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Case Study of a Typical Underperformed Waste Heat Recovery Power Plant and Remedial Measures

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ABSTRACT

With the development of technology, the efficiency of combined-cycle gas turbine (CCGT) has crossed the barrier of 60%. In this paper, a typical problem of low-output generation of a CCGT plant is analysed. During 1st four months of operation of the plant, the sustainable steam turbine power output was between 36 and 36.5 MW against the design output of 37.2 MW. This marginal drop is acceptable considering the degradation factor of gas turbines which in open cycle have completed almost 6 years of operation from that of the CCGT design period. But the output has gradually decreased and finally drops to the extent of 28–29 MW. In spite of several efforts the problem is not resolved till date. However, effort is on, as the plant has lost a considerable power generation vis-a-vis financial loss. The individual efficiency or performance of each constituents of CCGT like gas turbine (GT), Heat recovery steam generator (HRSG), condenser, steam turbine (ST), etc. have been studied based on the parameters of pre- and post-corrective actions and design criteria. The maintenance that has been carried out in GT, HRSG and condenser has given a sign of some improvement which has been shown in various charts of the report. However, to achieve the designed sustainable output of the CCGT system on the basis of this analysis, a few major corrective actions in condenser and in the ST is felt necessary.

Keywords: Brayton cycle, combined-cycle gas turbine, condenser, heat recovery steam generator, rankine cycle

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INTRODUCTION

A combined-cycle power plant (CCPP) is a combination of heat engines generally gas turbine (GT) and steam turbine (ST), that work sequentially or simultaneously from same heat source to produce mechanical work which in turn is converted into electrical energy through electrical generators. In this system, after producing mechanical work with the help of a GT, the residual enthalpy in the exhaust flue gas is now allowed to recover in the heat recovery steam generator (HRSG) to produce steam for further generation of electrical energy through ST generator. Whereas in single GT unit this remaining heat energy in the exhaust flue gas is wasted into the atmosphere. So, by integrating a combined-cycle gas turbine (CCGT) the overall efficiency of the thermal power plant can be increased up to 50–60%.

Many investigators have carried different analysis on the performance of CCGT with various operating parameters. Their papers illustrate the research, development, and projects related to the combine cycle operation. Mohanty *et al.* [1] have done a research to analyse the performance of a

CCGT subjected to varying operating condition**.** Matlab simulation has been performed which provides the information about the effects of various operating parameters such as maximum temperature and pressure of Rankine cycle, turbine inlet temperature and pressure ratio of Brayton cycle on the network output and efficiency of the combined cycle. Singhal *et al*. [2] have done similar kind of work to analyse the effect on the performance of a CCPP due to changing operating parameters such as GT compression ratio, ambient air temperature and turbine inlet temperature. Ersayin *et al.* [3] studied performance of a power plant by collecting actual operating data from power plant control unit. Efficiencies of each of the constituents of the power plant are obtained from energy and exergy calculations which indicate that combustion chamber has the most exergy destruction rate compared to the other components. Kaushik *et al.* [4] tried to compare the efficiencies of coal- and gas-operated thermal power plants obtained by energy and exergy analysis. The study suggest from the results of exergy analysis that main energy loss in case of coal-based thermal power plant occurs in boiler and in case of gas-based combined-cycle thermal power plant occurs in combustion chamber. Wang *et al.* [5] analysed the effect of installation of a low-pressure economiser in a coal-fired power plant equipped with desulphurization system. The research concludes that by installing a low-pressure economiser waste heat from flue gas can be recovered; also fuel consumption in standard coal equivalent and CO_2 emission can be reduced. Ganjehkaviri *et al.* [6] studied the effect of steam quality at the turbine outlet on the power output of a CCPP. In the study three cases of different steam quality at turbine outlet are taken to compare the results. The results obtained from exergy analysis suggests that steam with 88% quality at turbine outlet leads to efficient, economic, and environmental friendly generation. Rout *et al*. [7] analysed the power output, thermal efficiency, and specific steam consumption of a conventional steam power plant considering three cases of Rankine cycle namely regenerative cycle, superheater cycle, and cogeneration cycle. The research work concludes that with an increased turbine inlet temperature, power output, thermal efficiency are highest in cogeneration and superheater power plant, respectively and steam consumption is least in cogeneration steam power plant making it more efficient.

In this paper the performance analysis of waste heat recovery project (WHRP) of Lakwa Thermal Power Station (LTPS) is carried out which utilizes combined-cycle of gas and steam turbine. In this particular power station, the ST output is designed to achieve 62% of that of the mother unit by integrating double pressure HRSG. The degradation factor of GT operating in open cycle had been considered by the original designer at the time of designing the ST plant. From three numbers of 20 MW GT operating in parallel as such is designed in 2006 to generate 37.2 MW from the ST. Subsequently after 5–6 months of operation, the operating parameter records shows a gradual decrease of generation and finally the output goes down to 28.5 MW from the achieved sustainable generation of 36–36.5 MW. As a case study, various operating status of the decreasing generation tendency has been reviewed such as GT heat output, HRSG performance, condenser performance, subsequent rise of the turbine exhaust steam temperature, etc. to find out the root cause of low-output generation.

METHODOLOGY

The low-performance issue of a CCPP of LTPS may be a result of low performance of any of its constituents like GT, HRSG, ST, and condenser, etc. To find out the root cause of the present low-output issue of WHRP unit, study of major constituents of combine cycle is necessary. Therefore performances of these constituents of WHRP are evaluated with mathematical calculations based on the design data and actual operating data collected before and after the various maintenance operations performed on the constituents.

Gas Turbine

The exhaust flow of GT [8] is required to calculate the heat input to HRSG. Therefore, efficiency of HRSG is dependent on the flow rate of the exhaust flue gas to the steam generator. Hence GT exhaust flow rate is calculated based on the operating data to compare it with the design flow rate. To improve the exhaust flow some corrective actions had been performed. Offline washing of GT axial air compressor is done along with the replacement of air inlet filters to increase the air flow through the compressor which leads to the improvement in the GT exhaust flow. With water wash, the axial air compressor gets clean which in turn helps in delivery of sufficient compressed air for optimised combustion. After this maintenance, again operating parameters are collected to calculate the exhaust flow (Tables 1, 2).

$$\text{Exhaust flow of GT} = \text{Inlet air flow} + \text{Fuel flow} - \text{Air losses} \quad (1)$$

$$\text{Inlet Air Flow} = \text{Cqk}\times\text{CIIF}\times\left[\frac{3.5\times R^{1.42857}\times(1-R^{.28571})}{\text{CTIF}}\right]^{.5} \quad (2)$$

where Cqk = Bellmouth discharge constant obtained from factory mechanical run test, CTIF=Ambient temperature in degree Rankine, CIIF = Ambient pressure – Total inlet duct loss (Compressor inlet absolute impact pressure at inlet flange in PSI), CPIA = Ambient pressure – Bellmouth discharge pressure (Compressor absolute static pressure at inlet annulus after bellmouth, in PSI), R = CPIA/CIIF, Air losses = 0.25% of inlet air flow.

Table 1. *Operating parameters of gas turbine (GT) before axial compressor cleaning.*

Parameters	GT 5	GT 6	GT 7
Atmospheric pressure (mbar)	997	993.95	997
Ambient temperature (°C)	36	36	36
Total inlet duct loss (mm of water column)	180	160	123
Bellmouth discharge pressure (kg/cm^2)	.1954	.19	.143
Gas turbine speed	100.34%	100.33%	100.33%
Fuel consumption (kg/s)	1.55	1.57	1.55
Cqk	1060	1078	1078.80

Table 2. *Operating parameters of gas turbine (GT) after axial compressor cleaning.*

Parameter	GT 5	GT 6	GT 7
Atmospheric pressure (mbar)	997.11	997.11	997.11
Ambient temperature (°C)	36	35	36
Total inlet duct loss (mm of water column)	170	150	155
Bellmouth discharge pressure (kg/cm^2)	0.19	0.17	0.1
Gas turbine speed	100.34%	100.36%	100.37%
Fuel consumption (kg/s)	1.498	1.497	1.4968
Cqk	1060	1078	1078.80

Heat Recovery Steam Generator

The percentage of energy recovered in each of the three HRSG is calculated to analyse the efficiencies of steam generators on the basis of the design data provided and the actual data collected during operating condition prior to maintenance and after the maintenance. The exhaust flue mass temperature is observed as 122°C in HRSG 1, 121°C in HRSG 2, and 120°C in HRSG 3. All the modules in HRSG are inspected and it has been observed that there is a deformation between modules and casing that is side clearance between the modules and casing has increased. This may indicate that a part of GT exhaust flow might be bypassing the HRSG modules resulting in the increase of HRSG exhaust temperature. Therefore, new baffles are welded as a corrective action to minimize the escape of flue gas through the side clearance so that flue gas can be flown across the modules for proper heat transfer to obtain optimised generation of steam. After the maintenance, again the exhaust flue mass temperature is observed and it is found as 116°C in HRSG 1, 118°C in HRSG 2, and 115°C in HRSG 3 (Tables 3–5).

Table 3. *Thermodynamic properties of steam in HRSG based on the design data.*

	High pressure (HP)	**Low pressure (LP)**
Steam flow	41.2 TPH	11TPH
Steam pressure	63kg/cm^2	5kg/cm^2
Steam temperature	463°C	202°C
Specific enthalpy	795.17 kcal/kg	682.7 kcal/kg
Feed water flow	41.2 TPH	11 TPH
Feed water temperature	105°C	105°C
CPH flow	51.74 TPH	
CPH inlet temperature	45°C	
CPH outlet temperature	100°C	

Table 4. *Thermodynamic properties of steam based on actual data before maintenance.*

	HRSG 1	**HRSG 2**	**HRSG 3**
HP steam flow rate	40.62 TPH	40.7 TPH	40.26 TPH
HP steam pressure	61.6 kg/cm^2	61.2 kg/cm^2	60.5 kg/cm^2
HP steam temperature	459.5°C	474°C	472°C
Specific enthalpy of HP steam	3325.88 kJ/kg	3359.59 kJ/kg	3355.56 kJ/kg
LP steam flow rate	9.2 TPH	8 TPH	8.6 TPH
LP steam pressure	4.85 kg/cm^2	4.5 kg/cm^2	4.5 kg/cm^2
LP steam temperature	211.05°C	213.76°C	215.12°C
Specific enthalpy of LP steam	2877.78 kJ/kg	2888.88 kJ/kg	2894.44 kJ/kg
Flow rate of feed water	54.13 TPH	54.13 TPH	54.13 TPH
Feed water temperature	104.79°C	104.79°C	104.79°C
CPH flow rate	53.79 TPH	53.09 TPH	53.78 TPH
Temperature at CPH inlet	71.15°C	73.03°C	72.99°C
Temperature at CPH outlet	96.87°C	96.62°C	95.85°C
GT exhaust flow	111.698 kg/s	112.175 kg/s	100.548 kg/s
GT exhaust temperature	490°C	503°C	504°C
Specific enthalpy at GT exhaust	122.6682 kcal/kg	126.1894 kcal/kg	126.4607 kcal/kg

$$\text{Heat input to HRSG} = \text{GT exhaust flow} \times \text{Specific enthalpy of GT exhaust gas} \tag{3}$$

$$\text{Enthalpy of HP steam at the outlet of HRSG} = \text{HP steam flow rate} \times \text{Specific enthalpy of HP steam} \tag{4}$$

$$\text{Enthalpy of LP steam at the outlet of HRSG} = \text{LP steam flow rate} \times \text{Specific enthalpy of LP steam} \tag{5}$$

$$\text{Enthalpy at the CPH outlet} = \text{Flow rate at CPH outlet} \times \text{Specific enthalpy at CPH outlet} \tag{6}$$

$$\text{Enthalpy of feed water at the inlet to HRSG} = \text{Feed water flow rate} \times \text{Specific enthalpy of feed water at inlet to HRSG} \tag{7}$$

$$\text{Enthalpy at the CPH inlet} = \text{Flow rate at CPH inlet} \times \text{Specific enthalpy at CPH inlet} \tag{8}$$

$$\text{Energy output from HRSG} = (4) + (5) + (6) \tag{9}$$

$$\text{Energy input to HRSG excluding the energy of GT exhaust gas} = (7) + (8) \tag{10}$$

$$\text{Energy recovered in HRSG} = (9) - (10) \tag{11}$$

$$\text{Percentage of energy recovered in HRSG} = \frac{\text{Energy recovered in HRSG}}{\text{Heat input to HRSG}} \tag{12}$$

Heat input to HRSG (energy in GT exhaust) at design condition = 56.22 MW

Table 5. *Thermodynamic properties of steam-based on actual data after maintenance.*

	HRSG 1	HRSG 2	HRSG 3
HP steam flow rate	41.92 TPH	41.5 TPH	40.90 TPH
HP steam pressure	60.68 kg/cm^2	60.44 kg/cm^2	59.82 kg/cm^2
HP steam temperature	456.1°C	468°C	461.7°C
Specific enthalpy of HP steam	3316.41 kJ/kg	3345.25 kJ/kg	3330.9 kJ/kg
LP steam flow rate	8.5 TPH	8.8 TPH	8.8 TPH
LP steam pressure	4.96 kg/cm^2	4.97 kg/cm^2	4.89 kg/cm^2
LP steam temperature	211.8°C	214.76°C	214.95°C
Specific enthalpy of LP steam	2876.58 kJ/kg	2881.27 kJ/kg	2883.78 kJ/kg
Flow rate of feed water	50.42 TPH	50.30 TPH	49.7 TPH
Feed water temperature	104.65°C	104.65°C	104.65°C
CPH flow rate	54.09 TPH	53.31 TPH	54.08 TPH
Temperature at CPH inlet	50.79°C	51.64°C	52.54°C
Temperature at CPH outlet	98.44°C	98.35°C	97.95°C
GT exhaust flow	111.698 kg/s	112.175 kg/s	100.548 kg/s
GT exhaust temperature	490°C	503°C	504°C
Specific enthalpy at GT exhaust	122.6682 kcal/kg	126.1894 kcal/kg	126.4607 kcal/kg

Steam Turbine

Along with isentropic efficiencies total power developed by HP and LP steam are calculated. During the study, the ST has not subjected to any maintenance operation. Therefore, performance analysis of ST is done based on the design data and data collected during operating condition prior to maintenance (Tables 6, 7).

Table 6. *Design values of ST for calculation.*

	HP steam	LP steam
Steam pressure at turbine inlet	60 kg/cm^2	4.5 kg/cm^2
Steam temperature at turbine inlet	460°C	200°C
Specific enthalpy at turbine inlet	3328.8 kJ/kg	2858.8 kJ/kg
Steam pressure at turbine outlet	.098 kg/cm^2	.098 kg/cm^2
Steam temperature at turbine outlet	45.031°C	45.031°C
Specific enthalpy at turbine outlet	2341.59kJ/kg	2341.59kJ/kg
Ideal outlet steam enthalpy	2137.12kJ/kg	2251.15kJ/kg
Steam flow rate	122.84 TPH	31.64 TPH

Table 7. *Operating parameters of ST collected before maintenance.*

	HP steam	LP steam
Steam pressure at turbine inlet	59.103kg/cm^2	4.7550 kg/cm^2
Steam temperature at turbine inlet	461.3°C	210.87 °C
Specific enthalpy at turbine inlet	3331.82kJ/kg	2879.9kJ/kg
Steam pressure at turbine outlet	0.239 kg/cm^2	0.239 kg/cm^2
Steam temperature at turbine outlet	60.233 °C	60.233 °C
Specific enthalpy at turbine outlet	2522.06kJ/kg	2522.06kJ/kg
Ideal outlet steam enthalpy	2263.59kJ/kg	2357.58 kJ/kg
Steam flow rate	122.499 TPH	24.5668 TPH

Isentropic efficiency of HP steam = $\frac{\text{Inlet enthalpy} - \text{outlet enthalpy}}{\text{Inlet enthalpy} - \text{Ideal outlet enthalpy}}$ (13)

Power generated by HP steam per tonne = $\frac{\text{Inlet enthalpy} - \text{outlet enthalpy}}{3600}$ (14)

Total power generated by HP steam = HP steam flow rate × power generated by HP steam per tonne (15)

Isentropic efficiency of LP steam = $\frac{\text{Inlet enthalpy} - \text{outlet enthalpy}}{\text{Inlet enthalpy} - \text{Ideal outlet enthalpy}}$ (16)

Power generated by LP steam per tonne = $\frac{\text{Inlet enthalpy} - \text{outlet enthalpy}}{3600}$ (17)

Total power generated by LP steam = LP steam flow rate × power generated by LP steam per tonne (18)

Total power generated = total power generated by HP steam + total power generated by LP steam (19)

Condenser

To have an idea about the performance of the condenser, its efficiency is calculated based on the design data and actual data (Table 8). It has been observed that turbine exhaust temperature is considerably increased to around 60°C against the design temperature of 45°C. Also, there is a fall of condenser vacuum from design value -0.95 to -0.76kg/cm^2. In spite of increasing the water flow by operating the standby cooling water pump in parallel with the operating pumps, no such improvement on condenser performance has been observed. So, it reflects the possibility of fouling of condenser tubes. With the above observation it has come to a conclusion that there may be hard scale deposition in the condenser water tubes which may be the reason for reduced heat transfer, i.e. low performance of the condenser. Corrective action of chemical cleaning followed by hydrojet cleaning has been carried out for cleaning of the condenser. Offline water fill test of the condenser has been carried out for detection of presence of leakage, if any. Also, online helium leak detect test is done to find out if there is any air ingress to the entire turbine system which may be the reason of the fall of the condenser vacuum. But in both the cases no leakage or external air ingress is observed.

Amount of heat exchange,

$Q_1 = \dot{m}_h h_{fg} = \dot{m}_c C_{pc} (t_2 - t_1)$ (20)

Condenser efficiency, $\eta = \frac{t_2 - t_1}{t_s - t_1}$ (21)

Table 8. *Design and operating parameters of condenser before and after maintenance.*

	Design data	Operating data before maintenance	Operating data after maintenance
Cooling water inlet temperature, t_1	32°C	30°C	27.7°C
Cooling water outlet temperature, t_2	41.14°C	40°C	36.45°C
Vacuum inside the condenser	-0.95kg/cm^2	-0.76 kg/cm^2	-0.892 kg/cm^2
Saturation temperature, t_s	45.031°C	65.88°C	50.7°C

RESULTS

In this study, it is tried to evaluate the performances of the constituents by calculating the exhaust flow of GT, efficiencies of HRSG, condenser, ST along with the output of ST, based on the design data and operating data collected before and after the maintenance. The results obtained from these calculations give us an idea about the causes lead to the low generation of the ST. Therefore, the performances of each of the constituents are thoroughly discussed below.

GT Performance

The design value of GT exhaust flow is 107.45 kg/s and the exhaust flow rate of GTs based on operating parameters prior to and after the maintenance are shown in the Table 9 and 10. With reference to the Table 9, it has been observed that exhaust flow of GT unit 5 and 6 are at par with design exhaust flow but it is considerably low in case of GT unit 7. After the corrective action that is axial air compressor cleaning as mentioned before and replacement of old air filters, the exhaust flow of the GT unit 5 and unit 6 have been increased as shown in Table 10. But the GT unit 7 is still not in the desired level, whereas the HRSG 3 efficiency after maintenance is found almost equal to that of the design efficiency. This may probably because of erroneous reading of GT exhaust flow of unit 7 which need further verification. As such, considering efficiency factor of HRSG, it may be concluded that the GT exhaust flow is not affecting much on the low output issue of the turbine unit.

HRSG Performance

Percentage of energy recovered in HRSG prior to maintenance as shown in Table 11 is much below than that of the design value of 78.3%. After the corrective actions of HRSG as discussed before, it is found that the percentage of energy recovered in the three steam generators has been improved. Therefore, after rectification of the internal problems of HRSG, it can be concluded that the effect of HRSG performance on the low generation output issue is reduced considerably.

Condenser Performance

Efficiency of the condenser is calculated based on the design parameters and the parameters obtained before and after the maintenance and provided in the Table 12. It is observed that the condenser performance is very poor as compared to the design value. Therefore the condenser had gone for corrective actions like cleaning of condenser tubes as discussed before. But even after this maintenance, there is not much improvement in condenser efficiency. The probable causes of the low efficiency of the condenser are discussed below:

- Heat exchange is not proper probably due to formation of hard scaling in condenser tubes resulting in less condensation of exhaust steam from turbine which further leads to fall of condenser vacuum.
- Probabilities of leakage of HP steam to LP turbine section through wear out interstage gland seals. This may lead to increase of temperature and pressure in the LP section and subsequent increase of exhaust steam parameters. The increase of enthalpy at the turbine exhaust has adverse effect on condenser performance as the condenser is now required to condense steam having enthalpy more than that of the design value. Even after increasing the cooling water flow by operating the stand by 3rd cooling water pump in parallel to improve the condenser performance, the required condensation is not achieved which leads to the fall of condenser vacuum.

ST Performance

Isentropic efficiencies of HP and LP steam along with the total power generated by the steam are calculated based on the operating parameters to compare these with the design values. During the study, the ST is not subjected to any kind of corrective actions. Therefore performance of the ST cannot be analysed completely.

However, with reference to the Table 13, it is seen that total power generated by the steam in the turbine is around 29.99 MW against the design value of 38.208 MW. Considering the generator loss and transformer loss at full we get design output as 37.2 MW. The output is further decreasing in course of time and comes down to around 28 MW.

Table 9. *Exhaust flow of GT prior to maintenance.*

	Exhaust flow in kg/s
GT 5	109.4
GT 6	109.048
GT 7	81.25

Table 10. *Exhaust flow of GT after maintenance.*

	Exhaust flow in kg/s
GT 5	111.698
GT 6	112.175
GT 7	100.548

Table 11. *Percentage of energy recovered in HRSG.*

	Before maintenance	After maintenance
HRSG 1	70.89%	73.73%
HRSG 2	67.56%	71.58%
HRSG 3	75.36%	78.4%

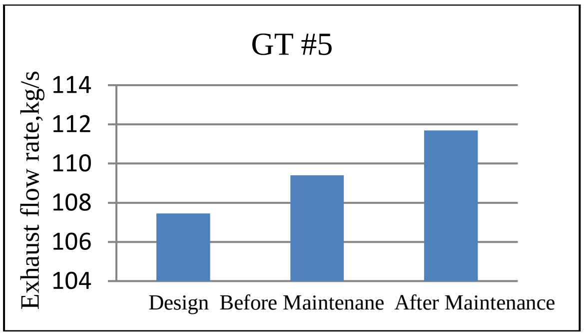
Table 12. *Efficiency of condenser.*

	Efficiency (%)
Based on design condition	70.14
Based on data before maintenance	27.87
Based on data after maintenance	38.04

Table 13. *Calculation results of steam turbine.*

	Based on design data	Based on operating data prior to maintenance
Isentropic efficiency of HP steam	82.85%	75.8%
Power generated by HP steam	33.664 MW	27.55 MW
Isentropic efficiency of LP steam	85.11%	68.51%
Power generated by LP steam	4.544 MW	2.44 MW
Total power generated	38.208 MW	29.99 MW

In Figure 1, bar diagrams of the various results of the performances of constituents provided in the above tables are shown.



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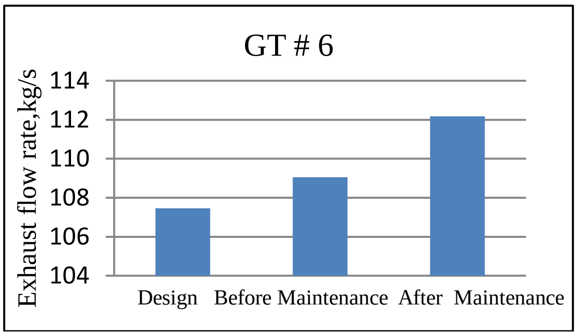

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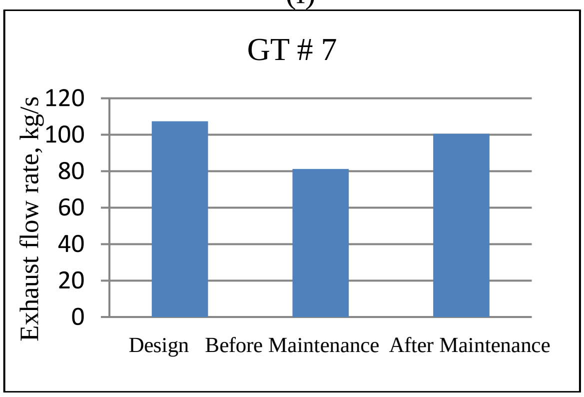

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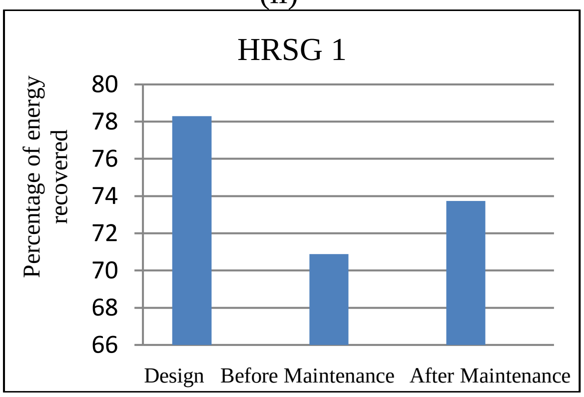

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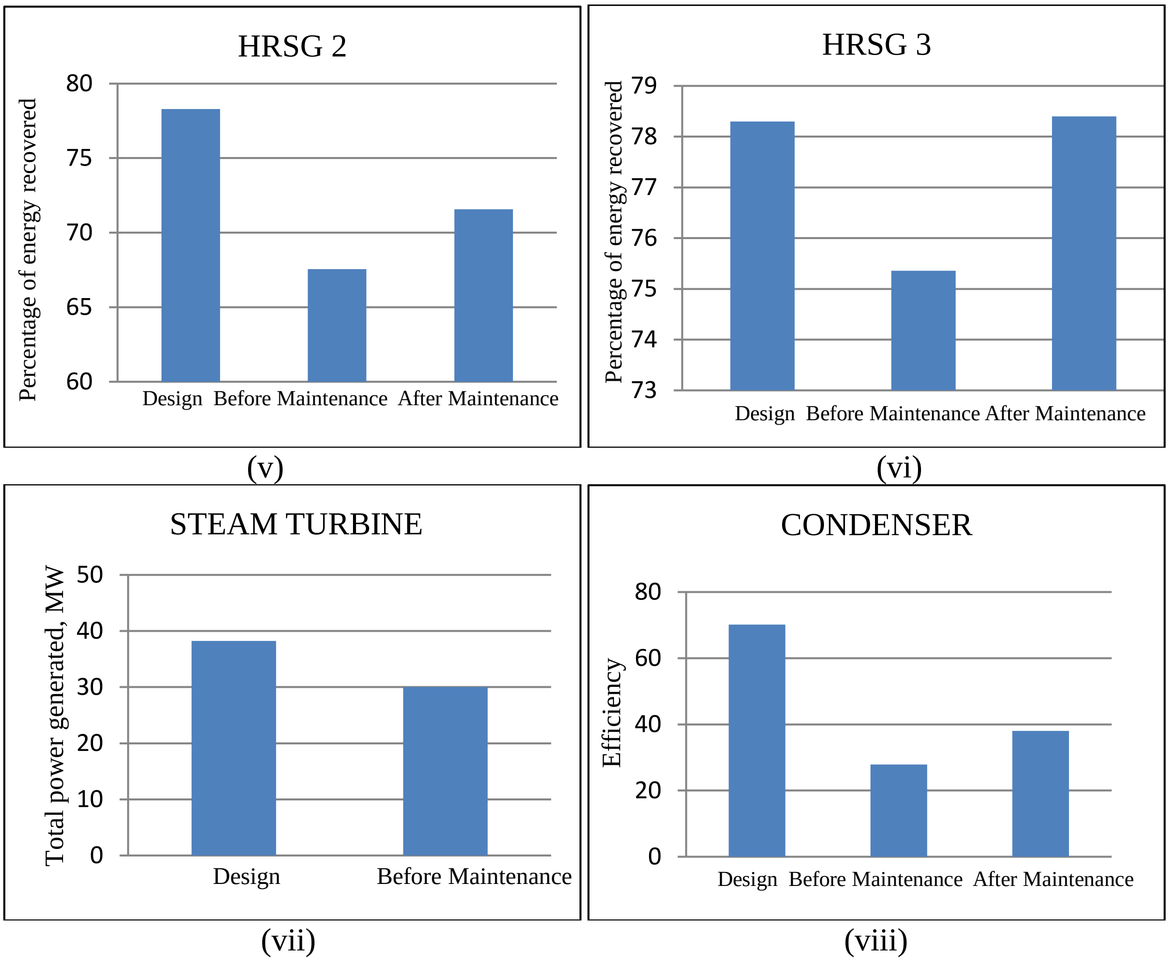

Fig. 1. *Bar diagrams of the various results of the performances of constituents.*

CONCLUSION AND DISCUSSION

The low-output problem particularly in this ST generator unit is a typical one. The suspected causes of decreasing generating capacity of ST have been analysed as mentioned above and subsequently corrective action had also been taken departmentally time-to-time. Except a marginal improvement in generator power output, the low-output problem still exists. As such, this has lead to further in-depth study of the operating behaviour of the CCGT unit. This is worthwhile to mention here that while analysing operating parameters of the ST unit during the commissioning stage, it has been noticed that the unit was exposed to rotor stalling phenomenon for quite a number of times. The restoration of the turbine almost in each occasion took more than 60 hours as recorded. It has been observed from the record that during the rotor stalling condition, rotating the turbine rotor either manually or by auto system was possible only with hard effort. This is due to hot bend rotor got hold up internally either in seals of gland or in seals of diaphragms. In the later stage, the rotor stalling problem was eliminated.

During initial observation, it appears that the typical rotor stalling behaviour during commissioning stage may not have any link with the turbine low-output issue. However, with no performance improvement in spite of taking various corrective measures, a careful overall analysis considering rotor stalling issue is also felt necessary. In this context, the increase of LP steam injection pressure has also taken into consideration. For confirmation of the behaviour of increased

LP pressure, the LP steam system has again been studied. During the study, the LP pressure is found very high, i.e. the pressure increased up to HRSG safety valve popping limit of 5.6 kg/cm². As per earlier record, the LP steam injection pressure and steam flow at the beginning was nearing to design value. This increased LP pressure (back pressure) behaviour along with fall of steam flow indicates that there is definitely flow restriction either upstream part of turbine LP cylinder or internal of LP turbine section. As per the earlier analysis of this report, the GT exhaust flow, HRSG efficiency are found almost nearing to that of the design values. Hence performance of ST needs to be verified in respect to this abnormal increase of LP pressure and subsequent link with low-output issue. Silica deposition in the internals of the turbine for this case study is not considered as the record of parameters of steam quality shows silica content well within that limit. Therefore, there is every likelihood of the damage of interstage gland seals and other internal seals during the aforementioned rotor stalling period. The rotor to seal rubbing during ST restoration process by barring is definitely going to increase clearance between rotor and the gland seal or other seals. With such increase in seal clearance, steam leakage along the rotor from HP turbine section to LP turbine section is inevitable. As such increase of exhaust steam temperature or loss of condenser vacuum will occur as an after effect of rotor stalling. So, the conclusion can be drawn that the ST needs to open up to replace interstage gland seals and other seals and for inspection of deposition if any. The corrective action is expected to eliminate the internal steam leakage and will improve condenser vacuum and subsequently improve the power output.

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