International Journal of Technical Research and Applications e-ISSN: 2320-8163,
www.ijtra.com Special Issue 19 (June, 2015), PP. 12-17

ENERGY AUDIT OF COMBINED CYCLE POWER PLANT: A CASE STUDY

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Abstract-
This study involves the complete energy audit of the combined cycle 450 MW power plant. It includes a detailed energy balance and performance analysis of different cycle components that includes Gas Turbine, Steam Turbine and HRSG Unit. Firstly, Gas Turbine has been analysed as a single independent unit. Various parameters that affect its performance include the fuel GCV, compressor performance, and ambient air conditions etc. In this study, all such factors are examined for their effect on the overall plant performance. Secondly, Steam Turbines section has been examined which also includes the condenser and the entire feed-water circuit. Lastly, HRSG unit is considered where an indirect method of energy balance has been employed, where prime focus would be at the various losses occurring during the run time. Heat losses have also been addressed which might occur due to insufficient or improper insulation of the entire instalment. In short, the prime purpose of this study is to evaluate the plant performance and pin point areas which require improvement.

I. INTRODUCTION

To improve and monitor the efficiency of a power plant, energy audit is carried out with an aim to save fuel and reduce emissions. For a thermal power plant, the specific objective is to improve the heat rate and reduce auxiliary power consumption of the power plant. In contrast to the performance guarantee test, the energy audit is carried out by using high precision equipments and is necessary for longer operations of the plant and variation in quality of fuel. Such a detailed energy analysis has never been conducted before in Pakistan. A mainstream plant has been selected for an operational analysis and its impact on the improvement in terms of the energy exported on the grid.

The present situation of Pakistan in terms of energy generation is not encouraging. There are serious energy shortfalls which are posing great concerns to the economic situation of the country. Therefore, the objective of the present study is to drive the attention of the authorities to implement such exercise of energy audit of the existing power plant as an example that should be practiced widely in the energy generating sector, locally. Tackling energy scarcity issue in Pakistan by improving grid output of Rousch power plant to give an idea of the impact it would have on the country, thus justifying this energy audit. The reason for the selection of this plant is the complexity it offers, being a combined cycle plant. In addition, access to technical information and design values were readily available from the plant site, unlike in many other plants present in the country.

II. MATERIALS AND METHODS

The power plant under consideration is 450 MW combined cycle power project. The plant consists of two gas turbines of 150 MW each and one steam turbine of 150 MW. Due to the availability of water from the nearby canal, this plant has been designed to operate on it. Average total electricity generated by the plant is about 10541.61 MWh; whereas power exported to the grid is 10222.67 MWh and auxiliary consumption is about 15.6 MW for every 450 MW generated. Natural gas consumption is about 79.16 MMSCF per month.

The plant has been divided into two parts of bottoming and topping cycle. The analysis commences with the topping cycle where compressor performance is evaluated, which is then linked to the overall gas turbine (GT) performance. Afterwards, bottoming cycle is taken into account which includes steam turbine performance and the entire heat recovery steam generator (HRSG), feedwater and condenser section associated with it.

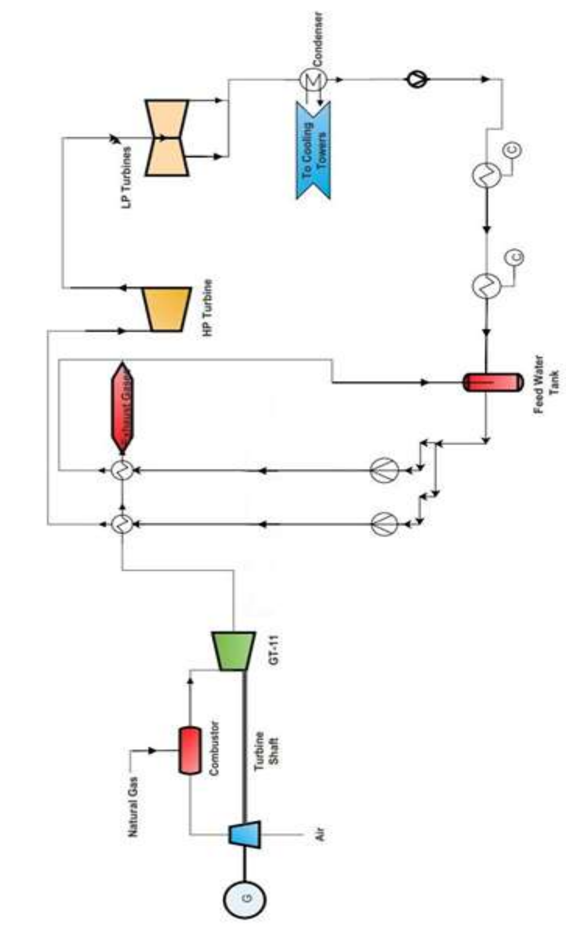

FIGURE 1: PFD of Rousch Power Plant

A. *Topping Cycle*

The topping cycle comprises of the entire gas turbine unit. This includes the compressor, the combustor and the turbine itself.

1. *Compressor*

The compressor is a ten stage axial flow type, which takes the air at ambient conditions, filter and then compress it to achieve ten times higher pressure than the atmospheres. Its efficiency can be calculated using the relation:

$$\eta_c = T_1 \frac{\{(P_2/P_1)^{(1-\frac{1}{Y})} - 1\}}{T_2 - T_1} \qquad (1)$$

The results of the data are shown in Table 1.1.

Parameter	Unit	Comp-1 (DAY-1)	Comp-2 (DAY-1)	Comp-1 (DAY-2)	Comp-2 (DAY-2)
Ambient Air Pressure	*bar*	1.003	1.003	1.009	1.009
Pressure drop across filters	*mBar*	4.56	4.51	4.62	4.55
Inlet Air pressure (P_1)	*bar*	0.9984	0.9985	1.0044	1.0045
Discharge Air pressure (P_2)	*Bar*	9.9640	9.9660	9.9710	9.9750
Air inlet temperature (T_1)	°C	30	30	30	30
Air outlet temperature (T_2)	°C	334.3	336.1	340.0	341.1
Efficiency	%	92.56	92.02	90.58	90.27

Table 1.1 Gas Turbine's Compressor Operating Data

2. *Combustor*

Performance of Combustor firstly requires a detailed analysis of the fuel using the gas chromatogram tabulated in Table 1.2 [1]. Afterwards, calorific value of the fuel is assessed and tabulated in Table 1.3 [1]. The formula, used to evaluate the basic enthalpy released during the combustion of fuel, makes use of the gross calorific value (GCV) value of the fuel, which is natural gas (methane).

Parameter	Actual (% mol)	Design (% vol)
Methane	88.01	≥90.00
Ethane	0.996	≤15
Propane	0.24	Not Applicable
Hydrogen	0.0	≤1.0
Sulphur Dioxide	0.0	Not Applicable
Nitrogen	7.51	≤10.0
Oxygen	0.0	≤0.1
Moisture(H20)	0.01	≤0.1
Carbon Dioxide	2.89	≤10.0
Other Hydrocarbons	0.344	Not Applicable

Table 1.2 Ultimate Analysis (% Mole)

Parameter	Units	Combustor -1	Combustor-2
Volumetric Flow	m^3/s	579.2	579.2
Density of fuel gas at 15°C	Kg/m^3	0.811	0.811
Calculated LHV at constant pressure	kJ/kg	40575	40575
Enthalpy of fuel gas at temperature at which flow is measured (H_i)	kCal/kg	48483.0	48483.0

Table 1.3 Gaseous Fuel – Natural Gas

3. *Gas Turbine*

The turbine section is the core of the complete operation of the topping Cycle. Some factors need to be taken care of during its operation. As for the startup, there is a particular situation that is potentially harmful, known as surging [6], which, if encountered more frequently, can potentially decrease the operating efficiency of the gas turbine. Another important aspect that must be taken care of is the turbine inlet temperature [1]**.** Since the basic thermodynamic relations do not depict a complete picture, the relation should not be always used to predict the actual behavior. Generally, it has been observed that during the operation, if the exiting gas temperature is higher than the design temperature from the combustion chamber, it has detrimental effects on the turbine section overall. As a consequence, there is a serious limit imposed on the lifetime of the equipment.

As shown in Table 1.4, several of the parameters collected are related in the evaluation of the turbine performance. The formula that was used in evaluating the operating efficiency comes from basic thermodynamics as given in [1]:

$$\eta_{th} = \frac{(860)(W_{out})}{H_i} \qquad (2)$$

Table 1.5 gives the results that were obtained by applying the above relation. A significant difference between the calculated and design heat rates can be observed.

Parameter	Unit	GT-1 (DAY-1)	GT-2 (DAY-1)	GT-1 (DAY-2)	GT-2 (DAY-2)
Relative Humidity	%	52.14	52.14	60.47	60.47
Barometer Pressure	mm Hg	760	760	761	761
Power generator	MW	146	148	147.8	148
Fuel Flow Rate	Kg/s	8.581	8.8	8.8	8.8
Flue gas Exit Temperature	°C	562.2	562.2	562.2	562.2
Compressed Air Discharge Pressure	kg/ cm^2	9.96	9.97	9.96	9.96
Compressed Air Discharge Temperature	°C	334.3	341.3	341.1	341.1
GT Exhaust Pressure	Bar	0.997	0.997	0.997	0.997
Air Suction Pressure	Bar	0.996	0.996	0.996	0.996
LHV of NG	kCal/ Nm^3	48483	48483	48483	48483

Table 1.4 Gas Turbine Parameters

Parameter	Load, W_{out} (MW)	Design Heat Rate (Kcal/KWH)	Calculated Heat Rate (Kcal/KWH)	Corrected Heat Rate (Kcal/KWH)	Efficiency, η_{th} (%)
GT-1 (DAY-1)	140.2	6550	7753	7653	46.43
GT-2 (DAY-1)	141.8	6550	7785	7680	46.24
GT-1 (DAY-2)	139.7	6550	7836	7763	45.94
GT-2 (DAY-2)	140.7	6550	7759	7521	47.62

Table 1.5 Gas Turbine Heat Rate & Efficiency

A. Bottoming Cycle

The bottoming cycle is much more complex and diverse than the topping cycle discussed before. There are several components that needs considerable attention namely steam turbine, HRSG, feedwater heating system and condenser. In this section, the goal is to calculate the steam turbine cylinder operating efficiency. Implications of the results are discussed later. Afterwards, feedwater parameters are measured and tabulated. Certain performance indicators are also discussed and mentioned both for condenser and feedwater. And finally, HRSG section is analyzed.

1. Steam Turbine

Steam Turbine forms the core of operations within the bottoming cycle. The operation of steam turbine is a function of many key plant design factors [2]. These factors include turbine type, pressure reheat designation, exhaust condition, extraction designations, extraction type, flow designation and shaft orientation. Each of the factors mentioned have been explained for ease of understanding.

The turbine type indicates the overall mechanical setup of the turbine and accounts for its mechanical efficiency. This also accounts for the shaft orientation which accounts whether the turbine is tandem arranged or has a cross compound arrangement, which in turn determines the mechanical efficiency of the turbine. Then the pressure reheat designation determines the available portion of heat that is made available by the steam to the turbine rotor. The flow designation is used in conjunction with it. Flow designation determines whether the turbine is single flow or double flow type. And it is imperative to be determined because the enthalpy determination process is dependent upon it. The exhaust conditions are also crucial to the overall auditing process because they determine the condenser operation and performance together with the overall work extracting potential of the turbine itself. Normally the condensing type turbine is employed, the same being employed at this power plant. All these factors determine the adiabatic, isentropic and thermal efficiency of the turbine.

Another important parameter that sets the operating efficiency is the extraction types. These extractions design the overall feedwater heater system and also contribute to the overall efficiency. Extraction designation is also important to determine the overall energy balance of the turbine. Furthermore, availability concepts of energy also hold much importance in this respect to determine the overall turbine cylinder efficiency. All factors explained above have been considered in determining the efficiencies shown in Table 1.7, which have been calculated from the measured parameters shown in Table 1.6 (b). Difference between the design values shown in Table_1.6 (a) and the performance parameters clearly shows a marked room for improvement, where the manufacturer's design values form the benchmark for the audit. The relation used in assessing the efficiency of the entire turbine section [3,4] is:

$$\eta_{t\,cyl} = \frac{h_{in}-h_{out}}{h_{in}-h_s} \tag{3}$$

This concludes our findings regarding the steam turbine section.

Parameter	Unit	Design Value
Rated output	MW	150.0
Heat rate	kCal/kWh	1550.0
Rated Steam Flow	Tonnes/h	462
HP inlet Pressure	kg/cm^2	68
HP inlet Temperature	^oC	530
HP exhaust Pressure	kg/cm^2	6.0
HP exhaust Temperature	^oC	225
LP inlet Pressure	kg/cm2	6.0
LP inlet Temperature	^oC	225
LP exhaust Pressure	kg/cm2	0.081
LP exhaust Temperature	^oC	44
Condenser Vacuum	Bar	0.081
CW inlet temperature of condenser	^oC	28.2
CW outlet temperature of condenser	^oC	40.1
CW flow through condenser	kg/s	5660
SH Spray	tonnes/h	2.31

Table 1.6 (a) Steam Turbine Generator Design Specifications

Parameter	Flow (tonnes/h)	Pressure (atm)	Tempera- -ture (^oC)	Enthalpy (kCal/kg)
Main Steam	461.7	60.0	530	3492.5
CRH-HPT exhaust	459.9	5.925	222.2	2898.6
LPT exhaust	459.6	0.091	43.3	2307.7
FW at eco inlet	460.558	77.01	106.0	450.1
SH Spray	2.31	60.0	350.0	3041.475

Table 1.6 (b) Steam Turbine Generator Measured Parameters

Parameter	Design at 0% makeup water
TG heat rate (kCal/kWh)	1624.827
TG Efficiency (%)	52.9
HRSG Efficiency (%)	84.8
Overall heat efficiency (%)	48.82
GCV of fuel (kCal/kg)	48483

Table 1.7 Steam Turbine Generator Performance Parameters

2. Feedwater Heating System

This study covers the closed type Feed water heaters and the data pertaining to the auditing of this Feedwater heater is shown in Table 1.8 [1].

Parameters	Unit	Design	Day 1	Day 2
Steam Inlet Temp.	°C	135.0	126.0	125.5
Steam Inlet Pressure	Bar	1.539	1.359	1.359
Drain Outlet Temp.	°C	106	105.5	105.5
Condensate inlet Temp.	°C	47.0	44.4	44.4
Condensate Inlet Pressure	Bar	3.2	3.169	3.18
Condensate Outlet Temp.	°C	105	104	104.4
Condensate outlet Pressure	Bar	1.210	1.208	1.211
Condensate flow	Tons/hr	515.0	512.2	513.4
Temperature Rise	°C	58.0	60.0	60.0
Steam Sat. Temp	°C	111.87	110.85	110.85
TTD	°C	6.87	5.85	5.85
DCA	°C	59.0	61.1	61.1

Table 1.8 Feedwater Heater Performance Parameters

3. Condenser

The condenser performance is another parameter that holds much importance in the overall power plant performance, which depends on parameters such as condenser arrangement, condenser operating pressure optimization, cleanliness factor, number of passes for the condenser and make up material properties [2].

The above parameters have been put in perspective during the energy audit of the plant. Nevertheless there are great many factors apart from the ones mentioned, but to a larger extent these serve to influence the overall operation. The key indicators for the condenser performance include the condenser effectiveness (ε), log-mean temperature difference (LMTD), and drain cooler approach (DCA) as given below:

1) Condenser Effectiveness

$$\varepsilon = \frac{\Delta T}{T_{sat} - T_{in}} \quad \textbf{(4)}$$

2) Log-Mean Temperature Difference

$$\text{LMTD} = \frac{\Delta T}{ln\frac{(T_{sat} - T_{in})}{(T_{sat} - T_{out})}} \quad \textbf{(5)}$$

3) Drain Cooler Approach

$$\text{DCA} = T_d - t_i \quad \textbf{(6)}$$

The results regarding condenser and its implications are discussed in the analysis section.

Particular	Unit	Design	Actual
Generation	MW	150.0	148.9
Frequency	Hz	50.0	49.8
Condenser Pressure	Bar	0.081	0.092
CW inlet Temperature	°C	28.2	30.3
CW outlet Temperature	°C	40.0	41.2
Sat. Temp	°C	41.3	43.3
Terminal Temp Difference	°C	1.3	2.1
CW Temperature Rise	°C	11.9	10.9
Condenser CW flow	Kg/s	5660	5558
CW Heat Pickup	kW	260.11	254.40
Condenser Effectiveness	%	90.8	83.2
Drain Cooler Approach	°C	77.3	75.2
LMTD	°C	5.15	5.98

Table 1.9 Condenser Performance Parameters

4. HRSG Unit

The HRSG unit does have auxiliary firing unit but presently, it is not used for auxiliary firing purposes. The Gas Turbine exhaust is directly fed into the HRSG for steam generating purposes. An indirect auditing method have been employed for the HRSG unit where the focus is on the losses [1], which includes co (carbon monoxide) loss, moisture loss in fuel and air, dry flue gas loss, blow down loss, surface heat loss. The first loss measures the amount of combustion that has been achieved in the combustion chamber of the Gas Turbine section. The moisture loss helps in determining the grade of fuel being burned. It helps in finding the energy gap that should have been used in generating steam. The dry flue gas loss is almost imperative because the heat potential of the flue gas cannot be exploited to the fullest. That would simply be a violation of the second law of thermodynamics.

All the losses which have been mentioned above are tabulated in Table 1.10.

Input/ Output Parameter	Percentage
HEAT INPUT IN FUEL	100
Various Heat losses in boiler	-
1. Dry flue gas loss	4.9
2. Loss due to hydrogen in fuel	0.0
3. Loss due to moisture in fuel	1.9
4. Loss due to moisture in air	1.2
5. Surface heat losses	7.2
TOTAL LOSSES	15.2
Boiler Efficiency	84.8

Table 1.10 Boiler Heat Balance

The relations for computing these losses are given in as follows:

1) Heat loss due to dry flue gas

$$L_{flue\ gas} = \frac{m * C_p * (T_f - T_a)}{GCV\ of\ fuel} \quad \textbf{(7)}$$

2) Moisture heat loss in air

$$L_{moisture} = \frac{AAS * humidity\ factor * C_p(T_f - T_a)}{GCV\ of\ fuel} \times 100 \quad \textbf{(8)}$$

3) Surface Heat losses

$$L_{surface} = 0.548[\{(\frac{T_s}{55.55})^4 - (\frac{T_a}{55.55})^4\} + 1.957(T_s - T_a)^{1.25} \times \sqrt{\frac{(196.85\ V_m + 68.9)}{68.9}}] \quad \textbf{(9)}$$

It should also be noted that during the efficiency tests, a number of parameters were kept constant. These include bunker level, auxiliary power of fans, flow rates, damper etc.

III. RECOMMENDATIONS

Some of the recommendations for the efficiency and heat rate improvements are as follows:

- Gas Turbine performance is a function of the ambient conditions, in particular the ambient air temperature. Turbine power output increases by 0.54% - 0.9% for every 1 degree decrease in the compressor inlet temperature. Corroborating this fact, the current operating parameters indicate that the fog cooling method [6] can result in marked improvement of the gas turbine unit.
- Minimize the re-heater spray flow rate. It is ideal to keep it at 0.5 % of main circuit mass flow rate. Because greater the steam spray, greater is the amount of steam that is made to bypass the HP & LP feed water heaters and therefore the cycle becomes less regenerative.
- Minimize the ambient wet bulb temperatures, which would decrease the condenser back pressure which would result in a net increase in the turbine output, and hence the heat rate.
- Low turbine heat rates can be possibly explained due to the removal of top heaters, since there was tube leak reported. This does increase the total flow rate through the turbine giving greater output. It also increases the total amount of heat that needs to be put into the system therefore reducing the efficiency.
- The turbine efficiency is also affected by the type of BFP and the method of flow control used. The plant was using constant speed motor driven BFPs and a throttle valve for controlling the flow. It is proposed that variable speed BFPs with adjustable frequency motors be used for a more better and controlled operation.
- In Cooling Tower fan power consumption, there is an energy saving potential by installing Automatic Temperature Controller (ATC).
- Boiler energy reduction is possible using variable speed drive and nano-fluids to enhance heat transfer processes [7].
- Exergy analysis is preferred as opposed to thermodynamic analysis of measuring efficiency because energy methods do not provide a comparison with the idealities [7].
- Osmotic filtration is practiced on fairly large scale at Gas Turbine plants. Reverse osmosis requires high pressure pumping. Even in the best-practiced system in the last effect (stage) of separation there is a reject stream to be maintained at 10 Kg/cm^2 artificially to have significant permeate rate. Normally this resistance is created by control valve. Finally, reject stream leaves to drain or to other point of use.

IV. CONCLUSION

As mentioned in the abstract, the purpose of this paper centers on the evaluation of the overall performance of the power plant. The findings conclude that the performance of the power plant is sensitive to the operation of condenser. Proper operation of the feed water circuit system is also worth a mention because this portion helps improving the exergetic efficiency of the cycle. In this manner, the efficient use of the HRSG unit must not be forgotten because the ultimate source of energy input to the bottoming cycle is the HRSG itself. Its auxiliaries should also be given special attention because the studies have shown that the operating flow rate do govern the operating characteristics of the heat exchangers, for instance the state of DNB (departure from nucleate boiling) can lead to tube burn out which can be catastrophic if not prevented. The set of recommendations proposed in the paper aim at providing greater ease in the operation and a greater adaptability to the fluctuating loads at the grid. They also ensure that the operating efficiencies improve and help on reducing the costs that add up to lessen the profits. For instance, the rated capacity of the plant is 450 MW but the operating inefficiencies do not allow to achieve the desired target. In short, there is a potential gap of about 15 MW that can be minimized by bringing in the suggested improvements and recommendations, Proper operating procedures will also allow industries to not only cut down on unnecessary expenses but will also help increasing the lifetime, reliability and the availability of the components that are installed at the units.

Conclusively, the objective which has been to show, on a quantitative scale, the performance of the power plant has been achieved. Moreover, it is intended to show that an energy conservative approach must be followed, given the sort of times that are coming ahead that brings serious concerns on energy and fuel resources. In addition, the figure that was quoted earlier referring to the amount of electricity additionally generated can be provided to far flung areas which are still not benefiting from electricity which includes slums and villages. This consolidates a step further on a long road to being a developed nation. The real reason why this topic is selected is to highlight this potential power generating capacity gap, which if lowered, could bring a drastic change on the lives on the people.

V. ACKNOWLEDGMENT

The authors are very grateful to Mr. Ian Whitehead (Rousch Plant Manager), Mr. Tom Scott (Rousch Operations Manager) and Mr. Razaq Anjum & Mr. Haseeb (Rousch Plant Engineers). In addition, authors express their utmost gratitude to Dr. Taqi Ahmed Cheema for reviewing the paper contents and details.

NOMENCLATURE

GT	Gas Turbine
ST	Steam Turbine
HRSG	Heat Recovery Steam Generator
GCV	Gross Calorific Value
MMSCF	Million Standard Cubic Feet
TTD	Terminal Temperature Difference
DCA	Drain Cooler Approach
P_2	Compressor outlet pressure
P_1	Compressor inlet pressure
T_2	Compressor outlet temperature
T_1	Compressor inlet temperature
Υ	$\frac{\text{constant pressure heat capacity}}{\text{constant volume heat capacity}}$
W_{out}	Power output at GT terminal (kW)
H_i	Heat input (kCal/h)
h_{in}	ST inlet enthalpy (kCal/kg)
h_{out}	ST outlet enthalpy (kCal/ kg)
h_s	Outlet isentropic enthalpy (kCal/kg)
ΔT	Average temperature rise in condenser (ºC)
T_{in}	Condenser inlet water temperature (ºC)
T_{out}	Condenser outlet water temperature (ºC)
T_{sat}	Saturation temp. at condenser pressure (ºC)
T_d	Drain outlet temperature (ºC)
t_i	Feedwater inlet temperature (ºC)
m	Mass of dry flue gas in kg/kg of fuel
C_p	Specific heat of flue gas in kCal/kg
T_f	Flue gas temperature (ºC)
T_a	Ambient temperature (ºC)
AAS	Actual mass of air supplied per kg of fuel
T_s	Surface temperature (K)
V_m	Wind velocity (m/s)

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