Performance Analysis of a Triple Pressure HRSG

Jee-Young Shin*, Young-Seok Son
School of Mechanical Engineering, Dong-Eui University, Busan 614-714, Korea
Moo-Geun Kim
School of Mechanical and Automotive Engineering, Inje University, Gyeongnam 621-749, Korea
Jae-Soo Kim
Plant Eng. & Const. Division, Hyundai Engineering and Construction Co., Ltd., Seoul 110-793, Korea
Yong-Joon Jeon
Hyundai Inst. of Construction Technology, Hyundai Engineering and Construction Co., Ltd., Yongin 449-716, Korea

Operating characteristics of a triple pressure reheat HRSG are analyzed using a commercial software package (Gate Cycle by GE Enter Software). The calculation routine determines all the design parameters including configuration and area of each heat exchanger. The off-design calculation part has the capability of simulating the effect of any operating parameters such as power load, process requirements, and operating mode, etc., on the transient performance of the plant. The arrangement of high-temperature and intermediate-temperature components of the HRSG is changed, and its effect on the steam turbine performance and HRSG characteristics is examined. It is shown that there could be a significant difference in HRSG sizes even though thermal performance is not in great deviation. From the viewpoint of both economics and steam turbine performance, it should be carefully reviewed whether the optimum design point could exist. 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 : Triple Pressure Heat Recovery Steam Generator, Combined Cycle Power Plant, Performance Analysis, Efficiency, Optimum Design

Nomenclature

A : Area
c_p : Specific heat
i : Enthalpy
$\dot{m}$: Mass flow rate
NTU : Number of transfer unit
$\dot{Q}$: Heat transfer rate
U : Overall heat transfer coefficient

* Corresponding Author,
E-mail : jyshin@dongeui.ac.kr
TEL : +82-51-890-1650; **FAX** : +82-51-890-2232
School of Mechanical Engineering, Dong-Eui University, Busan 614-714, Korea. (Manuscript **Received** September 23, 2002; **Revised** August 12, 2003)

Greek symbols

ε : Effectiveness

Subscripts

g : Gas side
s : Steam/water side
in : Inlet
max : Maximum
out : Outlet

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. (El-Masri and Foster-Pegg, 1996). 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 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 to 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 (Ganapathy, 1985 ; Manen, 1994 ; Pasha and Jolly, 1995 ; Ahn and Antonovsky, 2001). 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 are 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 researches 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 (Kehlhofer, 1991). 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 using a computer program, Gate Cycle, which is developed by GE Enter Software (2001). 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

A gas turbine by General Electric Company (model PG7251FB) having a power output of 190 MW at ISO condition with natural gas is selected as a model. The HRSG is a triple pressure 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℃ 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.

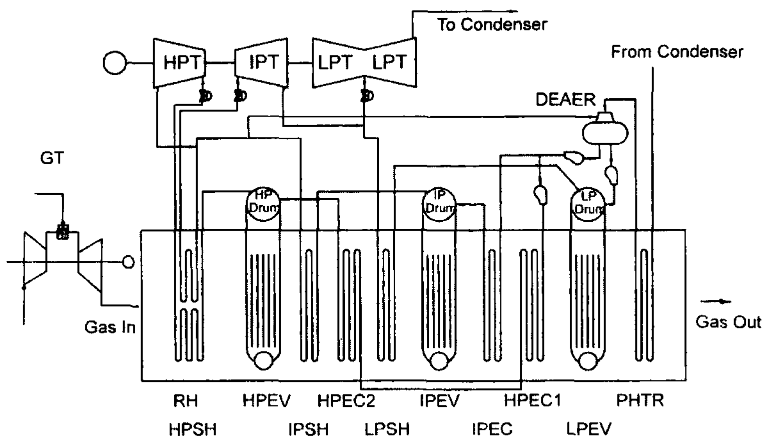


Fig. 1 A schematic of a combined cycle power plant with triple pressure reheat HRSG

3. Performance Analysis

In the HRSG, water (and/or steam) and exhaust gas of the gas turbine flow inside and outside of the tube respectively. Water (and/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).

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

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 heat transfer area of a HRSG together with other related parameters is calculated based on the design parameters as shown in Eq. (2).

$$A=\frac{(\dot{m}c_p)_{min}NTU}{U_{design}} \quad (2)$$

where $UTU=f(\varepsilon, (\dot{m}c_p)_{min}/(\dot{m}c_p)_{max})$ and $\varepsilon=Q/Q_{max}$. Q_{max} is the maximum heat transfer rate that could possibly be delivered by the heat exchanger (Incropera and DeWitt, 1996).

The heat transfer area can also be evaluated with the tube/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 (Kehlhofer, 1991) is carried out using the design parameters described as above. The procedure to determine the thermodynamic properties at each section (T, p, i) is repeated until the parameters converge to the assumed value. For illustration, the NTU at off-design operation is given by Eq. (3).

$$NTU=\frac{AU_{off-design}}{(\dot{m}c_p)_{min}} \quad (3)$$

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

$$\dot{Q}_{off-design}=\varepsilon\dot{Q}_{max} \quad (4)$$

The calculation has been carried out using the computer program, Gate Cycle version 5.40, developed by GE Enter Software (GE Enter Software,

Table 1 Design conditions and cycle parameters

Item	Unit	Data
Design conditions		
Temperature	℃	15
Altitude	m	0
Relative humidity	%	60
Cycle parameters		
HP and/or RH steam temp.	℃	565
HP steam pressure	MPa	10
IP steam pressure	MPa	2
LP steam pressure	MPa	0.2
Condenser pressure	kPa	5

2001). Off-design performance of gas turbine is estimated with the gas turbine performance curve sets in the library of the software. 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 (Spencer et al., 1963).

Design conditions such as ambient 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℃ in temperature, which is suitable for the conventional steam turbine with reheat section.

4. Results

The optimized arrangement of each section, e.g., HP superheater (HPSH), reheater (RH), and/or 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

Table 2 Cases with different arrangement of components in the high-temperature zone

Case	Order of heat exchangers
I	HPSH → RH
II	RH → HPSH
III	HPSH/RH (Parallel)

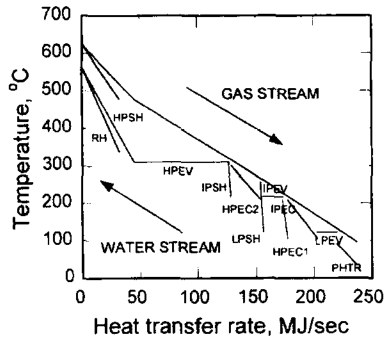


Fig. 2 Heat transfer rate-temperature diagram of the triple pressure HRSG (case III)

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 III) as described in Table 2 and foregoing 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. Figure 2 shows the relation between the heat transfer rate and exhaust gas temperature at the HRSG for the base model (case III). The orders of the components are HPSH/RH (parallel) for high-temperature zone and IPSH, HPEC2, LPSH for intermediate-temperature zone as shown in foregoing 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, 38.3 MW at HP, IP, LP turbine, respectively).

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

Table 3 Performance data of the HRSG and steam turbine (Changes in the arrangement of high-temperature components)

Item	Unit	Case I	Case II	Case III
HPSH				
Mass flow rate	kg/s	57.2	55.2	54.9
Outlet temp.	℃	565.3	546.3	565.0
Outlet pressure	MPa	10.0	10.0	10.0
Inlet temp.	℃	310.9	311.0	311.0
RH				
Mass flow rate	kg/s	65.6	64.0	63.7
Outlet temp.	℃	522.6	564.9	565.0
Outlet pressure	MPa	2.0	2.0	2.0
Inlet temp.	℃	338.0	325.0	337.9
IPSH				
Mass flow rate	kg/s	8.4	8.8	8.8
Outlet temp.	℃	310.8	311.0	311.0
Outlet pressure	MPa	2.1	2.1	2.1
Inlet temp.	℃	216.6	216.6	216.6
LPSH				
Mass flow rate	kg/s	6.9	7.3	7.4
Outlet temp.	℃	253.5	255.2	255.1
Outlet pressure	MPa	0.2	0.2	0.2
Inlet temp.	℃	121.6	121.6	121.6
LPT inlet (IPT outlet+LPSH outlet)				
Mass flow rate	kg/s	72.5	71.3	71.2
Temperature	℃	228.7	255.3	258.1
Heating steam to deaerator	kg/s	2.0	1.8	1.9
HPT power	MW	23.9	22.3	22.9
IPT power	MW	38.9	40.2	39.7
LPT power	MW	37.1	38.2	38.3
Total power, net	MW	97.1	97.9	98.1
Deviation in total power	%	–	0.8	0.9

In case I, 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 II, lower steam temperature at HPT inlet (HPSH outlet) is a good reason for the lower power output of HPT. Case III shows greater power output of HPT with higher steam temperature at HPSH outlet compared to case II. 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 temperature difference, approach temperature difference, etc., are set to constant. Case III 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 I and case II, respectively, as compared to case III (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 US $ 120/m^2 for HP components and US $ 100/m^2 for the remainder for simple evaluation, the difference in estimated price of the HRSG between case II and case III considering the area only is about six hundred thousand dollars (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.

Table 4 Area of each heat exchanger of the HRSG (m^2) (Changes in the arrangement of high-temperature components)

Case	I	II	III
HP part	95,152	95,521	90,595
IP part	18,773	19,201	19,272
LP part	24,104	24,665	23,950
Total area	138,029	139,387	133,817
Deviation in total area	3.1%	4.2%	–
Area of HPSH+RH	20,352	23,435	18,842
Deviation in area of HPSH+RH only	8.0%	24.4%	–

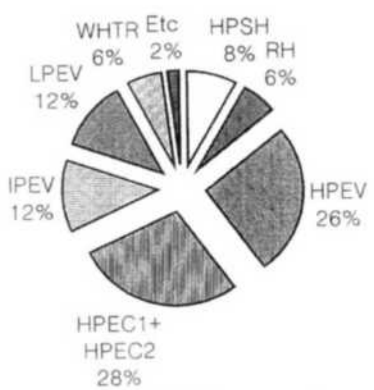


Fig. 3 Distribution of the heat transfer area of the HRSG (case III)

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 IV and V is about 5%, which is dominated by HP economizer (about 16700, 14400, 20700 m^2 for cases III, IV, and V, respectively) as shown in Table 6. If only the area of the 2nd HP economizer (HPEC2) is observed, the difference between cases IV and V 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 V is the lowest among cases III, IV, and V because the HP economizer lies in the back end of the IP part for case V. It means that, in case V, the temperature difference between gas and feedwater is the smallest among three cases because the water temperature at HP economizer outlet is

Table 5 Cases with different arrangement of components in the intermediate-temperature zone

Case	Order of heat exchangers
III	IPSH → HPEC2 → LPSH
IV	HPEC2 → IPSH → LPSH
V	IPSH → LPSH → HPEC2

Table 6 Area of each heat exchanger of the HRSG (m^2) (Changes in the arrangement of intermediate-temperature components)

Case	III	IV	V
HP part	90,595	87,879	94,655
IP part	19,272	19,672	18,816
LP part	23,950	24,779	24,883
Total area	133,817	132,330	138,354
Deviation in total area	1.1%	–	4.6%
Area of HPEC2	16,715	14,396	20,710
Deviation in area of HPEC2 only	16.1%	–	43.9%

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 V 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 7 Performance data of the HRSG and steam turbine (Changes in the arrangement of intermediate-temperature components)

Item	Unit	Case III	Case IV	Case V
HPSH				
Mass flow rate	kg/s	54.9	54.4	55.0
Outlet temp.	℃	565.0	565.0	564.9
Outlet pressure	MPa	10.0	10.0	10.0
Inlet temp.	℃	311.0	311.0	311.0
RH				
Mass flow rate	kg/s	63.7	63.7	63.5
Outlet temp.	℃	565.0	564.9	565.1
Outlet pressure	MPa	2.0	2.0	2.0
Inlet temp.	℃	337.9	329.6	338.0
IPSH				
Mass flow rate	kg/s	8.8	9.4	8.5
Outlet temp.	℃	311.0	259.8	311.0
Outlet pressure	MPa	2.1	2.1	2.1
Inlet temp.	℃	216.6	216.6	216.6
LPSH				
Mass flow rate	kg/s	7.4	7.5	7.5
Outlet temp.	℃	255.1	257.5	306.8
Outlet pressure	MPa	0.2	0.2	0.2
Inlet temp.	℃	121.6	121.6	208.0
LPT inlet (IPT outlet+LPSH outlet)				
Mass flow rate	kg/s	71.2	71.2	70.9
Temperature	℃	258.1	258.3	263.5
Heating steam to deaerator	kg/s	1.9	1.9	1.8
HPT power	MW	22.9	22.7	22.9
IPT power	MW	39.7	39.7	39.5
LPT power	MW	38.3	38.4	38.2
Total power, net	MW	98.1	97.9	98.2
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. 4~6.

As shown in Fig. 4, the power output of steam turbine decreases with higher ambient temperature except the region between 5 and 15℃. 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. 5, the flow rate of exhaust gas does not vary sharply with

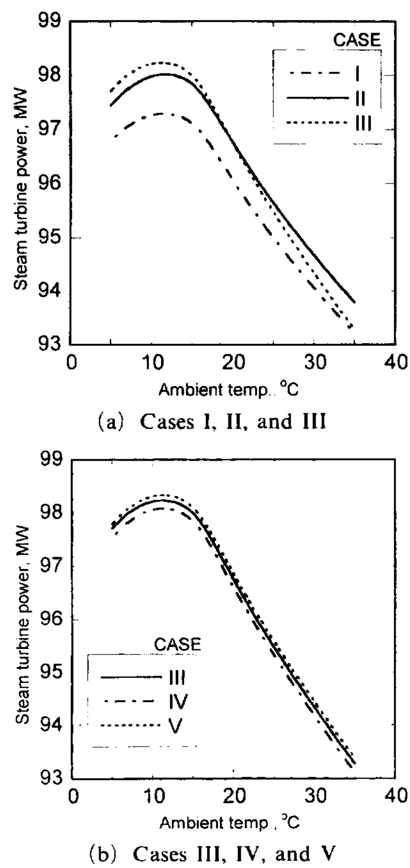


Fig. 4 Off-design performance of a steam turbine with ambient temperature

ambient temperature near 10℃ compared to the region above 15℃. But the temperature of exhaust gas varies gradually throughout the entire range of ambient temperature. Thus near 10℃, 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℃. 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 temperatuer or constant mass flow rate depending on each manufacurer.

The variation of the power outputs of cases II and III in Fig. 4(a) is also of interest. Near 20℃, the varitaion curves of case II and case III intersect. If the plant operates over 20℃ all the time, the configuration should be set as case II not as case III 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 a 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 III, IV, and V, the difference seems to be insignificant and the trend is nearly same as shown in Fig. 4(b).

The variation with gas turbine load is shown in Fig. 6. For cases I, II, and III, the slope differs from each other and the curves are quite linear as depicted in Fig. 6(a). However, we could see little difference for cases III, IV, and V as shown in Fig. 6(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 I is the worst choice considering the poor performance of the steam turbine, and case II 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.

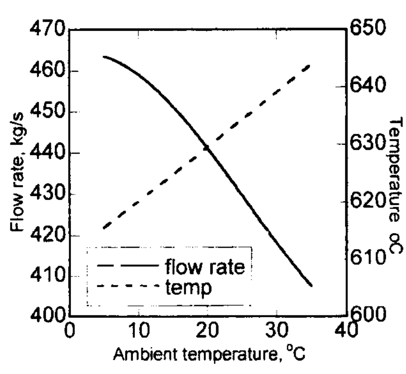
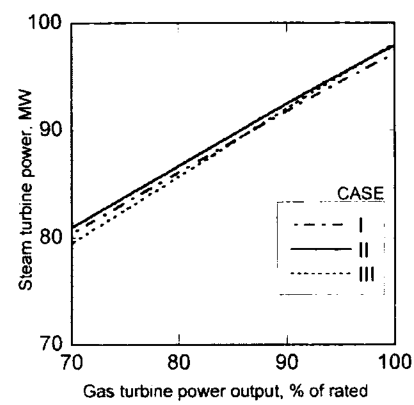
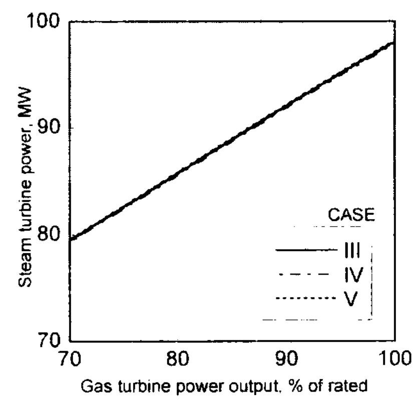


Fig. 5 Variations of the flow rate and temperature of exhaust gas of the gas turbine with ambient temperature



(a) Cases I, II, and III



(b) Cases III, IV, and V

Fig. 6 Off-design performance of a steam turbine with gas turbine load

5. Concluding Remarks

In this study, the operating characteristics of triple pressure reheat HRSGs are analyzed using a computer program, Gate Cycle, which is developed by GE Enter Software. The effects of the configuration of HP superheater and reheater on the thermal performance and economics of the plant are investigated.

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. Among the cases considered in this study, parallel configuration (case III) shows better performance in both investment cost of HRSG and thermal performance. 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.

From the viewpoint of both economics and heat recovery efficiency, it should be carefully reviewed whether the optimum design criterion could exist or not depending on the various configurations of HRSG. It is revealed that the off-design performance is also different depending on the arrangement of the heat exchangers. 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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