Characteristics** | **Descriptions** |
|---|---|
| Engine type | ***Steam turbine*** |
| Size | 50kW to 1,300MW, with larger sizes for central power generation |
| Electrical efficiency | 30-42% |
| Overall efficiency | 80% |
| Type of technologies | Two types of steam turbines are used in CHP applications non-condensing (back-pressure) turbines and extraction turbines. |
| Fuel | Because a steam turbine does not directly convert a fuel in to electrical energy, steam must be created via a boiler or heat recovery steam generator (HRSG).Boilers can accept many fuel types, including coal, oil, natural gas, agricultural, forest and urban biomass, rice husk, bagasse. |
| Thermodynamic cycle | Rankine cycle |
| Heat recovery | Because steam turbines produce electricity as a byproduct of steam generation, they are recovering useful heat. Exhaust steam from the turbine can also be used in a process or can be converted to hot water or for use in a chiller |

- Micro turbines

Micro turbines are similar in operation to combustion turbines, consisting of a compressor, a combustion chamber, and a turbine. However, most micro turbines contain a recuperator that transfers heat from the exhaust gas to the intake air. This preheating of the intake, or combustion, air reduces the amount of fuel required for ignition in the combustion chamber, thereby increasing energy efficiency.

Table **3.4**-Characteristic descriptions of micro-turbine.

| **Characteristics** | **Description** |
|---|---|
| Engine type | ***Micro turbine*** |
| Size | 25kW to 250kW |
| Electrical efficiency | 20-30% |
| Overall efficiency | 65-75% |
| Type of technologies | Not applicable |
| Fuel | Natural gas, biogas, propane, distillate oil,liquefied petroleum gas (LPG). |
| Thermodynamic cycle | Brayton cycle |
| Heat recovery | The recuperator on most microtturbine designs limits the amount of waste heat available for recovery. Heat can be recovered from the exhaust gas for water heating or low pressure steam. |

- Fuel cells

Fuel Cells Different from all of the other technologies discussed above, a fuel cell is an electrochemical device consisting of two electrodes, a negative anode, a positive cathode, and an electrolyte. In most configurations, hydrogen or another fuel enters through the anode and an oxidant, such as oxygen enters through the cathode. The hydrogen splits into protons, with a positive charge, and electrons, with a negative charge. The protons are allowed to pass through the electrolyte, while the electrons are forced through a circuit, generating electricity. The electrons then travel to the cathode, where both the protons and electrons react with oxygen. The two by-products in this system are water and heat.

Table **3.5**-Characteristic descriptions of fuel cell.

| **Characteristics** | **Description** |
|---|---|
| Engine type | ***Fuel cell*** |
| Size | 200kW to 2MW |
| Electrical efficiency | 40-70% |
| Overall efficiency | 65-80% |
| Type of technologies | Phosphoric acid (PAFC), alkaline (AFC), proton exchange membrane (PEM), molten-carbonate (MCFC), and solid oxide (SOFC). |
| Fuel | Primary fuel is hydrogen-can be reformed from many fuel sources, including natural gas, coal gas, propane and methanol. |
| Thermodynamic cycle | Not applicable, use of an electrochemical process. |
| Heat recovery | Heat created during electrical generation with varied grades of heat depending on the type of fuel cell (i.e., the PAFC and PEM produce lower grades of heat, and the MCFC and SOFC produce higher grades of heat) |

- Stirling Engines

A stirling engine operates in a closed cycle configuration. A fixed amount of gas, called a working fluid (e.g., air, hydrogen, helium), remains inside a chamber while the heating and cooling sources are external to the engine. The external heat source allows for a large range of fuels for operating the stirling engine. One configuration operates on four cycles: heating, expansion, cooling, and compression. During the heating cycle, the working fluid is heated via an external heat source on the "hot side." The working fluid increases in pressure and expands due to the increase in temperature and the fluid flows to the "cold side." As the hot fluid enters the "cold side," the gas cools and the pressure is reduced. One of the two pistons then compresses the fluid in the "cold side," forcing the fluid to return to the "hot side" and restart the cycle. The transfer of fluid between the "hot side" to the "cold side" causes the two pistons to move. The motion of the pistons is converted into rotational energy. The rotational motion is converted into the power output.

Table **3.6**-Characteristic descriptions of stirling engine.

| **Characteristics** | **Description** |
|---|---|
| Engine type | ***Stirling engine*** |
| Size | 55W to 55kW (reported) |
| Electrical efficiency | Around 30% (based on the STM power model 4-120) |
| Overall efficiency | Unknown |
| Type of technologies | Kinematic and free-piston |
| Fuel | A large range of fuels may be used, because the stirling engine is heated externally-fossil fuels, solar, nuclear, waste heat, and biomass. |
| Thermodynamic cycle | Stirling cycle |
| Heat recovery | Heat is recovered from exhaust gases. |

### 3.5-Fuel Sources

A range of fuel sources are available for CHP applications. The fuel sources are grouped by type (liquid, gaseous, solid, and renewable/solar) and are characterized below:

- Liquid fuel: has the ability to easily be stored on-site; often used in emergency generators.
- Gaseous fuel: natural gas is easy to obtain; depending on the location, biogas may already be produced as a waste product.
- Solid fuel: has the ability to easily be stored on-site; depending on the location, the fuel may already be on-site as a waste product.
- Renewable fuel: though other fuels included in the gaseous or solid categories may be considered renewable.

### 3.6-Cogeneration system using steam turbines

Steam turbine is one of the most versatile and oldest prime mover technologies still in general production. Power generation using steam turbines has been in use for about 100 years, when they replaced reciprocating steam engines due to their higher efficiencies and lower costs [3]. At the very beginning, steam turbine based cogeneration plant provides more power and heat. However, the consequence of the industrial progress pushes the needs to change the steam turbine based cogeneration to more efficient cogeneration systems such as reciprocating engine based or gas turbine based cogeneration plant. The general operating technique with steam turbine can be classified into two main categories, backpressure steam turbine and the condensing type steam turbine.

### 3.7-Steam Turbines for CHP application

Historically, steam turbines have been the primary power generation technology, providing mechanical and electric power and steam for a variety of industrial processes. In contrast with other CHP technologies, steam turbines usually generate electricity as a byproduct of steam generation. While steam turbines are competitively priced compared to other technologies, the costs of the boiler, fuel handling, and overall steam systems and the custom nature of most installations tend to drive up equipment costs.

Steam turbine cogeneration systems can produce all or a part of the energy requirement of the site, and the energy released at high temperature in the exhaust stack can be recovered for various heating and cooling applications. In the description of operating cycle with steam turbines, all the cogeneration plants mainly operate from two categories such as topping cycle and bottoming cycle. The type of cogeneration system found at Wonji-Shoa sugar factory is extraction –condensing steam turbine cogeneration system.

### 3.8-Equipment combinations in cogeneration systems

There are several units required in the cogeneration plant such as steam turbine, boiler, waste heat recovery system, condenser etc. However, the combinations of different components mainly depend on the required conditions. The following combinations are the established bases on the current applications in many industrial sectors such as paper industry, sugar mills, and rice husk mills.

Steam turbine and steam boiler based on fuel for cogeneration system are:

- ✓ Biomass fired with back pressure steam turbine
- ✓ Coal or lignite fired with back pressure steam turbine
- ✓ Liquid fuel fired with extraction and condensing steam turbine
- ✓ Natural gas fired with extraction and back pressure steam turbine

**3.8.1 Back pressure steam turbine**

In the back pressure steam turbine configuration below the high pressure steam is generated from the boiler and directed to the backpressure turbine to generate power for converting thermal energy into mechanical energy by electric generator. Part of the steam left over in the backpressure turbine still carries some thermal energy and lastly routes to the process consumer to provide hot steam for heating or drying applications and then back to the boiler to start next cycle (Premalatha,2008).

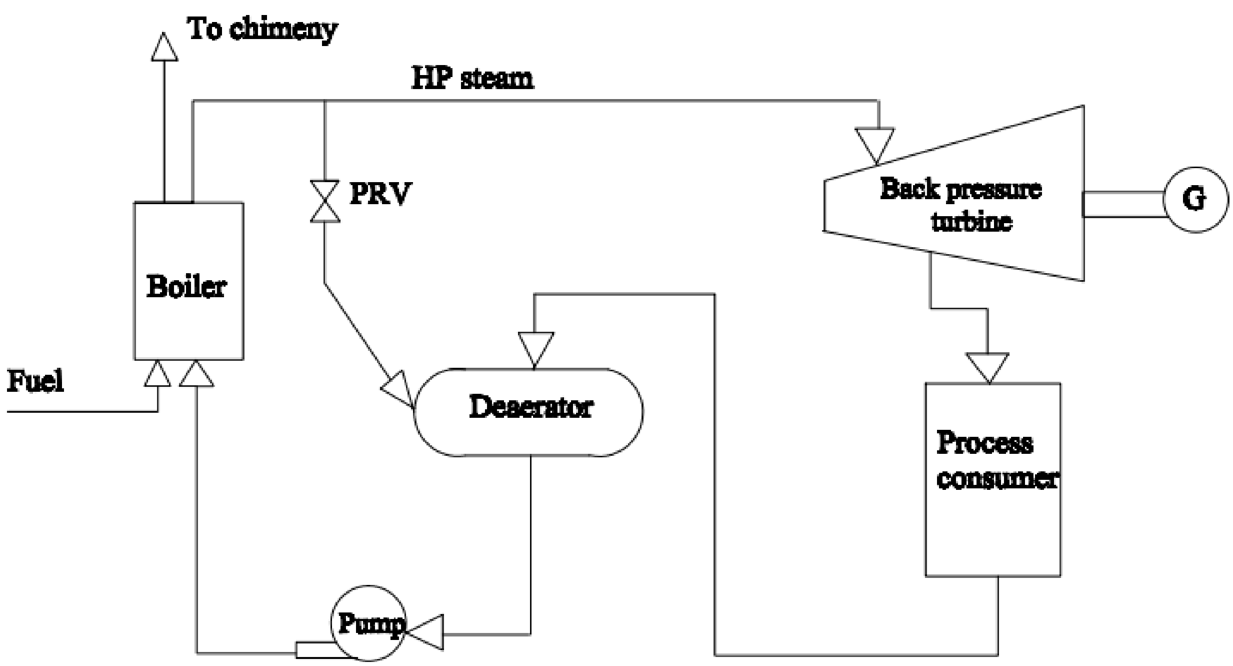


Figure **3.1**-Back pressure steam turbine cogeneration configuration

Advantages of back pressure steam turbine configuration:

- ✓ It is the priority choice for considering the highest efficiency of producing electricity and process heat simultaneously.
- ✓ It is the best option to employ when there is a fluctuating demand of electricity and heat at different period.

Disadvantage of back pressure steam turbine configuration:

- ✓ Steam flow rate is not adjustable.

### 3.8.2 Extraction condensing steam turbine

In second configuration (extraction condensing route), the high pressure and temperature steam expands through the turbine and extracted to process heat and some steam condensing to the condenser. At the first stage of steam extraction, part of the steam is sent to the process heater to supply process heat, and some steam flows to end-use devices to provide the necessary amount of thermal load (Premalatha,2008).

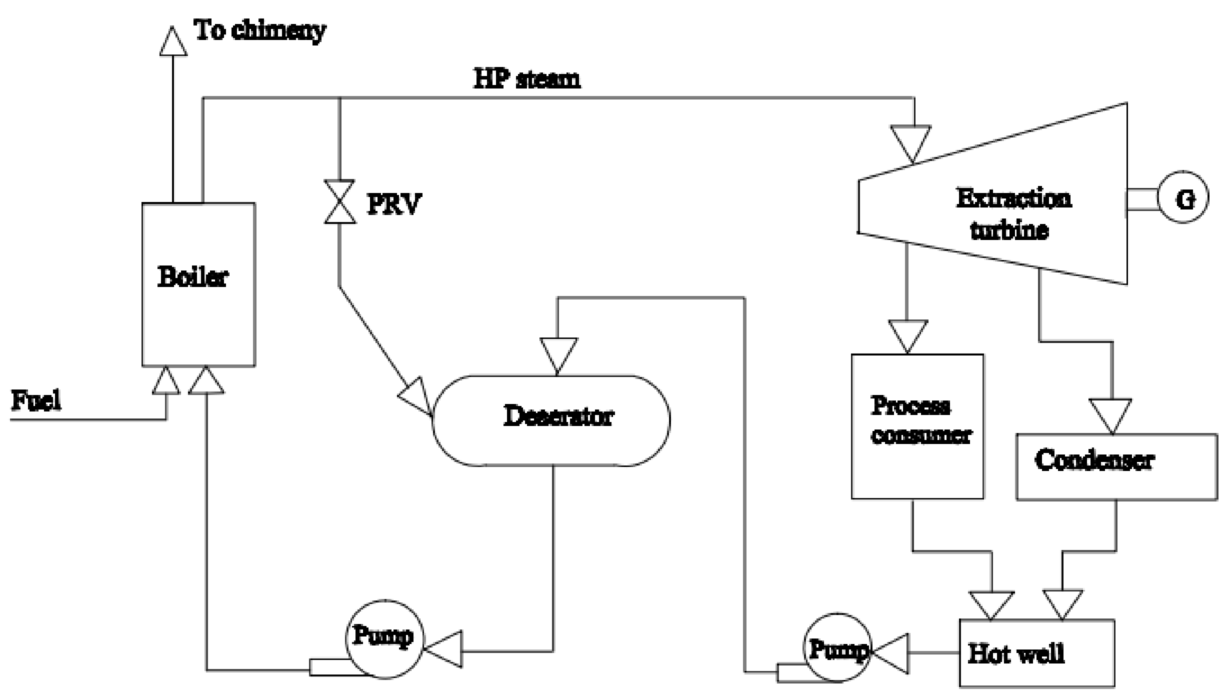


Figure **3.2**-Extraction condensing steam turbine cogeneration configuration.

Advantages of extraction condensing steam turbine configuration:

- ✓ There is a benefit from easily controlled power generation.
- ✓ Steam flow rate is adjustable.

Disadvantage of extraction condensing steam turbine configuration:

- ✓ Usually used for low or medium pressure steam.

**3.9-Sugar mill biomass**

The biomass considered in this model is sugarcane bagasse. Bagasse is the fibrous remain of sugarcane stalks after crushing and extraction of juice. Like any other biomass, it is composed of cellulose, hemicelluloses and lignin. It is estimated that approximately 270 kg of bagasse (50% humidity) is produced per ton of sugarcane processed (Garcia-Perez et al., 2002) or 1.25 kg of bagasse is produced per kilogram of sugar produced (Botha and Von Blottnitz, 2006; Singh et al., 2007).

Bagasse as feed has varying composition and heating values. Its characteristic property depends mainly on the climate, type of soil on which cane was grown, cane variety, harvesting method, amount of cane washing and the efficiency of the milling plant (Janghathaikul and Gheewala, 2004). The Gross Calorific Value (GCV) and the Net Calorific Value (NCV) of bagasse (at 50% moisture) are 9.314MJ/kg and 7.67MJ/kg respectively (Mbohwa, 2003). Oven dry bagasse has GCV of about 18 MJ/kg (Garcia-Perez et al., 2002).

The bagasse percentage in the cane can vary from 23% to 37%. On the average it is around 30%. This depends on the fiber percentage of cane, which normally ranges from 43 to 47%. The rest of the bagasse is made up of trapped dissolved matter, trash and water. The moisture content is also a large determinant of the bagasse percentage cane, since it constitutes about 45 to 55 % of the bagasse, soon after processing. The moisture content can be reduced by better de-watering, improved processing or by simply leaving the bagasse to dry. The bagasse is burnt in boilers that are normally designed to use both bagasse and/or fuel oil.

The average steam to bagasse ratio is normally 2.2. At a density of 130 kg/m3, storing bagasse takes a lot of space, hence the need to use boilers that burn as much of it as possible. The boilers generally range from 35 to 150 tons of steam per hour, at pressures varying from 15 to 82 bars and temperatures ranging from 300 to 525 degrees Celsius. The turbo-alternators that are used for electricity generation can be up to 35 MW. The possible capacity is increasing with the development of new technology in the area.

The high-pressure steam from the boilers is used in the turbo-alternators to produce electricity. It can also be used by the shredders, de-watering mills, drying-off mills, steam-feed pumps, for sugar processes (after pressure reduction) and by an ethanol plant. The usage areas vary and depend on the set up of the sugar factory. The factory processes use the low-pressure steam that is exhausted from the high-pressure steam processes. All these processes can be improved upon, resulting in less usage of steam and more electrical energy production.

The amount of steam used per ton of crashed cane, depends on how modern the plant is and whether or not the plant has steam or electrically driven critical machinery. Generally a figure of 50-60 per cent steam on cane is used, which is about 450 to 600 kg of steam per ton of cane. A reduction of this consumption level will reduce the need to redirect live steam to sugar processes. In the beetroot sugar industry, 300 kg of steam are used per ton of beetroot. There is room for the improvement of this figure in the sugar cane industry [2, 4, 12, and 14].

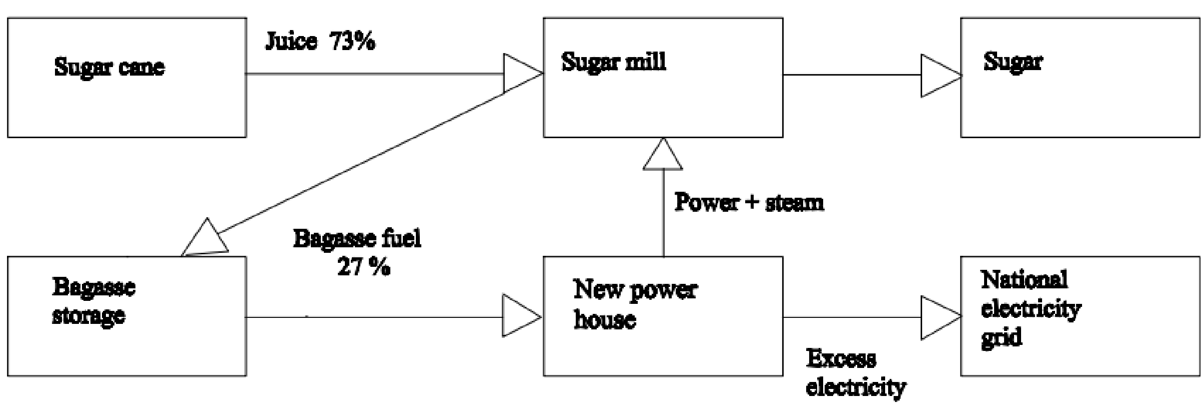


Figure **3.3**-Flow chart of power, steam and surplus electricity production from bagasse as fuel.

**3.10-Combustion model description**

The model is based on the Rankine cycle of a Combined Heat and Power (CHP) plant which is currently the most predominantly used cogeneration system in sugar mills. It consists mainly of the biomass combustor; steam generator (boiler) and turbo-alternator (see Fig. 2.2).

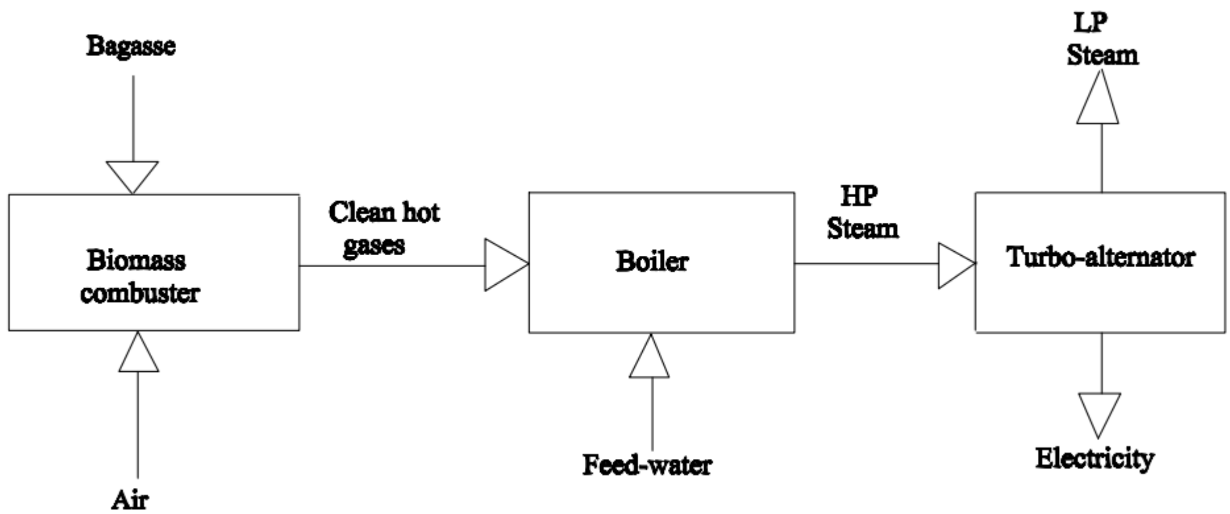


Figure **3.4** – Schematic representation of the combustion, steam generation and power generation process.

In this process, fuel (bagasse at 50% moisture) and combustion air are fed to the biomass combustor. Hot combustion gasses generated after combustion are cleaned to remove ash and particulate matter and then sent to the boiler for steam production. The combustor is assumed to operate at adiabatic conditions, in which case the net heat duty of the combustor is set to zero. This ensures that all the combustion energy is supplied by the only the bagasse. Combustion air is preheated with flue gas in the air preheater to raise its temperature to 250$^{o}$C prior to been fed to the combustor. Air is supplied in excess to ensure complete combustion and also to make sure that the percentage volume of oxygen in the flue gas meets environmental standards.

Cleaned hot combustion gases from the combustor are sent to the boiler from which heat is recovered to raise high pressure steam (HP Steam). The boiler is modeled as a series of three heat exchangers. Feed water enters the first exchanger (economizer) where its temperature is raised to the just below the saturation temperature and then sent to the second exchanger (evaporator) to produce saturated steam. The third exchanger raises the temperature of the saturated steam to HP steam at superheat conditions, depending on the degree of superheat temperature set. All heat for the steam generation is supplied by the hot combustion gasses.

HP steam from the boiler is expanded in a turbo-alternator; back-Pressure Steam Turbine (BPST) or Condensing Extraction Steam Turbine (CEST), depending on the pressure of the steam, to produce electricity and Low Pressure steam (LP steam) for the sugar mill. In this model, the turbine is modeled as an isentropic turbine in with an isentropic efficiency of 85% and a generator efficiency of 98%. Also this model assumes that all prime movers in the sugar mill are run by electrical motors; hence all the HP steam generated by the boiler is directed to the turbine.

### 3.10.1- Direct versus indirect combustion methods

Direct Combustion methods: The most common way of converting biomass to heat energy is through straight combustion, and this accounts for around 90% of all energy attained from biomass. However there are a number of different technologies available that can be used for biomass combustion divided into two groups: fixed bed combustion systems and fluidized bed combustion systems.

- ✓ *Fixed Bed Combustion* -There are two types of fixed bed combustion: underfeed stokers and grate firings. With these methods of combustion air is primarily supplied through the grate from below, and initial combustion of solid fuel takes place on the grate and some gasification occurs. This allows for secondary combustion in another chamber over the first one where secondary air is added.
- ✓ *Fluidized Bed Combustion Systems* - Fluidized bed furnaces operate in quite a different manner from fixed bed furnaces and have a number of advantages associated with them. There are two main types of fluidized bed furnace, Bubbling Fluidized Bed (BFB) and Circulating Fluidized Bed (CFB).
- Bubbling Fluidized Bed (BFB) Furnaces- The fundamental principle of a BFB furnace is that the fuel is dropped down a chute from above into the combustion chamber where a

bed, usually of silica sand, sits on top of a nozzle distributor plate, through which air is fed into the chamber with a velocity of between 1 and 2.5m/s. The bed normally has a temperature of between 800 and 900°C and the sand accounts for about 98% of the mixture, with the fuel then making up a small fraction of the fuel and bed material. There are two main advantages in terms of fuel size and type over more traditional fixed bed systems. Firstly they can cope with fuel of varying particle size and moisture content with little problem, and secondly they can burn mixtures of different fuel types such as wood and straw. BFB's are only a practical option with larger plants with a nominal boiler capacity greater than 10 MW.

- Circulating Fluidized Bed (CFB) Furnaces- If the air velocity is increased to 5-10m/s then a CFB system can be achieved, where the sand is carried upwards by the flue gases and a more thorough mixing of the bed material and fuel takes place. The sand is then separated from the gas in a hot cyclone or U beam separator at the top of the furnace and fed back into the combustion chamber where the whole process begins again. CFB's deliver very stable combustion conditions but it comes at a cost. Due to their larger size compared to other combustion methods the cost is relatively high and there are problems involved with fuel size, which must be very small, and the difficulties involved in running them at partial load. All of this means that they are really only feasible for plants with a boiler capacity of over about 30MW.

Indirect combustion methods: The indirect combustion methods have the capability to convert raw biomass into gases, liquid fuels, or solid fuels so that it can be used directly in the power plants for generating electricity. Biomass usually contains carbohydrates, which are composition of oxygen, carbon, and hydrogen and that can be broke down into different types of chemicals from which some are very useful fuels.

# CHAPTER FOUR

## 4. Data analysis

Proper consideration of the steam properties of the different components of the cogeneration unit at Wonji-Shoa Sugar Factory(WSSF) will be of great help to accomplish the evaluation task.The cogeneration unit has one bagasse combusted boiler with a nominal output pressure of 65bar and a design efficiency of 72%.The mass flow rate of steam entering the boiler is 13.78kg/s and that of the makeup water is 3.57kg/s.

Condensing extraction steam turbine is employed in the power generation unit of the sugar factory and its inlet temperature during the time of observation was 466.67˚C and has a design efficiency of 85%. The data gathered from the factory power generation control board are recorded in the table below.

Table **4.1**-Actual values of operating parameters at inlet and exit of different machines of WSSF taken from process control board.

| | $\dot{m}(kg/s)$ | $T(°C)$ | $P(bars)$ | $h(KJ/\mathrm{kg})$ |
|---|---|---|---|---|
| Inlet to pump-II | *17.36* | *69.02* | *0.3* | *288.96* |
| Inlet to boiler | *17.36* | *96.74* | *61.08* | *405.39* |
| Inlet to CEST turbine | *17.36* | *466.67* | *61.08* | *3412.53* |
| Inlet to LP process heater | *9.79* | *136.89* | *1.2* | *2743.7* |
| Exit of LP process heater | *9.79* | *104.7* | *1.2* | *439.02* |
| Thermal load (steam consumer) | *3.57* | *206.4* | *2.1* | |
| Inlet to condenser | *4* | *41.4* | *0.07* | *2576.12* |
| Inlet to pump-I | *4* | *38.81* | *0.07* | *197.55* |
| Exit from pump-I | *4* | *47.18* | *0.3* | *197.55* |
| Feed water to FWT | *3.58* | *25* | $1atm$ | |

Based on the data taken from the lab found at WSSF the flue gas analysis shows that the percentages of $CO_2$ and $O_2$ in the flue gas are *14.5*% and *7.3*% respectively.

Primary actual data taken from the data log book of the power generation unit found at WSSF are the following:

Cane crushing capacity of the factory =*5822TCD*

Working season days =*228days/year*

Maximum continuous rating (MCR) =*100%*

Set point boiler operating pressure =*65bar*

Condenser cooling water inlet temperature=*25ºC*

Maximum temperature difference of cooling water outlet and inlet=*10ºC*

Design point power output =*16MW*

Turbine isentropic efficiency =*85%* (taken from manufacturer's catalogue)

The cogeneration configuration of the WSSF is shown in the figure below. It shows the different components employed along with the path of steam at the inlet and exit of the components. Moreover the individual component name is given in a table .

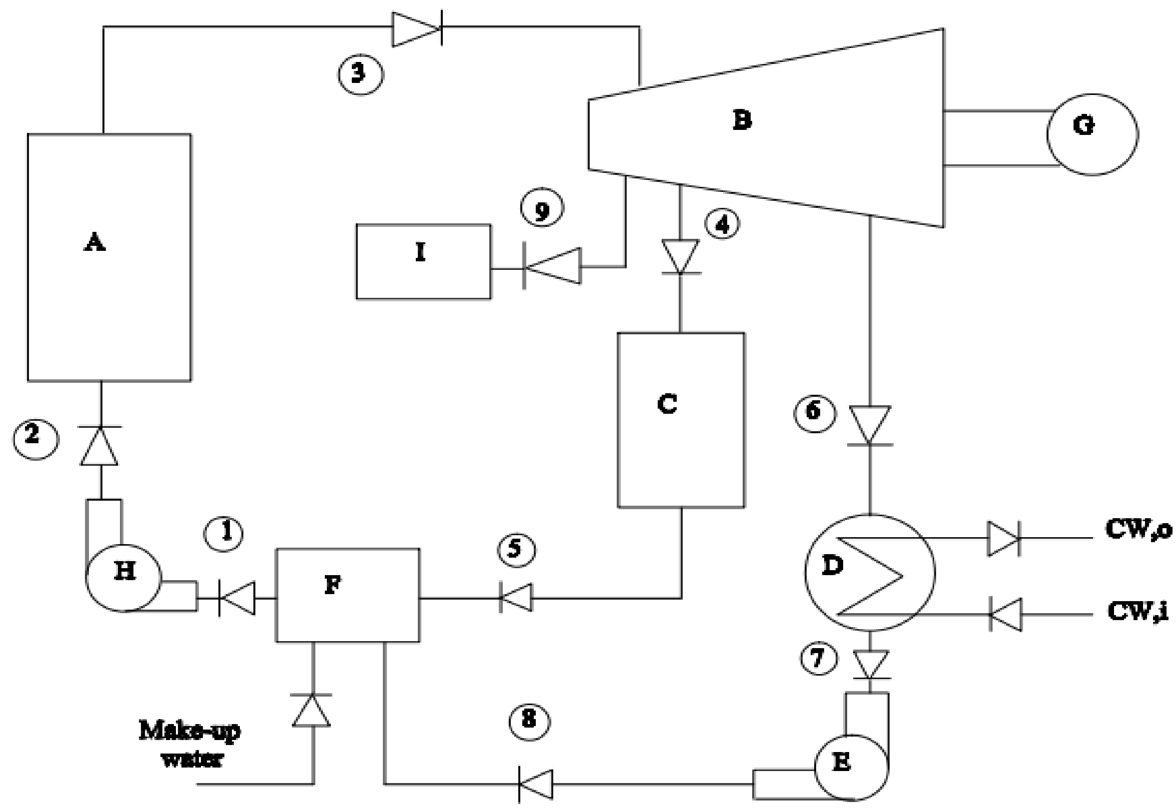


Figure **4.1**- Thermodynamic cycle of the cogeneration power plant at Wonji-Shoa sugar mill.

Table **4.2**- List of components in the cogeneration cycle

| LABEL | COMPONENTS |
|---|---|
| A | Boiler |
| B | Steam turbine (CEST) |
| C | LP-Process heater |
| D | Condenser |
| E | Condensate return pump |
| F | Feed water tank |
| G | Generator set |
| H | Boiler feed pump |
| I | Thermal load (steam consumer) |

Figure 4.1 and Table 4-2 describe the scheme of a modern $16MW$ bagasse based cogeneration plant located in WONJI,ETHIOPIA. This specific configuration applies to satisfy $16MW$ of electricity.

In order to have maximum efficiency, elimination of the additional steam flowing to distillery in the consideration of maximizing the power and process heat production is made, but the percentage of steam extraction from the condensing extraction steam turbine and some operating conditions are considered to be the same for the same study.

At first, high pressure and temperature steam is generated from the boiler, which is directly combusted with bagasse at an average calorific value of $7650KJ / kg$ , and thus the superheated steam is going to expand through CEST at $100\%$ of the total steam flow rate. In the condensing type steam turbine,*56.22%* of steam extracted from the CEST to the process heater ,*20.56%* to the thermal load(steam consumer) and the remaining *23.22%* steam directs to the condenser . The steam quality turns into saturated liquid after passing through the process heater and condenser at the corresponding pressure. The processed liquid water will pumped to the boiler pressure level, and then flows back to mixing chamber, at the meantime, the makeup water flows into the mixing chamber to replenish the water which adopts to centrifugal applications in the mills.

**4.1-Thermal analysis of cogeneration configuration**

Circular pump feeds water that enters to boiler from the mixing chamber where the makeup water mixes with the processed water from the process heater and the steam expands via CEST to condenser . The total fuels supply at $7.24kg / s$ in the present analysis, $72\%$ and rated combustion efficiency is considered as well since the fuel is not possible to have $100\%$ energy conversion .

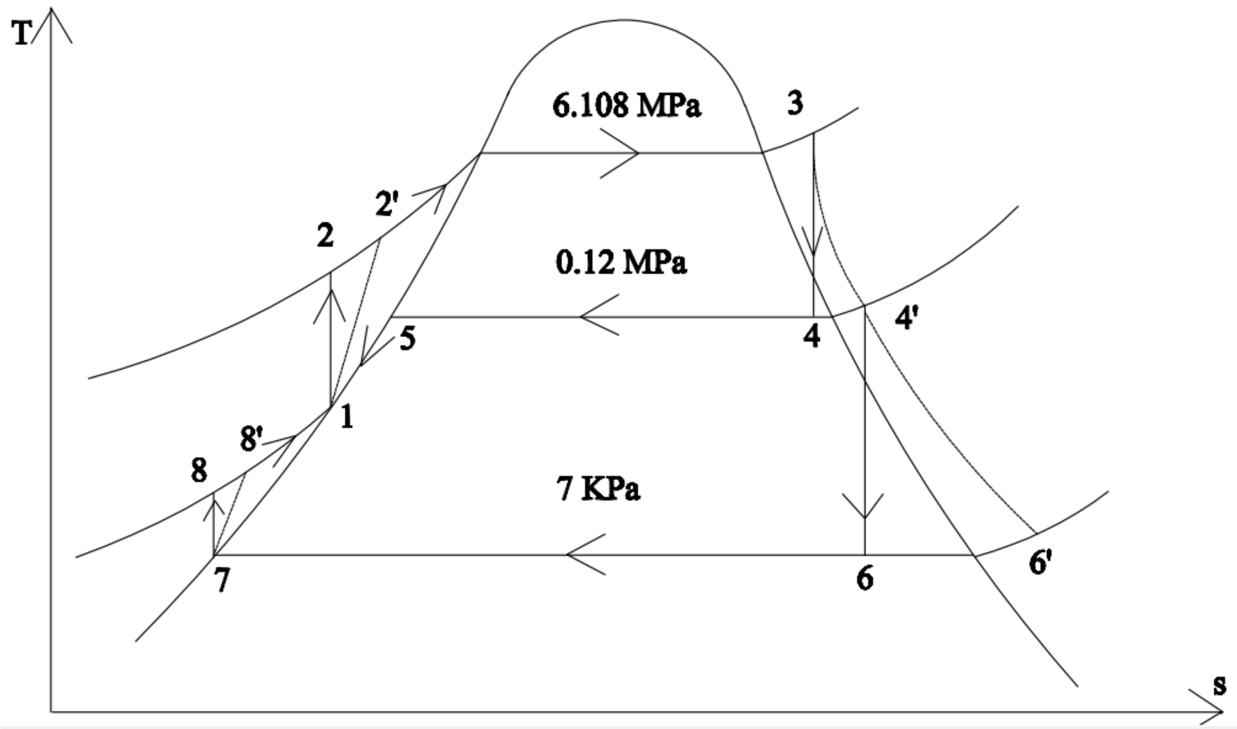


Figure **4.2**- T-S Diagram of the cogeneration configuration

Determination of actual values of thermodynamic properties at different state points as found by interpolation:

<u>*State point -1*</u>: Entrance to pump-II

(Data taken from process control board)

$T_1 = 69.02°C \quad , \; P_1 = 30KPa$

$$v_1 = v_f \; @ \; P = P_{sat}\left(30KPa\right) = 0.0010224m^3 / kg$$

$$h_1 = h_f \; @ \; P = P_{sat}\left(30KPa\right) = 288.96KJ / kg$$

<u>*State point -2'*</u>: Outlet from pump-II or entrance to boiler

Since state point-2' is in sub-cooled region properties at this state are taken to be as that of the saturated liquid at the same temperature.

$$h_{2'} = h_f \; @ \; T = T_{sat}\left(96.74°C\right) = 405.39KJ / kg$$

*State point-3*: Inlet to the turbine

Actual values,

$T_3 = 466.67°C$ , $P_3 = 6.108 MPa$

$s_3 = s_4 = 6.7622 KJ / kg * K$

$h_3 = 3412.53 KJ / kg$ , (as obtained from superheated steam table @ $T_3 = 466.67°C$ and $P_3 = 6.108 MPa$)

*State point-4*: Turbine outlet to the thermal loads or process heater (isentropic)

(Data taken from process control board)

$T_4$=104.7$^oC$, $P_4$= 120*KPa*

( $s_f = 1.359 KJ / kg * K$ , $s_g = 7.298 KJ / kg * K$ , $s_4 = 6.7622 KJ / kg * K$ )

Quality at state point-4 is thus,

$s_4 = s_f + x s_{fg}$ ,

$$6.7622 = 1.359 + \left[x\left(7.298 - 1.359\right)\right]$$

$x = 0.9097$ or $x = 90.97\%$

$h_f = 439.02 KJ / kg$ , $h_g = 2243.95 KJ / kg$

Enthalpy at state point-4 is thus,

$h_4 = h_f + x h_{fg}$ ,

$$h_4 = 439.02 + \left[0.9097\left(2243.95 - 439.02\right)\right]$$

$$h_4 = 2080.96 KJ / kg$$

State point-4’: Turbine outlet to the process heater (actual)

(Data taken from process control board)

$T_{4'} = 136.89^{o}C,\ P_{4'} = 120KPa$

$T_{sat} @ P\ (120KPa) = 104.7^{o}C,$

Since $T_4' > T_{sat} @ 120KPa$ , state point 4’ is in the superheated vapor region.

$h_4' = h\ @\ T_4'\ (136.89^{o}C)$ and $P_4'\ (120KPa) = 2743.7KJ/kg$ and $s_{4'}$=7.4812KJ/kg.K, (as obtained from superheated steam table).

*State point-5*:

$h_5 = h_f\ @\ P = P_{sat}(120KPa) = 439.02KJ/kg$

*State point-6*: Turbine outlet to condenser (ideal)

(Data taken from process control board)

$P_6 = 7KPa$

$T_6 = T_{sat}\ @\ P = P_{sat}(7KPa) = 38.81^{o}C$

$v_f = 0.0010074m^3/kg$

$s_6 = s_{4'} = 7.48124KJ/kg.K$ , $s_f = 0.55764KJ/kg*K,\ s_g = 8.2767KJ/kg*K$

$s_6 = s_f + xs_{fg}$

$7.48124 = 0.55764 + [x\ (8.2767 - 0.55764)]$

$x = 0.8969\ or\ x = 89.69\%$

$h_f = 162.75KJ/kg\ , h_g = 2571.34KJ/kg$

$h_6 = h_f + xh_{fg}$

$= 162.75 + \left[0.8969\left(2571.34 - 162.75\right)\right]$

$= 2323.01 KJ / kg$

<u>State point-6’</u>: Turbine outlet to condenser (actual)

$T_{6'} = 41.4^o C,\ P_{6'} = 7KPa$

Since $T_{6'} > T_{sat} @ 7KPa$, state point 6’ is in the superheated vapor region.

$h_{6'} = h @ T_{6'}\left(41.4^o C\right) \ and \ P_{6'}\left(7KPa\right) = 2576.18 KJ / kg$

<u>State point-7</u>: Inlet to pump-I

$P_7 = 7KPa,\ T_7 = 38.81^o C$

$h_7 = h_f @ P_{sat}\left(7KPa\right) = 162.75 KJ / kg$

$v_7 = v_f @ P_{sat}\left(7KPa\right) = 0.0010074 m^3 / kg$

<u>State point-8’</u>: Outlet from pump-I

$h_{8'} = h_f @ T_{sat}\left(47.18^o C\right) = 197.55 KJ / kg$

Assumptions

The following assumptions are made to simplify the present work before processing to the analysis aspect in the present analysis.

- ✓ Kinetic energy ,and the change in elevation (potential energy) of different components are negligible, in addition all the presented evaluations are found under steady flow conditions
- ✓ Ambient temperature and pressure are taken as 25$^o$C and 84.93KPa for an altitude of 1540m.

- ✓ Rated Boiler combustion efficiency is 72% (Taken from manufacturer's catalogue).
- ✓ Isentropic efficiency of the turbine is 85% (Taken from manufacturer's catalogue).
- ✓ Isentropic efficiency of pumps is 90% (Taken from manufacturer's catalogue).
- ✓ All the biomass fuels are burned directly under receiving condition with 50% moisture contains, and the evaluation of fuels heating values are based on ultimate analysis.
- ✓ There is no steam loss across steam turbine.
- ✓ Generator efficiency as per the manufacturer is 98.03 %.
- ✓ Medium pressure extracted steam is utilized for further process of sugar mill and low pressure exhaust steam from turbine is condensed in a condenser.

**4.2-Boiler efficiency analysis**

By definition, the fiber in bagasse is insoluble in water; it consists mainly of cellulose, hemicellulose, lignin and sulphur. Figure 4.3 present the average cane fiber composition according to Purchase (1995) and Rein (2007). For the calculations in this work it is adopted a bagasse moisture of $50\%$ and the cane fiber composition of $46.46\%$.

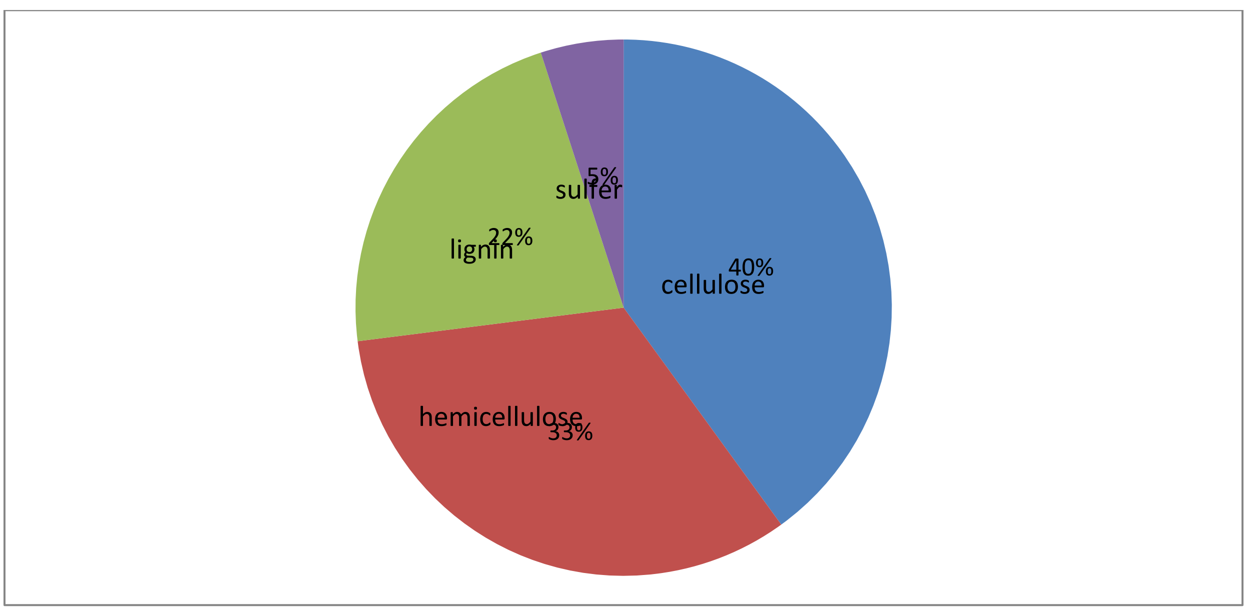


Figure **4.3**-Percentage composition of components in cane fiber (Purchase,1996)and (Rein,2007).

The recommended value of the steam to bagasse ratio , the ratio between the amount of steam produced in the boiler to the bagasse supplied to the boiler, for this specific case is 2.4 (Rein,2007) and the steam flow rate from the boiler output is $62.5TPH$ or $17.36(kg/s)$. Therefore the bagasse feed rate based on the recommended value of steam to bagasse ratio is,

$$Bagasse\ \ feed\ \ rate = (17.36/2.4)kg/s = 7.24kg/s$$

Assuming $7.24kg/s$ of bagasse fuel combustion,

$$50\%\ \ is\ \ moisture \Rightarrow 0.5 * 7.24kg/s = 3.62kg/s$$

$$46.46\%\ \ is\ \ \ fiber \Rightarrow 0.4646 * 7.24kg/s = 3.364kg/s$$

$$3.54\%\ \ is\ \ \ ash \Rightarrow 0.0354 * 7.24kg/s = 0.2563kg/s$$

The fiber is composed of

$$40\%\ cellulose\ \ \Rightarrow 0.4 * 3.364kg/s = 1.3456kg/s$$

$$33\%\ \ hemicellulose \Rightarrow\ 0.33 * 3.364kg/s = 1.1kg/s$$

$$22\%\ lignin\ \ \Rightarrow 0.22 * 3.364kg/s = 0.74kg/s$$

$$5\%\ sulfur\ \Rightarrow 0.05 * 3.364kg/s = 0.1682kg/s$$

The constituent ash of the bagasse is considered as $SiO_2$ , and due to this the constituent is large in the ash analysis. Other constituents of the bagasse such as sucrose and other soluble solids were neglected. Since the ash is inert material, it is considered that only the cellulose, hemicelluloses, lignin and sulphur react with the combustion air. The combustion reactions are taken from Wooley and Putshe (1996):

We have the following mass of reactants and products per kilogram of bagasse combusted.

Cellulose: $C_6H_{10}O_5 + 6O_2 \rightarrow 5H_2O + 6CO_2$

The table below shows the mass of reactants and products of the balanced combustion reaction of cellulose with oxygen per kilogram of bagasse combusted.

| $C_6H_{10}O_5$ [Kg/Kg fuel] | $O_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] | $H_2O$ [Kg/Kg fuel] | $CO_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] |
|---|---|---|---|---|---|
| 0.18584 | 0.1985 | 0.747 | 0.103 | 0.303 | 0.725 |

Hemicelluloses: $C_5H_8O_4 + 5O_2 \rightarrow 4H_2O + 5CO_2$

The table below shows the mass of reactants and products of the balanced combustion reaction of hemicellulose with oxygen per kilogram of bagasse combusted.

| $C_5H_8O_4$ [Kg/Kg fuel] | $O_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] | $H_2O$ [Kg/Kg fuel] | $CO_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] |
|---|---|---|---|---|---|
| 0.15332 | 0.039 | 0.1468 | 0.08363 | 0.2553 | 0.6114 |

Lignin: $C_{7.3}H_{13.9}O_{1.3} + 10.125O_2 \rightarrow 6.95H_2O + 7.3CO_2$

The table below shows the mass of reactants and products of the balanced combustion reaction of lignin with oxygen per kilogram of bagasse combusted.

| $C_{7.3}H_{13.9}O_{1.3}$ [Kg/Kg fuel] | $O_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] | $H_2O$ [Kg/Kg fuel] | $CO_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] |
|---|---|---|---|---|---|
| 0.10221 | 0.0564 | 0.2122 | 0.10374 | 0.2663 | 0.8836 |

Sulphur: $S + O_2 \rightarrow SO_2$

The table below shows the mass of reactants and products of the balanced combustion reaction of sulfur with oxygen per kilogram of bagasse combusted.

| $S$ [Kg/Kg fuel] | $O_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] | $SO_2$ [Kg/Kg fuel] | $N_2$ [Kg/Kg fuel] |
|---|---|---|---|---|
| 0.0164 | 0.0164 | 0.617 | 0.328 | 0.617 |

The excess air recommended for sugarcane bagasse boilers $30\%$ is according Lora and Zampieri (2008). This value is considered for the analysis. Heat losses due to incomplete chemical combustion are assumed as $2.51\%$ and losses due to radiation and convection to ambient air is considered in $0.1\%$ according to Rein (2007). As well as in the standard ASMEPTC 4, which considers carbon boilers, Rein (2007) indicates the Gross Heat Value (GHV) as calculation base for these losses.

In this analysis it is considered that the bagasse GHV is $9314 KJ / \mathrm{kg}$ and the mass flow of bagasse fed in the boiler is $7.24\ \mathrm{kg/s}$.

The performance parameters of boiler, like efficiency and evaporation ratio reduces with time due to poor combustion, heat transfer surface fouling and poor operation and maintenance. Even for a new boiler, reasons such as deteriorating fuel quality, water quality etc. can result in poor boiler performance. Boiler efficiency tests help us to find out the deviation of boiler efficiency from the best efficiency and target problem area for corrective action.

Thermal efficiency of boiler is defined as the percentage of heat input that is effectively utilized to generate steam. There are two methods of assessing boiler efficiency.

1) The Direct Method: Where the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.

2) The Indirect Method: Where the efficiency is the difference between the losses and the energy input.

Comparison of boiler efficiency determination the direct method versus indirect method:

### I. Boiler efficiency determination based on direct method

This is also known as 'input-output method' due to the fact that it needs only the useful output (steam) and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula

$$Boiler\ efficiency = \{Heat_{output} / \text{Heat}_{\text{input}}\} * 100\%$$

$$Boiler\ efficiency = \frac{Q(H-h)}{q * GCV_{fuel}} * 100\%$$

• Quantity of steam generated per hour $(Q)$ in $kg/hr = 17.36 kg/s = 62,496 kg/hr$

• Quantity of fuel used per hour $(q)$ in $kg/hr = 7.24 kg/s = 26,064 kg/hr$

• The enthalpy of steam $(H)$ $in\ KJ/kg = 2243.95 KJ/kg$

• The enthalpy of feed water $(h)$ $in\ KJ/kg = 176.06 KJ/kg$

• Type of fuel and gross calorific value of the fuel in $KJ/kg = 9314 KJ/kg$

Substituting the values in the efficiency equation we get:

$$\text{Boiler efficiency} = \frac{\frac{62,496 kg}{hr} * \left(\frac{2243.95 KJ}{kg} - \frac{176.06 KJ}{kg}\right)}{\frac{26,064 kg}{hr} * 9314 KJ/kg} = 0.5328 = 53.28\%$$

It should be noted that boiler may not generate $100\%$ dry steam, and there may be some amount of wet steam.

Advantages of direct method:

- ✓ Plant people can evaluate quickly the efficiency of boilers
- ✓ Requires few parameters for computation

- ✓ Needs few instruments for monitoring

Disadvantages of direct method:

- ✓ Does not give clues to the operator as to why efficiency of system is lower
- ✓ Does not calculate various losses accountable for various efficiency levels

### II. Boiler efficiency determination based on Indirect Method

There are reference standards for Boiler Testing at Site using indirect method namely British Standard, BS 845: 1987 and USA Standard is 'ASME PTC-4-1 Power Test Code Steam Generating Units'. Indirect method is also called as heat loss method. The efficiency can be arrived at, by subtracting the heat loss fractions from 100. The standards do not include blow down loss in the efficiency determination process. A detailed procedure for calculating boiler efficiency by indirect method is given below. The principle losses that occur in a boiler are:

- Loss of heat due to dry flue gas
- Loss of heat due to moisture in fuel and combustion air
- Loss of heat due to combustion of hydrogen
- Loss of heat due to radiation
- Loss of heat due to unburnt fuel in the flue gas

In the above, loss due to moisture in fuel and the loss due to combustion of hydrogen are dependent on the fuel, and cannot be controlled by design.

The data required for calculation of boiler efficiency using indirect method are:

- Ultimate analysis of fuel ( $H_2, O_2, S, C$, moisture content, ash content)
- Percentage of $O_2$ or $CO_2$ in the flue gas
- Flue gas temperature in $°C\ (T_f)$
- Ambient temperature in $°C\ (T_a)$ & humidity of air in $kg/kg$ of dry air.

- GCV of fuel in $Kcal/kg$
- Percentage combustible in ash (in case of solid fuels)
- GCV of ash in $Kcal/kg$ (in case of solid fuels)

<u>Major boiler losses</u>

*<u>1-loss due to sensible heat carried away by dry flue gas</u>*

$$\%\ heat\ lost\ due\ to\ dry\ flue\ gas = \frac{m * c_p * (T_f - T_a)}{GCV_{fuel}} * 100\%$$

$m$ = mass of dry flue gas in $kg/kg$ fuel

= $3.13 kg/kg$ fuel (calculated)

(m=combustion products from fuel: $CO_2$+ $SO_2$+Nitrogen in the actual mass of air supplied+ $O_2$ in flue gas. Water vapor in the flue gas is not considered.)

$c_p$ =Specific heat of flue gas= $0.23 Kcal/kg°C = 0.966 KJ/kg°C$

( i. e. $1 cal = 4.2 J$ )

$T_f =$ flue gas temperature (taken from control board)= $345.14°C = 618.14K$

$T_a =$ ambient air temperature = $25°C = 298K$

GCV of bagasse = $9314 KJ/kg$

After inserting the values obtained in the above equation

% heat loss due to dry flue gas

$$= \{[3.13 kg/kg * 0.966 KJ/kg°C * (345.14 - 25)°C] / [9314 KJ/kg]\} * 100\ \%$$

= 10.23%

*2-loss due to moisture in the fuel*

% heat loss due to moisture in fuel $= \dfrac{M * \{584 + c_p(T_f - T_a)\}}{GCV_{fuel}} * 100\%$

$M$ =Kilogram of moisture in kilogram of fuel(taken from data log book)

$= 0.5kg$

$c_p$ =Specific heat of superheated steam ( $0.45 Kcal / kg°C$ or $1.89 KJ / kg°C$ )

$*584$ is the latent heat corresponding to the partial pressure of water vapor.

% heat loss due to moisture in fuel

$= \{0.5kg * [584 + 1.89 KJ / kg°C * (345.14 - 25)°C] / [9314 KJ / kg]\} * 100\%$

= 6.33%

*3-loss due to $H_2$ in the fuel*

% heat loss due to $H_2$ in fuel $= \dfrac{9 * H_2 * \{584 + c_p(T_f - T_a)\}}{GCV_{fuel}} * 100\%$

$H_2 =$ $kg$ of $H_2$ in $1kg$ of fuel

$=$ $H_2$ in cellulose, hemicellulose and lignin

Mass of $H_2$ in $1kg$ of fuel $= 2 * \left\{ \sum \left[ \dfrac{M_{H_2,\ fiber}}{M_{fiber,\ component}} \right] * M_{fiber} \right\}$

$= 2 * \{[(0.18584 * 10) / 162] + [(8 * 0.153318) / 132] + [(13.9 * 0.102212) / 123.3]\}$

$= 0.0646 kg / kg$ of fuel

% heat loss due to $H_2$ in fuel

$= \{9 * 0.0646 kg * [584 + 1.89 KJ / kg°C (345.14 - 25)°C] / [9314 KJ / kg]\} * 100\%$

$= 7.36\%$

*<u>4-loss due to moisture in air</u>*

% heat loss due to moisture in air = $\dfrac{AAS * h.f * c_p\left(T_f - T_a\right)}{GCV_{fuel}} * 100\%$

Where,

$h.f$ = humidity factor for air taken as 0.14 (from psychrometric chart)

$AAS =$ the Actual mass of combustion air supplied to combust 1kg of the fuel, given by:

Actual mass of air supplied/kg of fuel $\left(AAS\right) = \left\{1 + EA/100\right\} * AIR_{theoretical}$

In which, excess air supplied $\left(EA\right) = \dfrac{O_2\%}{21 - O_2\%} * 100\%$,

Where, $O_2\% = 7.3\%$

$$EA = \frac{7.3}{21 - 7.3} * 100\% = 53.28\%$$

Theoretical air requirement

$$= \left\{\left[11.6 * C\right] + \left[34.8 * \left(H_2 - \frac{O_2}{8}\right)\right] + \left[4.35 * S\right]\right\} / 100 \; kg/kg \text{ of fuel}$$

Where, [ $C = 0.225$, $H_2 = 0.0646$, $O_2 = 0.3103$ , $S = 0.0164$ ] $kg/kg$ of fuel

Therefore, theoretical air requirement is

$$= \left\{\left[11.6 * 0.225\right] + \left[34.8 * \left(0.0646 - \left(\frac{0.3103}{8}\right)\right)\right] + \left[4.35 * 0.0164\right]\right\} / 100 \; kg/kg \text{ of fuel}$$

$= 3.58 \; kg/kg$ fuel

Now, the actual mass of air supplied $\left(AAS\right)/kg$ of fuel is

$$=\left\{1+\left(\frac{53.28}{100}\right)\right\}*3.58\ kg/kg\ \text{fuel}$$

$= 5.48\ kg/kg$ fuel

% heat loss due to moisture in air

$$= \left\{\left[5.48kg/kg*0.14*1.89KJ/kg{}^{\circ}C*\left(345.14-25\right){}^{\circ}C\right]/\left[9314KJ/kg\right]\right\}*100\%$$

$=4.9\%$

*5-heat loss due to unburnt fuel in fly ash*

% heat loss due to unburnt fuel in fly as = $\dfrac{Total\_ash/kg\_fuel*GCV_{ash}}{GCV_{fuel}}*100\%$

Where, mass of ash collected per kg of fuel= $0.0354\ kg/kg$ of fuel (as calculated)

$GCV_{fly\_ash} = 452.5Kcal/kg = 1900.5KJ/kg$ according to Canova [21]

%heat loss due to unburnt fuel in fly ash

$$= \left\{\left[0.0354kg/kg*1900.5KJ/kg\right]/\left[9314KJ/kg\right]\right\}*100\%$$

$= 0.72\%$

*6-loss due to radiation and convection from external surfaces of the boiler*

Loss due to radiation and convection is assumed to be $0.1\%$ of the energy input (Rein, 2007).

% loss due to radiation and convection= $0.1\%$

*7-loss due to incomplete chemical combustion*

Loss due to incomplete chemical combustion is assumed to be 2.51% of the energy input according to Rein .

% loss due to incomplete combustion=2.51%

Thus now we are in a position to calculate the combustion efficiency of the boiler by the indirect method.

$\eta_{boiler}$= 100-(i+ii+iii+iv+v+vi+vii)

$= 100\% - \{10.23 + 6.33 + 7.36 + 4.9 + 0.72 + 0.1 + 2.51\}\%$

= 67.85%

Therefore the efficiency of boiler through the direct and indirect method is 53.28% and 67.85% respectively. But the design point efficiency is higher than these, and it is 72%. Based on the evaluation the performance of the boiler is therefore not good.

**4.2.1-Pump-I Analysis**

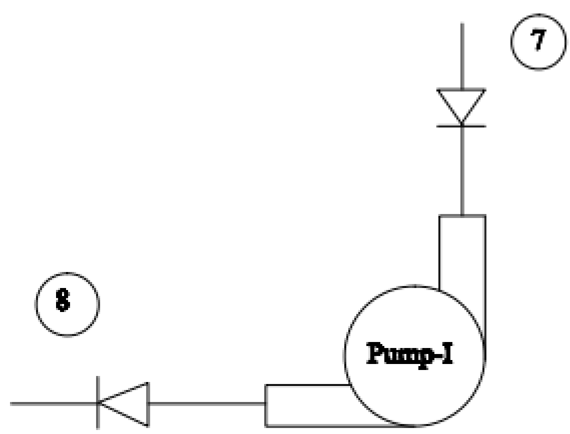


Figure **4.4**- Schematic representation of pump-I

**I-Pump-I efficiency determination based on the input/output or direct method**

$$Efficiency\ of\ pump-I = \frac{P_w}{P_s}$$

$Where, P_w = water\ power = QHf$ ,

$Q = Flow\ rate\left(m^3 / s\right)$

$H = Head\left(m\right)$

$f = 66.06 kg / m.s^2$

For the condensate return pump $\dot{m} = 4 kg / s$, taking density of water as $1000 kg / m^3$ ,

$Q = 0.004 m^3 / s$, and corresponding to this flow rate the value of the head H taken from the pump catalogue is : $H = 363.27 m$.

$P_s = shaft\ power\ provided\ to\ pump$

$P_s = 66.06 kg / m.s^2 * 0.004 m^3 / s * 363.27 m = 95.99 KW$

$P_s = 139.2 KW$

$Efficiency\ of\ pump - I = 99.59 KW / 139.2 KW = 0.6896 = 68.96\%$

**II-Efficiency determination of pump-I based on the energy balance or the indirect method**

The isentropic efficiency of pump-I is calculated as follows:

The process is adiabatic and incompressible, and the energy balance equation looks like

$$q - W_{P,I} = v_7 (P_8 - P_7) + (V^2{}_8 - V^2{}_7) / 2 + g(Z_8 - Z_7)$$

Based on the assumption the kinetic and potential energy terms are zero and adiabatic means '$q$' is also zero.

$$h_8 - h_7 = v_7 (P_8 - P_7)$$

$$0.0010074 * 10^3 m^3 / kg * \left(30 - 7\right) KPa = h_8 - 162.75 KJ / kg$$

Solving for $h_8$, we get

$$h_8 = 185.92 KJ / kg$$

Since state point 8' is in sub-cooled state, its properties are determined from saturated liquid line at the same saturation temperature and pressure.

Therefore,

$$h_{8'} = h_f @ T_{sat}(47.18°C) = 197.55 KJ/kg$$

$$\eta_{isentropic,\ pump\_I} = \left(\frac{h_8 - h_7}{h_{8'} - h_7}\right)$$

$$= \frac{(185.92 - 162.75)}{(197.55 - 162.75)}$$

$$= 0.6658 \ or \ 66.58\%$$

The efficiency of pump-I based on direct and indirect method is *68.89%* and *66.58%*. But the design point efficiency is higher then these and is *90%*. Based on the evaluation the performance of pump-I is therefore not good.

**4.2.2-Pump-II Analysis**

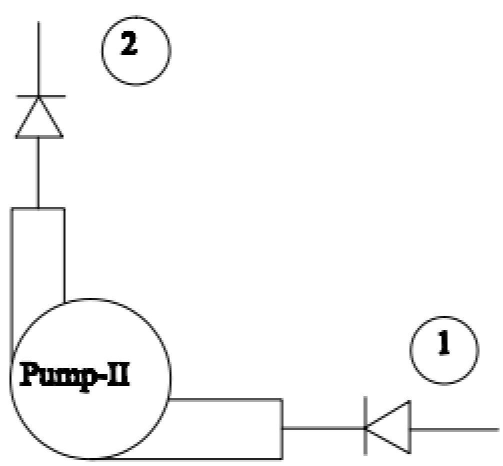


Figure **4.5**-Schematic representation of pump-II

**I- Efficiency determination of pump-II based on the input/output or direct method**

$$Efficiency\ of\ pump-II\ = \frac{P_w}{P_s}$$

$Where, P_w = water\ power = QHf$ ,

$Q = Flow\ rate\left(m^3/s\right)$

$H = Head\ (m)$

$f = 66.06 kg/m.s^2$

For the condensate return pump $\dot{m} = 17.364 kg/s$ ,taking density of water as $1000 kg/m^3$,

$Q = 0.01736 m^3/s$, and corresponding to this flow rate the value of the head H taken from the pump catalogue is : $H = 369.9m$.

$P_s = 66.06 kg/m.s^2 * 0.01736 m^3/s * 369.9m = 112.07 KW$

$P_s = shaft\ power\ provided\ to\ pump$

$P_s = 53.11 KW$

$Efficiency\ of\ pump-I\ = 112.07 KW / 153.11 KW\ = 0.732\ = 73.2\%$

**II- Efficiency determination of pump-II based on the energy balance or the indirect method**

The isentropic efficiency of pump-II is calculated in the same manner as that of pump-I.

The process is adiabatic and incompressible, and the energy balance equation looks like;

$$q - W_{P,II} = v_1(P_2 - P_1) + (V^2{}_2 - V^2{}_1)/2 + g(Z_2 - Z_1)$$

Based on the assumption the kinetic and potential energy terms are zero and adiabatic means '$q$' is zero.

$h_2 - h_1 = v_1(P_2 - P_1)$

$0.00107 * 10^3 m^3 / kg * (6.108 - 0.03) MPa = h_2 - 396.57 KJ / kg$

Solving for $h_2$,we get

$h_2 = 402.89 KJ / kg$

Since state point 2’ is in sub-cooled state ,its properties are determined from saturated liquid line at the same saturation temperature and pressure.

Therefore,

$h_{2'} = h_f @ T_{sat}(96.74°C) = 405.39 KJ / kg$

$$\eta_{isentropic,\ P-II} = \left(\frac{h_2 - h_1}{h_{2'} - h_1}\right)$$

$$= \left(\frac{402.89 - 396.57}{405.39 - 396.57}\right)$$

$= 0.7159$ *or* $71.59\%$

The efficiency of pump-II based on direct and indirect method is *73.2%* and *71.59%*. But the design point efficiency is higher then these and is *90%*. Based on the evaluation the performance of pump-II is therefore not good.

The overall heating supply can be written as

$\dot{Q}_{supply} = \dot{m}_{fuel} * LHV_{fuel}$

$= 7.24 kg / s * 2950 KJ / kg$

$= 21.35 MW$

The total steam produced is $62.5TPH$ or $17.63 kg / s$ and the assumption is $100\%$ steam flows to condensing steam turbine (CEST).

**4.3-Energy analysis of extraction condensing steam turbine**

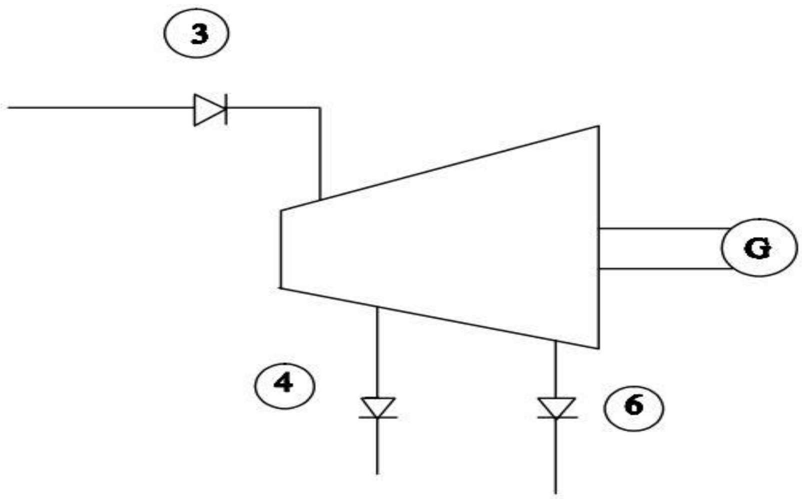


Figure **4.6** –Schematic representation of CEST turbine

**I- Efficiency determination of the turbine based on direct / input-output method**

1. Thermal power input

$\dot{E}_i = \dot{m}_3 \left( h_3 - h_{4'} \right)$

$= 17.36 kg / s * (3412.53 KJ / kg - 2743.7 KJ / kg)$

$= 11,610.88 KJ / s$

2. Thermal power output

$\dot{E}_o = \left( \dot{m}_3 - \dot{m}_4 - \dot{m}_9 \right) \left( h_{4'} - h_{6'} \right)$

$= (17.36 - 9.79 - 3.57) kg / s * (2743.7 - 2352.7) KJ / kg$

=1563.13*KJ/s*

3. Work done by turbine

$$\dot{W}_t = \dot{E}_i - \dot{E}_o$$

$$= 11,610.88 KJ/s - 1563.13 KJ/s$$

$$= 10,047.753.13 KJ/s$$

4.Efficiency of turbine

$$\eta_t = \frac{\dot{W}_t}{\dot{E}_i} * 100\%$$

$$= \frac{\dot{E}_i - \dot{E}_o}{\dot{E}_i} * 100\%$$

$$= \frac{11,610.88 KJ/s - 1563.13 KJ/s}{11,610.88 KJ/s}$$

$$= 0.8653$$

$$= 86.53\%$$

**II- Turbine efficiency determination based on energy balance or indirect method**

The isentropic efficiency of the turbine referring to figure 4.2 is given by:

Turbine isentropic efficiency= Actual power output/Ideal power output

$$= \frac{\dot{m}_3 (h_3 - h_{4'}) + \dot{m}_4 (h_{4'} - h_{6'})}{\dot{m}_3 (h_3 - h_4) + \dot{m}_4 (h_{4'} - h_6)}$$

$$= \frac{17.36(3412.53 - 2743.7) + 13.36(2743.7 - 2576.18)}{17.36(3412.53 - 2080.96) + 13.36(2743.7 - 2323.01)}$$

=0.7819 or78.19%

Overall cogeneration efficiency= $\frac{Enthalpy\ of\ steam\ across\ the\ expansion}{Enthalpy\ across\ heat\ supply}$

$$=\frac{\dot{m}_3\left(h_3-h_{4'}\right)+\dot{m}_4\left(h_{4'}-h_{6'}\right)}{\dot{m}_3\left(h_3-h_{2'}\right)}$$

$$=\frac{17.36(3412.53-2743.7)+13.36(2743.7-2576.18)}{17.36(3412.53-4055.39)}$$

$$=0.5632 \quad =56.32\%$$

## 4.4-Exergy analysis of condensing extraction steam turbine

Exergy is always evaluated with respect to a reference environment. The reference environment is in stable equilibrium, acts as an infinite system and is a sink or source for heat and materials. It experiences only internal reversible processes, in which its intensive properties (i.e. temperature $T_o$, pressure $P_o$) remains constant. In this analysis surrounding temperature and pressure are taken as $T_o$=25$^o$C (298 K) and $P_o$=84.93KPa .From saturated steam table at $T_o$=25$^o$C,and $P_o$=84.93KPa for an altitude of 1540m above sea level;

$$h_o = 303.208 KJ / kg.K$$
$$s_o = 1.712002 KJ / kg.K$$

1-Exergy input

$$\dot{\Psi}_{in} = \dot{m}_3\left\{\left(h_3-h_0\right)-T_0(s_3-s_0)\right\}$$

$$=17.36\left[(3412.53-303.208)-298*(6.83-1.712002)\right]$$

$$=27,056.81KW$$

2-Exergy out

$$\dot{\Psi}_{out} = \dot{m}_{4'}\left\{\left(h_{4'}-h_o\right)-T_o(s_{4'}-s_o)\right\}+\dot{m}_{6'}\{\left(h_{6'}-h_o\right)-T_o(s_{6'}-s_o)\}$$

$$=13.6*\left\{(2743.7-303.208)-298*(7.481-1.712)\right\}+4\left\{(2576.18-303.208)-298*(8.33632-1.712)\right\}$$

$$=16,456.73KW$$

3-Exergy efficiency ( 2nd-law efficiency) of cogeneration turbine

$$\eta_2 = \frac{\dot{\Psi}_{power}}{\left(\dot{\Psi}_{in} - \dot{\Psi}_{out}\right)}$$

$$= \frac{8054.83KW}{27,056KW - 16,456.73KW}$$

$$= 0.7598 \ = 75.98\%$$

Work done by the turbine is:

$$\dot{W}_{turbine} = \dot{m}_3 h_3 - \dot{m}_{4'} h_{4'} - \dot{m}_{6'} h_{6'}$$

$$= \left[(17.36 * 3412.53) - (13.36 * 2743.7) - (4 * 2576.18)\right] KJ / s$$

$$= 11,641.84 KJ / s$$

$$= 11.64 MW$$

Work input to the pump is:

$$\dot{W}_{pump} = \dot{m}_7 \left(h_{8'} - h_7\right) + \dot{m}_1 \left(h_{2'} - h_1\right)$$

$$= \left\{\left[4 * (197.55 - 162.75)\right] + \left[17.36 * (405.39 - 396.57)\right]\right\}$$

$$= 292.31 KW$$

The net work done by the turbine is therefor:

$$\dot{W}_{net} = \dot{W}_{turbine} - \dot{W}_{pump}$$

$= (11.64 - 0.2923)\ MW$

$= 11.35MW$

*Net electricity produced* = 11.35$MW$

**4.5-Process heater analysis**

The total steam produced is 17.3*kg/s* and the assumption is 100% of the steam flows to the extraction cum condensing steam turbine. Moreover it is known, as we go further through observation of the plant, that 56.22% of the steam outflow from the turbine i.e. 9.76*kg/s* is extracted to the process heater, 20.56% of the steam from the turbine i.e. 3.57*kg/s* is supplied as thermal load for end use consumers and the rest 23.22% of the steam outflow from the turbine i.e. 4*kg/s* is condensed in the condenser.

The process heat and the temperature at the out flow of the cold water being heated up can be expressed as:

$$\dot{Q}_{\text{process}} = \dot{m}_{4'} h_{4'} - \dot{m}_5 h_5$$

$Q_{process}$={(13.36*2743.7)-(9.79*540.75)}*KJ/s* =3.136$MW$

*Net heat to process*=3.136$MW$

The overall heating supply can be written as

$$\dot{Q}_{\text{supply}} = \dot{m}_{\text{fuel}} * LHV_{\text{fuel}}$$

$= 7.24kg / s * 2950KJ / kg$

= 24.35$MW$

*Total cogeneration efficiency= (Net electricity +Net heat to process)/heat supplied by fuel*

$\textit{Total cogeneration efficiency} = (11.35MW + 3.136MW) / 21.35MW = 67.85\%$

$\textit{Power generation efficiency} = \textit{Net electricity generated} / \textit{heat supplied by fuel}$

= 11.35$MW$/24.35$MW$ = 46.4%

## 4.6-Condenser analysis

The condenser consists of tubes of outside diameter 0.02134m with thickness 0.00277m. It is a shell- and tube design with double tube-passes cross flow arrangement. The length of one tube is 3m and its shell diameter is 2m. The number of tubes per pass is 36 and baffle spacing equal to 0.2m. The thermal conductivity of the tube material is 54 W/m-K. The inlet temperature of the circulation water holds the atmospheric temperature, $T_{amb}=25^oC$.

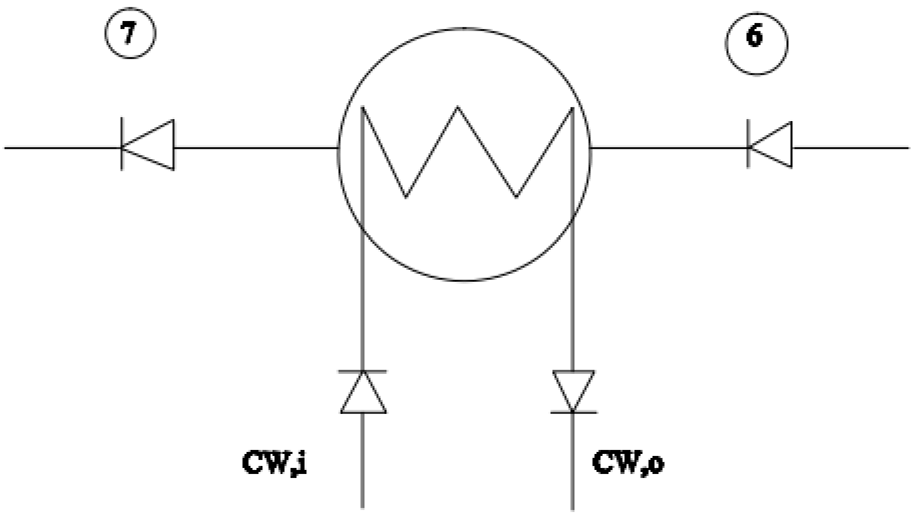


Figure **4.7**-Schematic representation of condenser.

The heat balance is as follows:

$$\dot{m}_7 h_7 - \dot{m}_6 \cdot h_6 \cdot = \dot{m}_{cw-2} * c_p * \left(T_{cw-2,o} - T_{cw-2.i}\right)$$

$$4kg/s\left(247.34KJ/kg - 162.75KJ/kg\right) = 8.64kg/s*4.2KJ/kg/K(T_{cw-2,o} - 298K)$$

$$T_{cw-2,o} = 307.32K = 34.32°C$$

The total heat transfer required is

$$\dot{Q} = 8.64kg/s*(4.2KJ/kg*K)*\left(307.32-298\right)K = 338.2KW$$

For counter flow heat exchanger

$$\text{LMTD}=\Delta T_m=\frac{\left(38.81^oC-32^oC\right)-\left(38.81^oC-25^oC\right)}{\ln\left[\dfrac{38.81^oC-32^oC}{38.81^oC-25^oC}\right]}=9.9^oC$$

$U=1419W/m^2*^oC$, for steam and water sides of shell and tube condenser.

$F=f\left(R,P\right)$, where $R$ and $P$ are empirical parameters

$$R=T_1-t_2/T_2-t_1=\left(38.81^oC-32^oC\right)/\left(38.81^oC-25^oC\right)=0.49$$

$$P=t_2-t_1/T_1-t_1=\left(32^oC-25^oC\right)/\left(38.81^oC-25^oC\right)=0.506$$

from chart of correction factor for $F$ taking the values of $P$ and $R$ we get $F=0.95$

$$\dot{Q}=UAF\Delta T_m$$

$$A=\dot{Q}/FU\Delta T_m=3.382*10^5W/\left(0.95\right)\left(1419W/m^2.°C\right)\left(9.9°C\right)=14.86m^2$$

**4.7-Feed water tank analysis**

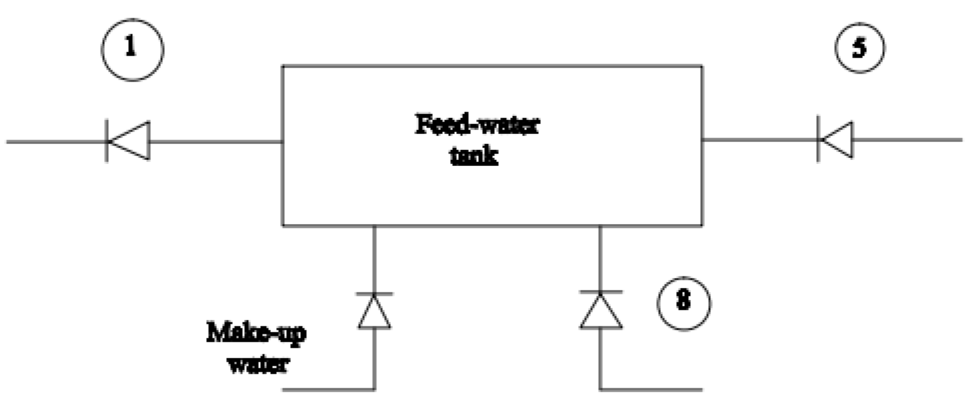


Figure **4.8**- Schematic representation of mixing chamber.

$$\dot{m}_5 h_5 + \dot{m}_8 h_8 + \dot{m}_{mw} h_{mw} = \dot{m}_1 h_1$$

$$\left( 9.79 kg/s * 540.75 KJ/kg \right) + \left( 4 kg/s * 240.49 KJ/kg \right) + 3.57 kg/s * h_{mw} = 17.36 kg/s * 396.57 KJ/kg$$

$$h_{mw} = 176.06 KJ/kg$$

$$T_{mw} = T_{sat} @ \; h_f = h_{mw} (176.06 KJ/kg)$$

$$T_{mw} = 42.02^o C$$

# CHAPTER-FIVE

## 5- Results and Discussion

### 5.1 Property diagram plot using EES

Engineering Equation Solver abbreviated as EES is a graphical interface mainly used in thermodynamics to plot property diagrams. The main inputs to this interface are the steam pressure, temperature and mass flow rate at each points in the cogeneration configuration. It has built-in steam tables and will automatically recognize a steam saturation and superheat point when given two properties like temperature and pressure.

The property diagram of temperature versus entropy is plotted using EES since it is an easier tool to plot diagrams.

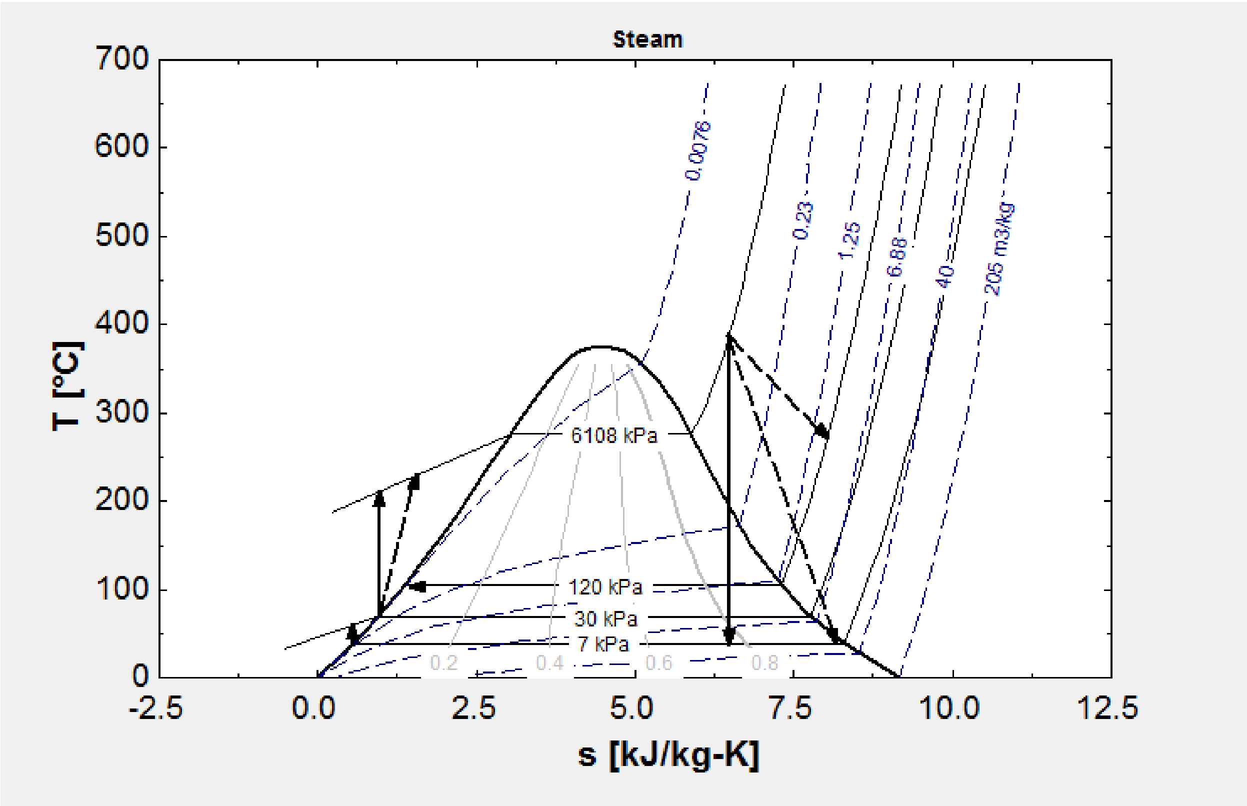


Figure **5.1**-Plot of T-s by EES software.

From the T-s diagram we can refer that after the combustion of bagasse in the bagasse fired boiler ,the water that enters to the boiler by boiler feed pump get heated and steam is formed at a pressure of 61.08 bar. This highly pressurized steam then goes to the condensing-extraction steam turbine to produce power and process steam. The steam after spinning the turbine blade about 78 % of the steam is extracted at 120kPa to the process heater. The rest 22% of the steam will go to shell and tube condenser at 7kPa.

At about 30kPa the feed water, the condensate pumped by condensate return pump and the saturated steam that leaves process heater mix in the mixing chamber. From the mixing chamber the water will be pumped by boiler feed pump-II to boiler entrance pressure. During mixing the saturated steam leaving the process heater will lose heat and consequently get its temperature reduced. The heat lost by the process steam will be equal to the heat gained by the feed water and the condensate. So in the T-s diagram it is clearly shown that the process steam at 120kPa will mix with the feed water at 30kPa and the condensate water at 7kPa. The resulting mixture of steam and water will reach a final pressure of 30kPa and from this pressure the water is pumped to boiler entrance pressure of 61.08bar.

### 5.2 Boiler losses and their representation

The Sankey or Band diagram shows the percentage fraction of the different boiler losses that account to the reduction of boiler combustion efficiency. The typical unburned combustible gases are $CO_2$, $SO_2$, $H_2$, $N_2$ and $O_2$.In large boilers usually CO can be found in significant amount in flue gases.

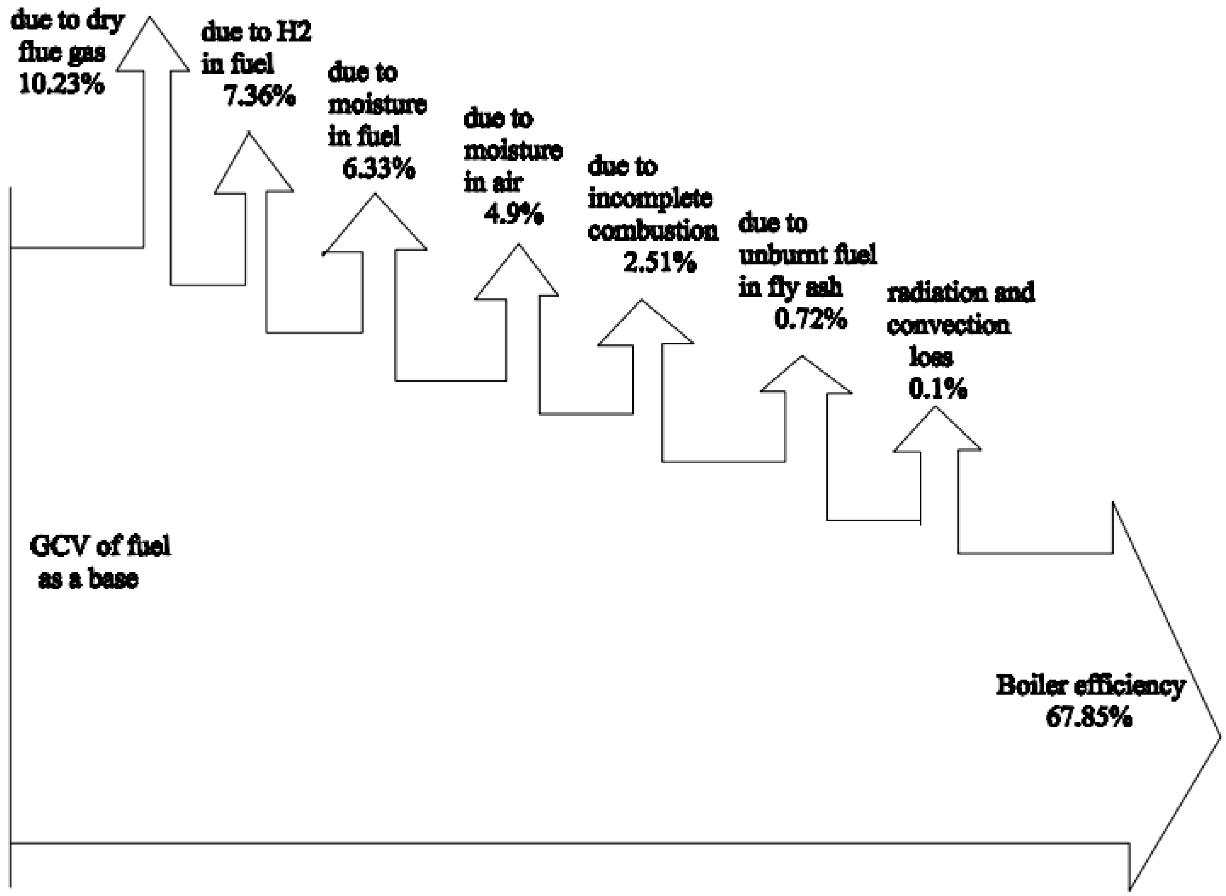


Figure **5.2**- Sankey diagram of the different boiler losses.

Unburned fuel can exit the furnace as bottom ash or fly ash. The heating value of the ashes can be measured in a specific laboratory test.

As can be seen from the result of the analysis the combustion efficiency of bagasse boiler is found to be 67.85% as compared to the rated or design boiler combustion efficiency of 72%. There is a 4.15% reduction in boiler combustion efficiency. This is mainly because of three factors:

1. The amount of moisture in bagasse will reduce the calorific value of the fuel. More over cane top and leaves and trashes are other options which have the potential of optimizing the amount of renewable energy that can be obtained from the sugar cane.
2. The level of excess air will affect the combustion efficiency.

3. The higher temperature of flue gas will affect the boiler efficiency as so much heat will be lost with it.

More over cane top and leaves and trashes are other options which have the potential of optimizing the amount of renewable energy that can be obtained from the sugar cane.

Flue gases leave the furnace in high temperature and thus they carry significant amount of energy from boiler process. To decrease flue gas losses, flue gas exit temperature should be decreased. However the acid dew-point of flue gases restricts the flue gas temperature to about 130-150$^{o}$C for sulfur containing fuels.

In recovery boilers the bottom ash is removed as molten ash in temperature of about 700-800$^{o}$C.The loss of sensible heat of ash is therefore of great importance in recovery boilers.

The main form of heat transfer from boiler to boiler room is radiation. It is proportional to the outer surface of the boiler and is usually 200-300W/m$^{2}$.K for a well insulated boiler having its outer temperature below 155$^{o}$C. Another possibility to determine the heat transfer losses is to use a table from the DIN 1942 standard. But in the case of bagasse fired boiler we assume a total radiation loss of 0.1% of the bagasse heating value as it is difficult to measure the exact value of the loss using long and approximate empirical relations.

**5.3 Condensate return and Boiler feed pump head curves**

A centrifugal pump performance curve is a tool that shows how a pump will perform in terms of head and flow. Pumps can generate high volume flow rates when pumping against low-pressure head or low volume flow rates when pumping against high-pressure head. The possible combinations of total pressure and volume flow rate for a specific pump can be plotted to create a pump curve. The curve defines the range of possible operating conditions for the pump. Pump curves plot data on a graph with x and y axes. The y-axis shows pump head in meters while the x-axis shows flow rate in m$^{3}$/s.

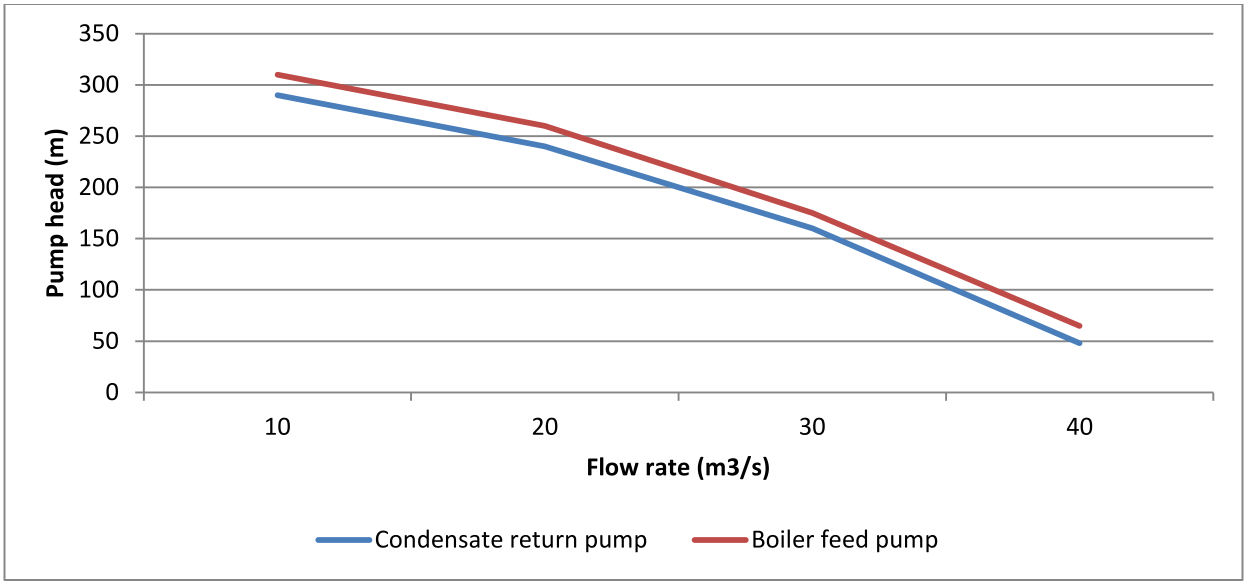


Figure **5.3**-Performance curve of boiler feed and condensate return pumps.

The rated efficiency of both the boiler feed-pump and the condensate return-pump is 92% and that obtained through analysis are 71.59% and 66.58% for the boiler feed-pump and the condensate return-pump respectively. From an efficiency and reliability point of view a centrifugal pump should be operated at or near its best efficiency point. Operating pumps at reduced capacities can cause various problems. Pumps will experience an excessive temperature rise and possible failure at some reduced flow rate if operated for an extended period of time. When the pump is run at a very low flow rate, the majority of the power input is converted to thermal energy, causing temperature rise. If the temperature rise continues, the liquid with in the pump can vaporize and cause thermal expansion of the internal parts. This could result in seizure of the rotating parts, complete pump failure or destruction of the pump.

Pump efficiency is the ratio of the liquid power delivered by the pump and the power delivered to the pump shaft. When selecting a pump, a key concern is optimizing pumping efficiency. It is good practice to examine several performance charts at different speeds to see if one model satisfies the requirements more efficiently than another. Whenever possible the lowest pump speed should be selected, as this will save wear and tear on the rotating parts.

## 5.4 Effect of varying temperature on energy and exergy efficiencies

The following figure presents the influences in increasing temperature on energy and exergy efficiency with firing fixed amount of bagasse at constant pressure 61.08bar and the temperature increases from 360$^{o}$Cto 460$^{o}$C.The consequences of increasing temperature provide the advantage in both energy and exergy efficiencies.

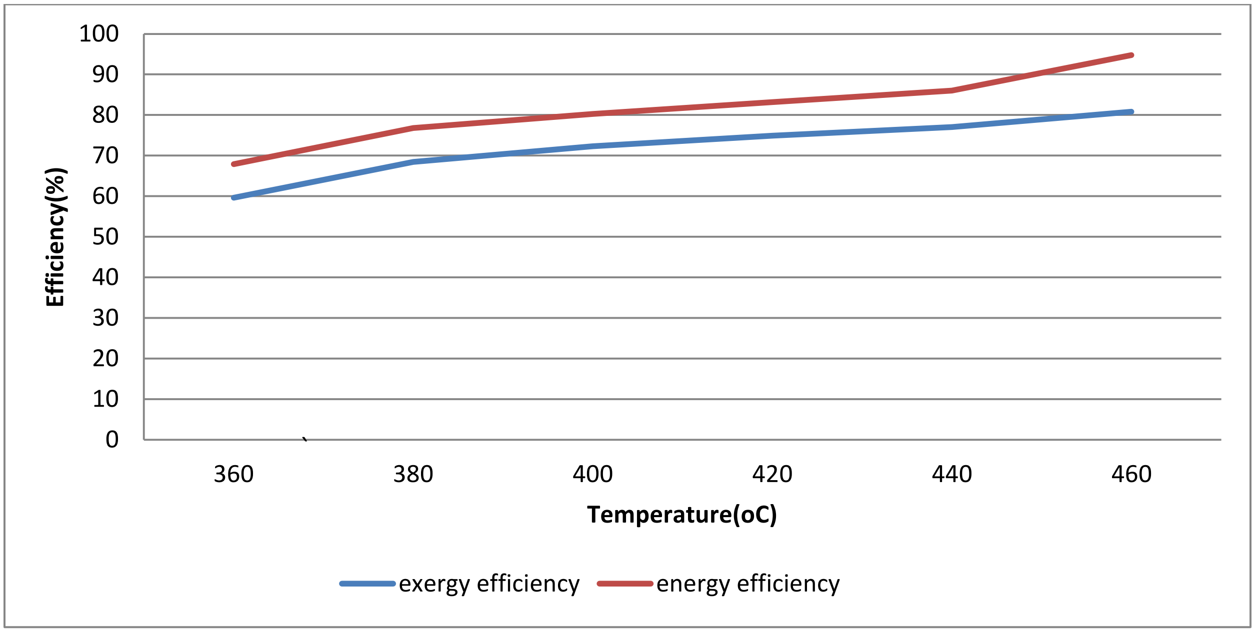


Figure **5.4**-Energy and exergy efficiencies of turbine versus turbine inlet temperature.

In particular, the energy efficiency is found to improve at maximum extent at 440$^{o}$Cand 61.08bar. After a specific temperature 380oC, the energy efficiency increase steadily at average 1.8% by every 20oC.

On the other hand, a similar result is also obtained from the current configuration in exergy efficiency; and is increasing steadily from 65% to 72% at average 1.1% increment.

According to the investigation, the exergy efficiency improves from higher temperature because the extent of increased exergy efficiency is steadily rising. However, the increased rate in energy efficiency is decreasing. The results draw attention to the more energy could lose as increasing temperature difference in the steam inlet condition even though the overall performance is improved.

**5.5 Studying the effects of boiler output temperature and pressure on overall plant's thermal efficiency**

Figure 5.5 shows the increasing and decreasing trend of overall efficiency with the temperature of boiler.

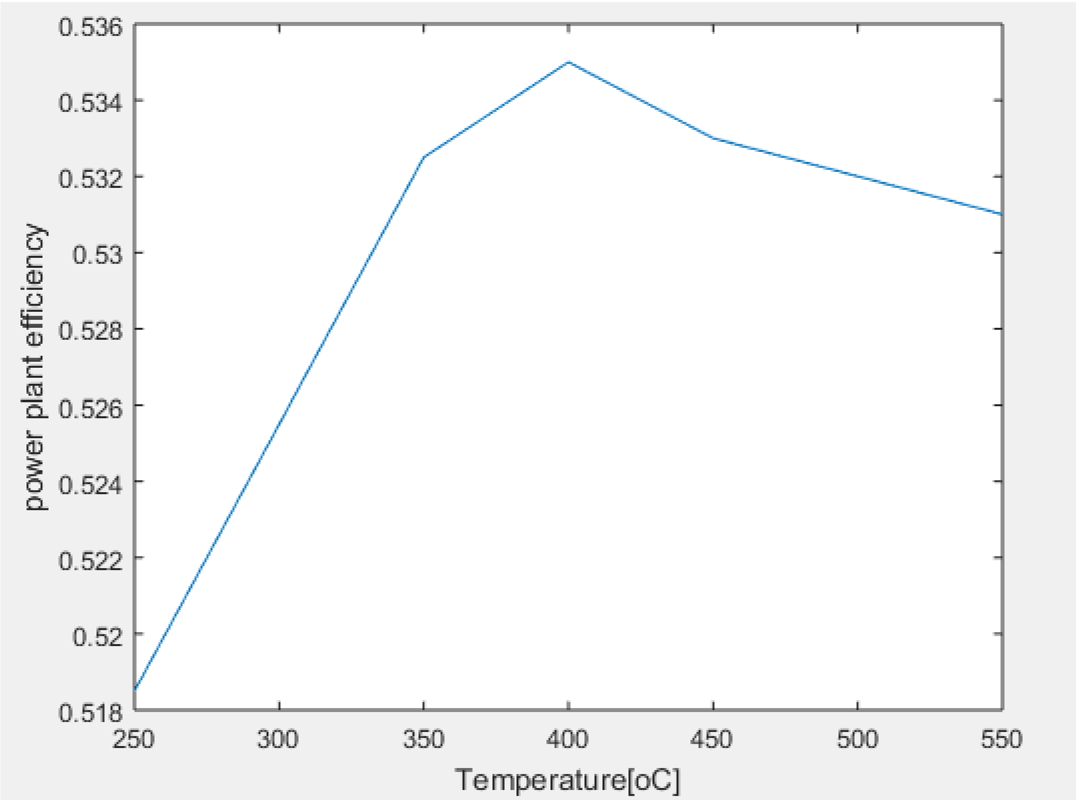


Figure **5.5**- Plot of overall thermal efficiency of the power plant versus outlet temperature of boiler.

If the temperature raises, the enthalpy of steam increases, the change of operation point leads to an increase of enthalpy difference across the expansion. Thus, the steam mass rate required by the turbines decreases and the steam mass rate through the 61.08to1.2 bar reduction station increases to fulfill the requirement of the process. If the temperature is reduced from the nominal one, no steam pressure reduction is necessary and the overall efficiency of the plant has a positive increasing trend with increasing temperature.

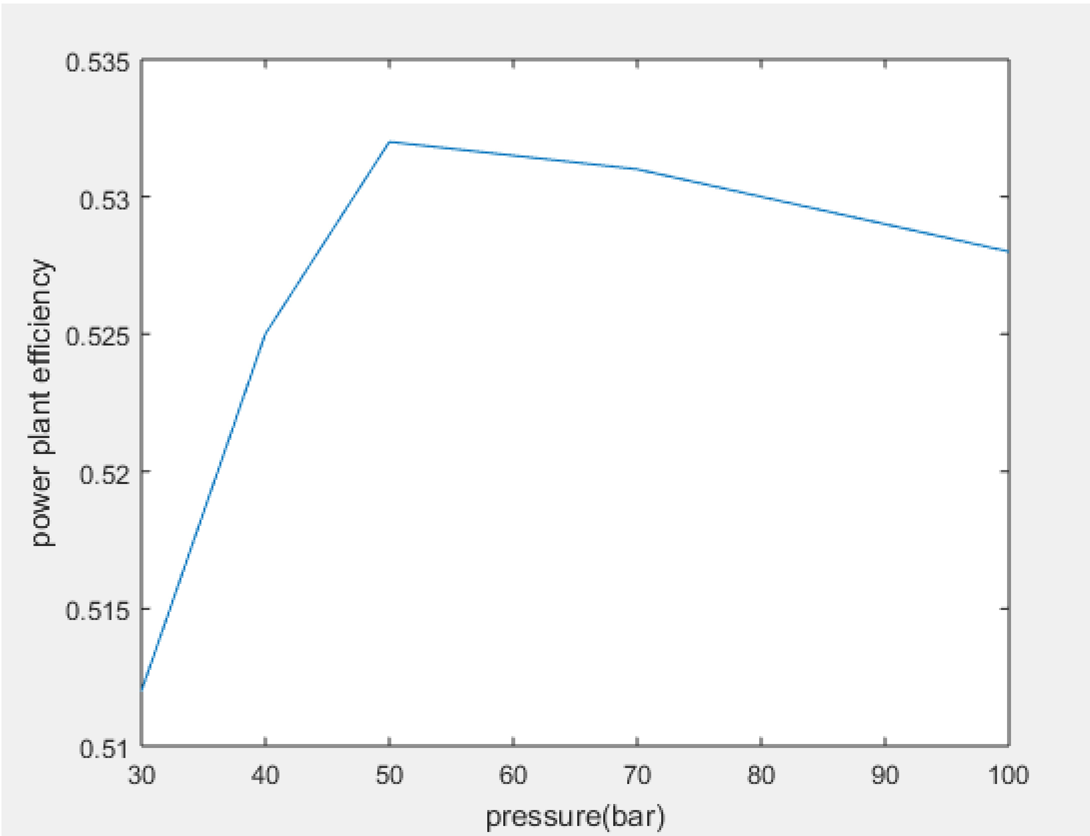


Figure **5.6**-Plot of overall thermal efficiency of the power plant versus operating pressure of boiler.

At low pressures, it appears that the overall efficiency of the power plant increases with pressure until reaching 50 bar (Figure 5.6), when pressure reduction of the steam starts to supply the process heat requirements. From this point on the efficiency decreases. Analyzing the thermodynamic aspect of the increase of boiler pressure while keeping the temperature constant implies a decrease of specific enthalpy of the 61.08bar head. The enthalpy of the turbine outlet decreases as well, and the result is an increase of the enthalpy drop in the expansion, with a consequent decrease of steam mass rate through the turbine. Consequently, the steam mass crossing the turbine increases, and the η decreases.

# CHAPTER-SIX

## 6-Conclusions

Studies show that the main use of bagasse is to provide a source of energy, and other applications are of secondary importance. In the beginning, bagasse was used only to provide heat to the sugar cane factories by burning it in boilers. However, improving technologies of CHP plant and the development of more efficient energy conversion technologies has allowed bagasse to provide not only energy for the factory, but also extra energy for the electricity grid.

Presented in this work is the first law of thermodynamics applied to determine bagasse boiler efficiency and the energy efficiency was evaluated using HHV as a calculation base. To determine the energy efficiency two methodologies were stated: input/output and energy balance. This last one has been proposed to calculate bagasse boilers, but it can be applied to others wet fuels boilers. In addition, it can be seen that the energy balance method gave more information about the bagasse boiler performance.

According to the research the efficiency of the boiler as calculated using the direct and indirect method are 53.28% and 67.85% respectively. It shows that these efficiencies are lower than the boiler's designed efficiency value, which is 72%.Therefore it is concluded that performance of the boiler is not good. Moreover the calculated values of the isentropic efficiency of the condensate return pump and the boiler feed pump based on direct and indirect method are 68.96% and 66.58% respectively for the condensate return pump and 73.2% and71.59% respectively for the boiler feed pump. It is shown that the isentropic efficiency of both pumps is lower than their designed efficiency value which is 90%.Therefore it is concluded that the performance of both pumps is not good.

The total cogeneration efficiency of the plant is found to be 67.85% and the power generation efficiency is found to be 53.16%. The overall cogeneration efficiency of the cogeneration power plant has a design limit of 80% and it is concluded that the power plant is performing at reduced overall cogeneration efficiency.

## 6.1-Recommendation

Bagasse, the fuel used in the sugar industry has a very high moisture content of 50%. Reducing the moisture content of the product improves its calorific value, and resulting in the overall improvement of plant efficiency. Various methods of moisture reduction are reported in the literature (Upadhiaya 1991),

1. Mechanical methods
2. Use of hot imbibition water
3. Chemical methods
4. Drying of bagasse

Other suggestions to improve boiler performance are:

- The amount of moisture in bagasse must be reduced using bagasse dryer to increase the calorific value of the fuel.

-More over cane top and leaves and trashes are other options which have the potential of optimizing the amount of renewable energy that can be obtained from the sugar cane.

-The incompleteness of combustion must be avoided as it will reduce boiler performance.

-The level of excess air must be as required as it will affect the combustion efficiency.

-The higher temperature of flue gas will affect the boiler efficiency as so much heat will be lost with it so that there must be some mechanism to recover this waste heat and to put it in to effect.

## REFERENCES


[1] Anderzej.W. , Michel.J, "Combined cycle and combined heat and power"In Control system, Robotics and Automation,volume 18, pages 305–308, New York, NY, 2002

[2] Branch, UNEP, —Technical study report on biomass fired fluidized bed combustion boiler technology. http://www.uneptie.org/energy .

[3] Cengel ., Boles.,"Thermodynamics An Engineering Approach" 4th Edition .

[4] Devki Energy Consultancy, Ltd. —Best practice manual cogeneration.India 2006. http://www.energymanagertraining.com.

[5] Derya.B.O., Onur .K., Duriye.B.," Exergy Analysis of a Cogeneration Plant",pages 154-169,2002.

[6] Gomez.M, Sebastián.F, Royo.J, —Co-firing versus biomass-fired power plants: GHG (Greenhouse Gases) emissions savings comparison by means of LCA (Life Cycle Assessment) methodology. 2011 Energy, pp. 229-237

[7] Head of Publications Service, —Greenhouse emission. Internal Energy OECO, 1999.

[8] John.A, Martin.S, —Co-generation application considerations. 2009. http://site.greenenrgy.com/prod_serv/products/tech_docs/en/downloads/GER3430G.pdf.

[9] Jayen,Veerapen, Milou ,Beerepoot,—Co-generation and renewables solution for low carbon energy future International Energy Agency

[10] McNish.T, Jacobson.A, Kammen.D.M, Gopal.A. Deshmukh.R, —Sweet carbon: An analysis of sugar industry carbon market opportunities under the clean development mechanism. Energy Policy 2009 , pp. 359-368.

[11] Power Generation Handbook, —Digital engineering library. McGraw -Hill 2004. http://www.digitalengineeringlibrary.com

[12] Mujeebu.A, Abdullah.Z, Ashok.S.A, ―Husk-fueled steam turbine cogeneration for a rice mill with power export—a case study energy sources. Part A: Recovery, Utilization, and Environmental Effects 2011, pp. 724-734

[13] Prachuab., Budit., "Exergetic and thermoeconomic analysis of rise-husk power plant in Thailand " pages 271-283,2000.

[14] Premalatha, Priya.M, Shanmuga.S, ―Efficient Cogeneration scheme for sugar industry .Journal of Science & Industrial Research 2008 ,pp. 239-242.

[15] Rosen.A, ―Reductions in energy use and environmental emissions achievable with utility-based cogeneration Simplified illustrations for Ontario. Applied Energy, 1998 , pp163 - 174 .

[16] Rosen C.,Aida.B., "Using exergy to understand and improve the efficiency of electrical power technology" 2009, pp. 220-235

[17] Rueddy B.V, Cliff Butcher, ―Second law of a natural gas-fired gas turbine cogeneration system. International Journal of Energy Research 2009, pp.128-136

[18] Thermal Energy Equipment Cogeneration, ―Energy efficiency guide for industry in Asia. http://www.energyefficiencyasia.org.

[19] Wang .Y, Dai.L, Gao.J, ―"Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry". Applied Energy 2009 86 (6), pp. 941-948.

[20] Zanova .A , ―Emission characterization and evaluation of natural gas-fueled cogeneration micro turbines and internal combustion engines. Energy Conversion and Management 2008, pp. 200-209

**<u>Appendix-EES model of the cogeneration configuration</u>**

```
// Bagasse-Fuelled Steam Turbine Cogeneration Configuration Modeling example

"Reference Condition"

T[0]=25 [C]

P[0]=101.31[Kpa]

h[0]=Enthalpy(Steam,T=T[0],P=P[0])

s[0]=Entropy(Steam,T=T[0],P=P[0])

EX_0=0

T_0=273+T[0]

“Pump Isentropic turbine efficiency 85%”

“State-1”

M_1=M_Steam                                  " 100% of total steam flow rate "

T[1]=69.02[C]

P[1]=30[KPa]

h[1]=Enthalpy(Steam,T=T[1],P=P[1])           "Determine enthalpy"

s[1]=Entropy(Steam,T=T[1],P=P[1])            "Determine entropy"

“State-2”

M_2actual=M_Steam

[2]=6108[Kpa]

T[2]=96.74[C]

h2_actual= Enthalpy(Water,T=T[2],)           "Determine Actual enthalpy “
```

h[2]=h[1]+0.85*( h2_actual-h[1]) "Theoretical enthalpy "

“State-3”

M_3= M_Steam

T[3]=466.67[C]

P[3]=P[2] "Same as state 2"

h[3]=Enthalpy(Steam,T=T[3],P=P[3]) "Determine enthalpy"

s[3]=Entropy(Steam,T=T[3],P=P[3])

“State-4”

M_4= 0.78*M_Steam

T4_actual=136.89[C]

P[4]=120[KPa]

s[4]=s[3]

x[4]=0.9097 "Saturated liquid"

h[4]=Enthalpy(Water,P=P[4],x=x[4]) "Determine enthalpy"

v[4]=Volume(Water,P=P[4],x=x[4]) "Determine specific volume"

h4_actual= Enthalpy(Steam,P=P[4],T=T4_actual])

s4_actual=Entropy(Steam,T=T4_actual,P=P[4]) "Determine Actual entropy"

“State-5”

M_5= 0.564* M_Steam

P[5]=P[4]

T[5]=104.7[C]

h[5]=Enthalpy(Water,P=P[5],x=0)

“State-6”

M_6=0.22*M_Steam "22 % extraction from CEST to condenser"

P[6]=7[Kpa]

s[6]=s[3]

x[6]=0.8036 "Saturated liquid"

h[6]=Enthalpy(Water,P=P[6],x=x[6]) "Determine enthalpy"

v[6]=Volume(Water,P=P[6],x=x[6]) "Determine specific volume"

h6_actual=Enthalpy(Water,P=P[6],x=1)

"State 7"

M_7=M_6

P[7]=7[Kpa]

x[7]=0 "Saturated liquid"

h[7]=Enthalpy(Water,P=P[7],x=x[7]) "Determine enthalpy"

v[7]=Volume(Water,P=P[7],x=x[7]) "Determine specific volume"

s[7]=Entropy(Water,P=P[7],x=x[7]) "Determine entropy"

T[7]=Temperature(Water,P=P[4],h=h[7]) "Determine Temperature"

"State 8"

M_8=M_6

```
P[8]=30[Kpa]

h_8actual=h[7]+v[7]*(P_7actual-P[7])                "Determine enthalpy via boundary work"

s[8]= s[7]                                          "Determine entropy"

T_8actual=47.18[C]                                  "Determine temperature"

//-------------------------------------Condensing ST work output-----------------------------------------
W_CEST=M_3*h[3]-M_4*h4_actual-M_6*h6_actual

//----------------------------------Energy Analysis-----------------------------------------------------
"Process Heat"


Q_boiler=M_3*h3_actual+M_2*h2_actual-               "Boiler"

Q_condenser=M_6*( h6_actual- h[7] )                 "Condenser"

—

"Pump1"

W_pump1=M_1*( h2_actual -h[1])

"Pump2"

W_pump2=M_7*( h8_actual -h[7])
```