Open Access Journal

Journal of Power Technologies 95 (4) (2015) 288–301

journal homepage:papers.itc.pw.edu.pl

Parallel feed water heating repowering of a 200 MW steam power plant

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Abstract

In this paper, feed water heating repowering of Shahid Montazeri steam power plant has been studied in three different modes. Efficiencies of energy and exergy have been selected as objective functions. Cycle-tempo software was used for simulations. In the first case, a low pressure heat recovery heat exchanger and a EGT-RLM600-PC gas turbine were used. Efficiencies of energy and exergy increase to 3.8% & 3.79% respectively and cooling tower water temperature difference increases to 0.652^{o}C. In the second case, a high pressure heat recovery heat exchanger and a Siemens (KWU) V64.3 gas turbine were used. Efficiencies of energy and exergy increase to 6.68% and 6.65% respectively. In the third case, both heat recovery heat exchangers are and Westinghouse-401 gas turbines were used. Efficiencies of energy and exergy increase to 8.93% & 9.05% respectively. This is the best plan in terms of efficiency and cycle power promotion.

Keywords: Mohammad Montazeri Power Plant, Repowering, Parallel Feed Water Heating, Exergy efficiency, Cycle-Tempo

1. Introduction

Thermal power plants with steam cycle currently provide 34.2% of the country's electricity grid needs [1]. The plants, which are over 18 years old, are among the main resources. Low efficiency of the plant, after the introduction of the gas turbine to industry, created the idea of combined cycle plants. The multi-year track record of the Iranian power industry shows that many power plants have used mazot (oil) as their main fuel. The following parameters are the issues that highlight the importance of the efficiency of power plants and the steam cycle power plants:

1. Environmentalists advocate preventive rules on the prevention of production and emission of pollutants
2. Limited availability of fossil fuel resources
3. Need for a more economic production cycle

New plants construction is inevitable as Iran is experiencing 0.7% growth in annual consumption of electricity [2], increasing population, increasing industrial needs for energy and the increasing influence of more electrical machines on our daily lives. Meanwhile, the plan to generate electricity from clean energy sources and other private sectors has been included in the agenda of the government. Nevertheless, Iran is deficient in terms of studies and planning in this field and the generation capacity of its power plants. Therefore, energy ministry launched a program to construct fossil fuel plants. In recent years, the government's approach has been directed toward combined cycle power plants and the most recently built plants are indeed combined cycle. However, the repowering of steam power plants is also on the agenda, involving simultaneous multi-objective design. The main benefits of repowering steam power plants include increased production capacity, efficiency of the new cycle, reduced pollutant emissions and longer useful life of the plant. In recent years, several studies have been conducted about steam power plants in Iran [3–7]. In the 1970s,

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when repowering was being widely discussed in North America and Europe, several executive projects were done in this field [8–12].

2. Repowering of Steam Power Plants

Iran's electricity production network faces a combination of issues with steam power plants. Many of these plants are reaching—or have reached—the end of their useful life. In addition, a significant number of steam power plants do not have acceptable returns in spite of their relatively short lifespan. Meanwhile, general experiences from other countries in similar situations can be used to achieve reasonable solutions. Repowering (in non-solid fuel steam power plants) refers to add-in gas turbine units of components of the steam cycle, which is an accepted way to extend the life of old steam cycle components [13]. Through these affordable operations, we will meet power and efficiency requirements. When repowering an existing steam plant, several goals are pursued. Among these goals we can point to:

1. Increasing the plant's productivity.
2. Increasing overall efficiency of the new cycle by optimizing fuel consumption in the existing boiler.
3. Improving environmental performance by reducing emissions of NOx and other gas pollutants.
4. Extending the useful life of the plant.
5. Sufficiency enhancement of operation.

2.1. Repowering Methods

Repowering methods have two categories which are applicable in fossil fuel power plants [8].

1. Repowering of non-solid fuel power plants
2. Repowering of solid fuel power plants

Considering the fact that in Iran many of the existing steam power plants work with non-solid fuel, we investigate the procedures associated with them. These methods can be divided into two main categories [9].

1. Complete repowering (HRBR)
2. Partial repowering (PR)

Partial repowering includes the following methods:

1. Hot wind box repowering (HWBR)
2. Feed water heating repowering (FWHR)
3. Supplementary boiler repowering (SBR)

2.2. Feed Water Heating Repowering

Boiler input feed water is pre-heated by hot exhaust gases from the turbine. Hence existing feed water heaters are replaced by heat exchangers and turbine exhaust gas is used instead of steam extracted from the steam turbine. In the feed water heating method, the only change in the steam power plant cycle is replacement of the feed water heater. This method can be considered the simplest repowering method. Since the gas cycle does not bear a close resemblance to the steam cycle, every production cycle can be used separately and they are independent of each other. In this method, if the gas turbine unit is switched off, the upper production limit can be maintained at the initial designed value. Fig. 1 represents this repowering method. One of the main limitations of this method is the increased steam flow that enters the condenser.

2.3. Parallel Feed Water Heating Repowering

As explained in the previous section, one of the main repowering limitations through feed water heating is the flow rate of steam through the turbine. As extracted steam is eliminated in the turbine, the flow rate of steam at the entrance of the turbine is reduced, but at later stages of the turbine, the flow rate will be at above the nominal value and this places limitations on steam turbine design. As an example, Scusa estimated the power of steam turbines connected to high pressure feed water heaters as being between 12% and 16% and for steam turbines connected to low pressure feed water heaters as being between 4% and 6% [10]. Loss of the condenser vacuum causes a reduction in power plant efficiency. The condenser vacuum bears a close relation with ambient temperature [11], the condenser vacuum increasing as the temperature falls. In the cold season, the flow rate of steam to condenser can increase and as a result extracted steam flow rate and cycle efficiency will increase. This occurs in repowering through parallel feeding of water heating. Another benefit of this project is warm steam heaters. A brief history of studies in this field is given below:

Shah Nazari et al. [14] focused on the repowering of a steam power plant. They described minor repowering methods and calculated new power plant cycle efficiency. They also cite technical constraints in slight and complete repowering.

Maqsoudi et al. [12] analyzed the repowering of parallel feed water and exergy for Shahid Rajai Power Plant. Having mentioned repowering, they carried out the necessary optimization using the genetic algorithm and ex-

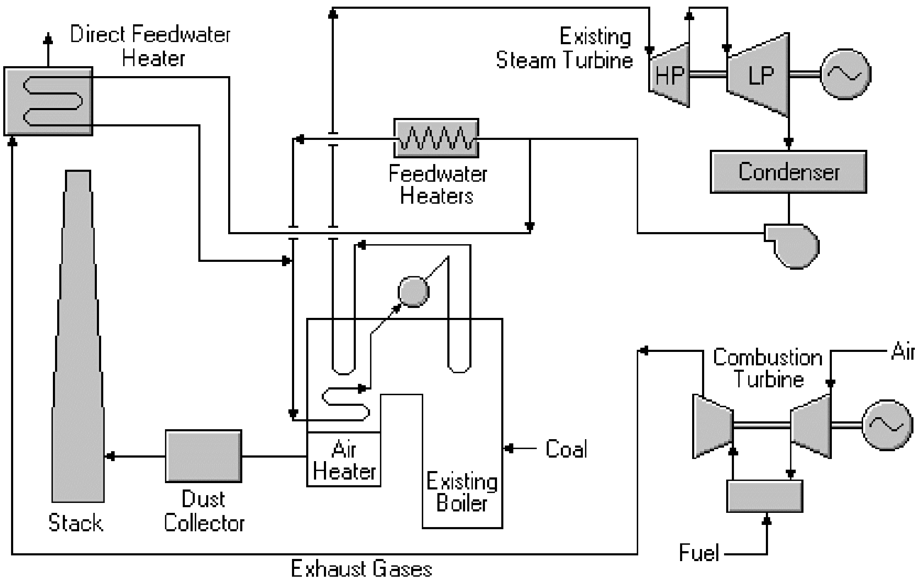

Figure 1: Schematic of feed water heating repowering [8]

ergy analysis. They concluded that a gas turbine has an optimal 25 MW capacity. In addition to increasing the plant power at a rate of 7%, exergy efficiency of the plant also increases 5%.

Zeki and Durmaz [13] analyzed the influence of hot wind box repowering on efficiency and CO_2 output. They conducted the study with simulation thermoflex software. This simulation showed an increase of 11% to 27% in generated power and a simultaneous reduction of 7% in generated CO_2.

Wolowicz et al. [15] investigated repowering simulation by feed water heating repowering of a power plant with unit capacity of 800MW in India. They used gas turbine model (A PG7161-EC) for the simulation. The results of this study showed that by this repowering, generated power increased 20% and thermal efficiency of the entire system at nominal load increased from 43.5% to 44.5%. Baqestani et al. [16] investigated the simulation of the Qazvin plant cycle with thermoflex software and offered the best way to deliver repowering using exergy and economic exergy analysis methods.

Frankle [17] studied the best ways of repowering for a 300 MW steam plant in Russia. He investigated all available repowering methods and suggested complete repowering. According to his calculations, plant efficiency will increase from 38% to 56.8%.

Hosseinalipoor et al. [18] performed a comparative economic analysis of steam plant repowering versus setting up new gas plants. The study starts off by introducing general ways of repowering steam power plants, then goes on to state the existing potential for repowering Iran's steam power plant assets. The analysis showed that the repowering of existing plants is more affordable.

Asadian and Samadi [19] compared all repowering methods to improve the performance of Lushan power plant. In this research hot wind box repowering (HWBR) and feed water heating repowering (FWHR) were studied using gas turbines in steam cycle and different states. It was determined that feed water heating repowering was the option of choice for the plant – after checking the results, and factoring in main plant fuel, investment costs for adjustments and independent operation and time of circuit exit and environmental aspects. The results showed that application of this method increased the power of the plant to 12 MW.

Haghighi and Tanazan [20] investigated the repowering of Be'sat power plant by feed water heating from technical and economic perspectives. Using thermodynamic simulation, they looked at the increase in generated power caused by repowering and the effect on cycle efficiency. Their finding was that repowering for this power plant is essential. Moreover, replacement of all heat converters before the feed water heater was shown to have more effect on efficiency and generated power.

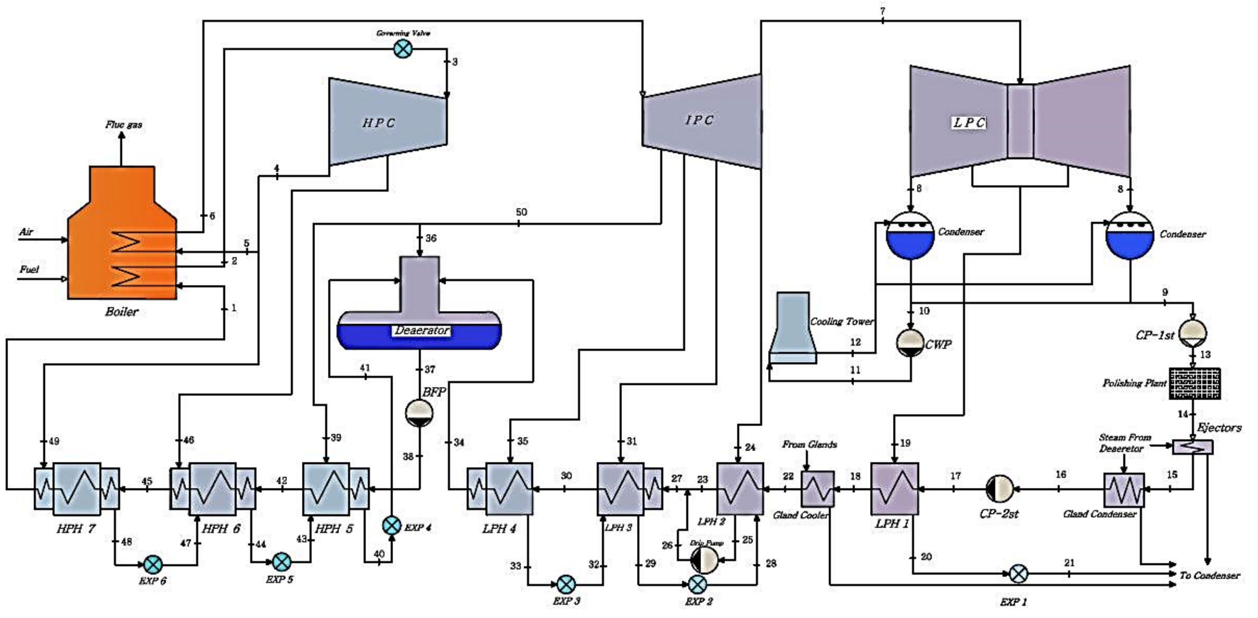

Figure 2: Schematic of Mohammad Montazeri steam cycle

Table 1: Total properties of the Cycle [11]

Operating conditions	value
Power produced, MW	200
Power consumption, MW	14
Volumetric flow rate of fuel (natural gas), Nm^3/h	$54{\cdot}10^3$
Heat rate, kJ/kWh	10448.6
Stem flow rate, main line, Ton/h	670
Steam pressure, main line, bar	130
Steam temperature, main line, °C	540
Water temperature, to boiler, °C	247
Stack gas temperature, °C	160
volumetric flow rate of air to burners, Nm^3/h	$9.6 \cdot 10^5$
Number of induced and draft fans	2
Number of burners	12
Combined pump/motor efficiency, %	95

3. Power Plant Cycle Description

Shahid Montazeri Power Plant of Isfahan is located 15 kilometers northwest of Isfahan, along the Isfahan-Tehran highway next to the Isfahan Refinery in 2.2 million m^2 of land. This power plant has 8 similar steam units, each with a capacity of 200 MW, whose technical specifications are presented in Table 1. The heat cycle of this power plant is a Rankine cycle, whose heat processes are briefly shown in Fig. 2. Each of the units in this power plant has a water-steam circuit as follows:

Distilled water with a temperature of 45.6°C enters the first-stage pumps from the condenser and is subjected to a pressure of up to 9 atm. Then, it enters the polishing plant and its purity increases after passing through the filters. In the next stage, its pressure increases up to 19.5 atm by second-stage pumps after passing the ejector and gland condenser and enters low pressure heaters after passing through low pressure heater No.1 and the gland cooler. In low pressure heaters, feed water is heated up to 157°C by the steam extracted from the turbines. The water leaving low pressure heaters enters the deaerator and is deaerated in this stage. It is then subjected to pressure of 180 atm by the boiler feed pumps and is directed into high pressure heaters. In the high pressure heaters, feed water is heated up to 245 °C and reaches 320 °C after passing through the economizer and absorbing the heat from the smoke leaving the boiler and enters the drum. In the boiler, water and steam re-enter the drum after water enters from the drum to wall tubes and after heat transfer through burners, and its steam is separated and directed into the super-heaters and is heated up to 540°C. The dry steam leaves the boiler at a pressure of 130atm and temperature of 540°C and enters the high pressure turbine. The steam leaving the high pressure turbine is reheated in the boiler up to 540°C and enters the intermediate pressure turbine with pressure of 24.5 atm and then enters the low pressure turbine after leaving the intermediate pressure turbine. The steam leaving the low pressure turbine is also mixed with the cooled water in the condenser and is converted to water and passes through the same route again.

3.1. Repowered Cycle

There are several ways to optimize parallel feed water heating repowering. Some of the main parameters considered for repowering and used as target functions are: final cycle efficiency, fuel consumption reduction, generated power increase and reduction of environmental pollutants. In this research, exergy efficiency is used as a target function and we will carry out optimizations knowing that exergy efficiency includes general conditions of the repowered power plant. Two heat recovery heat exchangers are used for the repowering. A low pressure heat exchanger is placed on the feed water path before the deaerator water entrance. This heat exchanger receives gland cooler water and passes it to the deaerator. A high pressure heat exchanger is placed on the feed water path next to the boiler feed water pumps. An increase in feed water temperature occurs in this heat exchanger at the boiler entrance. Considering the final explanations, new heat exchangers were installed parallel to existing heat exchangers. In Fig. 3, the project implementation is observed for Shahid Montazeri power plant. Due to the differences in condenser vacuum occurring through the year caused by fluctuations in environmental temperature, the steam flow rate/condenser should be controlled. When the environment temperature falls (in winter) the cooling power of the cooling tower will rise accordingly. In the feed water heating method, if cycle water heating by the gas turbine is increased, the steam flow rate for feed water heating will reduce and the steam flow rate in the final stages of the turbine and entering the condenser will also increase. Here, we compare condenser steam flow rate in various feed water flow rates to the heat recovery heat exchanger, assuming there are no restrictions on the steam flow rate of low and high pressure turbines.

1. The passed flow rate of the high pressure heat recovery heat exchanger is assumed to be 0 and the passed flow rate of the low pressure heat recovery heat exchanger is assumed to be variable from 10 kg/s to 140 kg/s and, in this case, the required steam flow rate for high pressure heaters is invariable and for low pressure heaters is variable.
2. In spite of the first case, the passed flow rate of the low pressure heat recovery heat exchanger is assumed to be 0 and the passed flow rate of the high pressure heat recovery heat exchanger is assumed to be variable at 10 kg/s to 140 kg/s and in this case the required steam flow rate for low pressure heaters is invariable and for high pressure heaters is variable and will be reduced by the passed water flow rate of the heat exchanger.
3. This is a combination of the previous cases. The passed flow rate of both heat recovery heat exchangers is variable at 10 kg/s to 140 kg/s. In this case the passed water measure is equal for both heat exchangers. Optimized gas turbines are used for feed water heating in the heat recovery heat exchanger. In the first case, gas turbine model EGT RLM-6000PC is used, in the second case gas turbine model Siemens (KWU) V64.3, and in the third case gas turbine model Westinghouse 401.

4. Assumptions made in repowering calculations

1. Entering steam pressure to extractions is calculated by Stodda law [21], Cycle-Tempo software does this process automatically.
2. Elected feed water heaters and all heat exchangers except the deaerator are considered as counter flow heat exchangers.
3. Sprays that are used to control temperature between the heat exchangers upside in the boiler in all simulations are ignored. In order to prevent an increase in steam and heaters' upper pipe temperature, these sprays are used and their water is supplied from feed water pump output.
4. In order to achieve optimal conditions, the generated charge by steam turbines in all cases has been fixed at 200 MW. The reason for this is prevention of steam cycle generation losses. All parameters are checked to compensate potential shortages in the cycle and we record necessary changes to offer the proper solution.
5. Since the condenser vacuum is one of the impressible parameters in the feed water heating method, it is assumed to be invariant. Therefore, in all cases, the required flow rate for circulated cooling water is calculated and a proper range is defined for it.
6. In the deaerator, the feed water degassing process is done by water saturation. For this purpose we use steam extracted from the intermediate pressure turbine. Non-saturated gases are separated

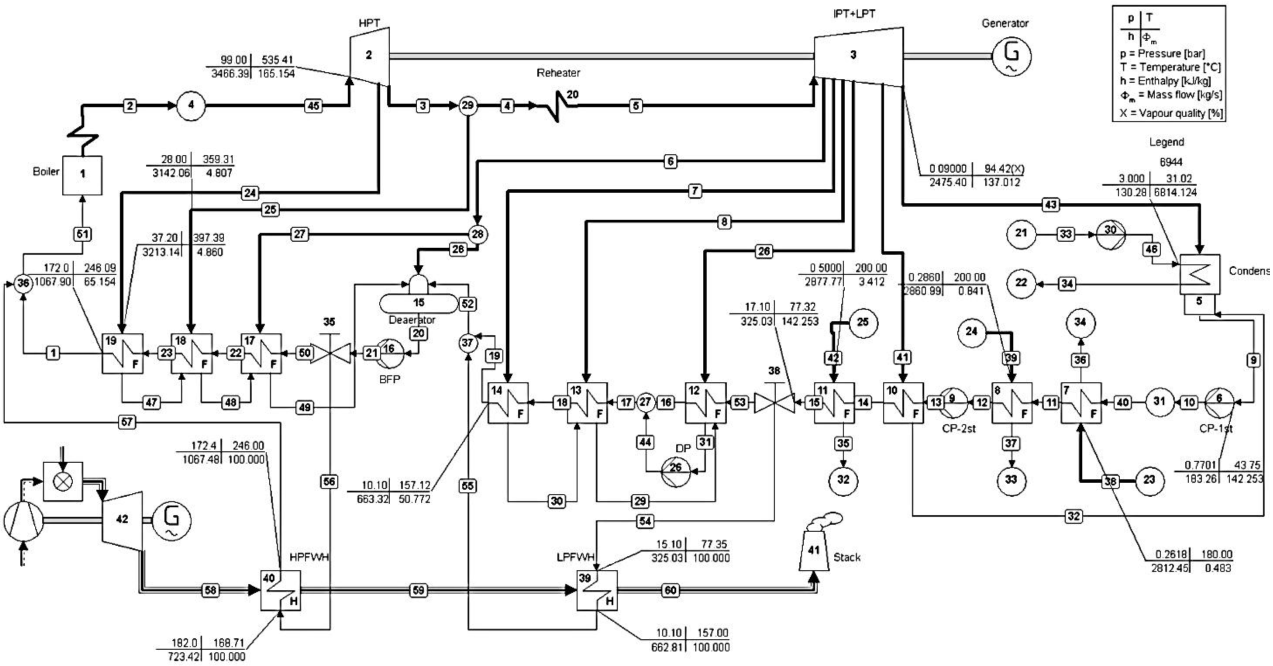

Figure 3: Schematic of repowered cycle in Cycle-Tempo

from the water and driven out of system. In simulations the pressure of this system is considered invariable and equal to the design value. So the input steam flow rate is a function of entrance feed water.

7. In order to preserve all thermodynamic parameters of the boiler, pressure and temperature on input and output are assumed to be identical to the first case.
8. Passed steam flow rate of all turbines stages is assumed to be without restriction.
9. Gas turbine with set capacity is used. For this purpose, we chose the best gas turbine from among several options.
10. Reduction in temperature of the exhaust gas is assumed to be 0 from the gas turbine to the high pressure heat exchanger and from the high pressure converter to the low pressure input and its temperature reduction equivalent is considered in heat exchangers. 800 kW heat loss for the low pressure heat recovery heat exchanger and 1020 kw for the high pressure heat recovery heat exchanger are considered [11].
11. All processes are done in steady state.
12. Exergy analysis is done based on the lower heat value of natural gas.
13. Environmental conditions for calculation of exergy are considered invariable: T=287.15 K , p = 101.325 kPa.
14. For environmental air, standard conditions defined in software are used.
15. Since we follow pure cycle efficiencies, consumption power of all pieces of equipment active in the cycle, including fans, pumps and other auxiliary power consumptions, are defined in the plan.

5. Dominant Equations

To perform the necessary calculations, we use exergy balance equations in a control volume and mass survival principle and thermodynamics first laws. To use the first law of thermodynamics the energy balance form should be used for standard volume. This equation is as below [22]:

$$\frac{dE_{C.V}}{dt} = \sum \dot{m}_i(h_i + \frac{v_i^2}{2} + gz_i) - \sum \dot{m}_e(h_e + \frac{v_e^2}{2} + gz_e) + \dot{Q}_{CV} - \dot{W}_{CV} \quad (1)$$

For exergy balance in a control volume the following equation can be used [22]:

$$sum\left(1 - \frac{T_o}{T}\right)Q_k + \sum(\dot{m}_i\psi_i) = \sum \psi_w + \sum(\dot{m}_o\psi_o) + \dot{I}_{des} \quad (2)$$

In control volume for irreversibility explanation the following equation is used [22]:

$$\dot{I}_{CV} = (\sum \dot{m}_i\psi_i - \sum \dot{m}_o\psi_o) + \sum(1 - \frac{T_o}{T})Q_{cv} - \dot{W}_{CV} \quad (3)$$

In exergy calculation of all cycle equipment, we should calculate all exergy flows. Exergy calculation for single-phase flows such as water or steam flow is carried out easily. For this action the following equation is used [21]:

$$\psi = (h - h_o) - T_o(s - s_o) \quad (4)$$

For transferred exergy by heat [21]:

$$\psi_Q = Q(1 - \frac{T_o}{T}) \quad (5)$$

The necessary calculations are done in order to select or design required heat exchangers. In this section, calculations are carried out to determine the required thermal capacity for converters. The rate of exchanged heat in these heat exchangers is equal to the required amount for feed water heating, which we will explain below:

$$\dot{Q}_{LPFWH} = \dot{m}_{LPFWH}(h_{out,LPFWH} - h_{in,LPFWH}) \quad (6)$$

$$\dot{Q}_{HPFWH} = \dot{m}_{HPFWH}(h_{out,HPFWH} - h_{in,HPFWH}) \quad (7)$$

To calculate cycle exergy efficiency the following equation is used [22]:

$$\eta_{2,cc} = \frac{\sum_{i=1}^{n} \dot{W}_i}{\sum_{i=1}^{n} \dot{EX}_f} \quad (8)$$

Due to the fact that the consumption fuel is natural gas, to calculate consumption fuel special exergy the following equation is used [23]:

$$\sum_{i=1}^{n} \dot{EX}_f = \dot{EX}_{f,st} + \dot{EX}_{f,gt} \quad (9)$$

$$\sum_{i=1}^{n} \dot{W}_i = \dot{W}_{st} + \dot{W}_{gt} \quad (10)$$

To calculate fuel exergy of cycle input the following equation is used [23]:

$$\dot{EX}_f = \psi_f \times \dot{m}_f \quad (11)$$

To calculate the exergy of natural gas we use the following equation [23]:

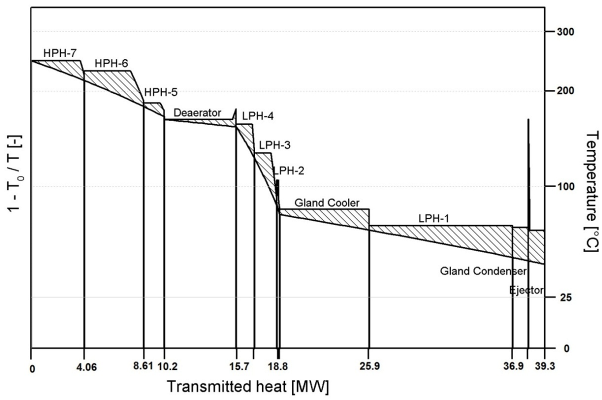

Figure 4: Value diagram of feed water along steam flash heaters

$$\psi_f = \xi \times LHV_f \quad (12)$$

where the amount of ξ depends on the chemical composition of the consumption fuel and is expressed in numerous experimental values in different references. In order to calculate the rate of cycle fuel input, fuel consumption in the steam cycle boiler and gas turbine must be calculated separately. To calculate steam cycle boiler fuel consumption we use the following equation [22]:

$$\dot{m}_{f,b} = \frac{\dot{Q}_{boiler}}{LHV_f} \quad (13)$$

In order to calculate Q_B, we use energy balance in the boiler [23]:

$$\dot{Q}_B = \frac{\dot{m}_{fw,i}(h_{o,B} - h_{i,B}) + \dot{m}_{reh}(h_{o,reh} - h_{i,reh})}{\eta_{1,B}} \quad (14)$$

$\eta_{1,B}$ value for Shahid Montazeri power plant boiler comes to 90.55 percent [11]. To calculate Q_{gt} , the energy balance in combustion chamber of the gas turbine is used [23]:

$$\dot{Q}_{gt} = \frac{\dot{m}_{air,i}(h_{o,c.ch} - h_{i,c.ch})}{\eta_{1,c.ch}} \quad (15)$$

6. Results View

Here we will see the total results for the repowering simulation by parallel feed water heating in the third case; afterwards results are presented separately for each of the triple cases. In Figs 4 and 5, illustrate the exergy

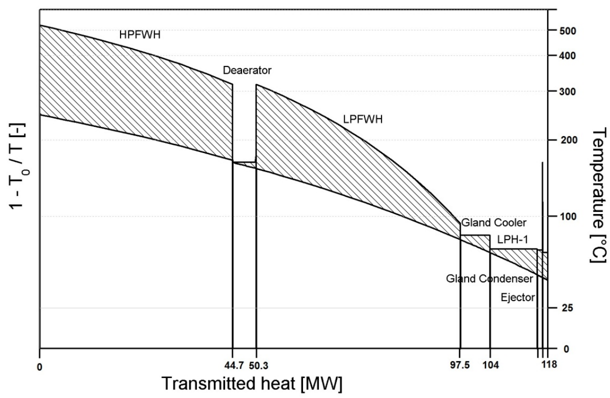

Figure 5: Value diagram of feed water along heat recovery heat exchangers

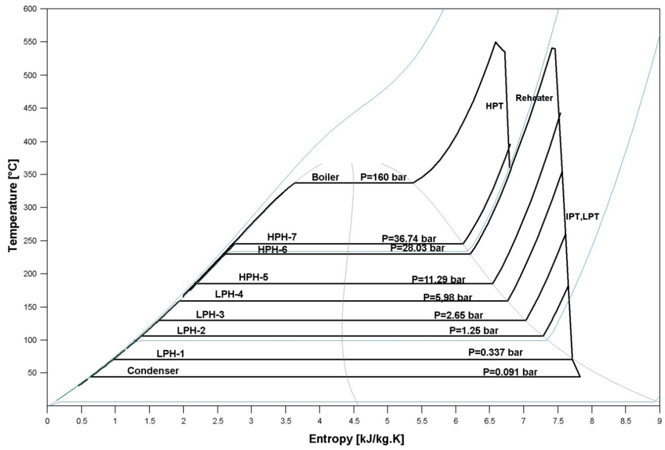

Figure 6: $T-s$ diagram of repowered cycle in third case

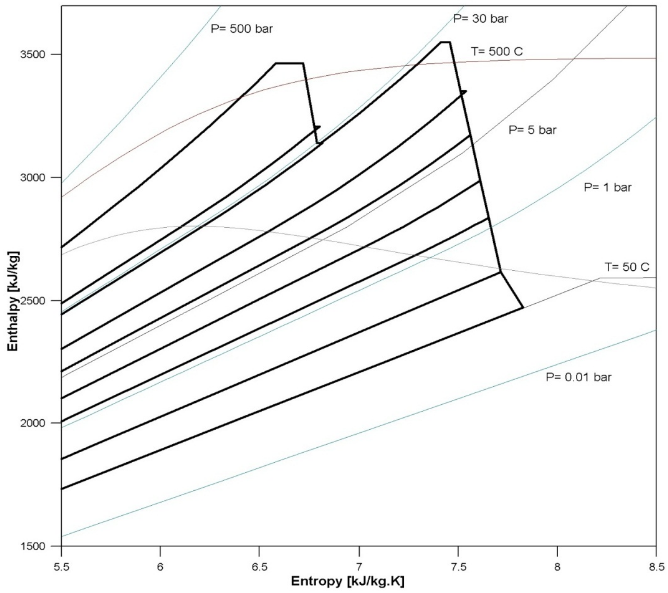

Figure 7: $H-s$ diagram (Mollier) of repowered cycle in third case

behavior value diagram for water heating track in steam heaters and heat recovery heat exchangers in the third case and the 140 kg/s flow rate for heat recovery heat exchangers. As you can see, in this case feed water heaters have a small share of feed water heating, and heat recovery heat exchangers have the maximum value of heating. Unlike the other heat exchangers in the cycle, the deaerator is known as a parallel flow heat exchanger, as can be observed well in Fig. 4, Using these diagrams, we can conclude that heat recovery heat exchangers have more exergy loss than steam flash heaters. Figs 6 and 7 show $T-s$ and $h-s$ diagrams for the repowered cycle by parallel feed water heating in the third case. The diagrams are used for recycle thermodynamic behavior analysis.

6.1. First case

In Fig. 8 we can see pure energy and exergy efficiency differences of cycle versus low pressure heat recovery heat exchanger passed flow rate. In this case exergy and energy efficiencies start from 68.33% and 64.3% respectively with 0 flow rate and increase to 34.9% and 35.9% respectively after the passed flow rate from the low pressure heat recovery heat exchanger increases to 140 kg/s. In Fig. 9 the required water flow rate changes in the cooling tower are seen in different passed flow rates of the low pressure heat exchanger in various temperature differences in the cooling tower. In Fig. 10 steam flow rate differences in the low pressure heaters and deaerator are shown against low pressure passed water flow rate changes. Clearly, entered steam flow rate to all of the heaters is reduced by the increase in passed flow rate from the low pressure heat recovery heat exchanger. In this case the entered steam flow rate to the deaerator is 2.16 kg/s. If the low pressure heat recovery heat exchanger passed water flow rate is 0, after an increase in passed flow rate of the low pressure heat recovery heat exchanger to 140 kg/s, the same amount will be reduced to 2.12 kg/s. After an increase in passed water flow rate of the heat recovery heat exchanger, the high pressure turbine steam flow rate is reduced (showing an increase in cycle efficiency) and the entered steam flow rate to the condenser increases (showing an increase in cooling power). In Fig. 11 we can see entered steam changes to the high pressure turbine and condenser and passed water from the low pressure steam flash heaters versus changes in passed water from the high pressure heat recovery heat exchanger. It is clear that an increase in the

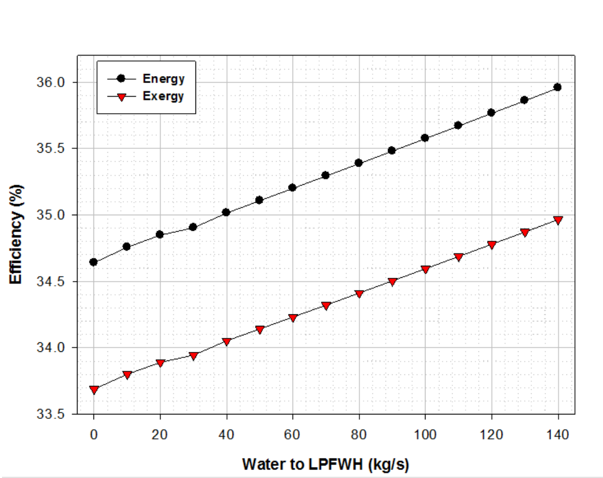

Figure 8: Variation of energy and exergy efficiencies of cycle across water mass flow rate along heat recovery heat exchanger in first case

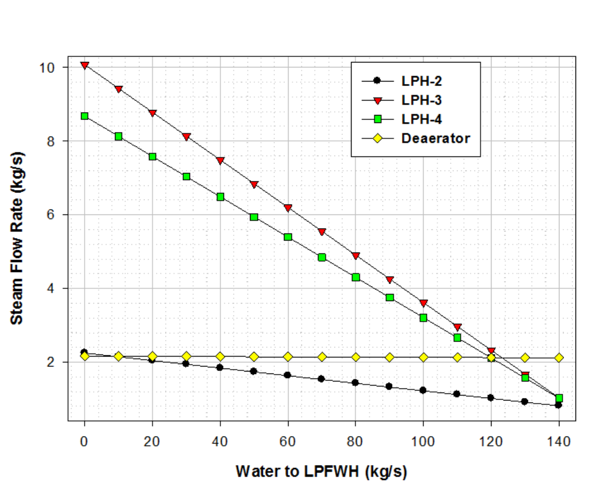

Figure 10: Variation of steam flow rate of steam flash heat exchangers across water mass flow rate along heat recovery heat exchanger in first case

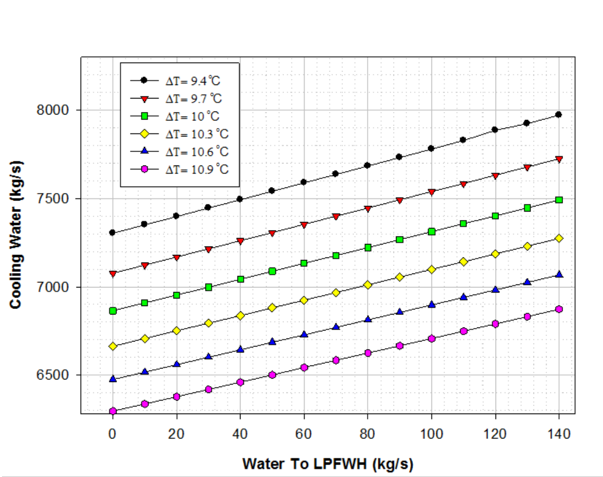

Figure 9: Variation of condenser cooling water across water mass flow rate along heat recovery heat exchanger in first case

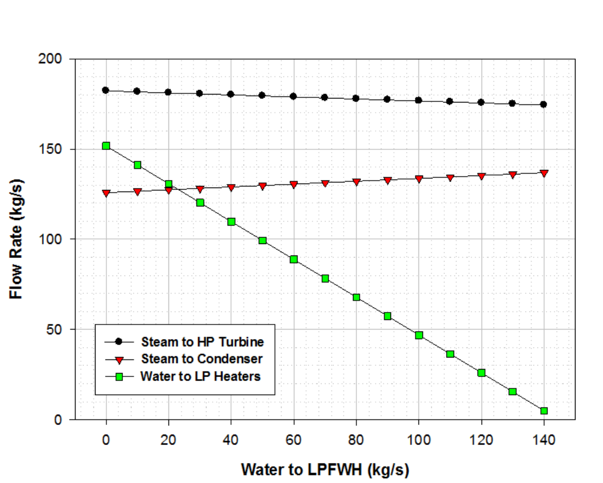

Figure 11: Variation of flow rates in major points across water mass flow rate along heat recovery heat exchanger in first case

passed water flow rate from the low pressure heat recovery heat exchanger will increase the entered flow rate to the condenser and will reduce the entered flow rate to the high pressure turbine.

6.2. Second case

In Fig. 12 we can observe changes in cycle energy and exergy efficiencies versus passed water flow rate changes of the high pressure heat exchanger, which starts from 33.8% and 32.84% respectively when the flow rate is 0 and increases to 36.1% and 35.0% when the passed flow rate from the high pressure heat recovery heat exchanger increases to 140 kg/s. In Fig. 13, changes in the cooling tower required water flow rate in various passed water flow rates from the high pressure heat recovery heat exchanger are seen in various temperature differences in the cooling tower. If we assume the available flow rate of rotatory water in the cooling water is 0, and in the high pressure heat recovery heat exchanger a passed water flow rate of 100 kg/s, a temperature difference equal to 10.3^{o}C is needed for the cooling tower water. As we can see, in this case despite a greater increase in cycle efficiency than in the first case, a lower temperature increase is required. Therefore, the first case is better than the second case. In Fig.14 we can see the steam flow rate changes to the deaerator and high pressure heaters versus passed water flow rates from the high pressure heat recovery heat exchanger. As we can see, the increase in passed water flow from the high pressure heat recovery heat exchanger will reduce the

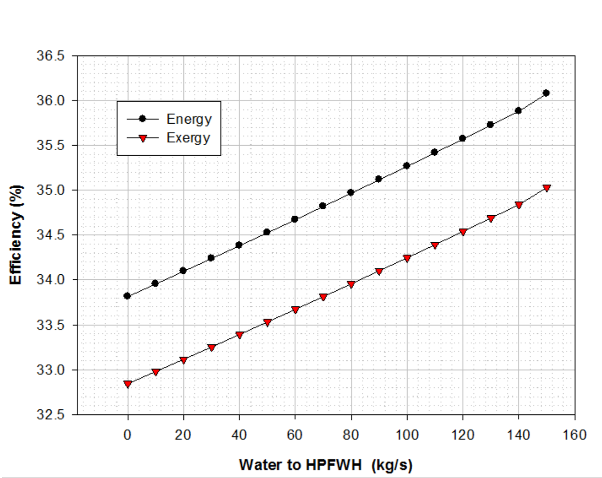

Figure 12: Variation of energy and exergy efficiencies of cycle across water mass flow rate along heat recovery heat exchanger in second case

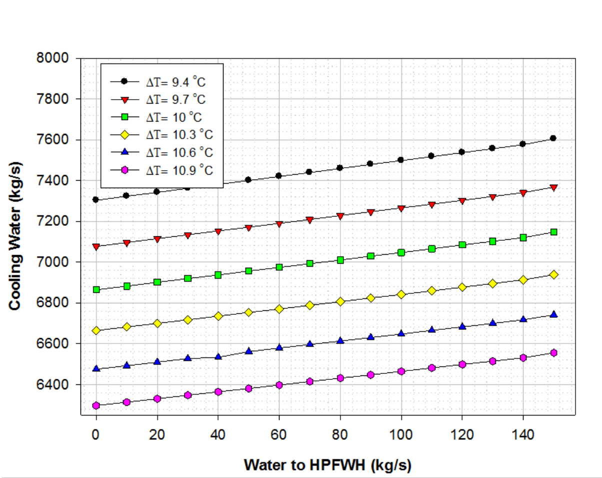

Figure 13: Variation of condenser cooling water across water mass flow rate along heat recovery heat exchanger in second case

required steam flow rate of high pressure heaters and will increase the required steam flow rate of the deaerator. The required steam flow rate for the deaerator in normal condition (simple cycle) is 2.16 kg/s which, after the passed water flow rate from the high pressure heat recovery heat exchanger increases to 140 kg/s, will increase to 2.65 kg/s. In Fig. 15 we can see the entered steam changes to the high and low pressure turbines, condenser and passed water from the high and low pressure steam flash heaters versus changes in passed water from the high pressure heat recovery heat exchanger. It is clear that an increase in passed water flow rate from the high pressure heat recovery heat exchanger will increase the entered flow rate to the condenser, intermediate pressure turbine and low pressure heaters and will reduce the entered flow rate

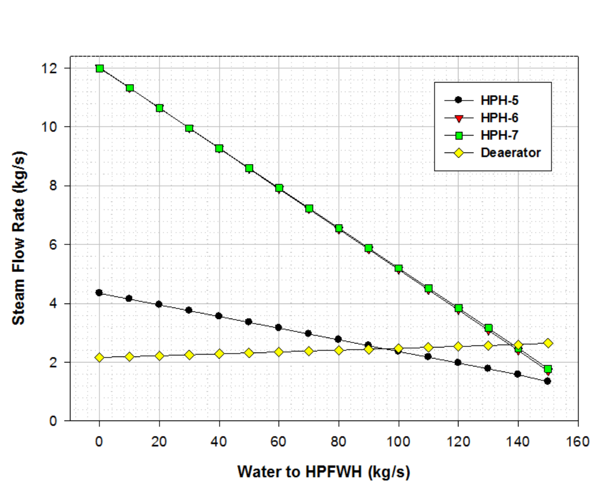

Figure 14: Variation of steam flow rate of steam flash heat exchangers across water mass flow rate along heat recovery heat exchanger in second case

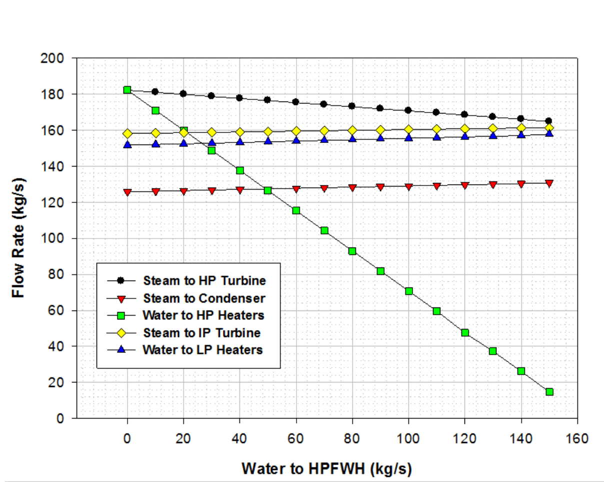

Figure 15: Variation of flow rates in major points across water mass flow rate along heat recovery heat exchanger in second case

to the high pressure turbine and high pressure heaters.

6.3. Third case

In Fig. 16 we can see net efficiencies for cycle energy and exergy versus the passed water flow rate from the heat recovery heat exchangers. In this case, energy and exergy efficiencies start from 34.4% and 33.296% respectively. When the flow rate is 0 and after the passed flow rate from heat recovery heat exchangers increases to 140 kg/s, it will increase to 37.4% and 36.3%. In Fig. 17 the required water flow rate of the cooling tower in various passed water flow rates from the heat recovery heat exchangers is seen in various temperature differences in cooling tower water. If we assume that the available flow rate for rotatory water

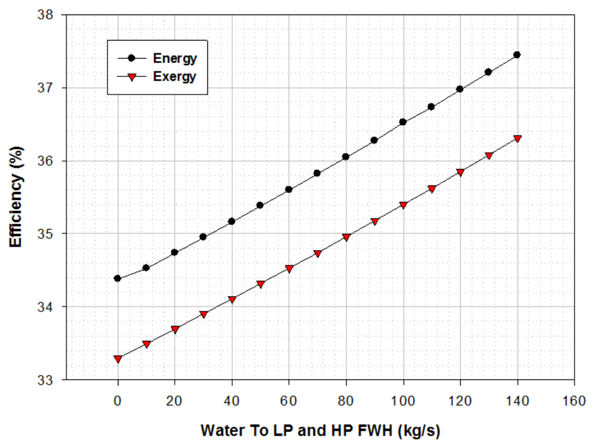

Figure 16: Variation of energy and exergy efficiencies of cycle across water mass flow rate along heat recovery heat exchangers in third case

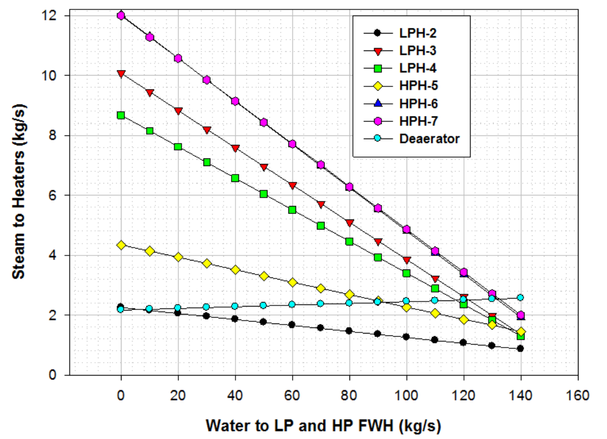

Figure 18: Variation of steam flow rate of steam flash heat exchangers across water mass flow rate along heat recovery heat exchangers in third case

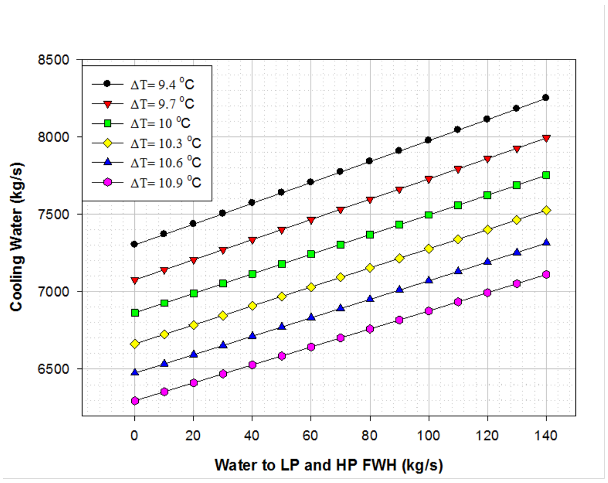

Figure 17: Variation of condenser cooling water across water mass flow rate along heat recovery heat exchangers in third case

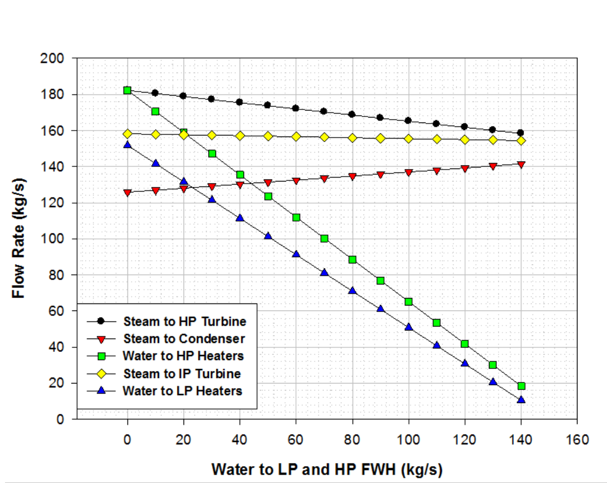

Figure 19: Variation of flow rates in major points across water mass flow rate along heat recovery heat exchangers in the third case

of the cooling tower is invariable, a temperature difference of 10.9^{o}C is required for cooling tower water when the passed flow rate from heat recovery heat exchangers is 100 kg/s, which is more than the values in the last two cases and shows that, in the third case, the required power for the cooling tower is more than in the first and second cases. In Fig. 18 the required steam flow rate changes for all of heaters and deaerators can be seen versus differences in passed water flow rate from heat recovery heat exchangers. By increasing the passed water flow rate from heat recovery heat exchangers, the steam flow rate of all heaters will be reduced. While the required steam flow rate for the deaerator increases, the entered water flow rate to the deaerator increases. In this case, the entered steam flow rate to the deaerator will change from the first case (normal cycle: 2.16 kg/s) to 2.56 kg/s when the water flow rate is 140 kg/s for heat recovery heat exchangers.

In Fig. 19 the entered steam flow rates to the high and low pressure turbines, condenser and the passed water from the high and low pressure heaters can be seen versus changes in passed water from the heat recovery heat exchangers. When the passed water flow rate from the heat recovery heat exchangers increases, we will have an increase in the entered steam flow rate to the condenser solely and the other cases will be decreased.

In Figs. 20 and 21, the exergy and energy efficiency changes are seen respectively versus changes in the passed water flow rate from the heat recovery heat ex-

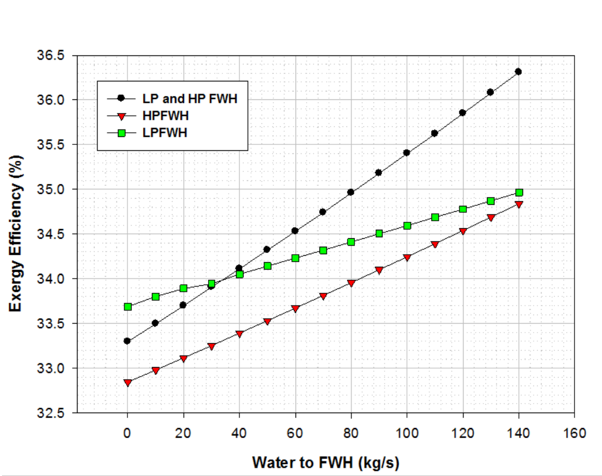

Figure 20: Variation of exergy efficiencies of cycle across water mass flow rate along heat recovery heat exchangers in all cases

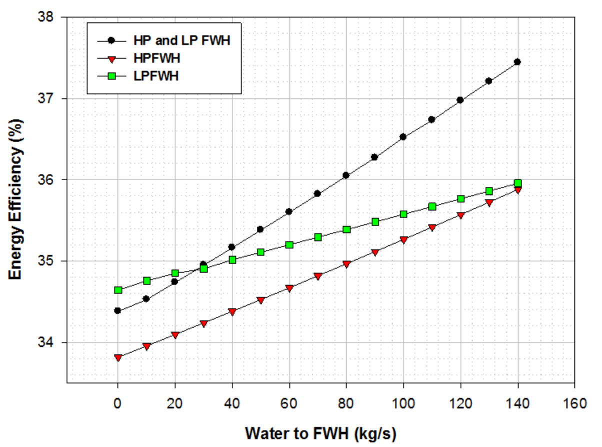

Figure 21: Variation of energy efficiencies of cycle across water mass flow rate along heat recovery heat exchangers in all cases

changers in all three cases. As we can see, the third case has a steeper gradient of increase in efficiencies than the other two. The starting points of these lines are dependent on optional gas turbines. Nevertheless, the efficiency increase rate is not dependent on the turbine model alone.

In Fig. 22 the required water flow rate changes in the cooling tower are seen versus changes of the passed water flow rate from the heat exchangers in all three cases. For the purposes of calculating the required water flow rate for the cooling tower in this figure, the possible temperature difference for the cooling tower is assumed at 10°C. The second case is seen to have a less negative effect on condenser vacuum.

In Fig. 23 changes of the entered steam flow rate to the high pressure turbine are seen versus changes of

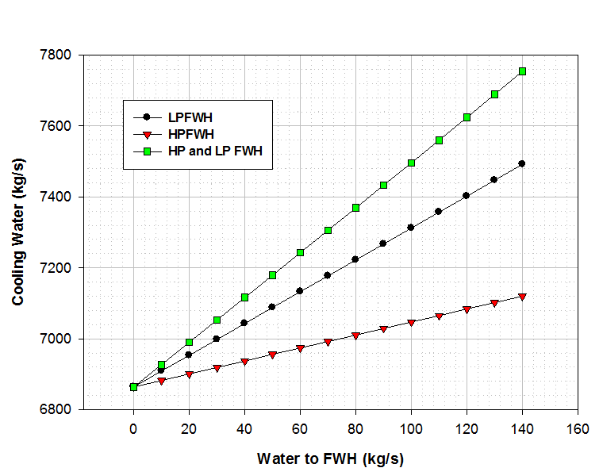

Figure 22: Variation of cooling water across water mass flow rate along heat recovery heat exchangers in all cases

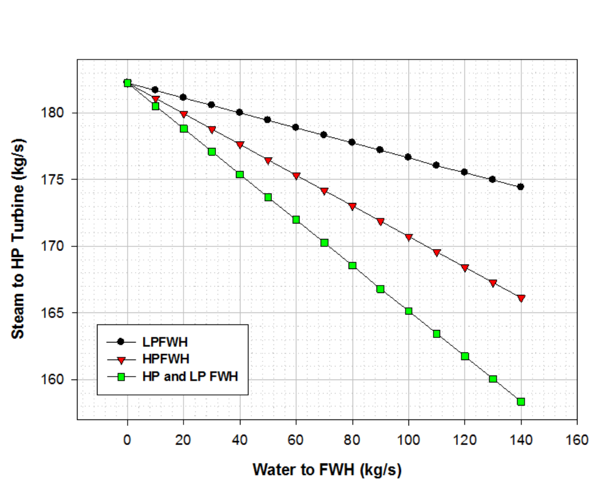

Figure 23: Variation of inlet steam to HP turbine across water mass flow rate along heat recovery heat exchangers in all cases

the passed water flow rate from the heat recovery heat exchangers in all three cases. The entered steam flow rates to the high pressure turbine, which is 182 kg/s in a simple cycle, for cases 1 to 3 are: 174, 165 and 158 kg/s respectively, with a 140 kg/s passed water flow rate from the heat recovery heat exchangers. Reduction of this parameter entails a steam generation decrease in the existing boiler. A reduction in steam generation in addition to a reduction in fuel consumption leads to an increase in combustion efficiency in the boiler and reduces pollutant emissions and consumption in the power plant. In light of the above descriptions, the third case with its implementation of parallel feed water heating repowering, is the best case for minimizing boiler costs.

7. Conclusion

This article presents an analysis of parallel feed water heating repowering of Shahid Montazeri power plant in Iran. For the repowering of a steam power plant, a gas turbine and two heat recovery heat exchangers were to be replaced by high and low pressure steam flash heaters. Three cases for repowering were suggested separately and in every case, using appropriate turbine, the results were checked. Energy and exergy efficiencies of the repowered cycle were selected as target functions. In every case the new net efficiency values of exergy and energy and changes in condenser vacuum, were analyzed carefully. Meanwhile having condenser vacuum as a non-variable, we considered the entered steam flow rate to the condenser in the case of changes in temperature difference in respect of the cooling water. Cooling tower modeling in the power plant is carried out at 16.1°C and in this case, temperature differences for cooling tower water circulation is 10°C. With environmental temperature, increasing the cooling tower temperature difference is less possible. Therefore in addition to calculating the required temperature difference in every case, we can get cycle efficiency changes in different environment temperatures at different steam extraction flow rates. Calculations related to efficiencies and required temperature differences in cooling tower flow rates from heat exchangers passed water in 100 kg/s were made. In the first case, the heat recovery heat exchanger is used. This project, in spite of its affordability, will have the most negative effects on condenser vacuum. In this case an EGT-RLM6000-PC gas turbine is used. Generated power, energy efficiency, exergy efficiency and water temperature difference in the cooling tower increased to 43.0 MW, 3.8%, 3.79% and 0.652°C, respectively. In the second case, the costs are higher, because high-pressure heat exchangers are more expensive to produce than low pressure heat exchangers. By implementing this project and using a Siemens V64.3 (KWU) gas turbine, the cycle power, energy and exergy efficiency and water temperature difference increased to 63 MW, 6.68 %, 6.65% and 259°C, respectively. In the third case a Westinghouse gas turbine was used and both high and low pressure heat exchangers were eemployed. In this case, the cycle power, energy efficiency, exergy efficiency and water temperature differences increased to 85.9 MW, 8.93%, 9.05% and 0.87°C respectively. This case is the best project in terms of cycle efficiency and power increase, but it entails the most negative effect on condenser vacuum.

Acknowledgments

Authors thank Mr. Lotfi and Mr. Abbasi in Mohammad Montazeri Steam Power Plant For Their help to in preparing technical information.

References

[1] Iran detailed statistics of electricity industry, for production management in 2013, in Persian.

[2] Iran Detailed statistics of electricity industry, for strategic management in 2013, in Persian.

[3] A. Singh, D. Kopecky, Repowering considerations for converting existing power plants to combined cycle power plants, in: 2002 International Joint Power Generation Conference, American Society of Mechanical Engineers, 2002, pp. 607–613.

[4] J. Feenstra, P. Kamminga, Mechanical and process design of the flevo power station conversion project, in: Publication ASME 89-GT-38, American Society of Mechanical Engineers, 1986.

[5] J. Schretter, S. Williams, J. Brandett, Two financial reasons why repowering is compelling compared to other power supply options, in: ASME Turbo Expo 2003, collocated with the 2003 International Joint Power Generation Conference, American Society of Mechanical Engineers, 2003, pp. 787–792.

[6] P. Ploumen, J. Veenema, Dutch experience with hot windbox repowering, in: ASME 1996 International Gas Turbine and Aeroengine Congress and Exhibition, American Society of Mechanical Engineers, 1996, pp. V004T10A013–V004T10A013.

[7] R. K. Tawney, J. A. Bonner, A. M. Elgawhary, Economic and performance evaluation of combined cycle repowering options, in: ASME Turbo Expo 2002: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2002, pp. 457–464.

[8] W.C. Stenzel, Sopocy, D.M. Pace, "Repowering Existing Fossil Steam Plants", www.Sepril.com.

[9] D. H. Cooke, Modeling of off-design multistage turbine pressures by stodola's ellipse, in: Energy Incorporated PEPSE User's Group Meeting, Richmond, VA, Nov, 1983, pp. 2–3.

[10] L. M. Romeo, J. M. Escosa, Repowering con turbina de gas de una central termoeléctrica de carbón de 335 mw, Ingeniería Química 31 (184) (2005) 50–56.

[11] Archive of Mohammad Montazeri Power Plant.

[12] K. M. MEHRABANI, S. S. F. YAZDI, A. MEHRPANAHI, S. N. N. ABAD, Optimization of exergy in repowering steam power plant by feed water heating using genetic algorithm, Indian J. Sci. Res 1 (2) (2014) 183–198.

[13] M. Z. Yilmazoglu, A. Durmaz, Hot windbox repowering of coal-fired thermal power plants, Turkish Journal of Engineering & Environmental Sciences 37 (2013) 33–41.

[14] M. R. Shahnazari, D. Foroughi, H. Fakhrian, Repowering of lowshan power plant, in: IGTC Conf., Tokyo, 2003.

[15] M. Wolowicz, J. Milewski, K. Badyda, Feedwater repowering of 800 mw supercritical steam power plant, Journal of Power Technologies 92 (2) (2012) 127–134.

[16] M. Baghestani, M. Ziabasharhagh, M. Khoshgoftar Manesh, Evaluation of repowering in a gas fired steam power plant based on exergy and exergoeconomic analysis, in: World Renewable Energy Congress, 2011, pp. 8–13.
[17] M. Fränkle, Srs: The standardized repowering solution for 300mw steam power plants in russia, Siemens Power Generation Germany. Report.
[18] A. Mehrpanahi, S. Hossienalipour, K. Mobini, Investigation of the effects of repowering options on electricity generation cost on iran steam power plants, International Journal of Sustainable Energy 32 (4) (2013) 229–243.
[19] M. Asadian, R. Samadi, Comparison of lowshan power plant repowering options for better plant application, in: 16th Iranian Soc. of Mech. Eng. Conf.(ISME), 2008.
[20] R. Haghighi, M. Tanassan, Feed-water repowering in beast power plant: Technical and costing aspects, Heat transfer and environment (2008) 187–192.
[21] R. E. Sonntag, C. Borgnakke, Fundamentals of thermodynamics, John Wiley & Sons, United State, 2009.
[22] M. Ameri, P. Ahmadi, A. Hamidi, Energy, exergy and exergoeconomic analysis of a steam power plant: a case study, International Journal of Energy Research 33 (5) (2009) 499–512.
[23] A. Bejan, G. Tsatsaronis, M. J. Moran, Thermal design and optimization, John Wiley & Sons, 1996.
[24] V. Nayyeri, S. Sayed-Mahmoodi, Use of fresh air in hot windbox power plant repowering method, in: 21th Iranian Power System Conf.(PSC), 2006, in Persian.
[25] S. Sanaye, M. Ziabasharhagh, M. Ghazinejad, Optimal design of gas turbine chp plant with preheater and hrsg, International Journal of Energy Research 33 (8) (2009) 766–777.

Nomenclature

0 Reference conditions of ambient

$\bar{R}$ World Constant for gases

$\dot{I}_{des}$ Destroyed exergy, kW

η_1 First low efficiency

η_2 Second low efficiency

ψ specifi exergy, kW/kg

ε Chemical exergy of fuel

ξ Exergy coefficient of fuels

a air

$c.s$ Control surface

$c.v$ Control volume

c_ch Combustion chamber

des destroyed

f fuel

fw Feed water

g gas

i inlet

o outlet

r relative

reh reheat

B Boiler

BFP Boiler feed water pump

CP-1st First stage Condensate pump

CP-2st Second stage Condensate pump

E Total energy, kJ

e Specific energy, kJ/kg

Ex Flow exergy, MW

G Gibbs function

h Specific enthalpy, kJ/kg

HPFWH High pressure feed water heater

HPH High pressure heater

LHV Lower heating value

LPFWH Low pressure feed water heater

LPH Low pressure heater

m Mass flow rate, kg/s

P pressure, bar

Q Heat, kW

s Specific entropy, kJ/kgK

st Steam turbine

T Temperature, °C

t Time, s

v Velocity, m/s

W Work, kW

Z Elevation, m