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Exergy and Efficiency Analysis of Combined Cycle Power Plant

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Abstract— **Combined cycle power plants with a gas turbine topping cycle and steam turbine bottoming cycle are widely used due to their high energy efficiencies. Combined cycle cogeneration has the possibility to produce power and process heat more efficiently, leading to higher performance and reduced greenhouse gas emissions. The objective of the present work is to evaluate the performance of a combined cycle cogeneration configuration based on energy and exergy analyses approaches. The effect of operating conditions on combined cycle efficiency, combined cycle cogeneration efficiency, power outputs and exergy destruction are investigated. The operating conditions investigated include gas turbine pressure ratio, gas turbine inlet temperature, process heat load and ambient conditions. It is demonstrated that a combined cycle cogeneration unit, operates more efficiently and produces less carbon dioxide than two separate, power production and process heat systems. The exergy analysis identifies the sources of irreversibility in the system and aids in the evaluation of losses and outputs by examining their quality. Exergy analysis of the combined Brayton/Rankine power cycle of NTPC (National Thermal Power Corporation) Dadri India is done. Theoretical exergy analysis is carried out for different combined cycle power plant which consists of a gas turbine unit, heat recovery steam generator without extra fuel consumption and steam turbine unit. The results pinpoint that more exergy losses occurred in the gas turbine combustion chamber reaching 35% of the total exergy losses, while the exergy losses in the other plant components are between 7% and 21% of the total exergy losses at 1400°C turbine inlet temperature and pressure ratio 10. The paper also considered the effect of the pressure ratio, turbine inlet temperature, pressure drop in combustion chamber and heat recovery steam generator on the exergy losses in the plant. Significant exergy loss is observed on changing pressure ratio and inlet temperature.**

Index Terms— Exergy Analysis, Combined Cycle Power Plant, Efficiency Analysis, Exergy Destruction, First Law Efficiency, Second Law Efficiency, Irreversibility.

1 Introduction

India is a rapidly developing economy, with a need for reliable and efficient supply of electricity and to be a power sufficient country is one of its prime concerns. The present installed capacity of electricity in India is 147,402.21MW which gives the per capita consumption of power in 2009-10 as calculated by the Central Electricity Authority about 720 kWh. While the per capita consumption of power in developed countries like U.S. is 13,338 kWh. The National Electricity Policy envisages that the per capita availability of electricity will be increased to over 1000 units by 2012. So large number of new power projects are currently in progress. Thermal power plants account for 75% of the installed capacity and gas power plant accounts 10% of the total power capacity of India. Because of this higher efficiency gas turbine and combined cycle power plants are becoming more and more attractive with regard to reduced fuel consumption and less emissions. As many as 56 thermal power stations in the country were left with a critical coal stock of less than seven days at the end of October 2008. High generation, inadequate linkages and most importantly insufficient coal supply were the major reasons behind the critical stock of coal at these power stations. In Singrauli region, there are many power plants like LNCO power 1320MW, ESSAR power 1200MW and NTPS Vindhyachal which are suffering due to coal crises. NCL (CIL) stopped supplying coal to LANCO power and ESSAR power. In western region Korba (CG) and Dahanu (Maharastra) stations received coal supply of only 77 percent of their linkages, the Chandrapur and Khaperkheda II (Maharastra) stations had insufficient linkage. The Ennore (North Chennai) and Mettur power stations in south received coal supply of 64 percent, 66 percent and 75 percent of their linkages, respectively. Kothagudem (AP) station suffered due to high generation and inadequate linkage. Coal supply to the Kahalgaon (Bihar) and Farakka station in the eastern region received only 69 percent and 62 percent of their linkage respectively. IB Valley (Orissa), Talcher TPS, Tenughat (Bokaro) and Budge plants suffered due to insufficient linkages.

A basic gas turbine cogeneration system consists of a gas turbine cycle (compressor, combustion chamber and expander), a heat recovery system for steam production and steam turbine. Fuel is introduced into the combustion chamber of the gas turbine where combustion takes place with compressed air coming out from the compressor. Hot exhaust gases from the gas turbine are the waste heat sources for process heat production. The quantity and quality of process heat produced depend on the temperature of the hot exhaust gases entering the

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heat recovery system and the resulting temperature of the steam produced. Steam produced can be used either for process heat or electric power that is generated by a steam turbine. Dadri power plant (NTPC), steam is used for power production. To sum up, Dadri power plant is based on gas turbine electric power production by steam turbine. NTPC (National Thermal Power Corporation) was set up in the public sector in the 1975. Only PSU (Public Sector Unit) to achieve excellent rating in respect of MOU (Millions of Unit) targets signed with Government of India each year. Today NTPC contributes more than 3/5th of the total power generation in India. The schematic diagram of NTPC Dadri (Gas Unit) combined power cycle is shown in figure 1. The gas turbine (Siemens AG, Germany, V 94.2, 131.3 MW) is shown as a topping plant, which forms the high temperature loop, whereas the steam plant (BHEL India, 146.5 MW, two cylinder condensing reaction 2x22 number of stages HP) forms the low temperature loop.The connecting link between the two cycles is the heat recovery steam generator (BHEL Trichy, Vertical forced Circulation) working on the exhaust of the gas turbine. A gas turbine cycle consists of an air compressor (Siemens KWU, Type Multistage Axial Flow, Number of stages: 16), a combustion chamber (Vertical Silo Type) and a gas turbine. The turbine's exhaust-gas goes to a heat-recovery steam generator to generate superheated steam. That steam is utilized in a standard steam power cycle, which consists of a turbine, a condenser (BHEL, Rectangular, SCD-1200) and a pump (BHEL Hyderabad, Vertical Mixed Flow). Both the gas and steam turbines drive electric generators.

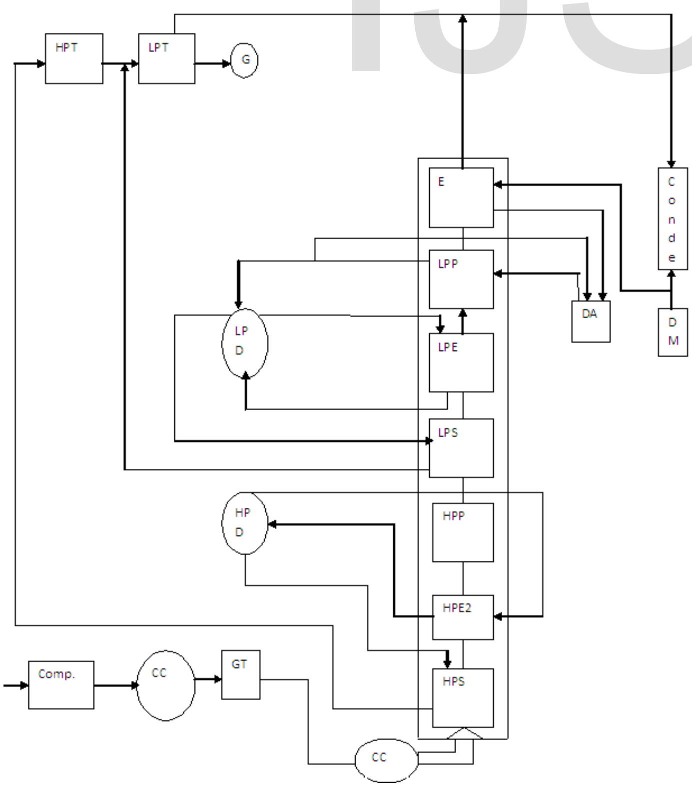

Fig. 1 Schematic diagram of NTPC Dadri.

2 Exergy Analysis

Exergy is a generic term for a group of concept that defines the maximum possible work potential of a system. In an open flow system there are three types of energy transfers across the control surface namely working transfer, heat transfer and energy associated with mass transfer and /or flow. The work transfer is equivalent to maximum work which can be obtained from that form of energy. The exergy of heat transfer (Q) from the control surface at temperature T is determined from maximum rate of conversion of thermal energy to work W_{max} is given by:

$$W_{max} = Q\,(1 - T_0/T)$$

Exergy is defined as the maximum work that may be achieved by bringing a system into equilibrium with its environment. Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design and improvement of energy systems. Exergy analysis is based on both first and second law of thermodynamics. Exergy analysis can clearly indicate the location of energy degradation in a process. The exergy method is a useful tool for furthering the goal of more efficient energy resource use, for it enables the locations, types and magnitudes of wastes and losses to be determined. Many engineers and scientists suggest that the thermodynamic performance of a process is best evaluated by performing an exergy analysis in place of conventional energy analysis because exergy analysis appears to provide more insights and to be more useful in furthering efficiency improvement efforts than energy analysis. The main purpose of exergy analysis is to identify the causes, types, location and to calculate magnitude of thermal losses.

3 Exergy Analysis Calculation

Energy analysis is based on first law of thermodynamics, which is related to the conservation of energy. While exergy analysis is based on second law of thermodynamics which state the conservation of mass and degradation of the quality of energy along with the entropy generation in the analysis design and improvement of energy systems. The second law analysis, i.e. the exergy analysis, calculates the system performance based on exergy, which is defined as the maximum possible reversible work obtainable in bringing the state of the system to equilibrium with that of environment. In the absence of magnetic, electrical, nuclear, surface tension effects, and considering that the system is at rest relative to the environment, the total exergy of a system can be divided into two components: physical exergy and chemical exergy.
The second law efficiency is defined as:

$$\eta_{2nd} = \frac{\text{actual thermal efficiency}}{\text{maximum possible thermal efficiency}}$$

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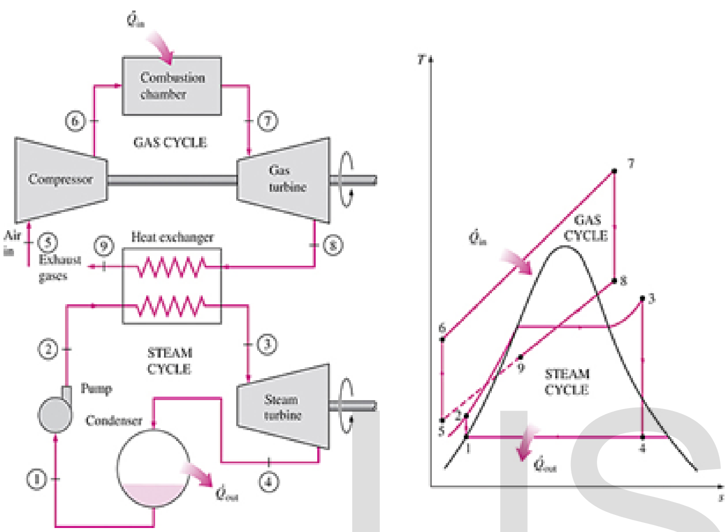
A combined cycle power plant as shown in figure 2(a & b) includes both the brayton cycle and rankine cycle. It joins operation of the gas turbine at the "hot end" and the steam turbine at the cold end. As shown in the diagram 2(b),which is operating temperature and entropy diagram, gas turbine operates on the brayton cycle, i.e., intake air compressed nearly isentropically from point 5 to 6, combusted at constant pressure from point 6 to 7, and then expanded nearly isentropically in the gas turbine from 7 to 8, exhausting gas from point 8, the brayton cycle applied to gas turbine is an open cycle , temperature at point 8 is higher than 1000°C and does not form a closed loop with the inlet air of point 1.[1]



(a) (b)

Fig. 2(a & b) T-S diagram of combined cycle power plant.

The useful products of a CCPP are electrical energy, W_E the thermal energy, Q_P in the form of superheated steam. The thermodynamics performance is the based on the first law efficiency and is defined as:

$$\eta_{1st} = \frac{W_E + Q_P}{E_F} \tag{1}$$

For simple cycle W_E is equal to W_{GT} and for combined cycle W_E is equal to W_{GT} + W_{ST}. Actually the efficiency of a CCPP is reduced by the various inherent losses. One unavoidable loss is heat lost by radiation and convection, while a second is the internal loss caused by irreversible processes as discussed in the second law of thermodynamics.
The exergy of the steam / water is defined as: [2]

$$\varepsilon = m(h - T_O s) \tag{2}$$

Where ε is exergy, h is enthalpy, To (K) is the ambient temperature and s is the entropy (kJ/kg K).The exergy of steam / water produced is:

$$B_p = m[(h_i - h_o) - T_o(s_i - s_o)] \tag{3}$$

Where m is the mass of steam /water, s_i is the entropy of inlet condition for the process, s_o is the entropy of outlet condition for the process and T_o is the temperature of environment. [2][1]. The first part of the above mentioned equation represents the energy of the process heat which is:

$$Q_p = m(h_i - h_o) \tag{4}$$

The process heat generator (HRSG) consists usually of three main parts an economizer, an evaporator and the last is superheater, all working at a process heat pressure. The water at T_c enters the economizer and exits at T_f at saturated liquid state. The saturated liquid enters the evaporator and exits at the same temperature at saturated vapor state; finally the saturated steam enters the superheater and exits at superheat temperature. [2][13] Energy balance at the system consisting of superheater and evaporator is:

$$m(h - h_f) = m_a(1 + r_{fa})(h_t - h_{pp}) \tag{5}$$

Where m_a is the mass of air in the gas turbine engine, r_{fa} is the fuel air ratio used in the combustion process, h_t is the enthalpy of gas mixture at turbine exit, and h_{pp} is the enthalpy of gas mixture at pinch point temperature. [13] To have a better assessment, some useful ratios such as process heat exergy factor and power to heat ratio are defined as: [2]

$$\varepsilon_p = \frac{B_P}{Q_P} \tag{6}$$

$$r_{ph} = \frac{W_E}{Q_P} \text{ For simple cycle,}$$

Where, W_E and Q_P are electrical energy in (kJ) or (kWh) and thermal energy of process heat respectively.

$$r_{ph} = \frac{W_E}{Q'_P} \text{ For combined cycle,}$$

Where, $Q'_P = Q_P - W_{ST}$,W_{ST} is electrical energy of steam turbine in (kJ) or (kWh).

Exergy is essential concept in second law analysis. If less exergy is consumed, a cycle can produce more efficiency. Therefore, by using exergy to evaluate the power plant cycles, a more accurate performance of system can be obtained. The exergy factor of generated steam / water (ε_s) and the exergy factor of fuel input (ε_f) can be expressed as follows: [2]

$$\varepsilon_s = \frac{B_P}{Q_P}$$

$$\varepsilon_F = \frac{B_f}{Q_f} = 1$$

$$\eta_{2nd} = \frac{\eta_{1ST}}{\varepsilon_f} * [(r + \varepsilon_s)/(r + 1)] \tag{7}$$

4 Efficiency of Components of Power Plant

Compressor efficiency, superheater efficiency and economizer efficiency of gas turbine of combined cycle is given by following equation which is taken from NTPC technical directory. [20]

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$$T_{inst} = C_{in}T * \left(\frac{P_2}{P_1}\right)^{\frac{\Upsilon-1}{\Upsilon}}$$

$$\text{Compressor Efficiency} = \eta = \left[\frac{T_{inst} - T_{ci}}{T_{co} - T_{ci}}\right] * 100$$

$$\text{Superheater Efficiency} = \eta = \frac{H_{is} - H_{os}}{m * C_p * (T_{is} - T_{os})} * 100$$

$$\text{Economiser Efficiency} = \eta = \frac{H_{ie} - H_{oe}}{m * C_p * (T_{ie} - T_{oe})} * 100$$

5 Exergy Destruction and Exergy Loss

Exergy is only conserved for reversible process but partially consumed in an irreversible process. Thus, exergy is never in balance for real processes, it is a measure of quality and quantity of energy. For real process the exergy input always exceeds the exergy output. This imbalance is due to irreversibility which is called exergy destruction. Unlike energy, exergy is not conserved but it is destroyed by irreversibility within system. This irreversibility may be classified as internal and external irreversibility. Main sources of internal irreversibility are friction, unrestrained expansion, and chemical reaction. External irreversibility arises due to heat transfer through a finite temperature difference. Exergy is lost when the energy associated with a material or energy stream is rejected to the environment. [10][3][17]

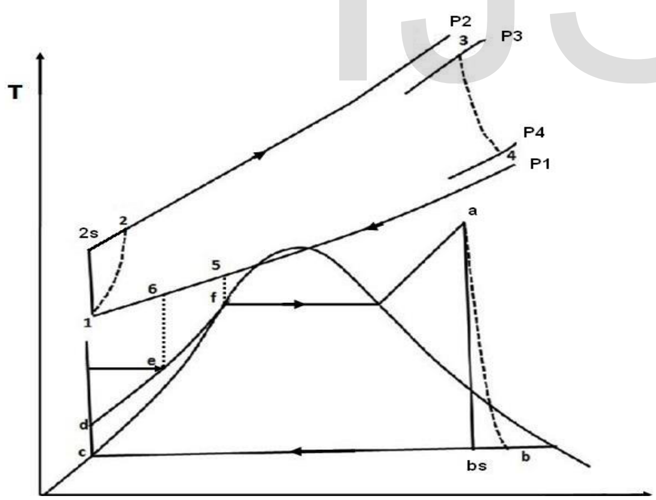

Fig. 3 Temperature entropy diagram of brayton / rankinecombuned cycle power plant.

5.1 Gas Turbine (GT):- Exergy loss due to irreversibility in gas turbine is given by:

$$I_{GT} = m_g * T_0 * (s_4 - s_3)$$

$$\text{where}(S_4 - S_3) = C_{pg} \ln\frac{T_4}{T_3} - R_g \ln\frac{P_4}{P_3}$$

$$R_g = C_{pg} * \frac{\Upsilon - 1}{\Upsilon}$$

5.2 HRSG: - The exergy loss due to irreversibility in heat recovery steam generator is given by:

$$I_{HRSG} = T_{0*}\Delta S_0 = T_0\left[m(s_a - s_e) + m_g(s_6 - s_4)\right]$$

$$I_{HRSG} = C_{PG} * \ln\frac{T_6}{T_4} - R_g * \ln\frac{P_6}{P_4}$$

5.3 Steam Turbine (ST):- The exergy loss due to irreversibility in steam turbine is given by:

$$I_{ST} = m_s * (s_b - s_a) * T_0$$

Since isentropic efficiency of ST is,

$$\eta_{ST} = \frac{h_a - h_b}{h_a - h_{bs}}$$

$$h_b = h_f + x_b * h_{fg}$$

$$s_b = s_f + x_b * s_{fg}$$

5.4 Exhaust Loss:- The exergy loss due to irreversibility in exhaust is given by:

$$I_{exh} = \int_{T_6}^{T_0}\left(1 - \frac{T_0}{T}\right)dQ = m_g * C_{pg}\left[(T_6 - T_0) - T_0 \ln\frac{T_6}{T_0}\right]$$

5.5 Compressor: - The exergy loss due to irreversibility in compressor is given by:

$$I_C = m_a * (s_2 - s_1) * T_0$$

$$(s_2 - s_1) = C_{pa} \ln\frac{T_2}{T_1} - R_a \ln\frac{P_2}{P_1}$$

$$R_a = C_{pa} * \frac{\Upsilon - 1}{\Upsilon}$$

Where T_1 is compressor inlet temperature in GT and T_2 is compressor outlet temperature in gas turbine (in K).

5.6 Combustion Chamber (CC):-The exergy loss due to irreversibility in combustion chamber is given by:

$$I_{CC} = T_0\left[\left(m_g C_{pg} \ln\frac{T_3}{T_0} - m_g R_g \ln\frac{P_3}{P_0}\right) - \left(m_a c_p \ln\frac{T_2}{T_1} - m_a R_a \ln\frac{P_2}{P_1}\right)\right]$$

5.7 Total Exergy Destruction:-

$$I_{TOTAL} = I_C + I_{COMB} + I_{GT} + I_{HRSG} + I_{ST} + I_{EXH}$$

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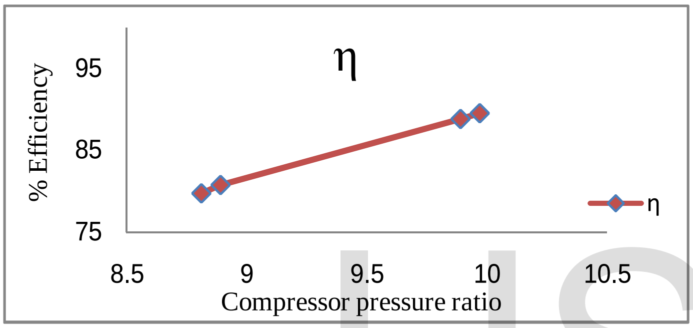
6 Results and Discussion

Based upon the methodology developed and equation derived here, the efficiency of each component of combined cycle power plant NTPC Dadri is calculated and shown here graphically. Exergy loss due to irreversibility in different components of gas / steam combined power cycle is shown graphically.

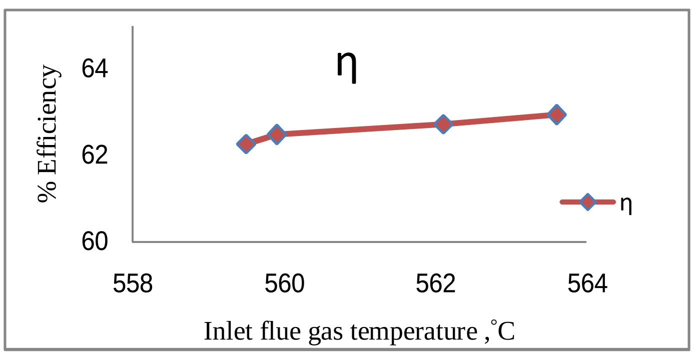
6.1 Compressor Pressure Efficiency

This graph shows variation of compressor efficiency with respect to compressor pressure ratio. In Dadri plant compressor ratio is varied between 8 to 10 as compressor ratio increases, efficiency of compressor also increases.



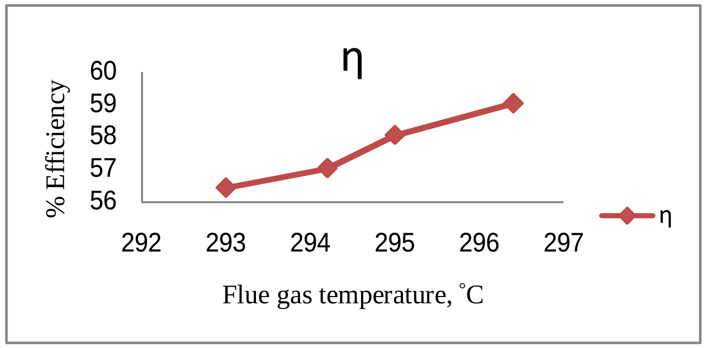
6.2 HP Superheater Efficiency Variation

This graph shows variation of efficiency w.r.t flue gas temperature. Superheater efficiency increases by increasing input steam pressure and input flue gas temperature. HP Superheater efficiency of HSRG of Dadri power plant is around 62% to 63%.



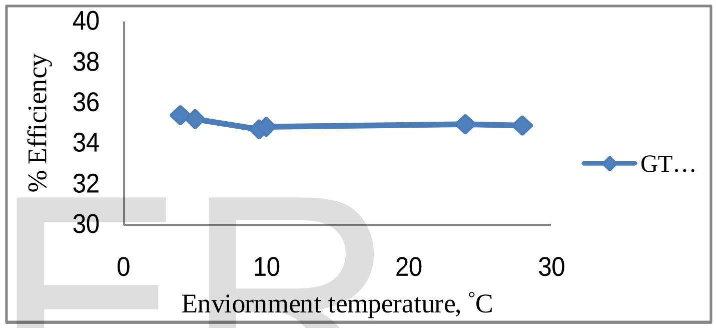
6.3 HP Economizer Efficiency Variation

This graph shows the variation of efficiency w.r.t inlet flue gas temperature. In NTPC Dadri HP economizer efficiency varies between 56% to 59%. As inlet flue gas temperature increases with low steam pressure, efficiency of HP economizer also increases.



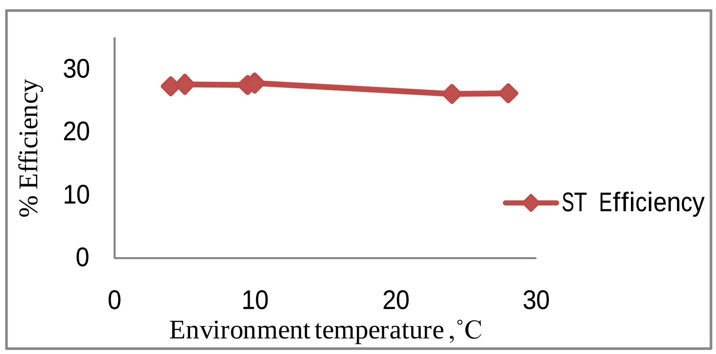
6.4 GT Efficiency Variation w.r.t Environment Temperature

This graph shows variation of efficiency of gas turbine w.r.t environment temperature in °C. Gas turbine efficiency of combined cycle power plant Dadri is about 35% to 40% level.



6.5 ST Efficiency Variation w.r.t Environment Temperature

This graph shows variation of efficiency of steam turbine w.r.t environment temperature in ˚C steam turbine efficiency of combined cycle power plant Dadri is about 25% to 30% level



6.6 First Law and Second Law Efficiency Variation of CCPP

First law efficiency in the current investigation comes out to be 45%. This indicates that plant is producing energy by using 45% of fuel but in real it is not true because first law tells only about capabilities of power plant which it can attain. However, when the same is calculated with the help of 2nd law, quality of energy is known. It means that if our second law effi-

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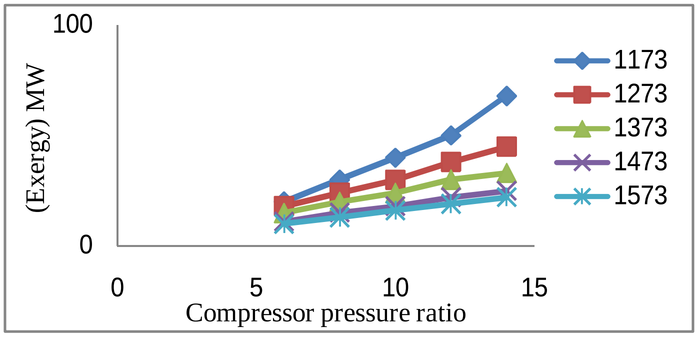
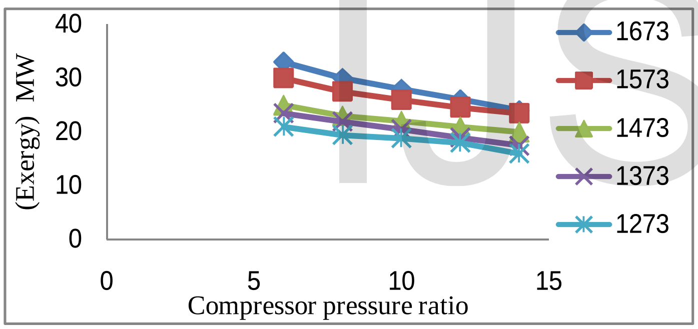
ciency is high, we can use the efficiency more qualitatively. Losses will low and less heat will be dissipated to the atmosphere. For Dadri power plant, second law efficiency is around 24%. To improve second law efficiency, HRSG unit's capacity can be increased by using economizer. More energy thus coming out from exhaust gas can be extracted for steam turbine.



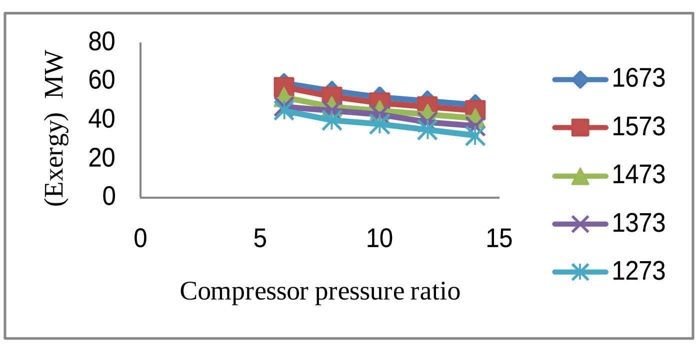
6.7 Graph Showing Exergy Loss in HRSG at Different TIT

Graph is plotted between exergy loss and compressor pressure ratio in a heat recovery steam generator for varying turbine inlet temperature. As pressure ratio increases, the gas turbine exhaust temperature decreases and thereby decrease in exergy destruction in WHRB. In HRSG 15% to 31% of exergy loss is calculated.



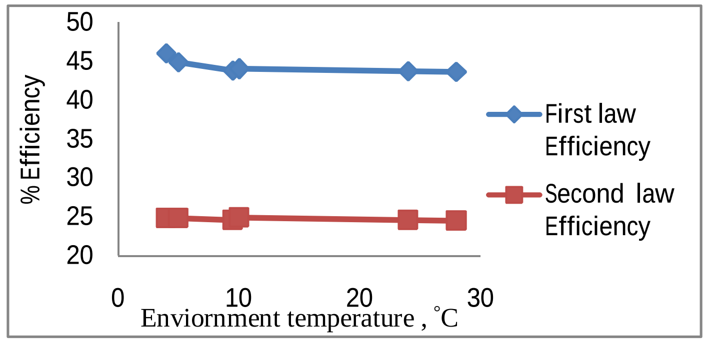
6.8 Graph Showing Exergy Loss in Gas Turbine at Different TIT

Graph shows the variation of exergy loss in gas turbine with variation in compressor pressure ratio and turbine inlet temperature. As pressure ratio increases, the exergy destruction in the turbine for all cases increases, because the entropy at turbine inlet decreases. In gas turbine, 2-7% of exergy loss is calculated.



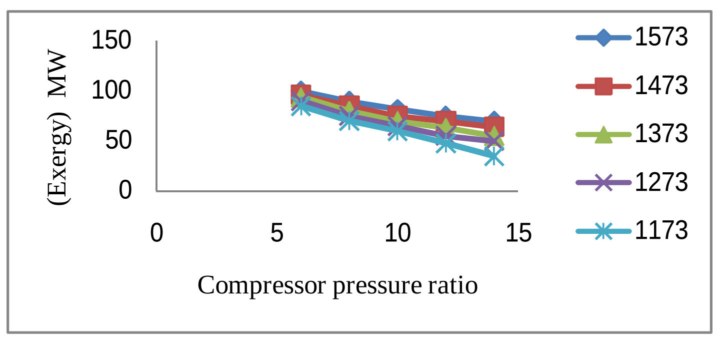
6.9 Graph of Exergy Loss in Steam Turbine at Different TIT

Graph shows the variation of exergy loss in steam turbine with variation in compressor pressure ratio and turbine inlet temperature. At low turbine inlet temperature (900˚C) the exergy loss in the steam turbine is lower and as the turbine inlet temperature (1400˚C) increases, the exergy loss is increasing rapidly. There is similar pattern in exergy losses that can be seen at temperature 1000 to 1400˚C but at low turbine inlet temperature, the pattern is slightly different. At any particular turbine inlet temperature as the compressor pressure ratio increases the exergy loss decreases slowly. In steam turbine 5% to 15% exergy loss is calculated.



6.10 Graph of exergy loss in combustion chamber at different TIT

Graph shows the variation of exergy loss in combustion chamber with variation in compressor pressure ratio and turbine inlet temperature. As pressure ratio increases, mass flow rate of air decreases so exergy destruction rate also decreases. At any particular turbine inlet temperature as the compressor pressure ratio increases the exergy loss decreases. In combustion chamber 35% to 50% of exergy loss is calculated.



7. Conclusion

For Dadri power plant, the first law efficiency is around 45% and that associated with second law is 24%. NTPC Dadri can use 45% of its energy source to convert to real work. The rest is consumed by system losses and irreversibility.

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To improve combined cycle power plant efficiency:-

1. The gas turbines are major power producer so it is essential to improve the efficiency of gas turbine.
2. Heat recovery steam generators are the main equipment for combined cycle power plant. Therefore, proper designing of HRSG is required to use all energy coming from exhaust gases.
3. More exergy losses occur in the combustion chamber due to combustion irreversibility and this must be reduced with the aid of new advances in the technology.
4. At higher turbine inlet temperature and lower pressure ratio, the exergy losses in the combustion chamber, heat recovery steam generator and steam turbine are higher.
5. At lower turbine inlet temperature and higher pressure ratio the exergy losses in the gas turbine is higher.

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Table 1 Compressor efficiency analysis of gas turbine.

FREQUENCY: 50 Hz	**COMPRESSOR EFFICIENCY**				
	Day				
PARAMETER	**SYMBOL**	**10/12/2012**	**14/01/2013**	**13/04/2013**	**25/05/2013**
Comp. Inlet Temp ˚C	t1	14	13.94	36.25	36.25
Comp. Inlet Temp K	T1	287	286.94	309.25	309.25
Comp. Outlet Temp ˚C	t2	331	331	347	351
Comp. Outlet Temp K	T2	604	604	620	624
Ambient Pressure (mbar)	-	993	993	993	993
Comp. Inlet Negative Pressure (mbar)	-	8.12	8.29	16.04	16.04
Comp. Inlet Pressure	p1	0.985	0.985	0.985	0.985
Comp. DischargePressure	p2	9.827	9.742	8.685	8.758
Compression Ratio	p2/p1	9.976	9.890	8.817	8.891
Gamma	ϒ	1.401	1.401	1.401	1.401
	ϒ-1/ϒ	0.286	0.286	0.286	0.286
Instant Temperature	T_{2s}	571.04	568.70	557.14	563.93
Compressor Efficiency	η	89.574	88.839	79.758	80.77
Load (KW)	-	133.6	133.6	115.6	115.6
Gas Flow Rate (m^3/hr)	-	41.89	41.89	38.16	39.28
Instant Heat Rate (GCV) (Kcal/Kwhr)	-	3002.53	3002.53	3002.53	3002.53

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Table 2 Exergy and efficiency analysis of component of heat recovery steam generator.

Equipment		M_{fg}(kg/s)	M_s(kg/s)	P	T °C (fg)	T °C (st)	h(kj/kg)	H (kW)	s(kj/kg.K)	ε	Mfg*Cp*ΔT	η (%)
HP- Superheater	I	471.59	61.75	66	559.5	281.5	2777.42	171505.7	5.83	1028.4	70370.6598	62.28031
	O	471.59	61.75	65	476.6	530.1	3487.17	215332.7	6.92	1411.2		
HP- Economizer	I	471.59	61.75	100	294.1	156.9	663.91	40996.4	1.89	96.91	60948.2916	57.04968
	O	471.59	61.75	97	222.3	281.3	1227.43	75767.2	3.06	309.3		

Table 3 Exergy analysis for steam cycle of combined cycle power plant.

Exergy analyses for steam cycle of power plant							November					
Equipment		Environment condition		T=	4°C		277 K		h			
			M(kg/s)	T	P(Mpa)	h(kj/kg)	H(Kw)	Q_p(Kw)	s(kj/kg.K)	ε(kj/kg)	Bp (kW)	ΣBp (kW)
HRSG	HP	I	11.06	31.4	0.11	151.35	1673.931	34616.69	0.4964	13.8472	185.98	
		O	11.06	435.5	5	3281.25	36290.63	-	6.7685	1406.376	1578.5	
	LP	I	3.67	31.4	0.11	151.35	555.4545	10129.46	0.4964	13.8472	185.98	
		O	3.67	226.6	0.55	2911.42	10684.91		7.1718	924.8314	1096.96	-2303.5
STG	HP	I	11.06	435.5	5	3281.25	36290.63	46668.26	6.7685	1406.376	1578.5	
	LP	I	14.72	226.6	0.55	2911.42	42856.1	-	7.1718	924.8314	1096.96	
	ST	O	14.72	36	0.06	2204.34	32447.88	-	8.3376	-105.175	66.96	2608.5
CONDENSER		I	14.72	34.7	0.05	2204.34	32447.88	-30510.1	8.3376	-105.175	66.96	
		O	14.72	3140	0.05	131.64	1937.741	-	0.4561	5.3003	177.43	110.47
CONDENSATE PUMP		I	14.72	31.4	0.05	131.64	1937.741	16.7808	0.4561	5.3003	177.43	
		O	14.72	31.4	1.14	132.78	1954.522		0.4561	6.4403	178.57	1.14