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# Off Design Performance Analysis of a Triple Pressure Reheat Heat Recovery Steam Generator

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**Abstract:**

Operating characteristics of a triple pressure reheat HRSG at design and off-design mode has been analyzed and the effect of exogenous variables such as power load, process requirements, and operating mode, etc., on the transient performance of the plant is studied. By changing the arrangement of High-Temperature and Intermediate-Temperature zone components of the HRSG, its effect on the Steam Turbine performance and HRSG characteristics is examined in this paper. It is shown that there could be a significant difference in HRSG sizes even though thermal performance is not in great deviation. From steam turbine performance point of view, it should be carefully reviewed whether the optimum design point could exist or not. Off-design performance could be one of the main factors in arranging components of the HRSG because power plants operate at various off-design conditions such as ambient temperature and gas turbine load, etc. It is shown that different Heat Exchanger configurations lead to different performances with ambient temperature, even though they have almost the same performances at design points.

**Key Words**: *Combined Cycle Power Plant, Triple Pressure Heat Recovery Steam Generator, Efficiency, NTU, effectiveness, Optimum Design*

## 1. Introduction

The gas/steam combined cycle has already become a well-proven and important technology for power generation due to its numerous advantages. The advantages include its high efficiency in utilizing energy resources, low environmental emissions, short duration of construction, low initial investment cost, low operation and maintenance cost, and flexibility in fuel selection, etc.[3] These features justify the fact that the combined cycle power plants are quite competitive in the power market.

The heat recovery steam generator (HRSG) is the component of the bottoming steam cycle, which absorbs energy of exhaust gas of the gas turbine and produces steam at subcritical pressures suitable for the process or for further electricity generation by a steam turbine. Power plant engineers can design their own HRSGs and the bottoming steam cycles at the initial stage. On the other hand, gas turbine is not made in order and steam turbine is selected according to the condition of the steam delivered from a HRSG. In this respect, the design of a HRSG is indispensable to the improvement of the overall system efficiency and power output, and to the reduction of the main equipment cost [4, 8, 9]. HRSGs are classified into single, dual, and triple pressure types depending on the number of drums in the boiler. Dual pressure HRSGs have been widely used because they showed higher efficiency than single pressure systems and lower investment cost than triple pressure HRSGs. Nowadays, however, the development of high efficiency gas turbines has triggered the use of a triple pressure HRSG. Overall plant efficiency can be improved up to nearly 60% if a triple pressure reheat HRSG and a high performance gas turbine is used. However, due to the complexity of the triple pressure HRSG, it is not easy to identify the effect of the change in design parameters on the overall plant performance.

In addition, many researchers have been focused on the design of triple pressure reheat or non-reheat HRSGs (Eisenkolb et al., 1996; Lee et al., 2002). The effect of the arrangement of the heat exchangers such as high pressure (HP) superheater, reheater, HP economizer, etc. should also be informed to the power producers. They have to know the operating characteristics of the power plant at various off-design conditions because they want to get the most efficient way of producing electricity [6]. Important parameters such as heat recovery capacity and efficiency should be carefully reviewed according to the variation of operating conditions.

In this study, the operating characteristics of triple pressure reheat HRSGs are analyzed in EXCEL Spread Sheets. The effects of the configuration of HP superheater and reheater on the thermal performance and economics of the plant are investigated. The arrangement of Intermediate-Temperature components such as intermediate pressure (IP) superheater, HP economizer, and low pressure (LP) superheater is changed to check its effect on the performance of a steam turbine. The off-design performance is also examined considering the operating ranges of the plant, for example, ambient temperature and gas turbine load.

## 2. Plant Description

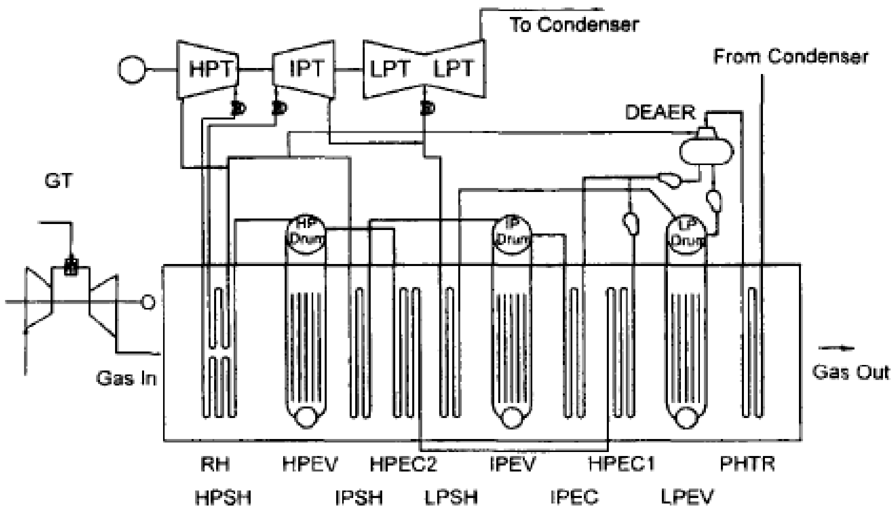


**Fig. 1** A schematic of a combined cycle power plant with triple pressure reheat HRSG

A gas turbine by General Electric Company [1] having a power output of 190 MW at ISO condition with natural gas is selected as a model. The HRSG is triple pressures reheat system which feeds HP, IP, and LP steam to the steam turbine for power production as shown in Fig. 1. The deaerator, which is heated by saturated HP steam, is used to supply the HP, IP and LP pumps with feed water. A condensate-water preheater is also included to fully utilize energy of the exhaust gas. The pressure level is set to a typical value and can be changed for other applications.

HP steam is heated in the HP superheater up to a temperature of 565°C, 100 bar which is at Subcritical Pressures and fed into the HP steam turbine (HPT). Steam expanded at HPT is mixed with the superheated IP steam at the exit of HPT, and heated at reheater. The exhaust steam after expansion in IP steam turbine (IPT) is mixed with the superheated LP steam.

## 3. Performance Analysis

In the HRSG, (water or steam) and exhaust gas of the gas turbine flow inside and outside of the tube respectively. Water or steam and exhaust gas exchange heat through a Cross-flow type Heat Exchanger at each section such as Economizer, Evaporator, and Superheater. For the simplicity of simulation procedure, each heat exchanger is assumed to be a Counter-flow type through the whole HRSG because the inlets of the water and the exhaust gas locate in the opposite side. The heat transferred at each module is calculated using the energy balance equation as described in Eq. (1).

$$\dot{Q} = \dot{m}_g (h_{g,in} - h_{g,out}) = \dot{m}_s (h_{s,out} - h_{s,in}) \qquad (1)$$

The gas side and water/steam side temperatures at the inlet and outlet of each section of the HRSG are determined using this energy balance equation

Using the well Known effectiveness –NTU method with overall heat transfer coefficient $U_{design}$, the area of HRSG together with other related parameters is calculated based on design parameters shown in equation

$$A = \frac{(\dot{m}C_p)_{\min} \text{ x NTU}}{U_{design}} \qquad (2)$$

$$\text{NTU} = f(\varepsilon,\ \frac{(\dot{m}C_p)_{\min}}{(\dot{m}C_p)_{\max}}) \qquad \varepsilon = \frac{Q}{Q_{\max}}$$

$Q_{\max}$ is the maximum heat transfer rate that could possibly be delivered by the heat exchanger [5].

The heat transfer area can be evaluated with the fin geometry data, and this area should match the one determined by the effectiveness-NTU method as above. Through the iterative procedure, the area is determined when the one equals to the other within a prescribed criterion.

Off-design performance analysis of the HRSG [6] is carried out using the design parameters described as above. The procedure to determine the thermodynamic properties at each section (**T , p , h***)* is repeated until the parameters converge to the assumed value. For illustration, the NTU at off-design operation is given by Eq. (3).

$$NTU = \frac{A \text{ x } \mathrm{U}_{\text{off-design}}}{(\dot{m}C_p)_{\min}} \tag{3}$$

Then, we can determine effectiveness as a function of NTU and actual heat transfer rate, ***as*** in Eq. (4). Also we can determine thermodynamic properties such as temperature, pressure, and enthalpy at each heat exchanger during off-design operation.

$$Q_{\text{off-design}} = \varepsilon \text{ x} \dot{Q}_{\max} \tag{4}$$

Off-design performance of gas turbine is estimated with the gas turbine performance curves. Steam turbine is assumed to operate on the sliding pressure mode during off-design operation. Performance of each section of steam turbine is analyzed using the well-known Spencer/Cotton/Cannon correlations [10]

Design conditions such as atmospheric conditions and cycle parameters are shown in Table 1. At this working condition, the exhaust gas of gas turbine is 451 kg/s in flow rate and 625°C in temperature, which is suitable for the conventional steam turbine with, reheat section.

**Table 1: Design Conditions and Cycle Parameters**

| **Parameter** | **unit** | **Data** |
|---|---|---|
| Design conditions | | |
| Temperature | ℃ | 30 |
| Altitude | m | 0 |
| Relative humidity | % | 65 |
| Cycle parameters | | |
| HP and/or RH steam temp. | ℃ | 565 |
| HP steam pressure | bar | 100 |
| IP steam pressure | bar | 20 |
| LP steam pressure | bar | 2 |
| Condenser pressure | bar | 0.05 |

**Table 2: Different Arrangement of Components in High Temperature Zone**

| **Case** | **Arrangement of Heat Exchangers in order** |
|---|---|
| 1 | High Pressure Superheater(HPSH) → Reheater(RH) |
| 2 | Reheater(RH) → High Pressure Superheater(HPSH) |
| 3 | High Pressure Superheater(HPSH)/Reheater(RH) (Parallel) |

## 4. Results

The optimized arrangement of each section, e.g., HP superheater (HPSH), reheater (RH), and HP economizer (HPEC), is of concern in this study. There are many design parameters to be concerned to fully understand the effect of the design parameters on the performance of a combined cycle power plant. However, we could see the effect of only one design parameter through the one cycle simulation due to the complexity of the combined cycle power plant. In this paper, the effects of the arrangements of components in the High and Intermediate Temperature zones are studied because they are the well-known parameters that could affect the system performance. Off-design performance is also included because it reflects the actual operation of the power plant itself.

The base model is the one with the parallel configuration of the HPSH and RH (case 3) as described in Table 2 and shown in above Fig. 1. The results are reviewed for three categories based on the arrangement of the components in the high-temperature and intermediate-temperature zones and its effect on the off-design performance of the steam turbine.

### 4.1 The effects of the arrangement of components in the High-Temperature zone

The position of the HPSH and RH could be a key factor in the steam cycle performance, and three cases are considered in this section as summarized in Table 2. The orders of the components are HPSH/RH (parallel) for High-Temperature zone and IPSH, HPEC2, LPSH for Intermediate-Temperature zone as shown in above Fig. 1. The heat transferred at the HRSG is 237.2 MW. Flow rate of condensate is 71.1 kg/s (54.9, 8.8, 7.4**)** kg/s of HP, IP, LP steam, respectively. Gross power output of the steam turbine is 100.9 MW (22.9, 39.7, and 38.3) MW at HP, IP, and LP turbine respectively)

**Table 3: Performance data of HRSG and Steam Turbine with changed arrangement of Components in high temperature zone**

| **Arrangement of Components** | | **case1** | **case2** | **case3** |
|---|---|---|---|---|
| **S.NO.** | **HPSH** | | | |
| **1** | Mass flow rate(Kg/s) | 57.2 | 55.2 | 54.9 |
| **2** | Outlet temp. (℃) | 565 | 546 | 565 |
| **3** | Outlet pressure( bar) | 100 | 100 | 100 |
| **4** | Inlet temp. (℃) | 310 | 311 | 311 |
| | **RH** | | | |
| **5** | Mass flow rate (kg/s) | 65.6 | 64.0 | 63.7 |
| **6** | Outlet temp. (℃) | 522 | 564 | 565 |
| **7** | Outlet pressure (bar) | 20 | 20 | 20 |
| **8** | Inlet temp. (℃) | 338 | 325 | 337 |
| | **IPSH** | | | |
| **9** | Mass flow rate (kg/s) | 8.4 | 8.8 | 8.8 |
| **10** | Outlet temp. (℃) | 310 | 311 | 311 |
| **11** | Outlet pressure( bar) | 21 | 21 | 21 |
| **12** | Inlet temp.( ℃) | 216 | 216 | 216 |
| | **LPSH** | | | |
| **13** | Mass flow rate (kg/s) | 6.9 | 7.3 | 7.4 |
| **14** | Outlet temp. (℃) | 253 | 255 | 255 |
| **15** | Outlet pressure( bar) | 2 | 2 | 2 |
| **16** | Inlet temp. (℃) | 121 | 121 | 121 |
| | **LPT inlet (IPT outlet + LPSH outlet)** | | | |
| **17** | Mass flow rate( kg/s) | 72.5 | 71.3 | 71.2 |
| **18** | Temperature(℃) | 228 | 255 | 258 |
| **19** | Heating steam to (kg/s) deaerator | 2.0 | 1.8 | 1.9 |
| **20** | HPT power (MW) | 23.9 | 22.3 | 22.9 |
| **21** | IPT power (MW) | 38.9 | 40.2 | 39.7 |
| **22** | LPT power (MW) | 37.1 | 38.2 | 38.3 |
| **23** | Total power, net (MW) | 97.1 | 97.9 | 98.1 |
| **24** | Deviation in total power% | - | 0.8 | 0.9 |

As seen in Table 3, the steam temperatures at HPSH and RH outlet can be set to 565°C only in the base model (case 3). The approach temperature difference (gas in - steam out) for the 2nd heat exchanger (RH in case 1 and HPSH in case 2) is set to 15°C for cases 1 and 2. Thus the outlet temperatures at RH (case 1) and HPSH (case 2) are limited to 523°C and 546°C, respectively, depending on the gas temperature at the 2nd heat exchanger inlet (the 1st heat exchanger outlet). In other words, for case1, the gas at the 2nd heat exchanger (RH) inlet has room to warm up the RH steam only to 523°C.

In case 1, the lower steam temperature at IPT inlet results in lower thermal efficiency, lower steam quality at the turbine outlet, and lower power output, if the steam flow rate is kept nearly the same. Also the lower steam temperature at LPT inlet is assumed to be the main reason for the lower power output of LPT. In case 2, lower steam temperature at HPT inlet (HPSH outlet) is a good reason for the lower power output of HPT. Case 3 shows greater power output of HPT with higher steam temperature at HPSH outlet compared to case 2. However, even though the steam conditions (flow rate and temperature) and the power outputs of each turbine are, to some extent, different in three cases, the difference in net power output of steam turbine is not significant. This result indicates that the optimization process in this study should be oriented to the minimization of certain criteria such as economical factor and heat transfer area, for example.

For comparison, only the arrangements are changed while any other parameters including pinch point (PP) temperature difference, approach (AP) temperature difference, etc., are set to constant. Case

3 shows that the heat transfer area of HP part (HPSH, RH, HPEVAP) is around 40% of the total surface as in Fig. 3 and its contribution in heat recovery is around 60% of total as estimated in Fig. 2. To say nothing of the important role of HP part in heat recovery such as steam generation, we should take notice of the cost. The cost of a HRSG is very sensitive to the HP part area because the price of materials for the high temperature part is about 20% higher than other parts, which is conjectured from the existing HRSG manufacturers' data (Dechamps, 1997).

The deviations in total heat transfer area are 3.1% and 4.2% for case 1 and case 2, respectively; as compared to case 3 (see Table 4). Looking into the HPSH and RH parts only, the deviations are 8.0% and 24.4%, respectively. It is not easy to assess the exact cost of the HRSG because the price is the proprietary information of each company. However, for example, if we use Rs 1200/$m^2$ for HP components and Rs 1000/ $m^2$ for the Intermediate Components for simple evaluation, the difference in estimated price of the HRSG between case 2 and case 3 considering the area only is about Rs sixty lakhs (4% of the total estimated HRSG price). This difference in the investment cost should not be ignored if we have nearly same power output as in this study.

### 4.2 The effects of the arrangement of components in the Intermediate-Temperature zone

Three cases considered in this section are illustrated in Table 5, which shows the arrangement of HP economizer, IP superheater, and LP superheater. The results reveal that the difference in total HRSG area between cases 4 and 5 is about 5%, which is dominated by HP economizer (about 16700, 14400, 20700 $m^2$ for cases 3, 4, and 5, respectively) as shown in Table 6. If only the area of the 2$^{nd}$ HP economizer (HPEC2) is observed, the difference between cases 4 and 5 is as large as 44%.

Looking into the gas temperature profile in the Intermediate Temperature zone of the HRSG, the gas temperature at HP economizer inlet in case 5 is the lowest among cases 3, 4, and 5 because the HP economizer lies in the back end of the IP part for case 5. It means that, in case 5, the temperature difference between gas and feedwater is the smallest among three cases because the water temperature at HP economizer outlet is fixed for three cases, and the heat transfer area should be enlarged in order to obtain the same thermal performance. It should be understood that the thermal performance of the HPEC2 (inlet and outlet temperatures and flow rate of water) was already set at design stage. As a result, the HP economizer of case 5 has the largest area among three cases. The economical factor will not be repeated here. The difference in power output is not significant even though the steam conditions are a little bit different (see Table 7).

**Table 4: Area of each Heat Exchanger of HRSG at High Temperature zone**

| **State Positions** | **case 1 ($m^2$)** | **case 2 ($m^2$)** | **case 3 ($m^2$)** |
|---|---|---|---|
| HP Part | 95152 | 95521 | 90595 |
| IP Part | 18773 | 19201 | 19272 |
| LP Part | 24104 | 24665 | 23950 |
| Total Area | 138029 | 139387 | 133817 |
| Deviation in Total Area | 3.1% | 4.2% | |
| Area of HPSH+RH | 20352 | 23435 | 18842 |
| Deviation in area of HPSH+RH | 8.0% | 24.4% | - |

**Table 5 Different Arrangement of Components in Intermediate Temperature Zone**

| **Case** | **Arrangement of heat exchangers in order** |
|---|---|
| 3 | Intermediate Pressure Superheater (IPSH)→High Pressure Economizer(HPEC2) →Low Pressure Super Heater (LPSH) |
| 4 | High Pressure Economizer HPEC2 → Intermediate Pressure Superheater( IPSH)→ Low Pressure Super Heater( LPSH) |
| 5 | Intermediate Pressure Superheater(IPSH)→ Low Pressure Super Heater (LPSH)→ High Pressure Economizer( HPEC2) |

**Table 6: Table 4: Area of each Heat Exchanger of HRSG at Intermediate temperature zone**

| State Positions | case 1 ($m^2$) | case 2 ($m^2$) | case 3 ($m^2$) |
|---|---|---|---|
| HP Part | 90595 | 87879 | 94655 |
| IP Part | 19272 | 19672 | 18816 |
| LP Part | 23950 | 24779 | 24883 |
| Total Area | 133817 | 132330 | 138354 |
| Deviation in Total Area | 1.1% | | 4.6% |
| Area of HPSH+RH | 16715 | 14396 | 20710 |
| Deviation in area of HPSH+RH | 16.1% | - | 43.9% |

**Table 7:** Performance data of HRSG and Steam Turbine with changed arrangement of Components in Intermediate Temperature zone

| Arrangement of the Components | | case 3 | case4 | case5 |
|---|---|---|---|---|
| **S.NO.** | **HPSH** | | | |
| **1** | Mass flow rate(kg/s) | 54.9 | 54.4 | 55.0 |
| **2** | Outlet temp.(℃) | 565 | 565 | 564 |
| **3** | Outlet pressure(bar) | 100 | 100 | 100 |
| **4** | Inlet temp.(℃) | 311 | 311 | 311 |
| | **RH** | | | |
| **5** | Mass flow rate(kg/s) | 63.7 | 63.7 | 63.5 |
| **6** | Outlet temp.(℃) | 565 | 564 | 565 |
| **7** | Outlet pressure(bar) | 20 | 20 | 20 |
| **8** | Inlet temp.(℃) | 337 | 329 | 338 |
| | **IPSH** | | | |
| **9** | Mass flow rate(kg/s) | 8.8 | 9.4 | 8.5 |
| **10** | Outlet temp.(℃) | 311 | 259 | 311 |
| **11** | Outlet pressure(bar) | 21 | 21 | 21 |
| **12** | Inlet temp(℃) | 216 | 216 | 216 |
| | **LPSH** | | | |
| **13** | Mass flow rate(kg/s) | 7.4 | 7.5 | 7.5 |
| **14** | Outlet temp.(℃) | 255 | 257 | 306 |
| **15** | Outlet pressure(bar) | 2 | 2 | 2 |
| **16** | Inlet temp.(℃) | 121 | 121 | 208 |
| | **LPT inlet (IPT outlet + LPSH outlet)** | | | |
| **17** | Mass flow rate(kg/s) | 71.2 | 71.2 | 70.9 |
| **18** | Temperature(℃) | 258 | 258 | 263 |
| **19** | Heating steam to deaerator(kg/s) | 1.9 | 1.9 | 1.8 |
| **20** | HPT power(MW) | 22.9 | 22.7 | 22.9 |
| **21** | IPT power(MW) | 39.7 | 39.7 | 39.5 |
| **22** | LPT power(MW) | 38.3 | 38.4 | 38.2 |
| **23** | Total power, net(MW) | 98.1 | 97.9 | 98.2 |
| **24** | Deviation in total power(%) | 0.2 | - | 0.3 |

### 4.3 Off-design performance:

Off-design performance with variations of the ambient temperature and gas turbine load condition is shown in Figs. 3 to Figs 5.

As shown in Fig. 3, the power output of steam turbine decreases with higher ambient temperature except the region between 5 and 15°C. This can be viewed from the fact that the variations of flow rate and temperature at the gas turbine exhaust at off-design condition affect the steam turbine performance. As shown in Fig. 4, the flow rate of exhaust gas does not vary sharply with ambient temperature near 10°C compared to the region above 15°C. But the temperature of exhaust gas varies gradually throughout the entire range of ambient temperature. Thus near 10°C, the power output of steam turbine increases even though the gas flow rate decreases as in Fig.3. It is presumed that the increase of the exhaust gas temperature dominates the steam production at the HRSG near the ambient temperature of 10°C. It should be reminded that variations of the flow rate and exhaust gas temperature of the gas turbine fully depend on the control algorithm of each gas turbine, which uses constant turbine inlet temperature or constant mass flow rate depending on each manufacturer

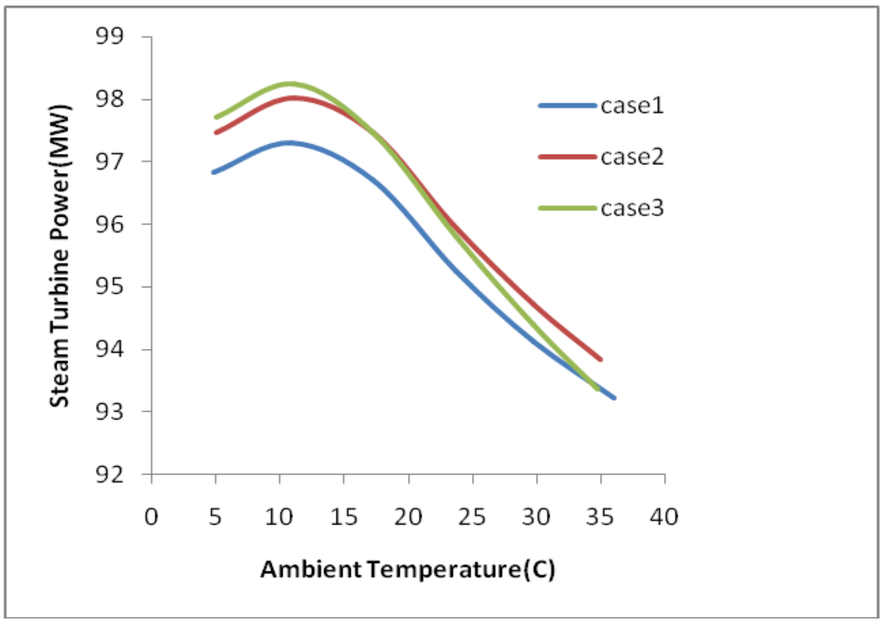


Fig. 3. a) off Design Performance of Steam Turbine at different case Arrangement of components in HRSG

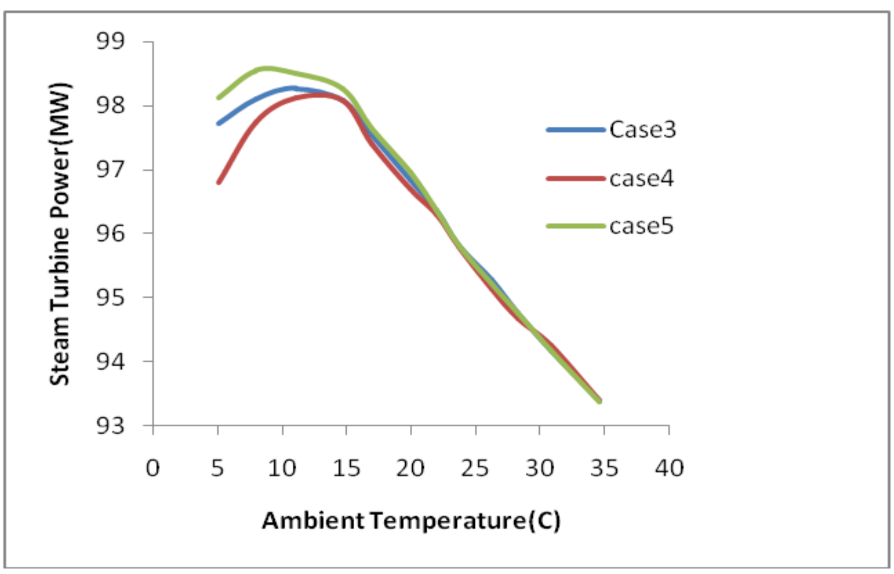


Fig. 3. b) off Design Performance of Steam Turbine at different case Arrangement of components in HRSG

with ambient temperature near 10°C compared to the region above 15°C. But the temperature of exhaust gas varies gradually throughout the entire range of ambient temperature. Thus near 10°C, the power output of steam turbine increases even though the gas flow rate decreases as in Fig. 4. It is presumed that the increase of the exhaust gas temperature dominates the steam production at the HRSG near the ambient temperature of 10°C. It should be reminded that variations of the flow rate and exhaust gas temperature of the gas turbine fully depend on the control algorithm of each gas turbine, which uses constant turbine inlet temperature or constant mass flow rate depending on each manufacturer.

The variation of the power outputs of cases 2 and 3 in Fig. 3(a) is also of interest. Near 20°C, the variation curves of case 2 and case 3 intersect. If the plant operates over 20°C all the time, the configuration should be set as case 2 not as case 3 for more power output. If the difference in the power output is not negligible, we should save the operating cost by choosing the right scheme suitable to actual working condition of that plant. This is one of the reasons why we have to study the off-design operating characteristics carefully. For cases 3, 4, and 5, the difference seems to be insignificant and the trend is nearly same as shown in Fig. 3(b).

The variation with gas turbine load is shown in Fig. 5. For cases 1, 2, and 3, the slope differs from each other and the curves are quite linear as depicted in Fig. 5(a). However, we could see little difference for cases 3, 4, and 5 as shown in Fig. 5(b). Considering lifetime revenue through electricity sales, the difference in off-design performance with gas turbine load could be an important factor. It means that we should select a suitable heat exchanger configuration considering the range of the operating conditions of the plant such as working temperature and/or load condition, etc. By way of illustration, case 1 is the worst choice considering the poor performance of the steam turbine, and case 2 is the worst choice with the high investment cost of a HRSG. However, the investment cost should be re-evaluated considering the annual revenue through the sales of electricity, which is also dependent on the economic factors such as net price per unit power, inflation rate, and interest rate, etc.

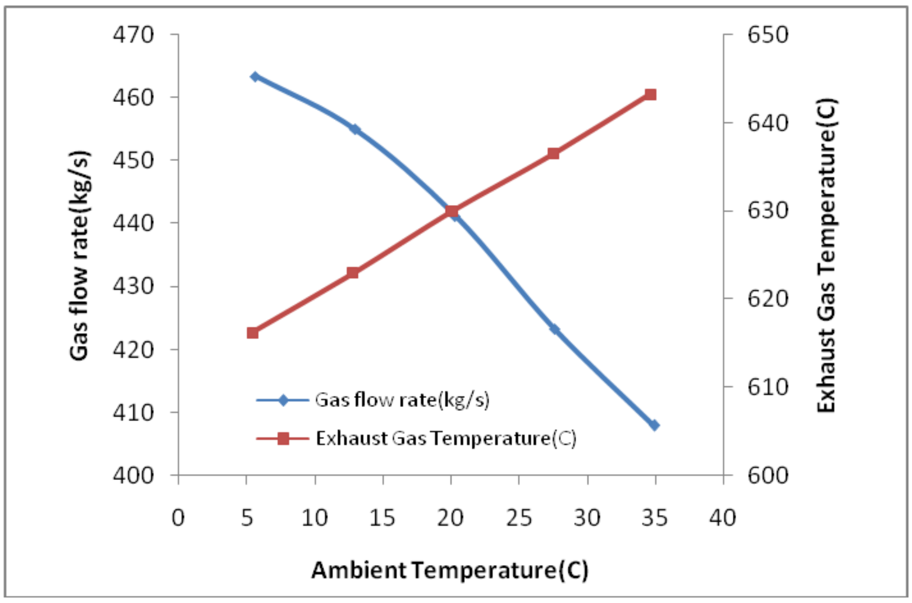


Figure 4: Variation of the steam flow rate and Exhaust Gas temperature of the Gas Turbine with ambient Temperature

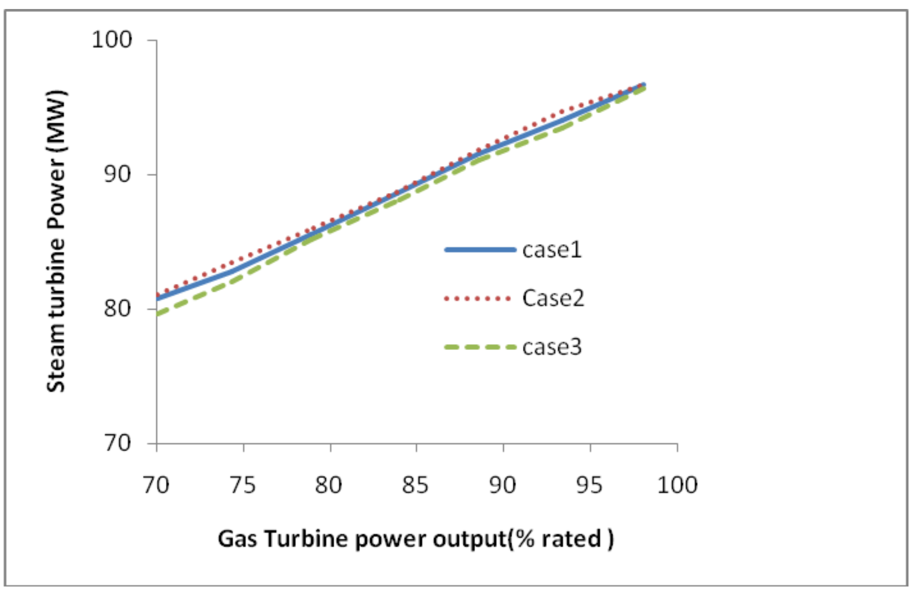


Figure 5: a) Off design performance of Steam Turbine with Gas Turbine load

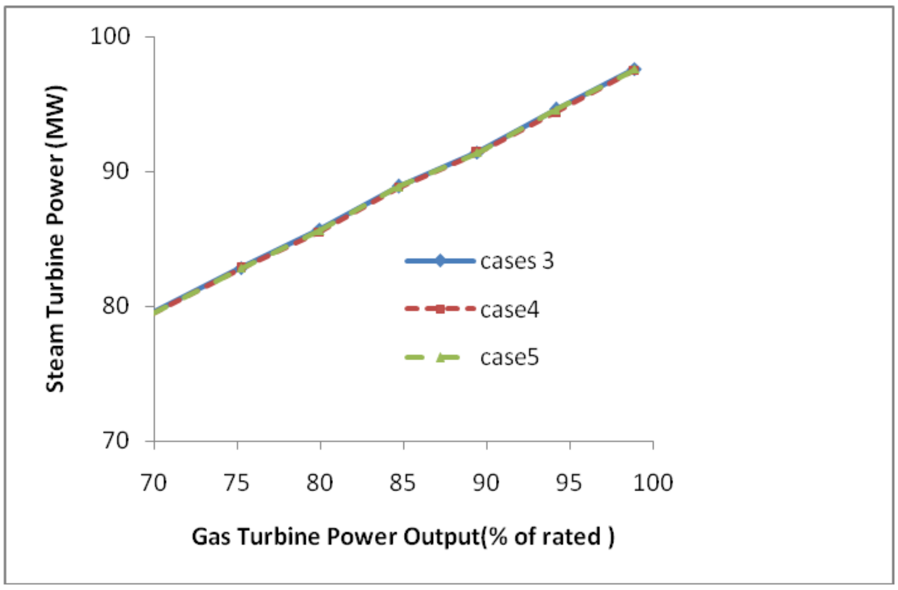


Figure 5: b) Off design performance of Steam Turbine with Gas Turbine load at different case arrangement of components in HRSG

## Conclusions:

In this study, the operating characteristics of triple pressure reheat HRSGs is analyzed at design and off design conditions and placing the components of HRSG at different locations and its effects on e thermal performance of the plant is analyzed.

1. It is shown that we could expect considerable difference in HRSG area and investment cost depending on the heat exchanger arrangement such as HP superheater and reheater even though the thermal performance is not in great deviation.
2. Among the cases considered in this study, parallel configuration (case 3) shows better performance in both thermal performance and investment cost of HRSG.
3. The arrangement of heat exchangers in the Intermediate-Temperature zone such as IP superheater, HP economizer, and LP superheater has also been investigated. There was a big difference in the area of HP economizer depending on the arrangement. It is recommended that HP economizer locate in the upstream of HRSG.
4. It is revealed that the off-design performance is also different depending on the arrangement of the heat exchangers.
5. It is revealed that the arrangement of HP superheater and reheater has a great impact on the steam turbine performance including off-design operation, while that of the components in the intermediate temperature zone has negligible impact even with **5%** change of the HRSG area.

Therefore, the effects of the configuration should be well predicted for the economic operation of the plant. The off-design performance could be a key factor in the optimum design considering the operating range of the plant, and should be reflected on the design in advance. The results of this study can be used as a guideline in estimating the optimum configuration of the triple pressure HRSG.

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