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Performance evaluation of a combined heat and power plant in an Indian sugar industry: A case study.

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Abstract

In this paper, a thermodynamic analysis of a combined heat and power plant is performed for a 20.70 MW bagasse fired power plant. Design data of a uniquely designed plant is used, which is located at Chikodi, India. It is integrated with sugar and ethanol production process to meet internal thermal and electric energy demand, whereas the surplus is exported. Energy and exergy formulations are developed for different components of the plant and also the combined heat and power system as a whole. A parametric study based on energy and exergy parameters is included. Energy analysis reveals that a lot of heat is lost through condenser whereas exergy analysis indicates that largest exergy destructions take place in the boiler. The combined heat and power system is found to be less sustainable at higher ambient temperatures in terms of exergy. A detailed discussion is provided, which facilitates and throws light on further improvements to be made in the present plant conditions. It is believed that the present analysis would be helpful for practicing engineers and designers in the field of sugar and power engineering.

Keywords: *energy analysis, exergy analysis, combined heat and power, cogeneration, sugar, bagasse, biomass.*

INTRODUCTION

India is a developing country and its energy market is one of the country's fastest developing sectors. Currently most of the energy demand in the country is met by using fossil fuels (coal, petroleum, natural gas). These conventional methods of power generation are dependent on non-renewable energy resources, which are depleting. They are disadvantageous due to their environmental impact. In view of this renewable energy sources offer environment-friendly processes of power generation and utilization. Moreover, increasing political and environmental pressures have prompted the development and utilization of renewable energy sources. Biomass is the second largest source of renewable energy [1]. There is a wide range of biomass resource available from agriculture, forestry, industrial and municipal waste etc. Biomass utilization is been practiced by numerous industries such as sugar industry, rice industry, paper and pulp, wood industry for many years as a waste disposal and energy recovery. Usually biomass based heat and power generation systems are utilized for energy requirements in the production processes of the industry.

In view of this combined heat and power (CHP) systems are found to be more advantageous, since they simultaneously generate power and process heat. It involves the production of both thermal energy, generally in the form of steam or hot water and electricity [2]. The efficiency of energy production can be increased from current levels that vary from 35 to 55% in the conventional power plants, to over 80% in CHP systems [3]. The present scope of work concentrates on sugar industry, which utilizes bagasse (a waste product) as fuel, in generation of power and process heat. Sugarcane production is one of the most important economical activities in India. Currently there are 500 to 600 sugar mills operating in India [4]. Now-a-days most of the sugar mills are not only self-sufficient in terms of energy requirements, but are also exporting power to the grid. In recent years, the complexity of power generating units has increased considerably. The general supply and environment situation requires an improved utilization of energy systems. Plant owners are increasingly demanding a strictly guaranteed performance. In this context, exergy analysis is found to be more useful than energy analysis. An energy analysis, which is based on first law of thermodynamics, has some limitations since it does not characterize the irreversibility of processes within the system. Exergy analysis, which is based on second law of thermodynamics, provides meaningful assessment of plant components in terms of efficiency. It helps in finding the locations, magnitudes and causes for losses in any energy system [5]. Previous literature reveals that a lot of research is carried out regarding exergy analysis of CHP systems. There are a range of technologies that can be used to cogenerate electricity and useful thermal energy. These technologies are generally classified according to the prime movers employed (i.e. gas turbines, diesel engines, steam turbines). The following section deals with the literature available regarding exergy analysis of CHP systems.

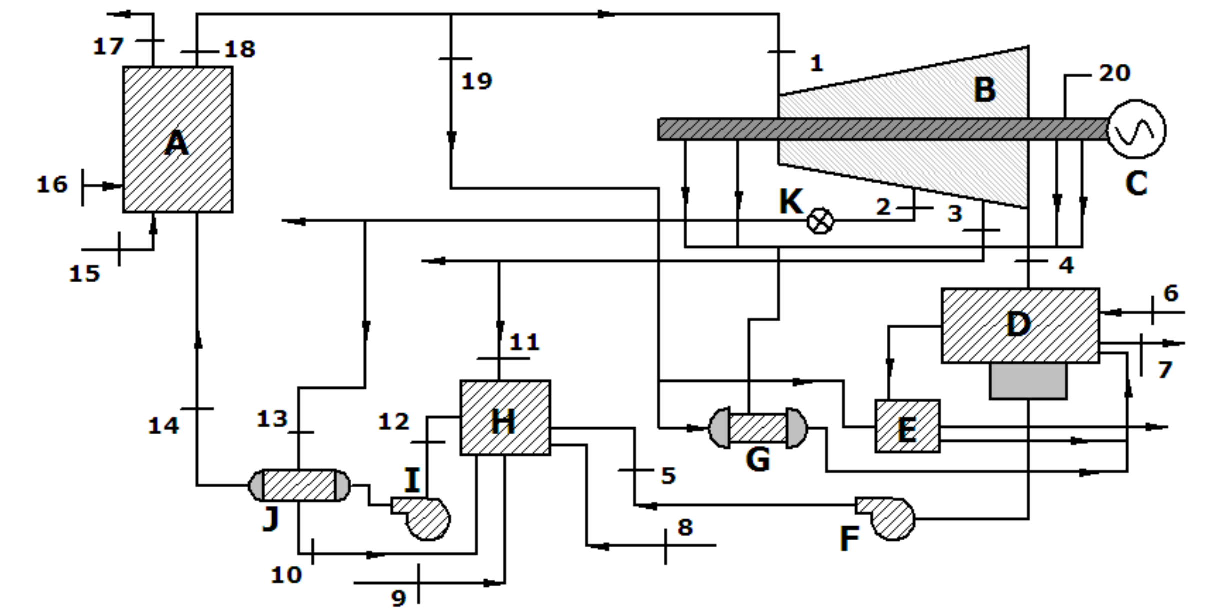
LITERATURE REVIEW

F. F. Huang [6], examined three systems using state-of-the-art industrial gas turbines based on first law and second law. It is found that first law is inadequate and second law analysis is inevitable. E. Bilgen [2] analyzed gas turbine based cogeneration system. The components considered for simulation of the results are gas turbine, heat recovery steam generator (HRSG). To simulate these systems, an algorithm has been developed. HRSG is found to be the least efficient component from exergy point of view. Ozgur Balli and Haydar Aras [7] conducted exergy analysis on micro-gas turbine (MGTCHP) driven combined heat and power system and found the energy and exergy efficiencies to be as 75.99% and 35.80% respectively. Yilmaz Yoru et al [8] performed exergy analysis of gas turbine based CHP system in a ceramic factory. The mean energetic and exergetic efficiency are found to be 82.30% and 34.70% respectively. Denilson Boschiero do and Espirito Santo [9] conducted exergy analysis of Internal combustion engine based cogeneration system. Thermodynamic analysis revealed that under the present plant conditions it is better to produce hot water than chilled water. Ayesogul Abusoglu and Mehmet Kanoglu [10-12] performed exergy analysis of diesel engine based CHP system. Exergetically waste heat boiler is found to be the least efficient component and major exergy destructions occurred in the diesel engine. Ozgur Balli and co-researchers [13] performed exergy analysis of steam turbine driven CHP system. The exergy efficiency of CHP system is found to be 38.16% with combustion chamber as the main contributor of exergy destructions. S.C. Kamate and P.B. Gangawati conducted [14] exergy analysis of cogeneration plants in sugar industries for BPST (back pressure steam turbine) and CEST (Condensate extraction steam turbine) systems in 2500 TCD (tonnes of cane crushed per day) sugar plant. It is proved that by increasing the steam pressure and temperature the exergy destructions decreased in the boiler, but on the contrary the exergy destructions increased in the turbine. Mehmet Kanoglu and Ibrahim Dincer [15] conducted performance assessment of gas turbine, diesel engine and steam turbine based CHP systems. It is found that the exergy efficiency of the diesel engine based CHP system is the highest followed by steam turbine and gas turbine based CHP systems.

Unlike past literature available the present work focuses on the performance evaluation of a custom made, steam turbine driven CHP system employed in a sugar factory. It is based on regenerative Rankine cycle using extraction-cum-condensing steam turbine (CEST) as prime mover with an output of 20.7 MW. The boiler is of travelling grate type, designed to fire moist bagasse (usually with 50% moisture content) with a capacity of 125 TPH (tons per hour) at 100% BMCR (boiler maximum continuous rating). This CHP plant is integrated with a sugar and ethanol plant in order to suffice the need of process heat and power, while the surplus is exported to grid. The main objective of this case study is to perform first law and second law analysis, using the design data of the actual CHP plant. A comparison of first and second law analysis is shown. The present study will demonstrate how second law (exergy) analysis helps in identifying major sources of losses and its magnitudes. It will provide some inputs to increase system efficiency and discuss the effect of system operation on the environment. A parametric study on how the system behaves under different operating conditions is carried out. The following sections deal with plant description and the analysis procedure employed.

Process description

A systematic layout of the combined heat and power (CHP) system is as shown in the figure 1.



A - boiler; B -- turbine; C -- generator; D -- condenser; E -- steam jet air ejector (SJAE); F-- condensate extraction pump (CEP); G -- gland steam condenser (GSC); H -- de-aerator; I -- boiler feed pump (BFP); J -- high pressure heater (HPH); K -- pressure reducing valve (PRV)

Fig 1: *Layout of the Combined Heat and Power (CHP) plant system.*

The main components of the plant are boiler, steam turbine, surface condenser, de-aerator, high pressure heater (HPH), boiler feed pump (BFP), condensate extraction pump (CEP), gland steam condenser (GSC), steam jet air ejector (SJAE). This plant is integrated with a sugar mill and distillery plant which suffices the need of thermal and electrical energy demands, whereas the surplus is exported to the grid. The bagasse a waste product from the sugar mill is used as fuel in the boiler. The sugar mill has a capacity of 5500 TCD (tones of cane crushed per day). Bagasse is fed to the boiler with an average moisture content of 50 wt%. The boiler is of travelling grate type with combustion air supplied by FD (forced draft) and SA (secondary air) fans. Steam generated in the boiler is fed to the turbine.

The turbine is of impulse-reaction type, 20.70 MW capacities, speed 6300 rpm, and condensing-extraction type with 2 extraction points for process steam as shown in the figure 1. The first extraction is at state point 2, used for distillery and HPH. Here the extracted steam pressure is at 16 bars (abs) which is reduced to 9 bars (abs) by using a PRV (pressure reducing valve).The second extraction is at state point 3 where steam is bled at 3 bars (abs) pressure for de-aeration and sugar mill requirements. The remainder is sent to exhaust which in turn goes to condenser. The surface condenser is shell and tube type maintained at a vacuum condition of 0.095 bars and the condensate temperature is 45^0C. To maintain vacuum condition steam jet air ejector (SJAE) is used which draws steam from state point 19. The turbine gland sealing arrangements are maintained in sub-atmospheric conditions by using gland steam condenser. The steam here is utilized again from state point 19. The condensates of GSC and SJAE are sent to surface condenser. The condensate from the surface condenser is sent to de-aerator which facilitates in removing the insoluble. The steam is cooled in the condenser using water from forced circulation cooling tower.

The required makeup water is supplied to the de-aerator from DM (demineralization) plant. The de-aerator is maintained at 1.5 bars (abs) pressure using steam from second extraction i.e. state point 3. The second extraction at state point 3 supplies steam to the sugar evaporation process and return condensate is fed back to the de-aerator at state point 9. A temperature of 105^0C is maintained for the condensate at the outlet of de-aerator. The temperature of the condensate is increased from 105^0C to 160^0C using high pressure heater, with heating media being supplied from first extraction. Thus the feed water is supplied to the economizer at 160^0C. The pressure, temperature and mass flow rate at different state points are detailed in table 2. The boiler consists of super-heater, economizer and air-heater due to which the exhaust flue gas temperature at the stack is reduced to 155^0C. The air pre-heater supplies air to the boiler at 180^0C. The boiler and other main parameters of the plant are shown in table 1. The turbine generates 6.5 MW in island mode of operation and the surplus is exportable. The following section deals with the analysis procedure employed.

4.0) Analysis procedure: The main process parameters of the plant are shown in table 1. This thermodynamic analysis would consider balances of mass, energy, exergy, exergy ratios, energy efficiency and exergy efficiency.

***Table 1:** Main Plant parameters at 100% BMCR and 100% TMCR condition.*

Sl no	Particulars	value
1	Bagasse consumption rate, kg s^{-1}	14.25
2	Steam/ bagasse ratio	2.43
3	Main Steam pressure (bar), temperature (^{0}C)	64 , 495
4	Total evaporation rate, kg s^{-1}	34.72
5	Feed water temperature, ^{0}C	160
6	Stack gas temperature, ^{0}C	155
7	Power output, kW	20700
8	Condenser pressure, bar	0.095

9	Condensate temperature, ^{0}C	45
10	Cooling water flow rate, kg s^{-1}	805.37
11	Cooling water temperature rise, ^{0}C	10
12	process heat (kW); E_2+E_3	58718.89
13	Power to heat ratio (PHR)	0.3525
14	Generator efficiency ,%	98
15	Turbine isentropic efficiency,%	95

Mass balance
For a steady state process, the mass balance for a control volume is given as,

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \text{ ……………………… (1)}$$

m indicates mass flow rate; suffix 'in' and 'out' indicate inlet and outlet respectively.

4.2) Energy balance: The energy balance for a control volume is given by,

$$\sum \dot{E}_{in} + \dot{Q} = \sum \dot{E}_{out} + \dot{W} \text{ …………………… (2)}$$

Applying the above energy balance equation for the different components of the CHP system, the heat loss in the component can be calculated. The following are the energy balance equations for the components considered.

For Boiler: $\dot{E}_{loss,boi} = \dot{E}_{14} + \dot{E}_{15} + \dot{E}_{16} - \dot{E}_{17} - \dot{E}_{18}$ ……………………… (3)

For turbine: $\dot{E}_{loss,tur} = \dot{E}_1 - \dot{E}_2 - \dot{E}_3 - \dot{E}_4 - \dot{W}_T$ …………………….. (4)

For condenser: $\dot{E}_{loss,cond} = \dot{E}_4 - \dot{E}_5$ ……………………… (5)

For high pressure heater: $\dot{E}_{loss,HPH} = \dot{E}_{12} + \dot{E}_{13} - \dot{E}_{10} - \dot{E}_{14}$ …………………… (6)

For De-aerator: $\dot{E}_{loss,DE} = \dot{E}_5 + \dot{E}_8 + \dot{E}_9 + \dot{E}_{10} + \dot{E}_{11} - \dot{E}_{12}$ …………………… (7)

The specific physical/flow energy for air and combustion gases with constant specific heat may be written as [7],

$$e_{ph} = C_{p(T)}T - C_{p(T_0)}T_0 = h_{(T)} - h_0 \text{ .... (8)}$$

Equation 8 is utilized to calculate the specific energy content of air/ combustion gas at state points 15 and 17.

4.3) Exergy balance: The general exergy balance equation is given by,

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} = \sum \dot{Ex}_{dest} \text{ ……………. (9)}$$

For a control volume at steady state exergy rate is given by,

$$\sum \left(1 - \frac{T_0}{T}\right)\dot{Q} - \dot{W} + \sum \dot{m}_{in}\, ex_{in} - \sum \dot{m}_{out}\, ex_{out} = \dot{Ex}_{dest} \text{ ……………. (10)}$$

Applying the above exergy balance equation the exergy destruction equations in the components considered is as given below.

For boiler: $\dot{Ex}_{dest,boi} = \dot{Ex}_{14} + \dot{Ex}_{15} + \dot{Ex}_{16} - \dot{Ex}_{17} - \dot{Ex}_{18}$ …………………………… (11)

For turbine: $\dot{Ex}_{dest,tur} = \dot{Ex}_1 - \dot{Ex}_2 - \dot{Ex}_3 - \dot{Ex}_4 - \dot{W}_G$ ……………………………… (12)

For condenser: the exergy destruction taking place in the condenser is given by [16]

$$\dot{E}x_{dest,cond} = \dot{m}_v \left\{ c_{p,v} \left[(T_{v1} - T_{cond}) - T_o \ln\left(\frac{T_{v1}}{T_{cond}}\right) \right] + h_{fg/T=T_{cond}} - T_0 s_{fg/T=T_{cond}} \right\} - \dot{m}_c c_{p,c} \left[(T_{c2} - T_{c1}) - T_0 \ln\left(\frac{T_{c2}}{T_{c1}}\right) \right]$$

…………..(13)

In the present context the modified form of exergy destruction equation for condenser is,

$$\dot{E}x_{dest,cond} = \dot{m}_4 \left\{ c_4 \left[(T_4 - T_5) - T_o \ln\left(\frac{T_4}{T_5}\right) \right] + h_{fg/T=T_5} - T_0 s_{fg/T=T_5} \right\} - \dot{m}_6 c_6 \left[(T_7 - T_6) - T_0 \ln\left(\frac{T_7}{T_6}\right) \right] \quad \ldots(14)$$

For high pressure heater: $\dot{E}x_{dest,HPH} = \dot{E}x_{12} + \dot{E}x_{13} - \dot{E}x_{14} - \dot{E}x_{10}$ ……………….. (15)

For De-aerator: $\dot{E}x_{dest,DE} = \dot{E}x_5 + \dot{E}x_8 + \dot{E}x_9 + \dot{E}x_{10} + \dot{E}x_{11} - \dot{E}x_{12}$ …………….. (16)

The total exergy destruction in the CHP system at the component level is given by,

$$\dot{E}x_{dest,tot} = \dot{E}x_{dest,boi} + \dot{E}x_{dest,tur} + \dot{E}x_{dest,cond} + \dot{E}x_{dest,HPH} + \dot{E}x_{dest,DE} \quad \ldots\ldots\ldots\ldots\ldots\ldots (17)$$

4.4) Specific exergy: The specific exergy for steam is as below,

$ex = (h - h_o) - T_o(s - s_o)$ ………………… (18)

where h,s and T are enthalpy, entropy and temperature respectively. Suffix 'o' indicate reference conditions.

But for an incompressible flow (like condensate and feed water flow) the specific exergy is given as [17],

$$ex_{in} = C\left[(T - T_o) - T_o \ln\left(\frac{T}{T_o}\right) \right] \ldots\ldots\ldots\ldots\ldots\ldots (19)$$

C indicates specific heat measured in kj kg^{-1} K^{-1}.

The specific exergy of air and combustion/ flue gas streams is given by [18],

$$ex_a = C\left[(T - T_o) - T_o \ln\left(\frac{T}{T_o}\right) \right] + RT_0 \ln\left(\frac{p}{p_o}\right) \ldots\ldots\ldots\ldots (20)$$

where R is the universal gas constant (R= 0.287 kj Kg^{-1} K^{-1}); p is pressure in bar.

The specific heat of air is a function of temperature [19],

$$C_{air} = 1.04841 - 0.000383719T + \frac{9.45378T^2}{10^7} - \frac{5.49031T^3}{10^{10}} + \frac{7.92981T^4}{10^{14}} \ldots\ldots\ldots (21)$$

The temperature is measured in Kelvin (K).

The specific heat of flue/combustion gases liberated by burning bagasse in a boiler is given as [20],

$C_{fg} = (0.27 + 0.00006T_{fg})$ ………………… (22)

Where T is measured in ^{0}C; the unit of specific heat is kj kg^{-1} K^{-1}.

Total exergy

The total exergy rate for any material stream is given by,

$\dot{E}x = \dot{m}ex$ ……………. (23)

4.6) Exergy input: The input to the CHP system in terms of exergy is the sum of physical/flow exergy of combustion air and the chemical exergy of fuel. Hence,

[Fuel exergy rate] = [physical exergy of air] + [chemical exergy of fuel]

$$\dot{E}x_{in} = \dot{E}x_{ph} + \dot{E}x_{ch} = \dot{m}_{15}\, ex_{15} + \dot{m}_{16}\, ex_{16} \quad \text{............. (24)}$$

Chemical exergy of bagasse

Specific chemical exergy of fuel can be stated as the maximum amount of energy obtainable when the fuel is brought from the environment state to the useful state by a process involving heat transfer and exchange of substances with the environment. Appendix C [18] suggests that for fossil fuels with mass ratio 2.67 > o/c > 0.667, which in particular includes wood,

$$\varphi_{dry} = \frac{1.0438 + 0.1882\left(\frac{h}{c}\right) - 0.2509\left(1 + 0.7256\frac{h}{c}\right) + 0.0383\left(\frac{n}{c}\right)}{1 - 0.3035\left(\frac{o}{c}\right)} \quad \text{.................. (25)}$$

The above equation gives (φ) which is the ratio of chemical exergy of dry solid fuels to the net calorific value (NCV) of fuel and is applicable for o/c ratio in the range specified above. This expression is estimated to be accurate within ±1%. The notations h,c,n,o indicate mass fractions of carbon, hydrogen, oxygen and nitrogen respectively. Equation 25 is on dry basis, but fuel (bagasse) from sugar mills to burn in the boilers has 50% moisture content. Taking moisture content of the fuel into consideration, the chemical exergy of the fuel is given by Kotas [18],

$$\varepsilon_o = \left[(NCV)_o + wh_{fg}\right]\varphi_{dry} + 9417s \quad \text{................ (26)}$$

w is the moisture content of the fuel which is taken as 0.5. For water substance at T_0= 298.15K, h_{fg}= 2442 kJ/kg, s is the sulfur content in the fuel (bagasse). But the sulfur content in the fuel is zero because it is an organic fuel. The net calorific value (NCV) or lower heating value (LHV) of the fuel is 7650 kJ/kg [20]. Hence equation 26 can be written as,

$$\varepsilon_o = [7650 + 0.5x2442]\varphi_{dry} \quad \text{................. (27)}$$

The ultimate analysis of bagasse yields [21], c= 0.4789, h=0.0592, n=0.33, o=0.4581. Now the ratio of o/c is 0.9565, which is in the range 2.67 > o/c > 0.667. Substituting the values in equation 25 leads to a value of φ_{dry}= 1.155. Hence the value of ε_o will result to 10246 kJ/Kg. This in turn, is very near to the experimental value of specific chemical exergy of bagasse 9890.70 kJ/kg [22]. The theoretically calculated chemical exergy and the practical value have an error of 3.59%, which are found to be in good agreement.

Exergy efficiency

The different components considered from the CHP system are usually heat exchangers except steam turbine (prime mover). The exergy efficiency of the CHP components and CHP system as a whole is as given below,

For boiler: $$\eta_{ex,boi} = \frac{\dot{E}x_{18} - \dot{E}x_{14}}{\dot{E}x_{in}} = \frac{\dot{E}x_{18} - \dot{E}x_{14}}{\dot{E}x_{15} + \dot{E}x_{16}} \quad \text{................ (28)}$$

For turbine: $$\eta_{ex,tur} = \frac{\dot{W}_G}{\dot{E}x_1 - \dot{E}x_2 - \dot{E}x_3 - \dot{E}x_4} \quad \text{............... (29)}$$

For condenser: The exergy efficiency of the surface condenser is given by [16],

$$\eta_{ex,cond}=\frac{\dot{m}_c\, c_{p,c}\left[\left(T_{c2}-T_{c1}\right)-T_0\ln\left(\frac{T_{c2}}{T_{c1}}\right)\right]}{\dot{m}_v\left\{c_{p,v}\left[\left(T_{v1}-T_{cond}\right)-T_o\ln\left(\frac{T_{v1}}{T_{cond}}\right)\right]+h_{fg/T=T_{cond}}-T_o s_{fg/T=T_{cond}}\right\}} \quad \ldots\ldots\ldots\ldots\ldots (30)$$

In the present context the exergy efficiency of condenser can be written as,

$$\eta_{ex,cond}=\frac{\dot{m}_6\, c_6\left[\left(T_7-T_6\right)-T_0\ln\left(\frac{T_7}{T_6}\right)\right]}{\dot{m}_4\left\{c_4\left[\left(T_4-T_5\right)-T_o\ln\left(\frac{T_4}{T_5}\right)\right]+h_{fg/T=T_5}-T_o s_{fg/T=T_5}\right\}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (31)$$

The exergy efficiency of a heat exchanger is defined as the ratio of exergy gained by the cold fluid to the exergy lost by the hot fluid. Hence the exergy efficiency for a heat exchanger is defined by [23],

$$\eta_{ex}=\frac{\dot{E}x_{cold,out}-\dot{E}x_{cold,in}}{\dot{E}x_{hot,in}-\dot{E}x_{hot,out}}=\frac{\Delta\dot{E}x_{cold}}{\Delta\dot{E}x_{hot}} \quad \ldots\ldots\ldots\ldots\ldots\ldots.. (32)$$

Applying the above mentioned principle for HPH and de-aerator we have,

For high pressure heater: $\eta_{ex,HPH}=\frac{\dot{E}x_{14}-\dot{E}x_{12}}{\dot{E}x_{13}-\dot{E}x_{10}}$ …………………….. (33)

But de-aerator is a direct contact heat exchanger and hence the modified equation is given by,

For De-aerator: $\eta_{ex,DE}=\frac{\left(\dot{m}_{12}-\dot{m}_{11}\right)ex_{12}-\left[\dot{E}x_5+\dot{E}x_8+\dot{E}x_9+\dot{E}x_{10}\right]}{\dot{m}_{11}\left(ex_{11}-ex_{12}\right)}$ ……………..... (34)

The CHP system is simultaneous generation of heat and electricity. Hence it gives two products as output i.e. process heat and power. Hence the exergy efficiency of a CHP system is given in general as,

$$\eta_{ex,CHP}=\frac{\dot{E}x_{out}}{\dot{E}x_{in}}=\frac{\dot{W}_G+\dot{E}x_p}{\dot{E}x_{in}} \quad \ldots\ldots\ldots\ldots\ldots\ldots.. (35)$$

Applying the above principle we have two different options of defining exergy efficiency equations for CHP system which are defined as below,

.

$$\eta_{ex,CHP-1}=\frac{\dot{W}_G+\dot{E}x_p}{\dot{E}x_{in}}=\frac{\dot{W}_T+\dot{E}x_2+\dot{E}x_3}{\dot{E}x_{15}+\dot{m}_{16}\,ex_{16,\exp}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (36)$$

Where $ex_{16,exp}$ is the experimental value of specific chemical exergy of bagasse i.e. 9890.70 kJ/kg.

$$\eta_{ex,CHP-2} = \frac{\dot{W}_G + \dot{E}x_p}{\dot{E}x_{in}} = \frac{\dot{W}_T + \dot{E}x_2 + \dot{E}x_3}{\dot{E}x_{15} + \dot{m}_{16}\, ex_{16,th}} \quad \text{................ (37)}$$

Where $ex_{16,th}$ is the theoretically calculated value of specific chemical exergy of bagasse i.e. 10246.00 kJ/kg.
The energy efficiency of the CHP system is as below,

$$\eta_{e,CHP} = \frac{\dot{W}_G + \dot{E}_p}{\dot{E}_{in}} = \frac{\dot{W}_G + \dot{E}_2 + \dot{E}_3}{\dot{E}_{15} + \dot{E}_{16}} = \frac{\dot{W}_G + \dot{E}_2 + \dot{E}_3}{\dot{E}_{15} + \dot{m}_{16}\, LHV} \quad \text{................. (38)}$$

Where LHV stands for lower heating value of fuel i.e. 7650 kJ/kg [20].

3.5) Improvement potential (IP): Another useful parameter employed here is the concept of an exergetic improvement potential (IP), which in the rate form, is as below [24],

$$\dot{IP} = (1-\eta_{ex})\left(\dot{E}x_{in} - \dot{E}x_{out}\right) \quad \text{.............................. (39)}$$

Total exergy destruction ratio (TExDR)
It is described as the ratio of total exergy destruction in the system to the total exergy input to the system as follows [25],

$$TExDR = \frac{\dot{E}x_{Tot,dest}}{\dot{E}x_{Tot,in}} \quad \text{.............................. (40)}$$

Applying equation 40 we have,

$$TExDR_I = \frac{\dot{E}x_{dest,tot}}{\dot{E}x_{15} + \dot{E}x_{16}} = \frac{\dot{E}x_{dest,tot}}{\dot{E}x_{15} + \dot{m}_{16}\, ex_{16,\exp}} \quad \text{............. (41)}$$

Where $ex_{16,exp}$ is the experimental value of specific chemical exergy of bagasse i.e. 9890.70 kJ/kg.

$$TExDR_{II} = \frac{\dot{E}x_{dest,tot}}{\dot{E}x_{15} + \dot{E}x_{16}} = \frac{\dot{E}x_{dest,tot}}{\dot{E}x_{15} + \dot{m}_{16}\, ex_{16,th}} \quad \text{............. (42)}$$

Where $ex_{16,th}$ is the theoretically calculated value of specific chemical exergy of bagasse i.e. 10246.00 kJ/kg.

3.6) Component exergy destruction ratio (CExDR): It is described as the ratio of exergy destruction of any component of the system to the exergy input to the system as follows [25],

$$CExDR = \frac{\dot{E}x_{i,dest}}{\dot{E}x_{Tot,in}} \quad \text{................................ (43)}$$

'i' indicates component.

3.7) Dimensionless exergy destruction ratio (DExDR): It is described as the ratio of exergy destruction of any component of the system to the total exergy destruction of the system as follows [25],

$$DExD = \frac{\dot{Ex}_{i,dest}}{\dot{Ex}_{Tot,dest}} \quad \ldots\ldots\ldots\ldots\ldots(44)$$

'i' indicates component.

3.8) Power to heat ratio (PHR): The combined heat and power plant generates two useful products namely process heat and electricity. In the present context the ratio of power developed by the turbine to the process heat supplied is called power to heat ratio (PHR). Mathematically,

$$PHR = \frac{\dot{W}_G}{\dot{Q}_p} = \frac{\dot{W}_G}{\dot{E}_2 + \dot{E}_3} \quad \ldots\ldots\ldots\ldots\ldots\ldots(45)$$

Assumptions

The assumptions made in the present analysis are as follows,

1. Only physical or flow exergy is taken into account.
2. The changes in kinetic and potential energies are neglected.
3. Ideal gas principles are applied to air and combustion products.
4. The combustion is assumed to be complete.
5. All the components are well insulated and heat loss to the environment is neglected.
6. The CHP system operates in a steady state.
7. Temperature difference between the component control volume and immediate surroundings is not considered.

The exergy destroyed in the plant's component is a function of entropy generated and the ambient air temperature surrounding the component. Temperature surrounding the component in a CHP system changes substantially in terms of location. For instance the temperature of the air surrounding the boiler and condenser, have large variation in the ambient conditions. Hence in the present analysis a natural-environment-subsystem model [26] is adopted for the reference condition. A detailed analysis for change in reference temperature is also given further.

RESULTS

Table 2 shows the energy and exergy rates at various state points in the CHP system. Table 3 shows the results of energy analysis and exergy analysis at reference conditions of p_0= 1.01325 bar and T_0= 25^{0}C. The exergy input to the CHP system is based on the experimental value of specific chemical exergy of bagasse i.e. 9890.70 kJ/kg. As it can be noticed, that maximum exergy destructions take place in the boiler, whereas minimum exergy destructions in the de-aerator. HPH is most efficient component of the plant followed by steam turbine in terms of exergy. Condenser is the least efficient component of the plant followed by boiler in terms of exergy. IP rates are found to be the highest for boiler and the least for HPH. The total heat loss for CHP system is 47928.37 kW whereas total exergy destruction is 104361.99 kW, which is more than twice. This indicates that energy analysis is less important than exergy analysis. Maximum heat loss takes place in the condenser followed by boiler. For HPH the exergy destruction is more than twice as compared to heat loss. For de-aerator the exergy destruction is more than four times as compared to heat loss. In condenser lot of heat loss takes place as compared to the exergy destruction rates. This indicates that energy analysis reveals that a lot of heat loss takes place through condenser whereas according to exergy analysis boiler is the main responsible component for exergy destruction.

Table 2: *Thermodynamic properties, energy and exergy rates at various state points of the CHP system*

St. pt.	Substance	$\dot{m}$ kg s^{-1}	p bar	T ^{0}C	C kJ/kg K	h kJ/kg	s kJ/kg K	ex kJ/kg	$\dot{E}$ kW	$\dot{Ex}$ kW
1	steam	34.50	64.00	490	2.40	3394.43	6.81	1368.63	117107.83	47217.73
2	process steam	4.98	16.00	300	2.24	3035.43	6.88	988.77	15116.44	4924.07
3	process steam	15.73	3.00	155	2.14	2771.93	7.10	659.71	43602.45	10377.23
4	exhaust steam	13.79	0.12	50	1.94	2591.29	8.07	190.01	35733.88	2620.23
5	condensate	14.01	-----	45	4.18	188.43	0.63	2.68	2639.90	37.62
6	cooling water	805.37	2.48	25	4.18	104.83	0.36	0.00	84426.93	0.00
7	cooling water	805.37	1.50	35	4.17	146.63	0.50	0.68	118091.40	551.19
8	makeup water	2.55	----	25	4.18	104.83	0.36	0.00	267.31	0.00
9	condensate	13.00	----	80	4.19	335.01	1.07	18.96	4355.13	246.55
10	condensate	3.60	----	105	4.22	440.15	1.36	38.55	1584.54	138.78
11	process steam	2.60	1.50	111	2.10	2693.40	7.22	545.42	7002.84	1418.09
12	condensate	35.76	----	105	4.22	440.15	1.36	38.55	15739.76	1378.54
13	process steam	3.60	9.00	200	2.65	2832.70	6.75	824.78	10197.72	2969.20
14	feed water	35.76	----	160	4.33	675.47	1.94	102.42	24154.80	3662.60
15	combustion air	52.50	1.07	180	1.02	462.06	----	35.62	8534.92	1870.18
16	fuel	14.25	----	----	-----	----	----	9890.70	109012.50	140942.47
17	flue gas	65.41	2.48	155	0.28	119.84	----	82.74	2526.98	5412.02
18	steam	34.72	66.00	495	2.41	3404.06	6.81	1378.26	118188.96	47853.18
19	process steam	0.22	14.00	495	2.18	3463.87	7.59	1205.63	762.05	265.23
20	power output	----	-----	-----	-----	-----	-----	---------	20700.00	20700.00

Boiler has DExDR and CExDR values as 0.8931 and 0.6526 respectively. This indicates that 89.31% of total exergy destruction and 65.26% of total exergy input takes place in the boiler. Turbine has DExDR and CExDR values as 0.0823 and 0.0601 respectively. This indicates that 8.23% of total exergy destruction and 6.01% of total exergy input takes place in the turbine. Condenser has DExDR and CExDR values as 0.0148 and 0.0108 respectively. This indicates that 1.48% of total exergy destruction and 1.08% of total exergy input takes place in the condenser. HPH has DExDR and CExDR values as 0.0053 and 0.0038 respectively. This indicates that 0.53% of total exergy destruction and 0.38% of total exergy input takes place in HPH. De-aerator has DExDR and CExDR values as 0.0045 and 0.0032 respectively. This indicates that 0.45% of total exergy destruction and 0.32% of total exergy input takes place in the de-aerator. The TExDR value is 0.7307. It means that 73.07% of total exergy input gets destructed collectively in the five main components of the CHP system. Since sustainability index (SI) is the reciprocal of TExDR or depletion number we have SI= 1.368. It is a very low value.

***Table 3:** Results of Exergy and Energy analysis of different components in the CHP system*

Sl no	particular	$\dot{Ex}_{dest}$ kW	η_{ex} %	$\dot{IP}$ kW	CExDR	TExDR	DExDR	Heat loss kW	Heat loss % total
01	Boiler	93210.05	30.94	64370.86	0.6526		0.8931	12571.24	26.23
02	Turbine	8596.20	70.65	2522.98	0.0601		0.0823	1955.06	4.08
03	Condenser	1546.88	26.31	1139.89	0.0108	0.7307	0.0148	33093.97	69.04
04	HPH	546.36	80.69	105.50	0.0038		0.0053	198.14	0.42
05	De-aerator	462.50	64.90	162.33	0.0032		0.0045	109.96	0.23
06	Total/ Avg	104361.99	54.69	68301.56			1.0000	47928.37	100.00
07	$\dot{Ex}_{in}$ kW	142812.65							

A parametric study is conducted for the CHP system. The isentropic efficiency of the steam turbine is found to be 95%. Table 4 indicates that as the condenser pressure increases the power output from the generator decreases. Hence a decreased condenser pressure is more advantageous from power output point of view.

***Table 4:** Effect of condenser pressure on generator output.*

Cond. pr. (bar)	0.075	0.095	0.120	0.150	0.1990
Power, kW	20852.10	20700.00	20622.22	20486.33	20419.02

Table 5 indicates that as the mass flow rate of steam at the turbine inlet increases the power output from the generator increases. This is due to the fact that the turbine inlet steam temperature, pressure and the speed of the turbine (i.e. 6300 rpm) is maintained constant at variable load conditions. Whereas the mass flow rate of steam to the turbine is varied.

Table 5: Effect of mass flow rate of steam at turbine inlet on generator output.

$\dot{m}_1$ kg s^{-1}	29.00	30.00	31.00	32.00	33.00	34.50
Power output, kW	16692.46	17425.22	18157.99	18890.76	19623.53	20700.00

Table 6 indicates that generator power output increases with increase in inlet steam pressure and temperature conditions. Process steam pressure and temperature conditions play a vital role. An increase or decrease in the process steam parameters would result in the variation of the enthalpy drop across the turbine. It is quite evident that a decrease in the pressure of the extracting steam would result in a higher enthalpy drop across the turbine resulting in higher work output from the turbine. But the sole purpose of CHP system is to suffice the need of process steam demand conditions of the industry. The present process steam parameters at state point 2 and 3 are being decided based on the demand conditions from the process. Hence maintaining the present conditions of the steam at state points 2 and 3 is inevitable.

Table 6: Effect of turbine inlet conditions on generator output.

Pr. P_1 (bar)	45	64	87	95	110
Temp. T_1 (^{0}C)	440	490	515	525	540
Power, kW	17753.11	20700.00	21775.25	22280.46	22966.66

Table 7 shows the exergy parameters and its effect on the variation in reference temperatures. It indicates that for boiler exergy destruction rates, IP rates increase as the reference temperature increases. Also a decrease in exergy efficiency of the boiler is observed with increase in reference temperatures. Also for turbine the exergy destruction rates and IP rates increase with reference temperatures. A fall in exergy efficiency of turbine is observed with increase in reference temperature. The condenser is the least efficient component at all values of reference temperatures.

Table 7: *Exergy rates (in kW) for the components of the CHP system at various reference temperatures*

Sl No	Parameter	Reference temperature (^{0}C)					
		15	20	25	30	35	40
01	$\dot{Ex}_{dest,boi}$	91924.11	92644.39	93210.05	93948.74	94622.28	95345.54
02	$\dot{IP}_{boi}$	62490.00	63526.25	64370.86	65444.69	66443.76	67514.17
03	$\eta_{ex,boi}$, %	32.02	31.43	30.94	30.34	29.78	29.19
04	$\dot{Ex}_{dest,tur}$	8373.34	8484.77	8596.20	8707.62	8819.05	8930.47
05	$\dot{IP}_{tur}$	2412.36	2467.37	2522.98	2579.19	2635.13	2691.64
06	$\eta_{ex,tur}$, %	71.19	70.92	70.65	70.38	70.12	69.86
07	$\dot{Ex}_{dest,cond}$	2570.42	2058.74	1546.88	1034.60	522.10	9.33
08	$\dot{IP}_{cond}$	2103.11	1617.55	1139.89	682.42	257.91	0.16
09	$\eta_{ex,cond}$, %	18.18	21.43	26.31	34.44	50.60	98.25
10	$\dot{Ex}_{dest,HPH}$	515.90	523.27	546.36	553.48	566.27	573.92
11	$\dot{IP}_{HPH}$	88.16	93.87	105.50	112.35	121.91	130.22
12	$\eta_{ex,HPH}$, %	82.91	82.06	80.69	79.70	78.47	77.31
13	$\dot{Ex}_{dest,DE}$	443.43	447.90	462.50	464.80	470.69	479.06
14	$\dot{IP}_{DE}$	134.04	144.67	162.33	174.57	190.72	212.12
15	$\eta_{ex,DE}$, %	69.77	67.70	64.90	62.44	59.48	55.72
16	$\sum \dot{Ex}_{dest,tot}$	103827.20	104159.07	104361.99	104709.24	105000.39	105338.32
17	$TExDR_{I}$	0.7258	0.7287	0.7307	0.7336	0.7362	0.7391
18	$TExDR_{II}$	0.7010	0.7038	0.7057	0.7085	0.7110	0.7137
19	$\eta_{ex,CHP-1}$, %	26.13	25.63	25.20	24.72	24.26	23..78
20	$\eta_{ex,CHP-2}$, %	25.23	24.75	24.34	23.87	23.43	22.97
21	$\eta_{e,CHP}$, %	67.56	67.56	67.56	67.56	67.56	67.56
22	$\dot{Ex}_{in,exp}$	143046.82	142932.22	142812.65	142715.92	142615.28	142519.04
23	$\dot{Ex}_{in,th}$	148109.85	147995.25	147875.68	147778.95	147678.31	147582.07

The exergy efficiency of condenser is found to increase with reference temperature. This is due to the fact that temperature difference between steam and cooling water decreases as the reference/dead state temperature increases. This decreases the exergy destruction rates and increases the exergy efficiency. For HPH exergy destruction rates, IP rates increase whereas exergy efficiency decreases with increase in reference temperature. For de-aerator exergy destruction rates, IP rates increase whereas exergy efficiency decreases with increase in reference temperature. Total exergy destruction ratios ($TExDR_I$ and $TExDR_{II}$), are found to increase with reference temperature, whereas exergy efficiency of CHP system is found to decrease with reference temperature. This indicates that operation of the CHP system at higher environmental temperature conditions is less sustainable. In turn the energy efficiency which is more than twice that of exergy efficiency remains constant with respect to reference temperatures. It is based on LHV (lower heating value) of the fuel. This makes the device to appear more efficient, which is misleading [27]. Using LHV incorporates losses occurring in the furnace –boiler system, energy lost with hot gases, incomplete combustion etc. Hence it clearly indicates that efficiencies based on energy can often be non-intuitive or even misleading [28].

DISCUSSION

An exergy analysis is usually aimed to determine the maximum performance of the system and/or identify the sites of exergy destructions. Identifying the main sites of exergy destruction, causes of destruction, and true magnitudes of destructions shows the direction for potential improvements for the system and components [29-30].Exergy appears to be a key concept, since it is a linkage between the physical and engineering world and the surrounding environment, and expresses the true efficiency of engineering systems which makes it a useful concept to find improvements. Exergy conscious utilization of energy resources would help advance technological development towards resource-saving and efficient technology can be achieved by improving design of processes with high exergetic efficiency. As it can be seen from the table 3 that highest IP rates exist for Boiler followed by turbine and condenser. The following paragraphs discuss different methods of improvement for locations with increasing order of magnitude in IP rates.

Auxiliary power consumption due to the running of Boiler fans (FD, ID and SA fan), boiler feed pump, circulating water pump; condensate extraction pump etc almost remains constant, when they are of constant speed drives. These equipments operate at a very less efficiency during part load conditions of the plant. Moreover higher auxiliary power consumption proves detrimental to the increase in the efficiency of the plant. Hence in the present case variable speed drive (VSD) machines are introduced, which operate at an optimum efficiency even at varying load conditions. This indicates that there is less scope for improvement in auxiliary power consumption. The IP rates for these components are not made available due to insufficient data. Another area that significantly benefits the plant performance is the design of the High pressure feed water heater (HPH) and de-aerator. A reduction of heating steam requirements for this equipment will result in a lower steam requirement of the turbine and thus also improve the heating rate. Since it is quite evident that it requires more than 2 units of heat to generate 1 unit of work, hence work is more useful than heat (particularly in terms of exergy). Electrical energy has equivalent amount of energy and exergy because it can in theory be totally converted into work. But process heat (i.e thermal energy) is not entirely utilizable because it has exergy content less than energy content [31]. Hence, optimization of process heating parameter offers scope for improvement. Moreover power to heat ratio (PHR) is a very important parameter for CHP plants, because it indicates the economic viability of the plant. According to Eastop and croft [32] the optimal power to heat ratio for extraction turbines is 0.3333.This is due to the reason that electricity is easily saleable in the market, but there are no purchasers for process heat. In the present case study the PHR value is 0.3525 at 100% BMCR and TMCR conditions. Hence this offers less scope for improvement.

Condenser contributes for the third highest IP rate. The condenser utilized here is water cooled operating with forced circulation cooling tower using an ID (induced draft) fan. The plant efficiency might be improved by lowering the condenser pressure or due to higher expansion of steam through the turbine. But this will decrease the quality of steam which in turn will be responsible for erosion of the turbine blade in the last stages. Right now the condenser operates at 0.095 bars with condensate temperature of 45^0C. For a cooling water temperature rise of 9- 10^0C if the condensate temperature is increased from 46^0C to 54^0C the exergy efficiency increased from 22.60% to 40.70% [16]. But increase in condensate temperature would increase condenser pressure and in turn would affect the turbine power output. Hence it seems quite necessary to strike a balance between the power output of turbine, exergy efficiency of condenser and prevention of turbine blade erosion in terms of condenser pressure or condensate temperature. Above all if an air cooled condenser is used (which uses ambient air to condense steam), then it would definitely have insignificant environmental impact due to reduced water consumption.

Thus replacing water cooled condenser with an air cooled condenser would be one of the options for improvement in the condenser, in terms of environmental impact. This idea however should be balanced with capital costs and operational benefits. Turbine is the second highest contributor for IP rates. Turbine blades made of higher cost steel alloys, effective blade design, 3-D airfoil designs by computational fluid dynamics, better steam sealing within the turbine, effective insulation to curb heat losses, flow passage optimization, barrel type designs, mono-rotor blocks, use of 9-12% chromium steels can ensure turbine efficiency improvement of 3-4% [33]. Also the isentropic efficiency of the turbine can be improved from 90% to a level of 95%. But in the present condition the isentropic efficiency of the steam turbine is found to be 95%. This indicates that there is less scope for improvement for steam turbine. An increase in turbine inlet steam pressure and temperature would increase the turbine power output. But this is quite disadvantageous because use of high pressure and high temperature steam at the inlet of the turbine would decrease the exergy destructions in the boiler but in turn would increase for a turbine. Literature available reveals [14] that when inlet steam conditions are above 61 bar and 475^0C, the benefits are found to be marginal (particularly in terms of exergy) though there is a substantial rise in the pressure and temperature. Boiler is the highest contributor of IP rates. It may be noted that a boiler energy use can be reduced by many ways like controlling excess air, enhancing heat transfer rate, improving combustion efficiency, use of more environmental friendly fuel, recovering waste heat, recovering condensate, optimizing blow down process, preventing leakage, providing proper insulation etc. [34].

The largest irreversibility occurs in the boiler, particularly due to heat transfer to the working fluid and losses due to flue gas emissions. Having been around for centuries, the technology involved in a boiler can be seen as having reached a plateau, with even marginal increase in efficiency painstakingly hard to achieve [35]. The main improvement is possible in the area of heat transfer to the working fluid, optimization of heat transfer area and configuration, effective arrangement of heating surfaces, better material selection etc. Effective augmentation of heat transfer areas with respect to heat recovery devices like economizer, air pre-heater etc. would reduce the stack gas temperatures. Temperature of the flue gases leaving the boiler range from 150-250^{0}C. In the present case the stack gas temperature is found to be at 155^{0}C. This stack gas temperature can be reduced further which would reduce environmental impact. But here a limit is imposed due to the problems of formation of acids, corrosion etc on the metal surfaces which are in contact with the flue gas. This limiting temperature is the dew point of sulfuric acid. As bagasse has low sulfur content the temperature is lower than in the case of boilers which use fossil fuels. For bagasse boilers the dew point is below 90^{0}C [36]. This means that the amount of heat which can be recovered from flue gas emissions is given by,

$$\dot{Q}_r = \dot{m}_{fg}.C_{fg}.\Delta T_{fg} \quad (29)$$

ΔT_{fg}= is the difference in actual temperature of the flue gas and dew point of sulfuric acid. With m_{fg}= 65.41 kg/s; C_{fg}= 0.279 kJ/kJ K; ΔT_{fg} = (155-90) = 65^{0}C, the value of Q_r is 1186.21 kW. This heat content can be recovered through heat recovery steam generator (HRSG) or fuel (bagasse) dryer. It would be quite advisable to use the heat for a dryer because of three specific reasons:

- An HRSG is also a heat exchanger which in turn would have very high destruction rates.
- Using a bagasse dryer would bring down the moisture content in the fuel and improve its burning characteristics.
- Using a bagasse dryer would reduce the flue gas temperature and its impact on the environment

The normal moisture content of bagasse is 50 wt%, which is used as fuel in the boilers. This fuel can be dried in two ways, by using super heated steam or by using waste flue gases. The disadvantages of using superheated steam are namely high generation cost of superheated steam, requirement of small particulate size of fuel, higher capital costs of the dryer. If a waste flue gas dryer is used then the options available are rotary type, flash type, fluidized bed type and belt conveyor type dryer. But however to ensure uninterrupted supply of dried fuel to the boiler the mechanism of the selected drier plays a vital role. Keeping these aspects into consideration a belt dryer is found to be advantageous due to the following reasons [37]. Firstly it is versatile in nature and can handle wide range of fuel. Secondly feed stock could be spread on moving perforated conveyor to dry the material in a continuous process. Thirdly problems of fire hazard and air emissions are negligible. Finally it can operate at high temperatures between 90^{0}C- 200^{0}C and the pressure drop across the dryer is very less.

A belt dryer if used would reduce the moisture content by a few %. This would recover much of the energy used during combustion for water evaporation. It would also be beneficial for decreasing the dimensions of the boiler and reducing the emissions of unburned solids. Above all generally biomass fuels have an auto-ignition temperature of 260-280^{0}C [37], hence the possibility of increase in amount of heat recovered would be from the range of 260- 280^{0}C as flue gas temperature to 90^{0}C i.e. dew point of sulfur for bagasse based boilers. The boiler used in the present case study is of travelling grate type. It could be replaced by innovative technology such as, bagasse –fluidized bed indirectly heated gasifier, hot gas air conditioning, molten carbonate fuel cells (MCFC) [38]. One major concern in gasification process is the moisture content in bagasse. The sugarcane bagasse leaves the milling process with 50% moisture content. This is a very high value and proves detrimental to start the process of gasification. Moreover literature [39] reveals that the exergy destroyed is nearly

6500 kJ/Kg of bagasse when the moisture content is 50%. In the present case study the bagasse flow rate is 14.25 kg/s, which results in value of 92625 kW. Hence in the present system a gasifier does not seem to be attractive in terms of exergy. A BIGFC (biomass integrated gasification with fuel cells power plant) appears attractive in terms of kWhr/ton of cane crushed. But in terms of internal rate of return, net present value, payback time etc steam cycles are found to be at acceptable levels [38]. But however since these technologies are in the development phase their capital costs are expected to decrease in the near future.

For the present case study considered a retrofitted bagasse dryer appears meaningful, since it decreases the waste emissions which have a greater potential in terms of environmental impact. A waste emission possesses exergy in two components: thermo-mechanical and chemical [40]. The potential impact of thermo-mechanical exergy on the environment is limited as compared to chemical exergy. Hence if bagasse dryer is introduced it would have far reaching consequences in terms of exergy compared to the capital cost employed. However it has to be noted that part of the irreversibility definitely exists which cannot be avoided due to physical, technological and economic constraints.

CONCLUSIONS

The following conclusions may be drawn from the present study:-

1. HPH is the most efficient component and condenser is the least efficient component in terms of exergy.
2. Highest exergy destructions take place in boiler, whereas lowest value of exergy destruction is found to be in de-aerator
3. Energy analysis indicates that lot of heat loss takes place through the condenser, whereas exergy analysis reveals that boiler is the main responsible component for irreversibility.
4. The CHP system is found to be less sustainable in terms of exergy with the increase in the ambient temperature conditions.
5. A thermodynamic trade-off between the power output of turbine, exergy efficiency of condenser and prevention of turbine blade erosion in terms of condenser pressure or condensate temperature offers scope for improvement.
6. Utilization of belt conveyor bagasse dryer would reduce environmental impact and increase boiler efficiency.
7. Finally, it may be stated that this study will definitely be useful to policy makers, engineers, industrial energy users and scientists in the field of energy and power engineering.

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Nomenclature

C—–- specific heat, ($kJ\ kg^{-1}\ K^{-1}$)
$C_{p,v}$ --- specific heat of vapor at constant pressure, ($kJ\ kg^{-1}\ K^{-1}$)
$C_{p,c}$ --- specific heat of cooling water at constant pressure, ($kJ\ kg^{-1}\ K^{-1}$)
C_{fg} --- specific heat of flue gas, ($kJ\ kg^{-1}\ K^{-1}$)
C_{air} --- specific heat of combustion air, ($kJ\ kg^{-1}\ K^{-1}$)
$C_{p(T)}$ – specific heat at constant temperature T, ($kJ\ kg^{-1}\ K^{-1}$)
$C_{p(To)}$ – specific heat at reference temperature T_o, ($kJ\ kg^{-1}\ K^{-1}$)
ex---- specific exergy of steam/vapor, ($kJ\ kg^{-1}$)
ex_{in} – specific exergy of incompressible fluid, ($kJ\ kg^{-1}$)
ex_a – specific exergy of air/ combustion gas, ($kJ\ kg^{-1}$)
ex_{ph} – specific physical exergy of air/combustion gas, ($kJ\ kg^{-1}$)
e_{ph} – specific physical exergy, ($kJ\ kg^{-1}$)
E – energy content, kW
E_{loss} – energy loss, kW
E_{in} – energy content at inlet, kW
E_{out} – energy content at outlet, kW
Ex --- total exergy content, (kW)
Ex_{Ph} --- total physical exergy content, (kW)
Ex_{ch} --- total chemical exergy content, (kW)
Ex_{in} --- total exergy content at the inlet, (kW)
Ex_{out} --- total exergy content at the outlet, (kW)
Ex_{dest} --- total exergy destruction, (kW)
$Ex_{dest,tot}$ – sum of exergy destruction in all components, kW
Ex_p --- total exergy content in the process steam, (kW)
$Ex_{hot,in}$ = total exergy at inlet of hot fluid, (kW)
$Ex_{cold,in}$= total exergy at inlet of cold fluid, (kW)
$Ex_{hot,out}$= total exergy at outlet of hot fluid, (kW)
$Ex_{cold,out}$= total exergy at outlet of cold fluid, (kW)
ΔEx_{cold}= rise in total exergy of cold fluid, (kW)
ΔEx_{hot}= fall in total exergy of hot fluid (kW)
h --- specific enthalpy, ($kJ\ kg^{-1}$)
$h_{(T)}$ – specific enthalpy at temperature T, ($kJ\ kg^{-1}$)
$h_{(To)}$ – specific enthalpy at reference temperature T_0, ($kJ\ kg^{-1}$)
h_0 --- specific enthalpy at reference condition, ($kJ\ kg^{-1}$)
h_{fg} --- latent enthalpy, ($kJ\ kg^{-1}$)
m --- mass flow rate, ($kg\ s^{-1}$)
m_{in} – mass flow rate at the inlet, ($kg\ s^{-1}$)
m_{out} – mass flow rate at the outlet, ($kg\ s^{-1}$)
m_{fg} – mass flow rate of flue gas, ($kg\ s^{-1}$)
m_c – mass flow rate of cooling water,($kg\ s^{-1}$)
m_v – mass flow rate of vapor, ($kg\ s^{-1}$)
p – pressure, (bar)
p_o – pressure at the reference condition, (bar)
Q – heat content, kW
Q_p – amount of process heat, kW
Q_r – amount of heat recovery, kW
R – universal gas constant, $kJ\ kg^{-1}\ K^{-1}$
s_{fg} – latent entropy, ($kJ\ kg^{-1}\ K^{-1}$)

s – sulfur content, (%)
s – specific entropy, (kJ kg^{-1} K^{-1})
T – temperature, (K)
T_0 – reference temperature, (K)
T_{c2} – cooling water outlet temperature, (K)
T_{c1} – cooling water inlet temperature, (K)
T_{cond} – temperature of the condensate, (K)
T_{fg} – temperature of flue gas, K
T_{v1} – temperature of the vapor at inlet of the condenser, (K)
W – work output, kW
W_G – turbine work output, (kW)
w -- moisture content, (% wt)

Abbreviations:
APH – air pre-heater
BPST – back pressure steam turbine
BMCR – boiler maximum continuous rating
BFP – boiler feed pump
CEP – condensate extraction pump
CEST—condensate extraction steam turbine
CHP --- combined heat and power
CExDR – component exergy destruction ratio
DExDR— dimensionless exergy destruction ratio
DE – de-aerator
GSC – gland steam condenser
HPH – high pressure heater
IP – improvement potential
LHV – lower heating value
NCV – net calorific value
PHR – power to heat ratio
SJAE—steam jet air ejector
TExDR – total exergy destruction ratio
TMCR – turbine maximum continuous rating

suffixes:
dest – destruction
in – inlet
out—outlet
loss – losses
boi – boiler
tur – turbine
cond—condenser/ condensate

Greek symbols:
η_{ex} -- exergy efficiency, %
$\eta_{ex,CHP-I}$ -- exergy efficiency of CHP system based on theoretical value of chemical exergy of bagasse
$\eta_{ex,CHP-II}$ -- exergy efficiency of CHP system based on experimental value of chemical exergy of bagasse
Φ_{dry} – ratio of chemical exergy and NCV of fuel on dry basis, (--)
ε_o – chemical exergy of bagasse, kJ kg^{-1}

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