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Thermodynamic Modeling and Optimization of a Dual Pressure Reheat Combined Power Cycle

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ABSTRACT: Heat recovery steam generator (HRSG) plays a key role on performance of combined cycle (CC). In this work, attention was focused on a dual pressure reheat (DPRH) HRSG to maximize the heat recovery and hence performance of CC. Deaerator, an essential open feed water heater in steam bottoming cycle was located to enhance the efficiency and remove the dissolved gasses in feedwater. Each of the heating section in HRSG is solved from the local flue gas condition with an aim of getting minimum possible temperature difference. For high performance, better conditions for compressor, HRSG sections, steam reheater and deaerator are developed. The CC system is optimized at a gas turbine inlet temperature of 1400◦C due to the present available technology of modern gas turbine blade cooling systems. The exergetic losses in CC system are compared with each other. The present DPRH HRSG model has been compared and validated with the plant and published data.

KEYWORDS: Combined cycle; dual pressure; deaerator; exergy analysis; heat recovery steam generator.

1. INTRODUCTION

Combined cycle (CC) power plants are gaining wider acceptance due to more and more availability of natural gas, now a days, because of their higher overall thermal efficiencies, peaking and two-shifting capabilities, fast start-up capabilities and lesser cooling water requirements. Heat recovery steam generator (HRSG) plays a very important role in recovering the sensible heat of gas turbine exhaust for generating steam, at required pressure and temperature, suitable to steam turbine for further power generation. Heat recovery from the HRSG has to be maximized for higher CC efficiency. Plant designs need to be adjusted to the specific project parameters to achieve optimum results. The optimization of the HRSG is particularly interesting for the combined plants design in order to maximize the work obtained in the vapor cycle. A detailed optimization of the HRSG is a difficult problem, depending on several variables. These include the number of pressure levels, the pressures, the mass flow ratio, and the inlet temperatures to the HRSG sections.

Recently, researchers paying much attention towards the heat recovery steam generator to improve the CC performance. Pasha & Sanjeev (1995) presented a discussion about the parameters which influence the type of circulation and the selection for HRSG. Ongiro et al (1997) developed a numerical method to predict the performance of the HRSG for the design and operation constraints. Ganapathy et al (1988) described the features of the HRSG used in the Cheng cycle system. In this system, a large quantity of steam is injected into a gas turbine to increase electrical power output. Subrahmanyam et al (1995)d iscussed about the various factors affecting the HRSG design for achieving

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highest CC efficiency with cheaper, economical and competitive designs and with highest requirements to meet the shorter deliveries.

Noelle & Heyen (2004) designed the once-through HRSG which is ideally matched to very high temperature and pressure, well into the supercritical range. Ragland & Stenzel (2000) compared four plant designs using natural gas with a view of cost benefits achieved through HRSG optimization. Casarosa et al (2004) determined the operating parameters means both thermodynamic and thermo economic analysis, minimizing a suitable objective function by analytical or numerical mathematical methods. The optimal design and operation of a HRSG is possible with minimization of entropy generation (Nag & De 1997). Reddy et al (2002) applied second law analysis for a waste HRSG which consists of an economizer, an evaporator and a super heater. Introducing multi pressure steam generation in the HRSG of a combined power plant improves the performance of the plant than that of a corresponding single pressure system (De & Biswal 2004).

Pelster et al (2001) compared the results of a reference CC with the dual and triple pressure HRSGs and also with and without steam reheating models. Bassily (2005, 2007) modelled a dual and triple pressure reheat CC with a preset in constraints on the minimum temperature difference for pinch points, the temperature difference for superheat approach, the steam turbine inlet temperature and pressure, the stack temperature, and the dryness fraction at steam turbine outlet without a deaerator in steam bottoming cycle. Srinivas (2009) suggested an improved location for a deaerator in a triple pressure HRSG. A literature review has shown that the DPRH HRSG has not been modelled and analysed parametrically with a deaerator. The purpose of this work is to optimize the DPRH HRSG in the CC with a deaerator. The variations in CC performance are plotted with the compressor pressure ratio, gas turbine inlet temperature, HRSG HP, steam reheat pressure and deaerator pressure. From the present thermodynamic modelling, the optimized conditions to air compressor, HP steam, LP steam, steam reheater and deaerator have been developed.

2. Thermodynamic model of the CC with DPRH HRSG The optimized model in HRSG improves the steam generation rate and hence steam turbine

output. Schematic flow diagram for CC is shown in figure 1. Simple gas cycle gives higher efficiency at lowcompressor ratio compared to the intercooled-reheat model and hence in this work simple gas cycle is selected as a topping cycle in CC (Srinivas et al 2007). Figure 2 shows the temperature-entropy diagram for the CC shown in figure 1. Steam cycle with a reheater is taken as a bottoming cycle in CC. In steam cycle a deaerator is located to gain higher efficiency as well as to remove the dissolved gasses in feed water. Condensate preheater is arranged in the HRSG as a last heat transfer surface to improve the heat recovery.

The temperature-heat transferred diagram for DPRH heat recovery is depicted in figure 3. TheCCis modelled withDPRHconfiguration in HRSG. An attempt has been made to improvethe heat recovery with parametric analysis. In this work, steam temperatures and low pressure (LP) are not getting fixed as in a regular method. Those are (except HP pressure and deaerator pressure) determined with the local flue gas temperature in heating sections of HRSG.

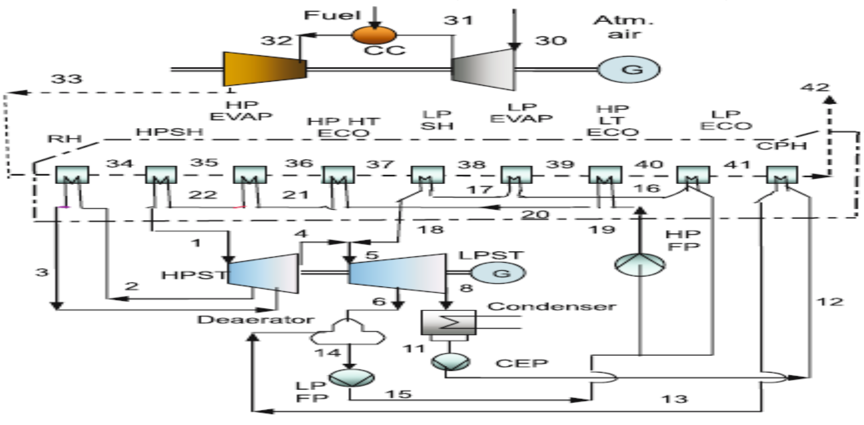

Figure 1. Schematic representation of CC with a dual pressure HRSG; HP: high pressure; LP: low pressure; HT: high temperature; LT: low temperature; GT: gas turbine; ST: steam turbine; RH: reheater; SH: superheater; EVAP: evaporator; ECO: economizer; CPH: condensate preheater; FP: feed pump; HRSG: heat recovery steam generator; CEP: condensate extraction pump.

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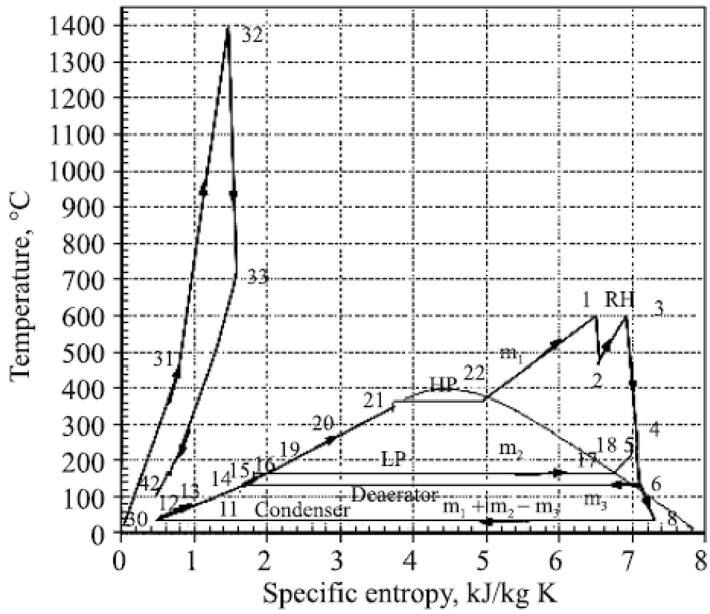

Figure 2. Temperature-entropy diagram of a DPRH CC.

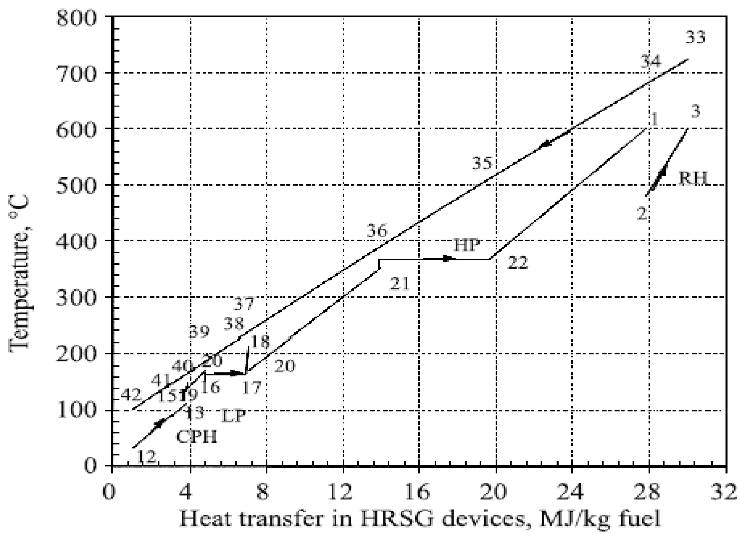

Figure 3. Details of heat recovery in a dual pressure HRSG.

Pinch point (PP)-minimum temperature difference between gas turbine exhaust leaving the evaporator and saturation temperature of steam in evaporators and terminal temperature difference (TTD)-temperature difference between exhaust entry and super heated steam in super heaters are maintained constant. Their choice is the consequence of a compromise between thermodynamic efficiency and investment costs (Franco & Russo 2002). 15K approach point (temperature difference between saturated steam and incoming water) is assumed in economizers.

3. Thermodynamic analysis of the CC

The assumptions made for the analysis of the CC are tabulated in table 1. The fuel to electricity efficiency of the CC is determined based on the lower heating value (50, 145 kJ/kg) of the fuel. The net work output of CC in percent of standard chemical exergy of the fuel (52, 275 kJ/kg) is expressed as exergy efficiency of CC (Kotas 1995). For an isentropic compression process, the change in the entropy for air is zero i.e. $\Delta_s = 0$. The isentropic efficiency (Eq. 1 and

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2) for compressor and gas turbine is determined from pressure ratio (r_c), specific heat ratio (γ) and polytrophic state efficiency (η_∞).

Isentropic efficiency of compressor, $\eta_c = \dfrac{r_c^{\left(\frac{\gamma-1}{\gamma}\right)}-1}{r_c^{\left(\frac{\gamma-1}{\gamma\eta\infty c}\right)}-1}$ (1)

Isentropic efficiency of gas turbine, η_{gt}, $= \dfrac{1-\left(\frac{P_{33}}{P_{32}}\right)^{\frac{\eta\infty gt(\gamma-1)}{\gamma}}}{1-\left(\frac{P_{33}}{P_{32}}\right)^{\frac{(\gamma-1)}{\gamma}}}$ (2)

In Eq. (1) and (2), $\eta_{\infty c}$ and $\eta_{\infty gt}$ are the polytrophic efficiencies of compressor and gas turbine respectively. For an isentropic compression process, the change in entropy for air is

Table 1. Assumptions made for thermodynamic evaluation of HRSG in CC.

Atmospheric condition	25°Cand 1.01325bar
Gas cycle pressure ratio and maximum temperature	14.5 and 1400°C
Inlet condition g=for hp steam turbine	200bar,600°C
TTD between flue gas and superhead steam	25
Approach point (AP) in LP and Hp economizers	15
Condenser pressure	0.05bar
Steam reheat pressure	50 percent of HP pressure
Pinch point(PP)-minimum temperature difference between the flue gas and the steam in the 2587vaporators	25
Degree of superheat(DSH) in superheater	50
Polytrophic stage efficiency for compressor and gas turbine	85 percent
Isentropic efficiency of steam turbine	90 percent
Generator efficiency	95 percent
Pressure drop in combustion chamber Pressure drop in HRSG, dearator and condenser in neglected Heat loss in HRSG turbine condenser, and deaerator is neglected	5 percent

zero, i.e. $\Delta_s = 0$.

$So_2,31' - So_2,30+3.76(SN_2,31' - SN_2,30)-R[\ln(0.21\times \frac{P_{31}}{P_{30}})+3.76\times \ln(0.79\times \frac{P_{31}}{P_{30}})]=0$ (3)

In Eq. (3), 'R' is the universal gas constant. The temperature of air after isentropic compression is estimated from the iteration of entropy change equation in compressor (Eq. 3). Then the actual temperature of compressed air is iterated from compressor isentropic efficiency (η_c) and enthalpies.

The combustion equation in gas turbine combustion chamber is

$CH_4+X(O_2+3.76N_2)\rightarrow CO_2+2H_20+(x-2)O_2+3.76xN_2$ (4)

In combustion equation (Eq. 4), '*x*' is the amount of air to be supplied at a fixed adiabatic flame temperature. The exergy balance equation for combustion chamber (Eq. 5) results the value of '*x*' (Eq. 6). The exergy balance equation in gas turbine combustion chamber,

$(h_{f,CH4}+hCH_4)+x(hO_2+3.76hN_2)_{31}$

$=(f,CO_2+hCO_2)_{32}+2(h_f,H_2O+hH_2O)32+(x-2)hO_{2,32}+3.76xhN_{2,32}$ (5)

From the above equation,

$X=\dfrac{(h_{f,CH_4}+h_{CH_4})-(h_{f,CO_2}+h_{Co_2})32-2(h_{f,H_20}+h_{H_20})32+2h_{O_{2,32}}}{h_{O_{2,32}}+3.76h_{N_{2,32}}-(h_{O_2}+3.76h_{N_2})31}$ (6)

The entropy change for isentropic expansion in the gas turbine is

$\Delta_s=0=S_{32}-S_{33'}$ (7)

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The entropy change for isentropic expansion in the gas turbine is where

$S_{32}=[\boldsymbol{s_{co_2}}+\boldsymbol{2s_{H_2}}+(x\text{-}2)s0_2+3.76xSN_2]_{32}$

$-R[\ln(\frac{1}{N_p}\times\frac{P_{32}}{P_0})+2\times \ln(\frac{2}{N_p}\times\frac{P_{32}}{P_{30}})+(x\text{-}2)\times \ln(\frac{(x-2)}{N_p}\times\frac{P_{32}}{P_0})+3.76xx\ln(\frac{3.76x}{N_p}\times\frac{P_{32}}{P_0})]$ (8)

And

$s_{33}^{''}=[\boldsymbol{s_{co_2}}+\boldsymbol{2s_{H_2}}+(x\text{-}2)s0_2+3.76xSN_2]_{33}$'

$-R[\ln(\frac{1}{N_p}\times\frac{P_{33}}{P_0})+2\times \ln(\frac{2}{N_p}\times\frac{P_{33}}{P_{30}})+(x\text{-}2)\times \ln(\frac{(x-2)}{N_p}\times\frac{P_{33}}{P_0})+3.76x\ln(\frac{3.76x}{N_p}\times\frac{P_{33}}{P_0})]$ (9)

The entropy change in the gas turbine is determined from Eq. (8) and (9). N_P is the total number chemical elements in the products of combustion. The iteration of entropy change equation (Eq. 7), results the gas turbine exhaust temperature. Then the actual temperature is iterated from the gas turbine isentropic efficiency *(η_{gt})* and enthalpies. The modelling of CC system is started with an arbitrarily chosen deaerator pressure. The exit state of deaerator is saturated liquid after direct mixing of bled steam with feed water. The LP pressure is evaluated from the saturation temperature. The saturation temperature of steam is a function of gas temperature, TTD and DSH (Eq. 10). The steam flow rate and gas temperature in the heating devices (HP, LP and deaerator) are determined from the heat balance equations. The saturation temperature of the LP evaporator,

The saturation temperature of the LP evaporator,

$TLP_{sat} = TLPex_{out} - TTD_{LP} - DSH_LP$. (10)

At HP evaporator the flue gas outlet temperature, $T_{36} = THP_{sat} + PP_{HP}$ (11)

At LP evaporator the flue gas outlet temperature, $T_{39} = TLP_{sat} + PP_{LP}$ (12)

At HPHT economizer, water outlet temperature, $T_{21} = THP_{sat} - AP_{HP}$ (13)

At LP economizer, water outlet temperature, $T_{16} = TLP_{sat}-AP_{LP}$. (14)

Thework interactions in gas and steam cycles are calculated to unit mass of fuel. To determine exergetic losses, the irreversibilities associated in all the components to be estimated for exergy analysis. In this analysis, the efficiency and losses are determined for 1 kmol of fuel. The chemical availability or exergy of a fuel is the maximum theoretical work obtainable by allowing the fuel to react with the oxygen from environment to produce environmental components of carbon dioxide and water vapor. When the difference in availability between states of same composition is evaluated, chemical contribution cancels, leaving just thermo mechanical contributions. This will be the case while doing availability analyses for compressors and turbines. However, the chemical contribution plays a main role in fuel gas combustor.

The chemical and physical exergy components are determined at each state by using the following equations.

Chemical exergy, $e_{ph}=\sum_k \boldsymbol{n_k \varepsilon_k^0}+R\boldsymbol{T_0}\sum_k \boldsymbol{n_k}\ \mathbf{ln}[\boldsymbol{P.x_k}]$

(15)

where x_k is the mole fraction of k^{th} component

physical exergy, $e_{ph}=h-\sum_k \boldsymbol{T_0 s_k}$ (16)

exerggy , $e=e_{ch}+e_{ph}$ (17)

The exergetic loss associated in each component is determined from exergy balance for the component/process. The exergetic loss due to mixing of fluids is added to the HRSG exergetic loss. The exergy efficiency is defined as the ratio of net work output from CC to the availability of fuel (Eq. 18).

exergy efficiency of CC, $\eta 2,cC=\left(\frac{\varepsilon_{CH_4}^0-i_{total}}{\varepsilon_{CH_4}^0}\right)\times \mathbf{100}$ (18)

IV. RESULTS AND DISCUSSIONS

CC with DPRH HRSG has been studied parametrically to find the maximum obtainable output and efficiency from the power system. Deaerator pressure at which steam is trapped to heat the feed water is a key parameter to get the optimum heat recovery from the exhaust. Sensitivity analysis has been carried out with the gas turbine inlet temperature, compressor pressure ratio, steam reheat pressure ratio and deaerator temperature ratio. All the plots are prepared at an air flow rate of 625 kg s^{-1}. Deaerator location is expressed in dimensionless form, defined as the ratio of difference in saturation temperatures of deaerator and condenser to the overall saturation temperature difference in the bottom cycle (Eq. 19).

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Figure 4 shows the effect of compressor pressure ratio for different values of gas turbine inlet temperature on CC power and efficiencies. In all the following figures (a) net electric power output, (b) fuel to electricity efficiency i.e. exergy efficiency and (c) exergy efficiency are developed against the CC key parameters. The difference in exergy of fuel and total exegetic losses gives the net work output (Eq. 18). The heating value of fuel is slightly less than the exergy of fuel. Generator efficiency has been considered in determining the fuel to electricity efficiency. Therefore, both the fuel to electricity efficiency and efficiency of CC are nearer to each other as shown in all the plots. The compressor pressure ratio is varied in such a way that the gas turbine exit temperature should not be less than 600◦C. Due to limitation of steam maximum temperature *(*600◦C*)*, the resulted range of feasible compressor pressure ratio is 8 to 12 at a gas turbine inlet temperature of 1200◦C. This range is 10 to 18 at 1300◦C and 14 to 26 at 1400◦C. The gas inlet temperature to HRSG is high at low pressure ratio and due to this reason; more quantity of steam can be generated in HRSG. This results in high output and efficiencies as shown. The pressure ratio corresponding to high performance increases from 10 to 14 with an increase in gas turbine inlet temperature of 1200◦C to 1400◦C.

Deaerator temperature ratio, $\theta_{deae} = \frac{T_{deae,sat} - T_{cond,sat}}{T_{HP,sat} - T_{cond,sat}}$ (19)

Figure 4 shows the effect of compressor pressure ratio for different values of gas turbine inlet temperature on CC power and efficiencies. In all the following figures (a) net electric power output, (b) fuel to electricity efficiency i.e. exergy efficiency and (c) exergy efficiency are developed against the CC key parameters. The difference in exergy of fuel and total exergetic losses gives the net work output (Eq. 18). The heating value of fuel is slightly less than the exergy of fuel. Generator efficiency has been considered in determining the fuel to electricity efficiency. Therefore, both the fuel to electricity efficiency and exergy efficiency of CC are nearer to each other as shown in all the plots. The compressor pressure ratio is varied in such a way that the gas turbine exit temperature should not be less than 600◦C. Due to limitation of steam maximum temperature (600◦C), the resulted range of feasible compressor pressure ratio is 8 to 12 at a gas turbine inlet temperature of 1200◦C. This range is 10 to 18 at 1300◦C and 14 to 26 at 1400◦C. The gas inlet temperature to HRSG is high at low pressure ratio and due to this reason; more quantity of steam can be generated in HRSG. This results in high output and efficiencies as shown. The pressure ratio corresponding to high performance increases from 10 to 14 with an increase in gas turbine inlet temperature of 1200◦C to 1400◦C.

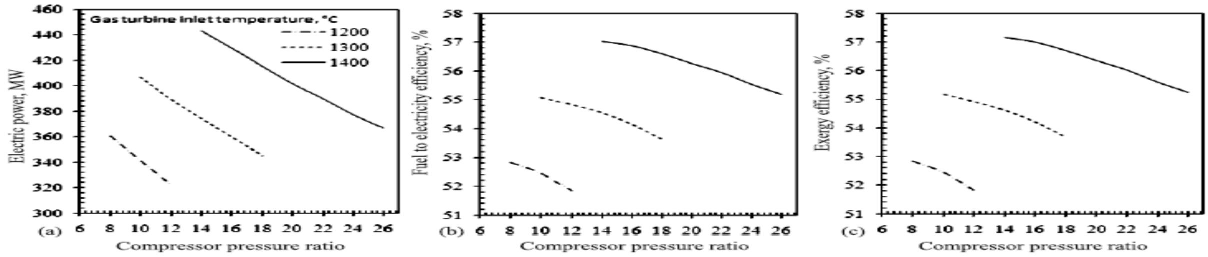

Figure 4. The feasible compressor pressure ratio at different gas turbine inlet temperatures and its effect on CC performance.

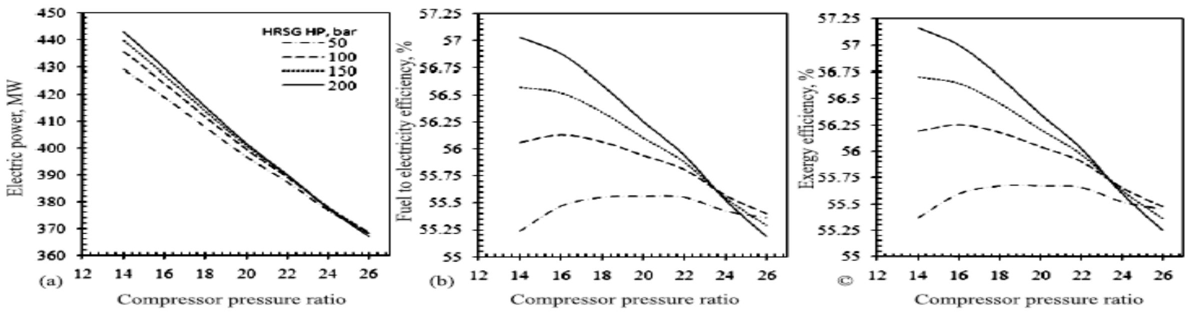

Figure 5. Effect of compressor pressure ratio on performance of CC for different values of HRSG HPs at a gas turbine inlet temperature of 1400◦C.

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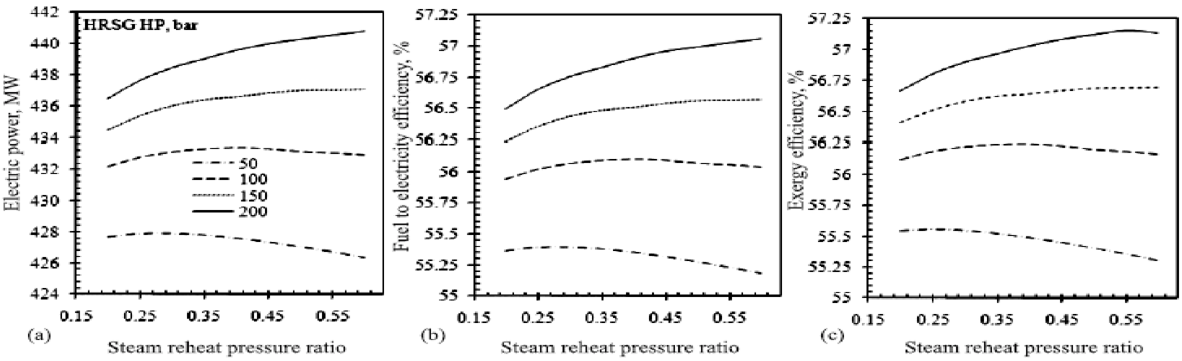

Figure 6. CC performance vs. steam reheat pressure ratio with HRSG HP.

Figure 5 presents the effect of compressor pressure ratio with HRSG HP on CC performance. The H combined cycle system will be the first power plant to achieve 60 percent efficiency and will operate at high temperature of 1430◦C. This has been achieved by advances in materials and cooling system technology. Therefore, all the performance characteristics curves are developed at a gas turbine inlet temperature of 1400◦C. At 1400◦C, the compressor pressure ratio is varied from 14 to 26 as discussed in the previous section. HRSG HP is varied from 50 to 200 bar. The power output is high at low pressure ratio irrespective of HRSG HP in the preselected pressure ratio range (14–26). But with respect to exergy and exergy efficiencies, the optimum pressure ratio decreases with an increase in HRSG HP. The effect of steam reheat pressure expressed in a fraction of HRSG HP has been depicted in Figure 6 at different values of HRSGHP. The optimum steam reheat pressure ratio increases with an increase in HRSG HP. But due to the limitation in dryness fraction (0·8) at steam turbine exit, the steam reheat pressure ratio is limited to 0·5 in the present work at HRSG HP of 200 bar.

Figure 7 generates the effect of deaerator temperature ratio as defined in Eq. (19) on CC performance at different sets of HRSG HPs. The optimum deaerator temperature ratio is found in between 0·25 and 0·3 with the variation of HRSG HP from 50 bar to 200 bar.

Figure 8 compares the exergetic losses of cycle components in percent of standard chemical exergy of fuel. The second low evaluation is useful to identify the major losses to focus the attention on the weak areas for further improvements. The major exergetic loss occurs in gas turbine combustion chamber in the order of 27·5 percent and the minor exergetic loss is in deaerator. With a steam injection in combustion chamber, this major exergetic loss can be decreased (Srinivas et al 2007, 2008). Next to GTCC, considerable level of exergy is destroyed in compressor, gas turbine, HRSG and exhaust in the range of 2 to 3 percent. The present DPRH HRSG model results are compared and validated with ABB–Alstom, greatest manufacturers of CC plants in the world and the literature results. This comparison is presented in table 2. The inlet conditions for DPRH HRSG are chosen from ABB plant. However, the present HRSG layout with a deaerator adopted here are quite different from those of comparison. The main differences we can find between the plant data and present work are the value of the HRSH HP and reheat pressures. It can be supposed that all the other differences that arise from the comparison (pressures and the flow rates) are a consequence of this primary characteristic that influences the properties of the thermal cycles deeply.

Figure 8. Comparison of exergetic losses in CC components in per cent of standard chemical exergy input of fuel.

In the present work, the HRSG LP is more than from the observations. It is due to inclusion of deaerator at a pressure below the LP level. Remaining all the results are in agreement with the plant and literature results.

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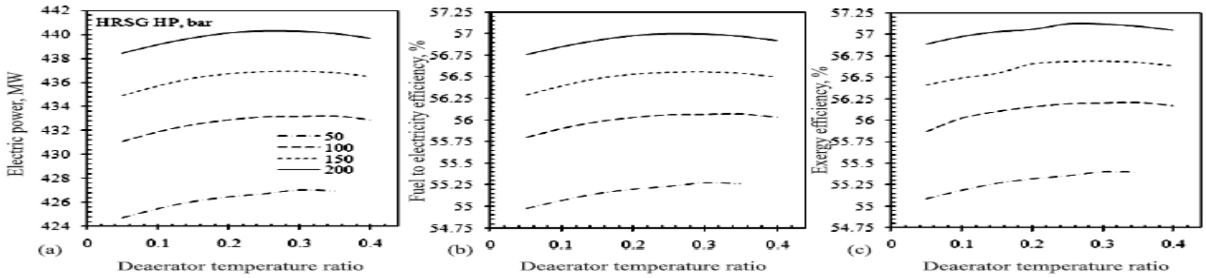
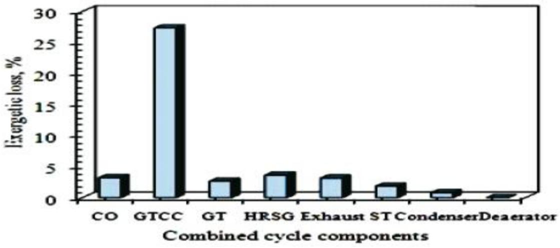


Figure 7. CC performance vs. deaerator temperature ratio with HRSG HP.



V. CONCLUSIONS

In a CC power generation system, a DPRH HRSG has been critically examined. For a given set of gas turbine inlet temperature, the feasible range of compressor pressure ratio has been developed. For a maximum steam temperature of 600◦C, the feasible compressor ratios are 8–12, 10–18 and 14–26 for a gas turbine inlet temperature of 1200◦C, 1300◦C and 1400◦C

Table 2. Comparison of heat recovery details with the plant data and published results at Tg,HRSGin = 650◦C and mg = 386·7 kg s−1.

	ABB	Franco & Russo (2002)	Present work
P_{HP}, bar	164	220	200
T_{HPSH}, °C	563	556	550
m_{HP}, kg s^{-1}	58·9	67·3	57·3
P_{RH}, bar	38·2	69·7	100
T_{RHin}, °C	–	374	434
T_{RHout}, °C	562	556	550
P_{LP}, bar	6·9	1·6	12·8
T_{LPSH}, °C	323	sat	242
m_{LP}, kg s^{-1}	11·4	4·53	9·76
$W_{net,sc}$, MW	97	67·3	87

respectively. Out of this range, the optimum compressor pressure ratio decreases with an increase in HRSG HP. The optimum deaerator temperature ratio is suggested at 0·25 to 0·3.

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