Gas Processing Journal

Vol. 5, No. 2, 2017
http://gpj.ui.ac.ir
DOI: http://dx.doi.org/10.22108/gpj.2018.111226.1031

Startup time improvement of combined cycle power plants based on stream splitting concept for heat recovery steam generators

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Article History
Received: 2018-05-27 Revised: 2018-07-29 Accepted: 2018-07-30

Abstract
One of the main components in combined cycles is the heat recovery steam generator (HRSG) which is responsible for energy recovery from gas turbine exhaust stream and produce superheated steam. In this paper, driving force plot and exergy analysis is used to study the behavior of heat exchangers of an HRSG in base case and off design conditions. The results indicated that performance of high pressure super heaters is far from their ideal operation. Based on this conclusion, a new configuration for the HRSG is proposed to boost its performance. This new configuration is based on split concept where it splits the hot flue gas to bypass the high pressure superheats. Therefore more energy is delivered to the high pressure evaporator and consequently steam production rate increases. Another advantage of this configuration is that it reduces start up time of the HRSG and consequently decreases start up time of the plant which results in higher flexibility of the system. To examine the applicability of the proposed approach, it is used to improve start up time in an existing combined cycle power plant. The results indicate that pressure and mass flow rate of HP stream reaches to its design point almost 20 minutes sooner. Finally, a sensitivity analysis is performed on the effect of split ratio on the power production. The results show that there is an optimum value for split ratio in which power production of steam turbine is maximized.

Keywords
Combined cycle power plant; HRSG; Stream splitting; Start up time

1. Introduction

Energy demand has increased rapidly during the last century due to huge population growth and technology improvement. Electricity is one of the most important forms of energy. Currently, thermal power plants have the highest share in electricity production throughout the world. To produce electricity, these power plants consume fossil fuels such as coal, natural gas, etc. which results in environmental pollution. To alleviate the situation, fuel consumption in these plants should be decreased. This could be done by increasing efficiency of the power plant. Different methodologies have been suggested to boost performance of power plants. Khoshgoftarmanesh et al. (2016) performed techno-economic analysis on different inlet air cooling systems to increase power production in Turbo-compressor stations. Bassily (2008) studied the effect of recuperated gas-reheat unit and compared its performance with simple gas turbines. He used this method to decrease exergy destruction in heat recovery steam generators (HRSG). The results indicated that this method has both thermodynamic and economic benefits. In another study, Bassily (2013) proposed a new technique of gas turbine blade cooling (steam-

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ISSN (Online): 2345-4172, **ISSN (Print):** 2322-3251

cooled technique). This technique allows the HRSG to be integrated with turbines which have higher turbine inlet temperature. Srinivas et al. (2008) analyzed steam injected gas turbines thermodynamically. Since a part of produced steam is sued in combustion chamber, power generation in steam cycle reduces slightly, but power generation in gas turbine increases. Heat recovery steam generators have a key role in combined cycle power plants which is recovering heat from gas turbine exhaust stream and produce steam for the steam turbine. Since it connects the Brayton and Rankine cycles, its improvement could have a considerable effect on the total system performance. Feng et al. (2014) proposed a mathematical model which has the ability to thermodynamically analyze different types of HRSG. They showed that heat exchangers layout is the most important parameter which affects the performance of the HRSG. Franco and Casarosa (2002) suggested using optimization process to achieve the optimum performance. Franco and Russo (2002) showed that pinch temperature is one of the most important parameters of system which should absolutely be considered as one of the decision variables in optimization processes. Based on their results, optimization process could increase efficiency of the plant up to 60%. Manassaldi et al. (2011) used mixed integer non-linear programming in GAMS to increase efficiency of an HRSG considering energy efficiency as their objective function. Tajik Mansouri et al. (2012) performed techno economic analysis on three kinds of HRSG including a dual pressure; a triple pressure with reheat and a triple pressure without reheat. They showed that HRSG systems with higher pressure levels have higher exergy efficiency, but their capital cost is also higher. Other researchers (Mohagheghi, & Shayegan, 2009; Nadir, & Ghenaiet, 2015; Woudstra, Woudstra, Pirone, & Stelt, 2010) reached to the same conclusion by using different methods. For example, Woudstra et al. (2010) used internal exergy efficiency, value diagram and exergy flow diagram concepts and Mohagheghi and Shayegan (2009) developed non-linear equation in general form and then solved it utilizing the hybrid Newton method. Another method to improve performance of the HRSG is to increase their performance during start up and shut down procedures. When start up time of a combined cycle decrease, it means that the plant could reach to its nominal capacity sooner and therefore energy loss during this time decreases. Alobaid et al. (2012) used simulation techniques to study the effect of fast start up gas turbines on the benson heat recovery systems. In another study, they used both static and dynamic simulations to analyze the startup procedure in a combined cycle power plant. Since most big power plants have triple pressure HRSG, many researchers focused on these systems (Alobaid, Postler, Ströhle, Epple, & Kim, 2008). Mertens et al. (2016) studied transient response of a triple pressure HRSG to the startup procedure. They considered geometry data, system descriptions and heat transfer calculations in their simulation. Triple pressure HRSG is also studied by other researchers (Zhang, Zheng, Yang, & Liu, 2016). Exergy analysis is a very beneficial tool to study energy systems. It shows the improvement possibilities in any system and helps researcher to focus on the equipment with lowest performance. Sheikhi et al. (2014) studied refrigeration cycle of Tabriz olefin plant using exergy analysis and found the equipment which has the lowest exergy efficiency. Sheikhi et al. (2015) used advanced exergy analysis to study the same olefin plant. They found that endogenous section is the main reason for exergy destruction of all components. Ghorbani et al. (2016) performed exergy and exergoeconomic analyses on a novel mixed fluid cascade natural gas liquefaction process. They showed that HX8 has the highest exergoeconomic factor, which is equal to 89.49%. Another beneficial tool is the pinch analysis which is perfect for studying heat exchangers network. Yoon et al. (2007) studied heat exchanger network (HEN) of an industrial ethylbenzene plant using pinch analysis. They changed the HEN with adding and removing some heat exchangers and reduced energy cost by 5.6%. Fernández-Polanco & Tatsumi (2016) integrated anaerobic digestion and wastewater treatment plant. They used pinch analysis to do it. They studied different types of combined heat and power system to be integrated with the main plant and concluded that micro gas turbines are not a good fit. Combined utilization of pinch and exergy analysis could be more beneficial. Quijera et al. (2013) performed combined pinch and exergy analyses on the integrated system of solar energy and heat pump. It helped them to reduce the amount of energy loss to the ambient. In another study, they performed the same analysis on diary process coupled with solar thermal energy (Quijera, & Labidi, 2013). Ghorbani et al. (2012) studied a refrigeration cycle in natural gas liquid recovery plant with combined pinch and exergy analyses. Using optimization techniques, they reduced power consumption of the compressor by 170 kW.

In the next step, they changed the refrigerant fluid to reduce power consumption of the system further. Arriola-Medellín et al. (2014) studied a power plant with this methodology and proposed a new design which increased total efficiency of the plant by 0.81%.

In this paper, a new HRSG layout is proposed to improve performance of the system. It is based on the split concept which is used in pinch analysis a lot. The split concept has been used before in boosting HRSG performance. Zebian & Mitsos (2014) proposed a method which benefits from split concept. In their method, flue gas recycling process is used to reduce heat exchange area and power required for recycling, simultaneously. To do so, they split the hot flue gas and diluted it with recycled gas to achieve tolerable temperature at the HRSG inlet. Using a multi-objective optimization process, required power and area was reduced by 18% and 12%, respectively. In this paper, the splitting occurs at the HRSG inlet, not at its outlet. This method results in higher steam production which results in higher power production. It also reduces thermal stresses in super heaters. Therefore it is a very beneficial method especially during the summers in which higher ambient temperature reduces power production in power plants. Both exergy and pinch analyses are applied on the system to study the effect of these changes on the plants performance and startup time and the results of the new method are compared with the base case condition.

System Description

Damavand power plant is selected as the case study to perform the analysis. It is located in Tehran, Iran. With total capacity of 2868MW, it is one of the biggest power plants in the Middle East. It is comprised of 12 gas turbine units (with 159 MW capacity for each gas turbine) and 6 Rankine cycles with a total capacity of 960MW. Also, there are 12 HRSGs in the plant. Schematic diagram of the plant is shown in Figure 1. For simplicity, only one gas turbine, one HRSG and one steam turbine are shown in the figure.

As shown in the figure, the system begins with Brayton cycle. In this cycle, first the air is compressed. Then it is delivered to the combustion chamber where the combustion process occurs. Hot flue gas goes to the turbine and produces power. Gas turbine exhaust stream has high quality energy which could be recovered. This is done by the HRSG. In the HRSG, there are different heat exchangers. The first one is the condensate preheater. It increases temperature of water which is come from the condenser. The next one is the deaerator which is responsible for removing oxygen and other dissolved gases from the water. After that, water is pumped to low pressure (LP) and high pressure (HP) sections. In LP section there are an evaporator and a super heater. Finally the LP steam is delivered to the LP section of the steam turbine. The HP stream goes through two economizers, an evaporator and two super heaters. Also between the super heaters there is a desuperheater which sprays some water to decrease its temperature. The HP steam finally reaches to the HP section of steam turbine. To increase temperature of the flue gas, a duct burner is also placed at the inlet of the HRSG. By burning a small amount of fuel, it increases energy content of the flue gas. After power production in steam turbine, the steam goes to the condenser where it condenses and finally it reaches to the HRSG again.

The problem with combined cycle power plants is that their performance deteriorates during the summer. This is due to increasing ambient temperature which affects performance of gas turbine and cooling towers. When ambient temperature increases, density of air decreases. Therefore more power is needed in the compressor. Also mass flow rate of the compressor reduces. Consequently, mass flow rate of flue gas reduces which decreases energy delivery to the HRSG. On the other hand, increasing ambient temperature results lower driving force in cooling towers. As a result, steam turbine back pressure increases which decreases power production in the steam turbine.

To solve these problems, a new method is proposed in this paper (Figure 2). In this method, configuration of the HRSG changes slightly. The flue gas stream is split before going to the super heaters. The new stream bypasses the two super heaters and goes directly to the evaporator. In other words, instead of transferring heat in super heaters, it gives its energy to the evaporator. Although this results in lower steam temperature at the end of super heaters, it increases mass flow rate of steam produced in evaporator.

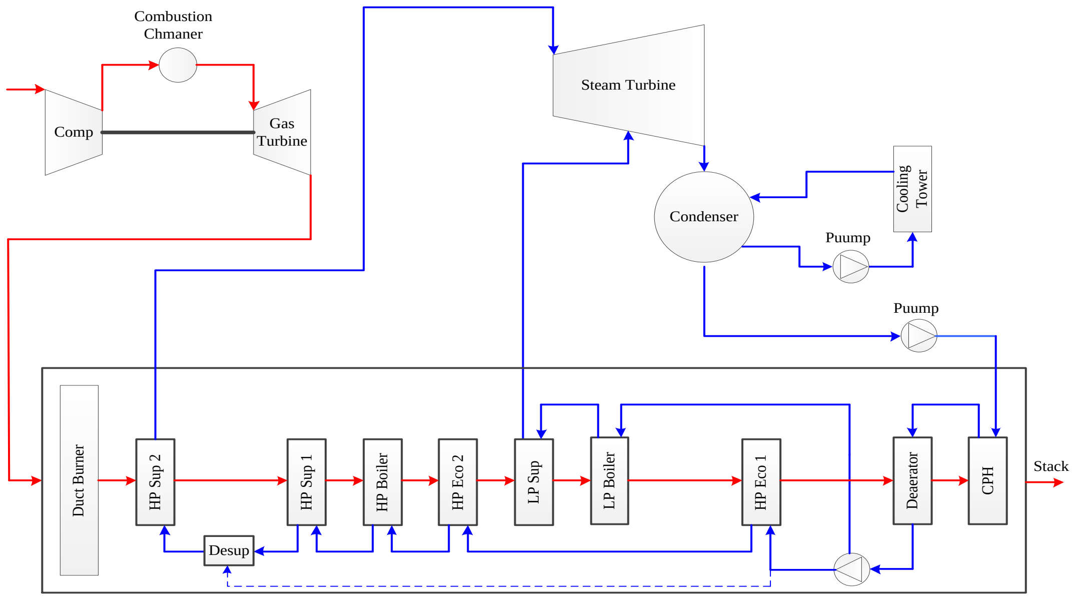

Figure 1. Schematic diagram of Damavand power plant

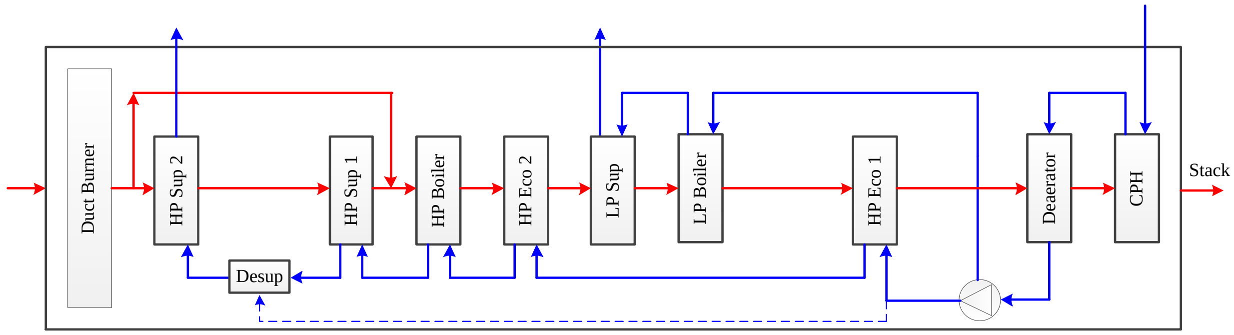

Figure 2. Schematic diagram of the new proposed method using split concept

2. Exergy analysis

Exergy is an indicator which shows the quality of energy. It could be transferred in three ways including heat, work and mass. Exergy of heat and work are calculated by:

$$\dot{Ex}^{Q} = \left(1 - \frac{T_0}{T_r}\right)\dot{Q} \quad (1)$$

$$\dot{Ex}^{W} = \dot{W}_{c.v.} \quad (2)$$

Specific exergy associated to mass transfer is computed using equation (3):

$$ex = ex_{ki} + ex_{po} + ex_{ph} + ex_{ch} \quad (3)$$

The first two terms are neglected due to their insignificant changes. Therefore only physical and chemical exergy should be computed. These two parameters are computed using equations (4) and (5):

$$ex_{ph} = (h - h_0) - T_0(s - s_0) \quad (4)$$

$$ex_{ch} = \sum_{i=1}^{N} y_i ex_i^{ch} + RT_0\left(\sum_{i=1}^{N} y_i\, ln(y_i)\right) \quad (5)$$

where h, s, y_i and R are enthalpy, entropy, molar fraction and universal gas constant, respectively. It should be mentioned that chemical exergy is defined for mixtures, such as flue gas or fuel. But calculating chemical exergy of fuel with equation (5) is a little bit complex. Therefore a simple formula is proposed to calculate chemical exergy of gaseous fuels (Dincer, & Cengel, 2001; Vandani, Bidi, & Ahmadi, 2015):

$$\xi = \frac{ex_{ch}}{LHV_f} \quad (6)$$

In equation (6), ξ is a constant and its value is different for different kind of fuels. Also LHV_f is the lower heating value of the fuel.

After calculating these parameters for each stream, exergy destruction and exergy efficiency of each component could be computed. Exergy destruction of each component is calculated by considering each component as a control volume and applying exergy balance to them as follows:

$$\dot{Ex}^{Q} + \sum \dot{m}_{in} ex_{in} = \dot{Ex}^{w} + \sum \dot{m}_{out} ex_{out} + \dot{Ex}^{D} \quad (7)$$

On the other hand, exergy efficiency of the components does not have a unique formula and there are different definitions for it. In this paper, we used fuel-product definition (Marmolejo-Correa, & Gundersen, 2012). Table 1 shows the required equations to calculate exergy destruction and exergy efficiency of each component (Marmolejo-Correa, & Gundersen, 2012).

3. Pinch Analysis

Pinch analysis is a methodology which is first proposed by Linnhoff (1974) after oil crisis in 20th century. Its goal is to maximize energy recovery in industrial processes to reduce their need to external energy sources. In this method, all streams are categorized into either hot or cold streams. Hot streams are those which should be cooled and lose their heat and cold streams are those which should be heated. The aim of the pinch analysis is to provide the required energy of cold streams by the hot streams. During the years, different kind of tools have been developed which can help the researchers to improve their process. The first one is the composite curve. In this plot, all hot streams are combined together and form a single line which represents all the hot streams in the process. The same thing happens to the cold streams too. These two lines are drawn in a same plot where its vertical axis shows the temperature and the horizontal axis shows the enthalpy. The lowest vertical distance between the two lines is called the pinch point. Figure 3 shows a sample of a composite curve for a process. The pinch point, divides the process into two regions, namely above the pinch and below the pinch. Based on this, three golden rules are defined as follows:

- No heat should be transferred across the pinch point
- Cold utility should not be used above the pinch
- Hot utility should not be used below the pinch

Although composite curve could calculate overall energy target, it cannot determine the amount of energy that should be delivered to the process at different temperatures. To solve this issue, grand composite curve is used. Figure 4 shows a sample of this plot. As can be seen, total energy demand calculated by composite curve could be delivered at different levels and grand composite curve specifies the amount of each of them. One of the most important goals of pinch analysis is to minimize total annual cost of the plant without having a negative effect on thermodynamic performance of the system. To do so, total area of heat exchangers should be minimized. There are different tools to achieve this goal. One of them is the driving force plot which was introduced by Linnhoff and Vredeveld (1984). As it is known, the required area for a heat exchanger is at its minimum value when it has vertical heat transfer. Driving force plot helps identify these heat exchangers. It is drawn by plotting the hot temperature versus cold temperature of the heat exchanger. A sample of this diagram is shown in Figure 5. Individual matches can be shown in the driving force plot. In an ideal match between streams, the heat exchanger will coincide with the driving force plot.

Table 1. Exergy destruction and exergy efficiency of each component

Component	Exergy destruction	Exergy efficiency
Heat exchangers	$\dot{Ex}^{D}_{HE} = \sum \dot{Ex}_{in} - \sum \dot{Ex}_{out}$	$\varphi = \frac{(\dot{Ex}_{out} - \dot{Ex}_{in})_{steam}}{(\dot{Ex}_{in} - \dot{Ex}_{out})_{flue\ gas}}$
Pump	$\dot{Ex}^{D}_{pump} = \dot{Ex}_{in} + \dot{W}_{pump} - \dot{Ex}_{out}$	$\varphi = \frac{\dot{Ex}_{out} - \dot{Ex}_{in}}{\dot{W}_{pump}}$
Total HRSG	$\dot{Ex}^{D}_{HRSG} = \sum \dot{Ex}^{D}_{i}$	$\varphi = \frac{(\dot{Ex}_{out} - \dot{Ex}_{in})_{steam}}{\dot{Ex}_{in,fluegas} + \dot{W}_{pump}}$

If a heat exchanger deviates from the driving force plot, it indicates that the match is not ideal and it has a criss-cross heat transfer. Therefore it should be modified (Linnhoff, & Ahmad, 1990). Generally, it could be said that driving force plot is a beneficial tool for analyzing heat exchangers individually.

4. Validation

Aspen energy analyzer Version 8.4. (2013) is used to perform the simulation and the results are compared with the actual data provided by the manufacturer's data sheet. Heat exchangers network simulated in software is shown in Figure 6. To validate the results, required area for each heat exchanger in the HRSG is calculated and then it is compared with the actual data provided by the manufacturer. The comparison is shown in Table 2. As it is shown, the difference between actual data and computed data are within a reasonable range.

Also based on data reported by the manufacturer, when pinch temperature of the plant is equal to 15°C, the total heat transfer area of HRSG is minimized. The same thing is simulated using ASPEN software and it is shown in Figure 7. As shown in the figure, total cost of the HRSG is at its minimum when minimum temperature difference is equal to 16°C. This is in good agreement with the manufacturer data.

Based on the presented data, it could be concluded that the simulation results are valid.

5. Results and Discussion

Three different scenarios are defined in this section and each one of them is simulated. The first scenario is the base case condition (design condition of the power plant).

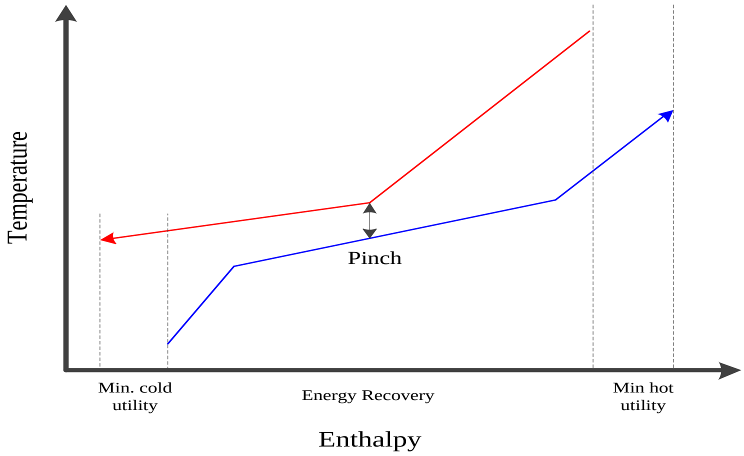

Figure 3. Composite curve for a sample process

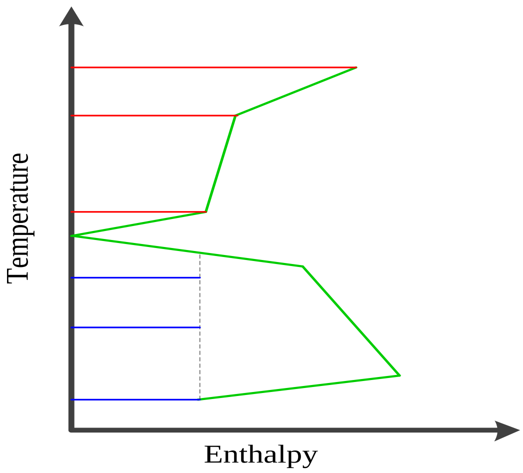

Figure 4. Grand composite curve for a sample process

In this scenario, temperature of flue gas is equal to 620°C at the HRSG inlet, while water temperature at HRSG inlet is 49°C. Also HP and LP mass flow rate of produced steam are equal to 66.51 kg/s and 8.84 kg/s and their temperature is equal to 523°C and 234°C, respectively. The result for this scenario is shown in Figure 8 and Figure 9. As can be seen in Figure 8, pinch temperature occurs at HP evaporator. Based on Figure 9, heat transfer in all heat exchangers is vertical which indicates that they are at close to their optimum performance, except for LP super heater. But heat transfer in this heat exchanger is very low compared with other heat exchangers. Therefore it could be neglected.

The second scenario is similar to the first one. The only difference between these two scenarios is that the second scenario is performance of the power plant during the summer. As mentioned before, during the summer, due to increase in ambient temperature, performance of gas turbine and cooling tower deteriorates. This results in lower mass flow rate of flue gas and lower energy delivery in the HRSG. Also, due to poor performance of cooling towers, power production in steam turbine decreases. Due to all these changes, total mass flow rate of produced steam from 75.35 kg/s in the first scenario, reaches to 68.98 kg/s. Figure 10 and Figure 11 show the results of simulation for the second scenario.

As can be seen in Figure 11, three heat exchangers are far from the DFP plot which indicates that their performance should be modified. These three heat exchangers include HP super heaters and LP super heater. As mentioned before, heat transfer in LP super heater is very low and it could be neglected.

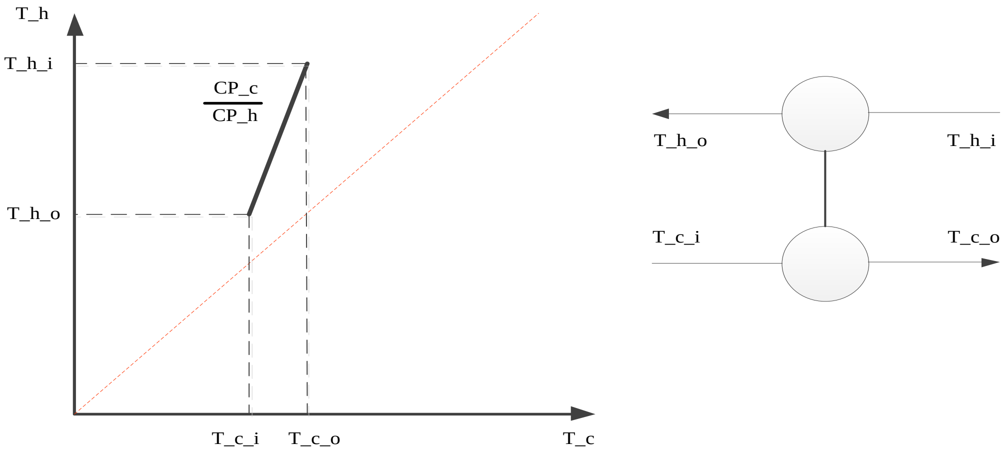

Figure 5. Driving force plot

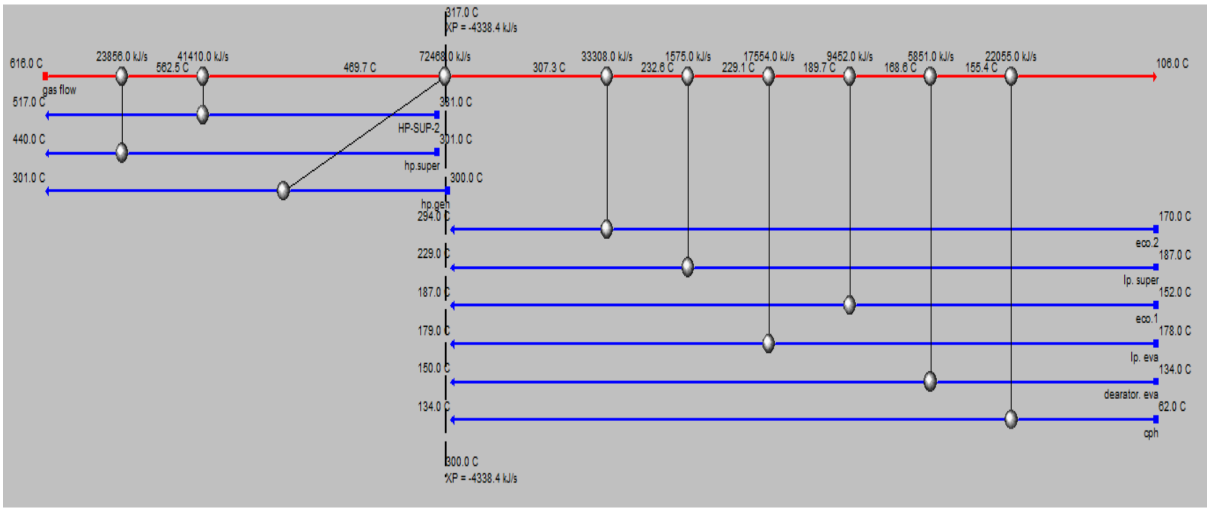

Figure 6. Heat exchanger network of the HRSG used in simulation

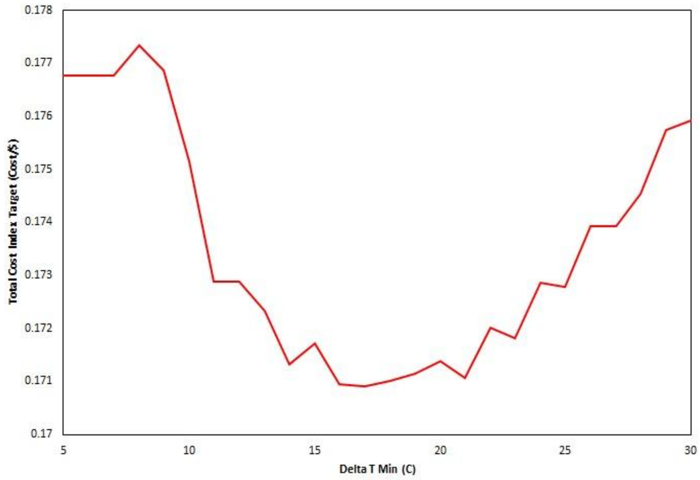

Figure 7. Variation of total cost of HRSG with minimum temperature difference

Table 2. Comparison between actual data and calculated area

Component	Actual data (m2)	Area calculated by software (m2)	Error (%)
HP sup1	6583	6721.243	2.1
HP sup2	6583	6721.243	2.1
HP eva	35180	36481.66	3.7
HP eco2	26000	25194	-3.1
LP sup	665	672.98	1.2
LP eva	13192	12849.01	-2.6
HP eco1	13192	12849.01	-2.6
DA	5491	5628.275	2.5
CPH	16508	17052.764	3.3
Total	123394	124170.181	-0.63

Table 3. Exergy efficiency of each component and total HRSG

	1st scenario	2nd scenario	3rd scenario
HP sup 2	88.87	88.88	75.37
HP sup 1	83.94	83.93	71.04
HP eva	86.40	86.39	85.65
HP eco 2	91.34	91.32	92.10
LP sup	76.45	76.45	76.98
LP eva	83.29	83.29	83.73
HP eco 1	89.70	89.69	90.39
DA	91.84	91.84	86.82
CPH	70.93	73.64	75.73
Pump	72.15	72.16	72.18
Total	81.6	81.26	81.77

The HP super heaters could be modified in three ways. The first one is to change their inlet and outlet temperature. The second one is to change mass flow rate (CP) of hot and cold streams by either splitting or changing streams and the third one is to do both of these methods simultaneously. Inlet temperature of flue gas could be changed by burning some fuel in the duct burner and changing mass flow rate could be done by splitting process. In this paper, the third method is used (changing both temperature and mass flow rate of the hot stream). The splitting concept is performed as it is shown in Figure 2. The flue gas stream is divided into two streams. The first stream passes thorough high pressure super heaters and the second stream bypasses these two heat exchangers and combines with the main stream at the HP evaporator inlet. It should be noted that mass flow rate of the second stream is equal to 15% of the main flue gas stream. This is the third scenario which is simulated and the results are presented in Figure 12 to Figure 14. Splitting concept is shown in Figure 12 and Figure 13 and Figure 14 show composite curve and driving force plot of this scenario.

As can be seen in Figure 14, after splitting, the two HP super heaters coincide with the DFP plot which indicates that the heat exchangers are in their ideal heat transfer condition. In this scenario, heat transfer of process is reduced due to splitting. Instead, this energy is delivered to the HP evaporator and mass flow rate of produced steam increases. Also due to fuel consumption in duct burner, flue gas temperature increases. It should be mentioned that in base case scenario, flue gas temperature could not increase more, because the metal of the super heaters could not bear higher temperature and thermal stresses would decrease its life time. But in the third scenario, it could be easily done thanks to the splitting process. When flue gas enters the HRSG with higher temperature, efficiency of the HRSG increases. This is shown in Figure 15. This is because in this case, flue gas leaves the HRSG with lower temperature which means that HRSG recovers more energy. Duct burner could not be operating in the first and the second scenarios due to thermal stresses in HP super heaters. Variation of operating condition of HP super heaters affects the performance of the LP super heater too. But in general, the effect of LP super heater in system performance is lower than the effect of HP super heaters. This is shown Figure 16. The figure is extracted from the power plant's document. As can be seen in this figure, effect of variation of HP steam temperature is higher than the LP steam temperature. Therefore inconsiderable changes in LP steam temperature could be neglected. Based on the obtained results, total steam production in summer reduces by 8.46% compared with the first scenario. This results in lower power production in steam turbine. But after using splitting concept, total steam production only reduces by 1.96%. Therefore performance of the third scenario is better than the second one. The proposed system has another advantage which is very important for combined cycle power plants. Currently, startup time of the combined cycles is a little lengthy which reduces flexibility of these power plants. To decrease this period of time, a lot of studies have been done during the recent years. The bottle neck for HRSG in startup time is the HP evaporator due to its wall thickness. The thicker the high pressure steam drum wall, the slower the HRSG startup. To resolve this issue, Benson evaporator is designed. The proposed system in this paper reduces the startup time further which is very beneficial. The effect of these changes on the startup time of the HRSG is shown in Figure 17 and Figure. As can be seen in Figure 17, after splitting, wall temperature of super heater and evaporator increases faster. Therefore they reach to their design point temperature sooner and consequently start up time decreases Figure 18 shows variation of mass flow rate and temperature of HP stream before and after splitting. As can be seen, pressure and mass flow rate of HP stream reaches to its design point almost 20 minutes sooner which is very considerable. This reduces startup time of the plants considerably. Considering the fact that capital cost of splitting process is very negligible, the new proposed system is of high value.

To get more insight about the effect of splitting process on the system performance, exergy analysis is carried out and exergy destruction and exergy efficiency of each component in the HRSG are calculated. To do so, results obtained in ASPEN software used in a Matlab program to calculate exergy destruction of equipment. Thermodynamic properties of each stream are computed using REFPROP (Lemmon, Huber, & McLinden, 2007). It also should be mentioned that atmospheric condition for exergy analysis were selected to be equal to 20˚C and 1.01 bar.

Figure 19 shows the value of exergy destruction in each component. As can be seen, the highest amounts of exergy destruction belong to the HP super heaters, HP evaporator and condensate preheater (CPH). Also exergy destruction of HP super heaters in the third scenario is higher than the first and second scenarios. This is because of burning fuel in duct burner which increases temperature difference between hot and cold streams. The amount of exergy destruction in other components is approximately equal in all the scenarios.

Also exergy efficiency of each heat exchanger in the HRSG is shown in Table 3. As can be seen, exergy efficiency of heat exchangers in the first and the second scenarios is almost the same. In general, due to higher ambient temperature in the second scenario, total efficiency of the HRSG in the second scenario decreases and from 81.6% in scenario1 reaches to 81.26% in scenario 2.

The biggest difference between exergy efficiency of heat exchangers is in the HP super heaters. Due to stream splitting, exergy efficiency of HP super heaters and HP evaporator decreased considerably. But since in this condition, more steam is produced, in general total exergy efficiency of the HRSG increases and reaches to 81.77%. As mentioned before, mass flow rate of bypass stream was considered to be 15% of the total mass flow rate of turbines exhaust stream. To analyze its effect on the system performance, a sensitivity analysis is performed. The result is shown in Figure 20. As it is shown, increasing the split ratio increases power generated by steam turbine. This is due to increasing mass flow rate of HP stream in HP evaporator. In a way that when split ratio increases from 2.5% to 15%, augmentation in generated power of steam turbines from 0.15 MW reaches to 0.9 MW. Also by increasing the split ratio, the graph reaches to an optimum value in which augmentation in generated power reaches to its maximum value.

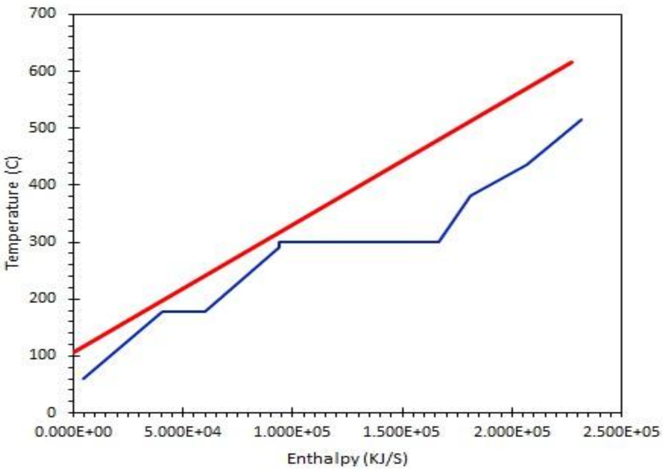

Figure 8. Composite curve for HRSG in base case condition

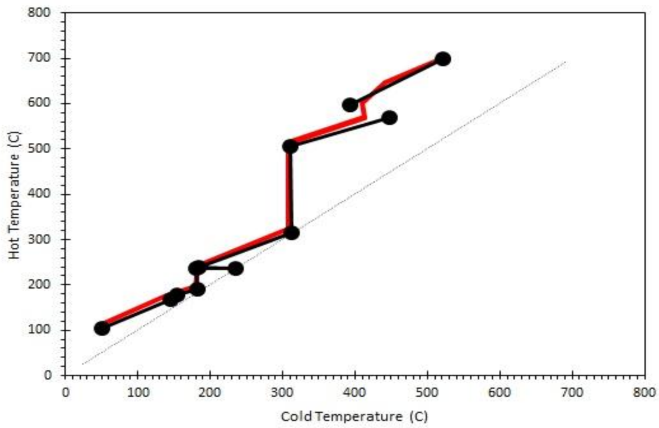

Figure 9. Driving force plot of the HRSG in base case condition

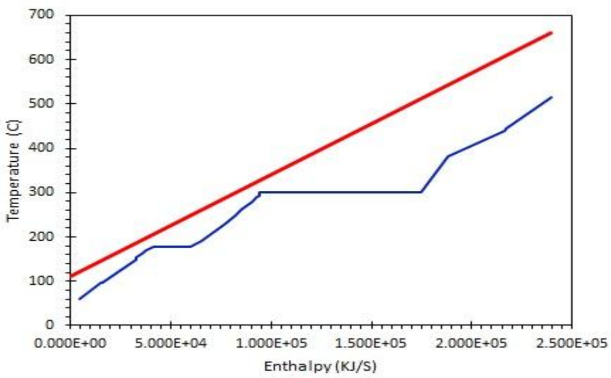

Figure 10. Composite curve for HRSG during summer

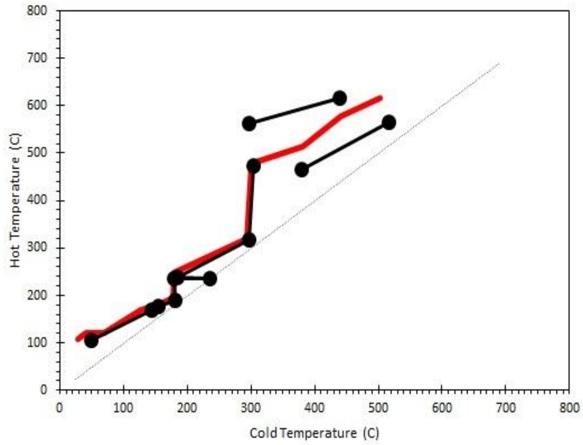

Figure 11. Driving force plot of the HRSG during summer

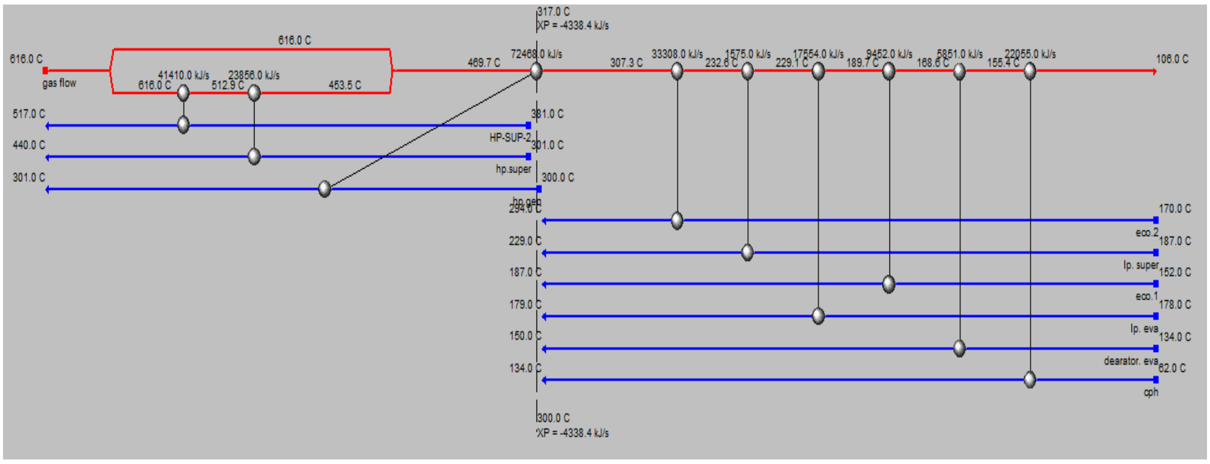

Figure 12. Heat exchanger network design after splitting

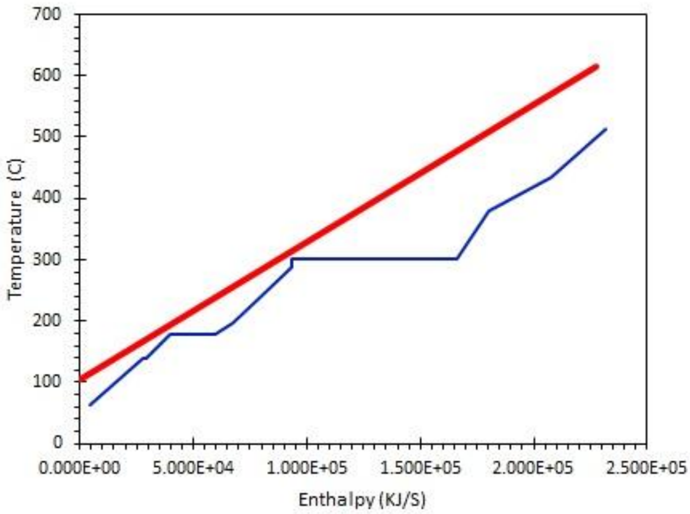

Figure 13. Composite curve of HRSG after splitting

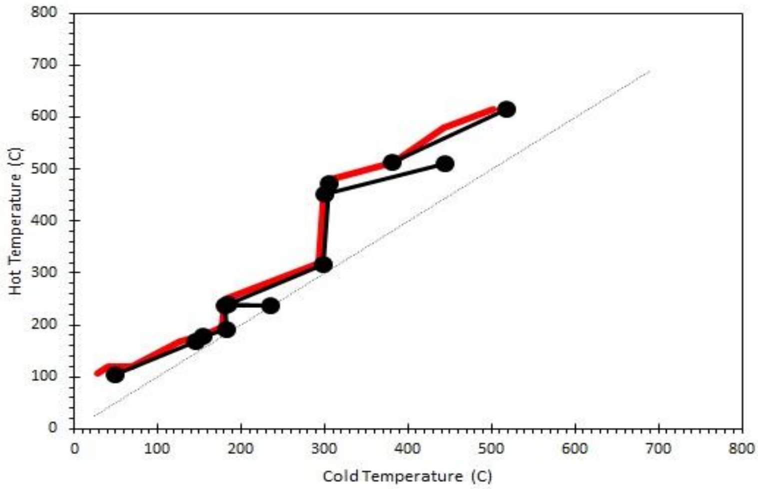

Figure 14. Driving force plot after splitting

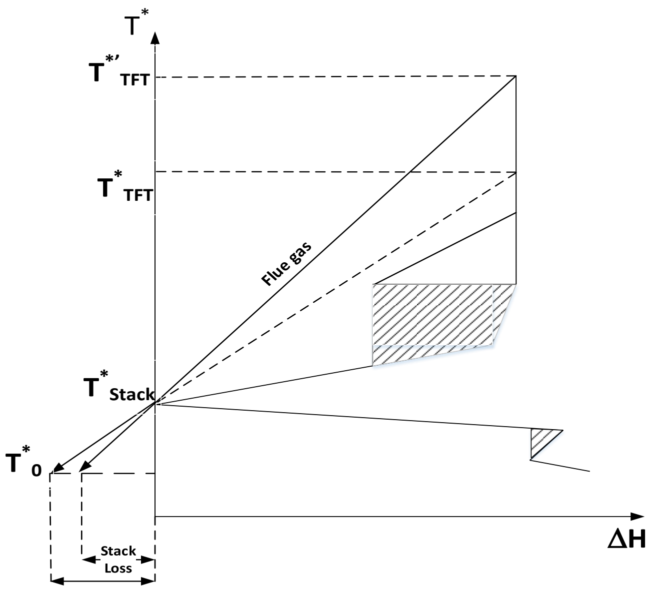

Figure 15. Effect of higher duct burner outlet temperature in the HRSG performance

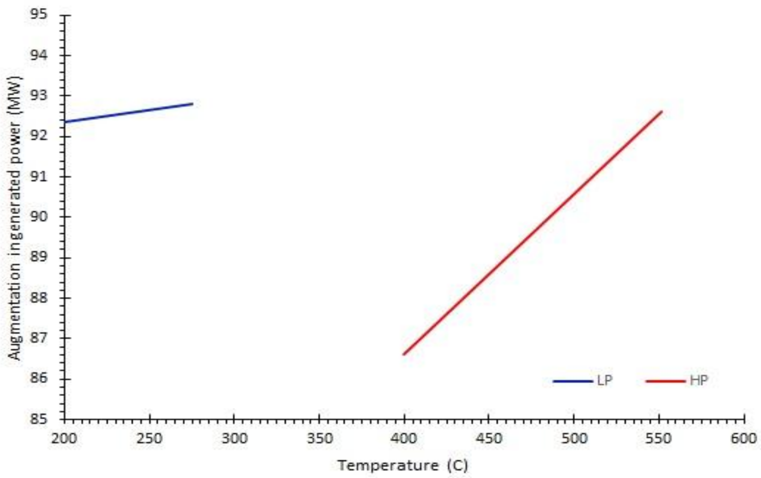

Figure 16. Effect of HP and LP steam temperature on generated power by steam turbine

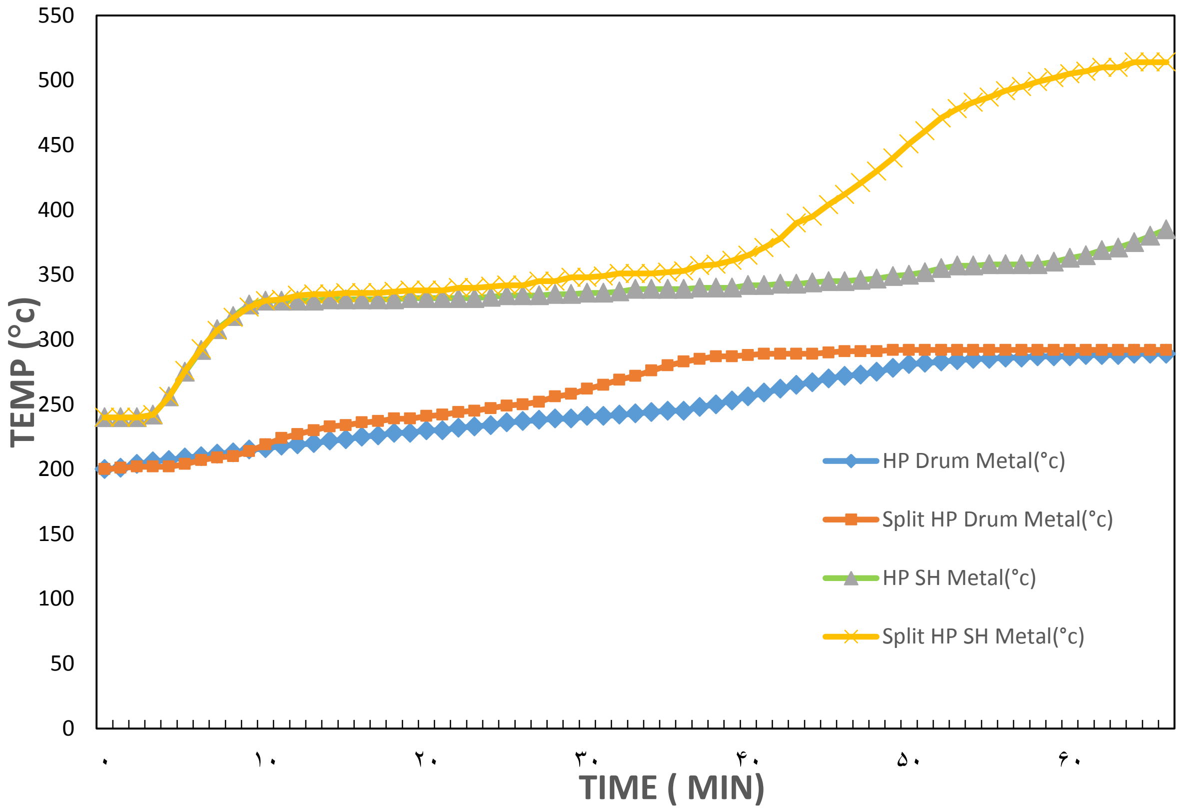

Figure 17. Effect of splitting process on temperature of evaporator and super heater wall

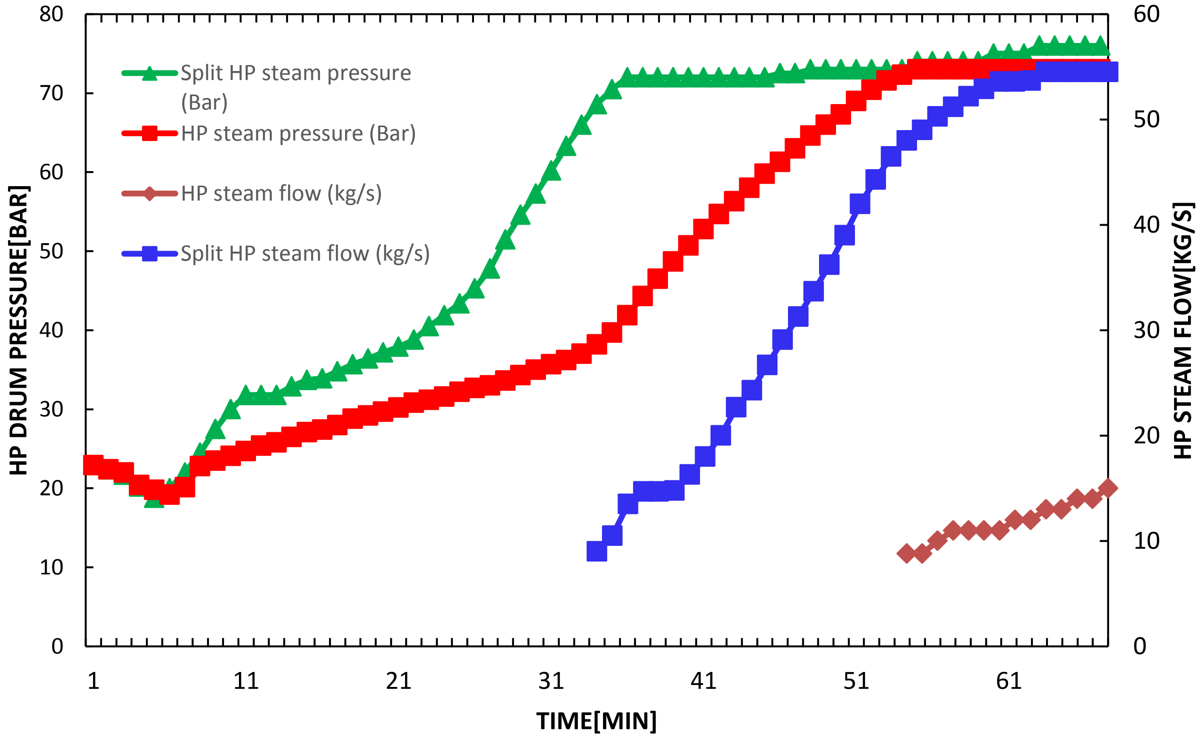

Figure 18. Effect of splitting on pressure and mass flow rate of HP stream

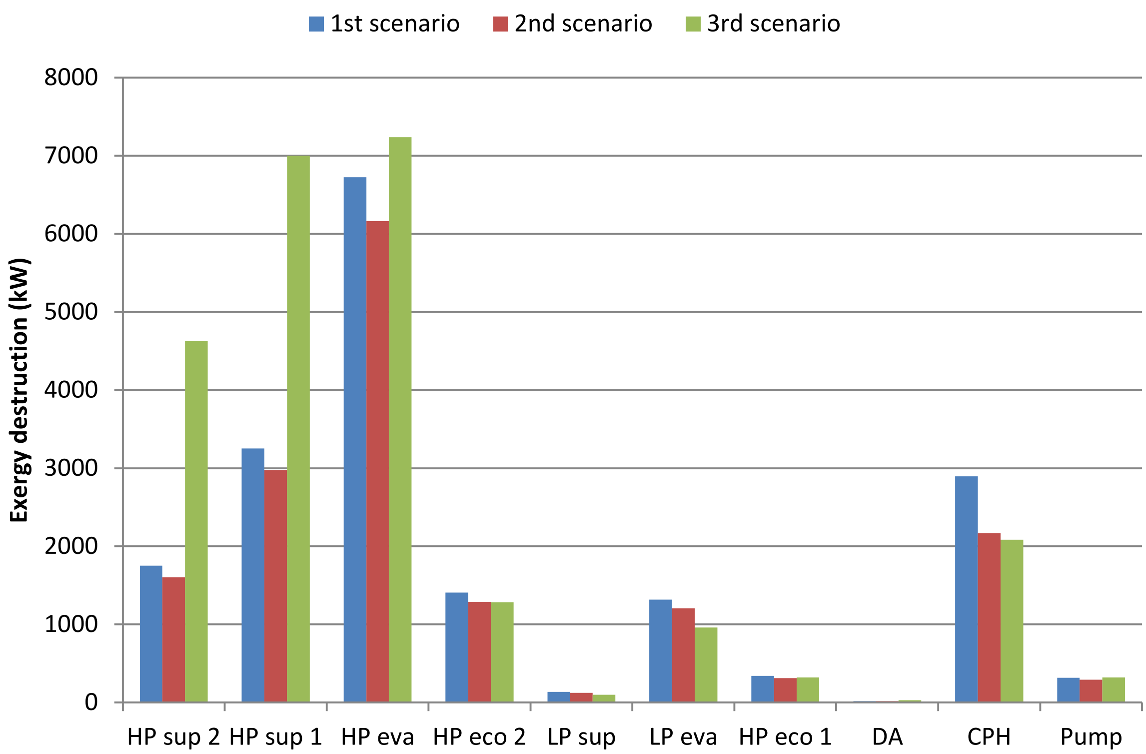

Figure 19. Exergy destruction in each component of the HRSG in all scenarios

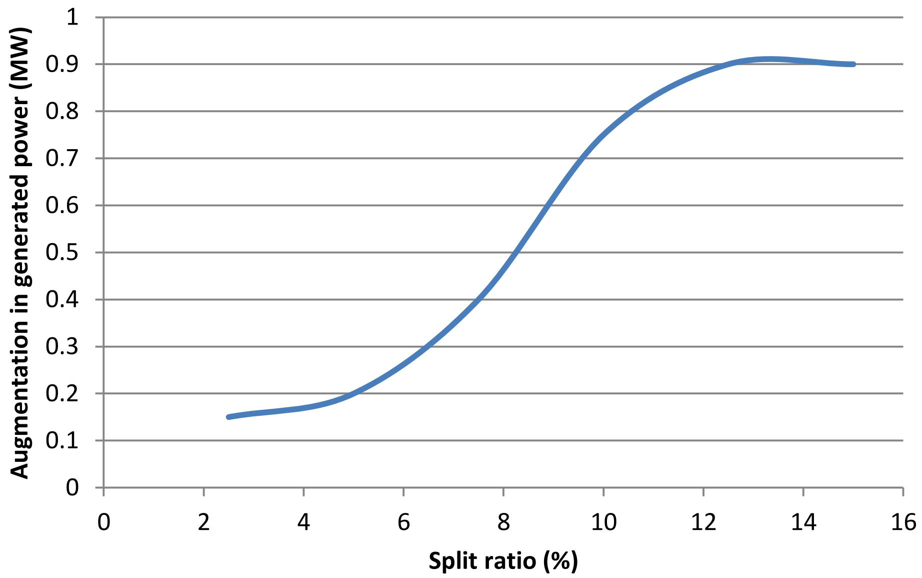

Figure 20. The effect of split ratio on generated power by steam turbine

6. Conclusion

In this paper, a new method to improve performance of the combined cycle power plants focusing on the HRSG is proposed. This method is very helpful especially during the summers when ambient temperature increases and as a result performance of the combined cycles deteriorates. Based on the results, this method improves heat transfer in HP super heaters and HP evaporator. It also increases mass flow rate of produced steam in the evaporator which results in higher power production in steam turbine. Also exergy analysis revealed that exergy destruction of the HRSG increases which is due to higher temperature difference between hot and cold streams. But in general, it could be concluded that this new method is a beneficial way to increase net power generated in combined cycle power plants and reduces its startup time which results in higher flexibility of the system. Finally, a sensitivity analysis is performed on the effect of split ratio on the power production of steam turbine. It showed that there is an optimum value for split ratio in which power production of steam turbine is maximum.

Nomenclature

$\dot{E}x$	Exergy flow rate (kW)
$\dot{E}x^D$	Exergy Destruction
$\dot{E}x^Q$	Exergy associated with heat
$\dot{E}x^W$	Exergy associated with work
ex	Specific exergy (kJ/kg)
ex_i^{ch}	Standard chemical exergy of i^{th} component
h	Specific enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)Q feed flow rate (mole/s)
$\dot{Q}$	Heat transferred (kW)
R	Gas constant (kJ/kg K)
s	Specific entropy (kJ/kg K)
T	Temperature (°C)
T_r	Heat transfer temperature
$\dot{W}$	Work rate (kW)
y	mole fraction
Greek letters	
ψ	Exergy efficiency
ξ	Chemical exergy/Energy ratio
Subscripts	
0	Reference environment condition
ch	Chemical
c.v.	Control volume
DB	Duct burner
eco	Economizer
eva	Evaporator
F	Fuel
HE	Heat exchanger
in	Inlet stream
ki	Kinetic
ph	Physical
po	Potential
out	Outlet Stream
sup	Super heater
Abbreviations	
CPH	Condensate preheater
DA	Deaerator
DFP	Driving force plot
HEN	Heat exchanger network
HRSG	Heat recovery steam generator
LHV	Lower heating value of fuel (kJ/kg)
LP	Low pressure
HP	High pressure

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