Modeling and Efficiency Analysis of Chps with Lower Optimum Cycle Pressure Ratio

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Abstract- Regenerators are installed with the gas turbine power plants, operating at high pressure ratios and low firing temperatures, to enhance their thermal efficiencies. In general, the enhanced investment for a regenerator installation is returned in twelve to eighteen months. A gas turbine based heat and power cohort systems are on hand with different configurations. Further their performance is affected by design and operating parameters. Complexity of system analysis is increased due to interdependency of components (i.e. air compressor, combustion chamber, gas turbine, regenerator and steam generator) and parameters. Mathematical modeling based upon exergy, energy and mass balance athwart the components is simulated using computer programming tool EES (Engineering Equation Solver). From the results it is found that in case of cogeneration cycle with regenerator optimum cycle pressure ratio is 15 with first law efficiency 85.49% and for cogeneration cycle without regenerator is 36 with efficiency 80.41%.

Keywords: *Cogeneration Cycle, Gas turbine, Exergy, CR, Regenerator.*

1 INTRODUCTION

Exergy is to represent the availability of useful work from a energy conversion system. But it is not possible to extract all of the work from a system. Therefore, tthermodynamic analysis (I^{st} or II^{nd} law) is for premeditating the effect of design and operating parameters on a newly designed thermodynamic cycle or existing one [1-26]. In literature different other techniques are also available for the analysis [18-25]. Out of these techniques energy analysis [18], exergy analysis [19] and graph theoretic analysis [20] are some of the most used techniques for the analysis. Dev et al [20] proposed that second law efficiency of the system is higher than first law efficiency. It is due to the reason that amount of available energy is always less than total amount of energy available with the system. It was further proposed that [21] the amount of available energy is to be compared with the standard environmental conditions. This is to be analysed for each kind of thermal system. At present a lot of different kinds of thermal systems are available for the energy conversion [22]. These systems comprise of fuel energy extraction system along with the system capable of converting chemical energy of fuel into mechanical energy. This mechanical energy is converted into electrical energy. Electricity is supplied to utility providers and revenue is generated. A lot of losses are related with the electricity supply therefore, it is preferred to establish the plants near to energy consumption site. For this purpose plant size should be small and pollution emission also should be minimum. Therefore, in literature it is suggested that [23] gas turbine based power plants are best for this purpose. At present Gas Turbine (GT) power plants are intrinsically associated with low thermal efficiency. Waste heat segregated in the environment with flue gas can be extracted for a number of functional works and additional gain in efficiency is assured. In the present effort parametric analysis of a 30 MW gas turbine cycle is performed in which some fraction of waste heat is transferred to pressurized air at compressor outlet. Remaining heat is transferred to pressurized water in steam generator with a view that heat rejected to atmosphere through stack is minimized. Schematic configuration of this cycle capable of providing both heat and power is shown in Figure 1. In the present analysis exergetic analysis of combined heat and power system (CHPS) shown in Figure 1 is carried out for different operating parameters.

2 EXERGY MODELING OF THE CHPS

Thermal systems design and analysis engross principles from thermodynamics, heat transmit, fluid mechanics, manufacturing and design. Here work, thermodynamics of a 30 MW cogeneration cycle (Figure 1) is studied. Ambient air enters the compressor and after compression its temperature and pressure is increased. Compressed air is passed through a regenerator where high temperature combustion gases coming out of gas turbine transfer their heat to the compressed air. After gaining heat, compressed air comes to combustion chamber and fuel is added. After burning with air, chemical energy of fuel is converted into thermal energy. Combustion products temperature is designed equal to TIT which is fixed by thermal stress limit of gas turbine blade material. Combustion product temperature is controlled by making A/F mixture a leaner or richer mixture. Gases coming out from gas turbine have large amount of thermal energy. Major part of this thermal energy is transferred to compressed air in regenerator and high pressure water in steam generator. Flue gas temperature at stack inlet is kept above dew point temperature of flue gases to avoid corrosion in stack. In present work, mathematical modeling based on mass, energy and exergy balance across each component is followed by execution of computer program in software Engineering Equation Solver (EES) for different cycle operating parameters.

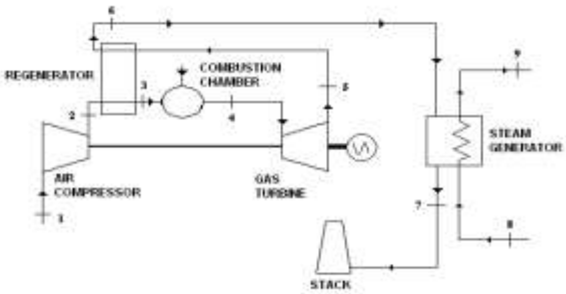

Figure 1 Schematic diagram of CHPS

3 THERMODYNAMIC MODELING

For recital of the plant at ISO day condition (15°C, 101.325 kPa, 60% relative humidity), assumptions underlying the cogeneration system model are as following:

1. The coordination operates at steady state of fluid flow.
2. Ideal-gas assortment principles are relevant for the air and the combustion products.
3. Fuel injected in combustion chamber is taken as Methane. The fuel is provided to combustion chamber at required pressure by throttling from a high-pressure source.
4. Heat transfer from the combustion chamber is 2% of the fuel lower heating value. All other components operate without heat loss.
5. For the present analysis, air is considered to be a combination of N_2 (77.48%), O_2 (20.59%), CO_2 (0.03%) and H_2O (1.9%).

Thermodynamic properties of the fluids are inbuilt functions of software EES. The energy equilibrium equations available in literature [5, 8, 9, 15, and 17] for various parts of the CHPS (Figure 1) are as follows:

3.1 Air Compressor

$$\eta_{SAC} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

$$s(T_{2s}, p_2) - s(T_1, p_1) = s^0(T_{2s}) - s^0(T_1) - R\ln\frac{p_2}{p_1} = 0 \tag{2}$$

$$W_{AC} = \dot{m}_a (h_2 - h_1) \tag{3}$$

3.2 Combustion Chamber

For complete combustion of methane the chemical equation takes the form

$$\lambda CH_4 + [0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O] \longrightarrow [1+\lambda][x_{N_2}N_2 + x_{O_2}O_2 + x_{CO_2}CO_2 + x_{H_2O}H_2O] \tag{4}$$

$$\lambda = \frac{n_F}{n_a} \quad \text{and} \quad 1+\lambda = \frac{n_P}{n_a} \tag{5}$$

$$x_{N_2} = \frac{0.7748}{1+\lambda} \tag{6}$$

$$x_{O_2} = \frac{0.2059 - 2\lambda}{1+\lambda} \tag{7}$$

$$x_{CO_2} = \frac{0.0003 + \lambda}{1+\lambda} \tag{8}$$

and

$$x_{H_2O} = \frac{0.019 + 2\lambda}{1+\lambda} \tag{9}$$

$$\dot{m}_a h_3 + \dot{m}_f LHV = \dot{m}_g h_4 + (1-\eta_{cc})\dot{m}_f LHV \tag{10}$$

$$\frac{P_4}{P_3} = (1 - \Delta P_{cc}) \tag{11}$$

3.3 Gas Turbine

$$\eta_T = \frac{T_4 - T_5}{T_4 - T_{5s}} \tag{12}$$

$$s(T_{5s}, p_5) - s(T_4, p_4) = s^0(T_{5s}) - s^0(T_4) - R\ln\frac{p_5}{p_4} = 0 \tag{13}$$

$$W_{GT} = \dot{m}_g (h_5 - h_6) \tag{14}$$

$$\dot{W}_{Net} = \dot{W}_{GT} - \dot{W}_{AC} \tag{15}$$

$$\dot{m}_g = \dot{m}_f + \dot{m}_a$$

(16)

3.4 Steam Generator

$$Q_p = h_6 - h_7 = h_9 - h_8$$

(17)

3.5 Regenerator

$$\varepsilon_R = \frac{T_3 - T_2}{T_5 - T_2} \quad (18)$$

$$m_a\left(h_2 - h_3\right) = m_p\left(h_6 - h_5\right)$$

(19)

$$\frac{P_3}{P_2} = (1 - \Delta P_R)$$

(20)

4 Exergy Analyses

The general exergy-balance equations available in literature [5, 8, 9, 15, and 17] and used for present analysis are represented by the expressions as below.

$$\dot{E}_W = \sum_{i=1}^{n} (\dot{E}_Q)i + \sum_{in} \dot{m}e - \sum_{out} \dot{m}e - \dot{E}_D$$

(21)

For single stream flow the above expression becomes.

$$\dot{E}_W = (\dot{E}_Q) + \dot{m}\, e_{in} - \dot{m}\, e_{out} - \dot{m}\, e_D$$

(22)

Specific exergy is given by

$$e = (C_{pa} + wC_{pv})T_a\left(\frac{T}{T_a} - 1 - \ln\frac{T}{T_a}\right) + (1+w)R_a T_a \ln\frac{P}{P_a}$$
$$+ R_a T_a\left[(1+w)\ln\left(\frac{1+w_a}{1+w}\right) + w\ln\left(\frac{w}{w_a}\right)\right]$$

(23)

$$e_f = \xi \times LHV$$

(24)

Where $\xi = 1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x}$ for the fuel $C_x H_y$

(25)

In this paper for the exergy analysis of CHPS, exergy destruction rate and the exergy efficiency for each component in the cycle (Figure 1) are shown in Table 1.The operating conditions for base case of the gas turbine power plant such as process heat, calorific value, output electrical power etc. are listed in Table 2.

5 Results and Discussion

As the cycle pressure ratio is increased then the cost of power plant equipments and components is also increased. With increase in cost of components and equipments per unit electricity generation cost is also increased. There are numerous methods to increase the efficiency of the power plant. In this line it is an effort to improve the thermodynamics of the power plant. It is achieved with the help of development in the thermodynamic cycles. These cycles are numerous and methods to improve their efficiency are also numerous. Exergy is based on the first and second laws of thermodynamics. Energy is conserved in any thermodynamic system, ideal or otherwise, while exergy is conserved for an ideal process which is not possible in real life. Therefore, exergy is not conserved for real processes or devices. In the present analysis exergy destruction in a cogeneration cycle represented in Figure 1 is studied for the change in cycle pressure ratio. Further it is found that exergy destruction is associated with specific heat which depends upon the concentration of constituents in combustion products. It is desirable to calculate the optimum cycle pressure ratio for the efficient design of the cycle.

Exergy in a CHP system is destructed in every part. Exergy destruction in combustion chamber is related to the amount of fuel consumed in combustion chamber. That is why exergy destruction in combustion chamber follows the same pattern as that of fuel consumed in combustion chamber. If heat is transferred from a high temperature to lower temperature then its quality goes down and exergy destruction takes place. Exergy (or available energy, or availability) is the maximum useful work that can be extracted from a quantity of energy and refers to the quality of energy. Thus, though the energy is conserved in the process of conversion, its quality deteriorates and less work can be obtained with each conversion. The various irreversible processes encountered within the combustor leads to certain degree of exergy loss. Several studies have indicated that the conventional combustion involves inherent thermodynamic irreversibility, which significantly limits the conversion of fuel energy into useful work [13, 14]. For typical atmospheric combustion systems, about 1/3rd of the fuel energy is discharged into the environment as heat. Most of irreversibility within the combustor is due to internal heat transfer between the products and reactants. Such heat transfer becomes inevitable in both premixed and diffusion flames, where highly energetic product molecules are free to exchange energy with unreacted fuel and air molecules [15]. The product and reactant molecules have large energy difference (i.e. temperature difference) and considerable entropy is generated when they interact. Internal heat transfer within the combustor is often difficult to be recognized as an efficiency problem, because it does not result in a direct energy loss from the combustion zone to the surrounding. Instead, internal heat transfer only degrades the exergy of the product flue gas and reduces its ability to produce useful work.

In actual practice it is difficult to manufacture a gas turbine with higher expansion ratio in comparison to lower compression. For this case regenerator lowers the optimum cycle pressure ratio for maximum first law efficiency. In case of cogeneration cycle with regenerator optimum cycle pressure ratio is 15 (Table 5) and for cogeneration cycle without regenerator is 36 (Table 4). With increase in cycle ratio from 5 to 36 it is found that exergy destruction in air compressor and gas turbine is increased by 24.51% and 23.04% respectively for cogeneration cycle without regenerator. While exergy destruction for combustion chamber and steam generator is decreased by 46.96% and 53.10% respectively. Exergy destruction in the cogeneration cycle without regenerator keeps on decreasing with increase in CR and 43.51% decrease in exergy destruction is with change in CR from 5 to36. First law efficiency is dependent upon mass of fuel injected in combustion chamber. Minimum fuel injected is at CR-36 that is 1.68% and corresponding value of first law efficiency is 80.41%.

Thermodynamic modeling of combined cycle components has done in last chapter for performance prediction. Analysis has been done based upon the dependent parameters like first law and second law efficiency, as a function of independent parameters such as compressor pressure ratio, turbine inlet temperature etc. Exergy, the essential concept in second law analysis, is always consumed or destroyed in any process. Therefore, by using exergy to evaluate the power plant cycles, a more accurate performance of the system can be obtained. Second law analysis gives much more meaningful evaluation by indicating the association of irreversibility or exergy destruction with work, combustion and heat transfer processes and allows thermodynamic evaluation of energy conservation options in cogeneration cycle, and thereby provides an indicator that points in the direction in which engineers should concentrate their efforts to improve the performance of thermal power plant.

6 Conclusions

Gas turbines are used extensively for by the power generation industry. Computer simulation tools are very effective tool for the analysis of any thermodynamic cycle. With the help of thermodynamic analysis it is easy to choose best thermodynamic cycle out of the cycles available in the market. The cycle performance is dependent upon a large number of parameters and cycle pressure ratio is one of them. For a CHP system it is desired to calculate the optimum cycle pressure ratio. Exergy analysis of the cycle cannot be neglected as it is more effective tool of analysis and the reasons for the inefficiency may be rectified. The both of the above mentioned objectives are achieved in the present work. Hence the proposed methodology may be used for the more complex systems also.

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TABLE I The exergy destruction rate and exergy efficiency equations for plant components

Components	Exergy Destruction Rate	Exergy Efficiency
Air Compressor	$e_{D,AC} = e_1 - e_2 + \dot{W}_{AC}$	$\eta_{ex,AC} = \frac{e_2 - e_1}{\dot{W}_{AC}}$
Combustion Chamber	$e_{D,CC} = e_3 + e_f - e_4$	$\eta_{ex,CC} = \frac{e_4}{e_3 + e_f}$
Gas Turbine	$e_{D,GT} = (e_4 - e_5) - \dot{W}_{GT}$	$\eta_{ex,GT} = \frac{\dot{W}_{GT}}{e_4 - e_5}$
Regenerator	$e_{D,R} = (e_2 - e_3) + (e_5 - e_6)$	$\eta_{ex,R} = 1 - \frac{e_{D,R}}{\sum_{i,R} e}$
Steam Generator	$e_{D,SG} = (e_6 - e_7) - \mathrm{m}_w(e_9 - e_8)$	$\eta_{ex,SG} = 1 - \frac{e_{D,SG}}{\sum_{i,SG} e}$

TABLE II Operating state of affairs for Cogeneration system

Name	Unit	Value
Output Power	MW	30
Process Heat	kJ	37722
Lower Heating Value of fuel	kJ/kg	50196.96
Pressure loss in regenerator air side	%	5
Pressure loss in regenerator flue gas side	%	3
Pressure loss in combustion chamber	%	5
Pressure loss in Steam Generator	%	5

TABLE III Value of different variables with change in cycle pressure ratio for the cogeneration cycle without regenerator

Cycle pressure ratio	**15**	**20**	**25**	**30**	**35**	**36**	**40**
$e_{D,CC}$ (kJ/kg)	29851	27759	26583	25896	25515	25466	25347
$e_{D,COMPRESSOR}$ (kJ/kg)	1969	2037	2107	2180	2255	2291	2332
$e_{D,SG}$ (kJ/kg)	9645	8491	7666	7068	6634	6563	6321
$e_{D,GT}$ (kJ/kg)	2808	2903	3001	3103	3208	3231	3317
$e_{D,CYCLE}$ (kJ/kg)	44273	41190	39357	38247	37612	37551	37317
Mass of Air (kg/s)	83.4	84.13	87.23	91.17	95.76	96.74	100.9
Mass of Fuel (kg/s)	1.83	1.75	1.70	1.69	1.68	1.68	1.69
Air-Fuel ratio	45.62	48.21	51.19	54.04	56.86	57.45	59.67
Cycle efficiency	74.07	77.60	79.47	80.27	80.41	80.41	80.08
Heat Rate (kJ/kg)	359.71	356.59	343.92	329.06	313.28	310.11	297.32