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# Energy and exergy analysis of Montazeri Steam Power Plant in Iran

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ABSTRACT

This paper aims at investigating steam cycle of Shahid Montazeri Power Plant of Isfahan with individual unit capacity of 200 MW. Using mass, energy, and exergy balance equations, all cycle equipment have been analyzed individually and energy efficiency, exergy efficiency, and irreversibility has been calculated for each of them as required. EES (Engineering Equation Solver) software is used for performing analyses. Values and ratios regarding heat drop and exergy loss have been presented for each equipment in individual tables. The results from the energy analysis show that 69.8% of the total lost energy in the cycle occurs in the condenser as the main equipment wasting energy, while exergy analysis introduces the boiler as the main equipment wasting exergy where 85.66% of the total exergy entering the cycle is lost.



**Contents**



## 1. Introduction

In many countries, steam power plant cycles have been discarded from the power generation cycle due to low efficiency, environment pollution, and especially insufficient fossil resources and have been replaced by the power plants with high efficiency and economic and technical justification. In Iran as well many research have investigated in recent years using new and green energies in spite of plenty available fossil fuels resources. Accordingly, using wind power, solar energy, nuclear power and other energies have been recommended. However, the power supply network's demand for high speed in installing and implementing power plants and good capabilities of steam power plants in this aspect has made using them justifiable. Also, availability of fossil fuels in Iran is another reason contributing justifiable utilization of these power plants. As up to 80% of the power generated in the world is produced through energy conversion from power plants working with fossil fuels (such as fuel oil, light crude oil, gasoline, and natural gas) and the rest from the other resources

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**Nomenclature**

| | |
|---|---|
| $B$ | Boiler |
| $\dot{I}_{des}$ | Destroyed exergy (kW) |
| $\dot{m}$ | Mass flow rate (kg/s) |
| $\bar{R}$ | World constant for gases |
| BFP | Boiler feed pump |
| $C$ | Condenser |
| CP-1st | First stage condensate pump |
| CP-2nd | Second stage condensate pump |
| CT | Cooling tower |
| CWP | Circulating water pump |
| DE | Deaerator |
| DP | Drip pump |
| e | Specific energy (kJ/kg) |
| E | Total energy (kJ) |
| EJ | Ejector |
| $Ex$ | Flow exergy |
| EXP | Expansion valve |
| G | Generator |
| $g$ | The gravity of Earth (m/s$^2$) |
| GC | Gland condenser |
| GCO | Gland cooler |
| GV | Governing valve |
| $h$ | Specific enthalpy (kJ/kg) |
| HPH | High pressure heater |
| HPT | High pressure turbine |
| IPT | Intermediate pressure turbine |
| LPH | Low pressure heater |
| LPT | Low pressure turbine |
| P | Pressure (bar) |
| PP | Polishing plant |
| $Q$ | Heat (kw) |
| $s$ | Specific entropy (kJ/kg K) |
| $T$ | Temperature (°C) |
| $v$ | Velocity (m/s) |
| $W$ | Work (kW) |
| $z$ | Elevation (m) |
| $Ql$ | Heat loss (kW) |

*Greec symbols*

| | |
|---|---|
| $\eta_1$ | First low efficiency |
| $\eta_2$ | Second low efficiency |
| $\psi$ | Specific exergy (kW/kg) |

*Subscripts and superscripts*

| | |
|---|---|
| 0 | Reference conditions of ambient |
| $a$ | Air |
| $des$ | Destroyed |
| $g$ | Gas |
| $i$ | Inlet |
| $o$ | Outlet |

such as dams, nuclear power, wind power, solar energy, geothermal power and other energies [1] and also as major power plants in our country are of steam cycle type [2], it is necessary to carry out enough research toward optimizing the cycle of these power plants. Using the first and second laws of thermodynamics as available and useful tools for analyzing energy and exergy of power conversion systems seems an appropriate method. In this way, one can know the extent of heat loss and irreversibility of the processes. Exergy analysis flourished in recent years is an appropriate method toward understanding the processes through which some solutions for optimal usage of existing power plants can be developed [3]. Exergy analysis is a useful tool to represent the difference between energy losses and internal irreversibilities in a process [4]. Kotas and Szargut [5,6] used exergy analysis method for thermal, chemical, and metallurgical analyses of power plants. Kotas [5] has used Grossman diagrams in which any single flow is defined by its own exergy to determine the flow exergy in a system. He also points to thermo-economic issues and economic optimizations of thermodynamic systems. Exergy analysis is an appropriate method for measuring performances of process components. By this method, exergy of the points in which energy conversion takes place can be obtained, efficiencies of cycle components are calculated, and the place where the largest losses happen can be also identified and the steps can be in turn taken to decrease them [7]. Rosen and Dincer [8] introduce exergy analysis the best tool for deciding on optimization of the cycle with respect to the input information.

Vosoogh [9] performed energy and exergy analyses on the boiler of a steam power plant and concluded that decreasing combustion excess air from the fraction 0.4–0.15, energy and exergy efficiencies respectively increase to 0.19% and 0.37%. Also, reducing the temperature of the smoke leaving the chimney from 137 °C to 90 °C, the above efficiencies respectively increase to 0.84% and 2.3%. Abdussaeid Ganji et al. [10] analyzed energy and exergy of the equipment in Heat Recovery Steam Generators in combined-cycle power plants in order to develop an optimization plan. Among the parameters investigated in this research were drum pressure and arrangement of heat exchangers in Heat Recovery Steam Generators for high and low pressure parts. Kaushik et al. [11] analyzed energy and exergy in Rankine cycle and combined-cycle power plants. Among the differences between this research and other similar cases are detailed description of the process and providing all the equations required for simulating each equipment in Rankine cycle and combined-cycle, which can be good references for future works. They concluded that while the largest energy loss happens in the condenser, the largest exergy loss occurs in the boiler and during combustion process. This can be due to the incomplete combustion process, inappropriate heat insulation and entropy generation in the device. Reddy et al. [12] revealed interesting results in exergy analysis. They found that combustion chamber of gas turbine produces the largest exergy in the whole cycle so that increasing pressure ratio in gas turbine compressor can decrease exergy loss in other equipment. Ameri et al. [13] analyzed exergy of the combined-cycle power plant (420 MW) of Neka, Iran. Their results showed that combustion chamber, gas turbine, duct burner, and heat recovery steam generator are the main sources of irreversibility which comprise 83% of the total exergy loss of the power plant. The main source of exergy loss was also recognized to be the combustion chamber of gas turbine. The second source of exergy generation was heat recovery steam generator whose improvement can reduce the exergy loss of the power plant. Isam Aljundi [14] performed energy and exergy analysis in alhossein power plant in Jordan and concluded that by preheating the air entering the boiler and reducing fuel to air ratio, exergy loss in boiler (known as the main factor of exergy loss) can be reduced. Ahmadi et al. [15] performed energy and exergy analyses in steam power plant of Hamedan and investigated the effect of environment temperature changes and unit load variations on total efficiency and concluded that preheating the air entering the boiler causes an increase in efficiency so that it is better that power plant always works in its nominal load in order to increase the efficiencies. They also calculated exergy drop costs in each equipment. Hossein Ordon et al. [1] performed thermodynamic analysis in 9 power plants in Turkey and proposed their best working states in order to increase the efficiency.

### 1.1. Power plant cycle description

Shahid Montazeri Power Plant of Isfahan is located 15 km far from northwest of Isfahan along the Isfahan–Tehran highway next to the Isfahan Refinery in a 2.2 million $m^2$ land. This power plant has 8 similar steam units each with a capacity of 200 MW whose technical specifications have been presented in Table 1. Heat cycle of this power plant is a Rankine cycle whose heat process has been briefly shown in Fig. 1.

Each of the units in this power plant has a water–steam circuit as follows:

The distilled water with a temperature of 45.6 °C enters the first-stage pumps from the condenser and is subjected to a pressure up to 9 atm. Then, it enters the polishing plant and its purity increases passing the filters. In the next stage, its pressure increases up to 19.5 atm by second-stage pumps after passing ejector and gland condenser and enters low pressure heaters after passing the low pressure heater No. 1 and 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. Then, it is subjected to the pressure 180 atm by the boiler feed pumps and is directed into high pressure heaters. In high pressure heaters, feed water is heated up to 245 °C and reaches 320 °C passing 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 entrance of the water from drum to wall tubes and 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 leaving the boiler with pressure of 130 atm and temperature of 540 °C 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 the same route again.

**Table 1**
Operating conditions of the power plant.

| Operating conditions | value | unit |
|---|---|---|
| Power produced | 200 | MW |
| Power consumption | 14 | MW |
| Mass flow rate of fuel | 54 | $Nm^3/h$ |
| Heat rate | 10448.6 | kJ/kW h |
| Stem flow rate, main line | 670 | Ton/h |
| Steam pressure, main line | 130 | Bar |
| Steam temperature, main line | 540 | °C |
| Water temperature, to boiler | 247 | °C |
| Stack gas temperature | 160 | °C |
| Inlet gas volumetric flow rate to burners | $9.6*10^6$ | $Nm^3/h$ |
| Number of induced and draft fans | 2 | – |
| Number of burners | 12 | – |
| Combined pump/motor efficiency | 95 | % |

## 2. Analysis

In order to analyze energy and exergy in water–steam heat cycle of the power plant, thermodynamic parameters required for all cycle points in design conditions are extracted from the information available in the power plant archive (they are presented in Table 2) [16] and the appropriate standard volume for each equipment is determined. Applying conservation of mass, energy, and exergy equations for each standard volume, each equipment's status is then determined in terms of energy and exergy efficiency and values of unknown parameters are obtained as well. This is fulfilled using EES software [17]. This software helps us to be able to appropriately obtain optimal values of the concerned parameters as well as optimal usage of thermodynamic capabilities available in its library. Following are calculations and descriptions regarding the application of first and second laws of

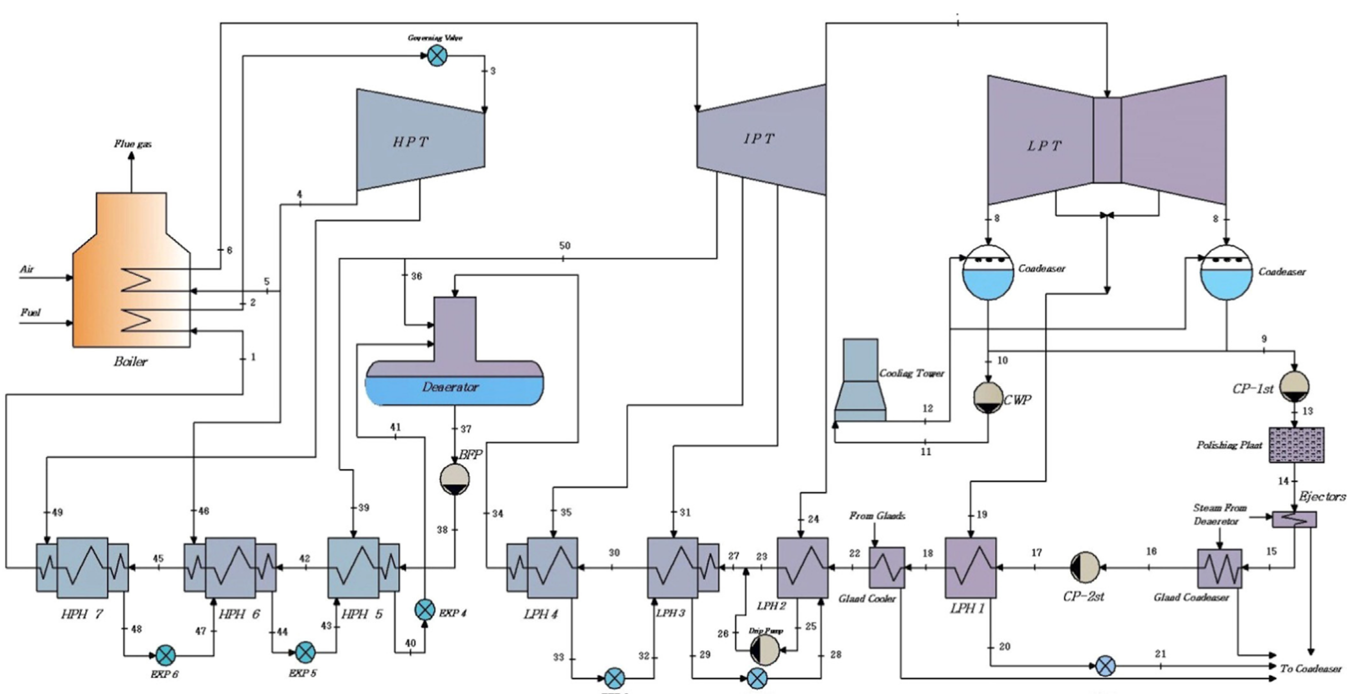

**Fig. 1.** Schematic diagram of power plant.

**Table 2**
Thermodynamic properties of points in cycle (refer to Fig. 1).

| Node | $T$ (°C) | $P$ (MPa) | $\dot{m}$ (kg/s) | $h$ (kJ/kg) | $s$ (kJ/kg K) | $\psi$ (kJ/kg) | $Ex$ (MW) |
|---|---|---|---|---|---|---|---|
| 1 | 519 | 17 | 179.4 | 1066 | 2.725 | 258.3 | 46,337 |
| 2 | 813.1 | 13 | 179.4 | 3444 | 6.574 | 1488 | 266,977 |
| 3 | 813.1 | 9.9 | 179.4 | 3477 | 6.731 | 1475 | 264,535 |
| 4 | 606.2 | 2.802 | 171.4 | 3079 | 6.714 | 1082 | 185,451 |
| 5 | 606.2 | 2.802 | 161.5 | 3079 | 6.714 | 1082 | 174,696 |
| 6 | 813.1 | 2.42 | 161.5 | 3552 | 7.452 | 1335 | 215,551 |
| 7 | 450 | 0.127 | 137.8 | 2828 | 7.622 | 559.8 | 77,137 |
| 8 | 318.2 | 0.017 | 133.1 | 188.4 | 0.6385 | 2.607 | 346.9 |
| 9 | 308.2 | 0.075 | 137.8 | 146.7 | 0.5049 | 0.6649 | 91.62 |
| 10 | 318.2 | 0.075 | 6944 | 188.5 | 0.6385 | 2.665 | 18,505 |
| 11 | 318.2 | 0.215 | 6944 | 188.6 | 0.6386 | 2.808 | 19,494 |
| 12 | 308.2 | 0.08 | 6944 | 146.7 | 0.5051 | 0.6713 | 4661 |
| 13 | 308.6 | 0.9 | 137.8 | 149.5 | 0.5114 | 1.561 | 215.1 |
| 14 | 308.6 | 0.51 | 137.8 | 149.1 | 0.5116 | 1.17 | 161.3 |
| 15 | 310.2 | 0.420 | 137.8 | 155.3 | 0.5319 | 1.308 | 180.3 |
| 16 | 312.2 | 0.31 | 137.8 | 163.6 | 0.5588 | 1.547 | 213.2 |
| 17 | 312.3 | 1.95 | 137.8 | 165.9 | 0.5608 | 3.228 | 444.8 |
| 18 | 322.5 | 1.73 | 137.8 | 207.9 | 0.6939 | 5.564 | 766.7 |
| 19 | 344.1 | 0.043 | 4.72 | 297.2 | 0.9671 | 13.42 | 63.35 |
| 20 | 329.1 | 0.035 | 4.72 | 234.4 | 0.7807 | 6.245 | 29.48 |
| 21 | 329.1 | 0.017 | 4.72 | 234.4 | 0.7807 | 6.227 | 29.39 |
| 22 | 325.1 | 1.6 | 137.8 | 219.1 | 0.7288 | 6.327 | 871.9 |
| 23 | 357.7 | 1.4 | 137.8 | 354.9 | 1.128 | 23.27 | 3207 |
| 24 | 443.2 | 0.13 | 5.5 | 2814 | 7.58 | 558.4 | 3071 |
| 25 | 378.5 | 0.13 | 17.79 | 441.5 | 1.366 | 38.63 | 687.1 |
| 26 | 379.3 | 1.45 | 17.79 | 445.8 | 1.374 | 40.64 | 723 |
| 27 | 360.1 | 1.4 | 155.6 | 365.2 | 1.156 | 25.02 | 3893 |
| 28 | 386.3 | 0.13 | 12.29 | 2697 | 7.298 | 525.9 | 6462 |
| 29 | 385.3 | 0.265 | 12.29 | 470.3 | 1.442 | 45.06 | 553.8 |
| 30 | 384.5 | 1.2 | 155.6 | 467.6 | 1.432 | 45.22 | 7035 |
| 31 | 532.3 | 0.265 | 5.12 | 2987 | 7.61 | 722.6 | 3700 |
| 32 | 430.5 | 0.265 | 7.169 | 2779 | 7.177 | 644 | 4617 |
| 33 | 430.5 | 0.598 | 7.169 | 664.4 | 1.917 | 97.42 | 698.4 |
| 34 | 432.1 | 0.9 | 155.6 | 671.5 | 1.933 | 99.85 | 15,535 |
| 35 | 644.2 | 0.598 | 7.169 | 3209 | 7.617 | 942.9 | 6760 |
| 36 | 720.3 | 1.248 | 1.8 | 3361 | 7.504 | 1129 | 2032 |
| 37 | 438.3 | 0.77 | 179.4 | 698.3 | 1.995 | 108.2 | 19,410 |
| 38 | 440 | 15.5 | 179.4 | 715.5 | 1.989 | 127 | 22,780 |
| 39 | 721.2 | 1.123 | 4.07 | 3365 | 7.557 | 1116 | 4543 |
| 40 | 459 | 1.123 | 22.01 | 2784 | 6.551 | 835.8 | 18,398 |
| 41 | 459 | 0.75 | 22.01 | 2809 | 6.78 | 792.3 | 17,439 |
| 42 | 454.5 | 18.00 | 179.4 | 777.8 | 2.13 | 147.4 | 26,439 |
| 43 | 511.6 | 1.123 | 17.94 | 2911 | 6.813 | 884.5 | 15,868 |
| 44 | 511.6 | 2.802 | 17.94 | 2830 | 6.265 | 966.5 | 17,340 |
| 45 | 489.3 | 17.5 | 179.4 | 930.7 | 2.455 | 203.3 | 36,472 |
| 46 | 603.6 | 2.802 | 9.94 | 3073 | 6.703 | 1079 | 10,724 |
| 47 | 507.6 | 2.802 | 8 | 2817 | 6.24 | 961.3 | 7690 |
| 48 | 507.6 | 4.151 | 8 | 1011 | 2.649 | 266.2 | 1810 |
| 49 | 679.2 | 4.151 | 8 | 3268 | 6.832 | 1235 | 9882 |
| 50 | 720.3 | 1.248 | 5.87 | 3361 | 7.504 | 1129 | 6625 |

thermodynamics to the equipment. The first law of thermodynamics and the main exergy balance equation respectively equal [11]:

$$\sum \dot{Q}_k + \sum \dot{m}\left(h_i + \frac{C^2{}_i}{2} + gZ_i\right) = \sum \dot{m}\left(h_o + \frac{C^2{}_o}{2} + gZ_o\right) + \sum \dot{W} \tag{1}$$

$$\sum \left(1 - \frac{T_0}{T}\right) Q_k + \sum (\dot{m}_i \psi_i) = \sum \psi_W + \sum (\dot{m}_o \psi_o) + \dot{I}_{\text{destroyed}} \tag{2}$$

Mass, energy, and exergy balance equations for a standard volume in steady state disregarding potential and kinetic energy terms are respectively expressed as follows [14]:

$$\sum \dot{m}_i = \sum \dot{m}_o \tag{3}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_o h_o - \sum \dot{m}_i h_i \tag{4}$$

$$\dot{W} + \sum (\dot{m}_i \psi_i) = \dot{Q} + \sum (\dot{m}_o \psi_o) \tag{5}$$

where the net exergy transfer by heat $\psi_Q$ at the temperature of $T$ equals [14]:

$$\psi_Q = Q\left(1 - \frac{T_0}{T}\right) \tag{6}$$

Specific flow exergy also equals:

$$\psi = (h - h_0) - T_0(s - s_0) \tag{7}$$

Therefore, the total exergy of a flow is expressed as:

$$Ex = \dot{m}\psi = \dot{m}((h - h_0) - T_0(s - s_0)) \tag{8}$$

This stage deals with the way above equations are applied to the main cycle equipment.

### 2.1. Boiler

In the boiler, chemical exergy of the fuel is released during combustion process and is transferred to the water through a complex heat transfer process and finally a part of this exergy is directed toward the turbine in exit steam. The boiler, the main equipment destroying the exergy in the whole cycle, has low exergy efficiency due to the reasons such as incomplete combustion process and high temperature difference in the heat transfer process. Although the boiler of the power plant can work on three types of fuel as fuel oil, gasoline, and natural gas, all the calculations in this paper have been done for natural gas. For performing calculations, first of all exergy of all flows should be expressed. To this aim, specific exergy and flow rate of each Stream, including input and exit flows, are needed. Given the Eq. (7) and having specific enthalpy and entropy and base temperature ($T_0$), calculating exergy of all steam flows is performed easily. For boiler energy balance and energy efficiency can be shown respectively [11]:

$$\dot{m}_f h_f + \dot{m}_a h_a = \dot{m}_g h_g + \dot{m}_1(h_2 - h_1) + \dot{m}_5(h_6 - h_5) + Ql_B \tag{9}$$

$$\eta_{1,\text{boiler}} = \frac{\dot{m}_1(h_2 - h_1) + \dot{m}_5(h_6 - h_5)}{\dot{m}_f h_f} \tag{10}$$

Exergy balance and exergy efficiency [11]:

$$\dot{I}_{\text{destroyed,HPT}} = \left(\dot{m}_f \psi_f\right) + (\dot{m}_a \psi_a) + (\dot{m}_1 \psi_1) + (\dot{m}_5 \psi_5) - \left(\dot{m}_g \psi_g\right) - (\dot{m}_2 \psi_2) - (\dot{m}_6 \psi_6) \tag{11}$$

$$\eta_{2,\text{boiler}} = \frac{\left(\dot{m}_g \psi_g\right) + (\dot{m}_2 \psi_2) + (\dot{m}_6 \psi_6)}{\left(\dot{m}_f \psi_f\right) + (\dot{m}_a \psi_a) + (\dot{m}_1 \psi_1) + (\dot{m}_5 \psi_5)} \tag{12}$$

However, further calculations are needed for calculating exergy of the fuel and input and exit air. These calculations are explained below.

### 2.2. Fuel exergy

To calculate the boiler fuel exergy, we should first obtain hydrocarbon compositions of this fuel. The results of this analysis are presented in Table 3.

Rosen [8] has stated an equation for calculating chemical exergy of natural gas in which total fuel exergy can be calculated given the molar ratio and exergy of each component:

$$\Psi_f = N_f \times \Psi_{ch,f} \tag{13}$$

where $N_f$ is the mole fraction of each component of hydrocarbon component and $\Psi_{ch,f}$ is the chemical exergy of each component according to Table 4.

### 2.3. Calculating chemical exergy of gases leaving the boiler

Gases leaving the boiler leave the economizer with a temperature of 350 °C and go toward Ljungström. Temperature of the smoke leaving the chimney equals 160 °C. Due to intense air leakage in Ljungström and that we have airflow rate after Ljungströms, standard volume boundary is considered the smoke inlet to Ljungström and air outlet from Ljungström. Kotas [18] states the following equation to express exergy of combustion products:

$$\Psi_g = \sum_i x_i \bar{\varepsilon}_{o,i} + \bar{R} T_o + \sum_i x_i \ln x_i \tag{14}$$

where $\bar{\varepsilon}_{o,i}$ is molar chemical exergy specific to each component that are shown in Table 5, and $x_i$ is mole fraction of each component of combustion products and $\Psi_g$ is in joule per mole.

In order to calculate the percentage of the existing elements in gases leaving the chimney, combustion process has been balanced. Results of this balance are presented in Table 6 assuming 5% excess air as well as complete combustion process [6].

### 2.4. Calculating exergy of the air entering the boiler

As the air entering the furnace is preheated to 350 °C, its exergy should be considered and calculated. To this aim, Eq. (10) and data in Table 5 are used. Environment air composition have been considered according to Table 7 in this paper.

### 2.5. Turbine

As the turbine here is of three-stage type with high pressure turbine, intermediate pressure turbine, and low pressure turbine, the equations are individually applied to each one and finally their energy and exergy resultant is calculated. Respective equations for high pressure turbine are as follows [11]:

Energy balance and thermal efficiency or the efficiency of first law of thermodynamic:

$$W_{\text{HPT}} = \dot{m}_3(h_3 - h_{49}) + (\dot{m}_3 - \dot{m}_{49})(h_{49} - h_4) - Ql_{\text{HPT}} \tag{15}$$

$$\eta_{1,\text{HPT}} = \frac{W_{\text{HPT}}}{\dot{m}_3(h_3 - h_{49}) + (\dot{m}_3 - \dot{m}_{49})(h_{49} - h_4)} \tag{16}$$

Exergy balance and exergy efficiency:

$$W_{\text{HPT}} = \dot{m}_3(\psi_3 - \psi_{49}) + (\dot{m}_3 - \dot{m}_{49})(\psi_{49} - \psi_4) - \dot{I}_{\text{destroyed,HPT}} \tag{17}$$

$$\eta_{2,\text{HPT}} = \frac{W_{\text{HPT}}}{\dot{m}_3(\psi_3 - \psi_{49}) + (\dot{m}_3 - \dot{m}_{49})(\psi_{49} - \psi_4)} \tag{18}$$

Results from energy and exergy analyses for turbines are observed in Table 8.

### 2.6. Condenser

Power plant condensers are of direct spray type that steam leaving the low pressure turbine, feed water heater condensates No. 1, gland cooler, gland condenser and ejectors enters it and inlet route of first-stage condensate pumps is derived from it. As the

**Table 3**
Volume fraction of the natural gas components [15].

| Component of natural gas | Volume fraction (%) |
|---|---|
| Methane ($CH_4$) | 98.57 |
| Ethane ($C_2H_6$) | 0.63 |
| Propane ($C_3H_8$) | 0.1 |
| Butane ($C_4H_{10}$) | 0.05 |
| Pentane ($C_5H_{12}$) | 0.04 |
| Nitrogen ($N_2$) | 0.6 |
| Carbon dioxide ($CO_2$) | 0.01 |

**Table 4**
Chemical exergy of fuels [8].

| Kind of fuel | | $\Psi_{ch,f}$ (kJ/kmol) |
|---|---|---|
| Methane | ($CH_4$) | 832,400 |
| Ethane | ($C_2H_6$) | 1,362,200 |
| Propane | ($C_3H_8$) | 2,153,200 |
| Butane | ($C_4H_{10}$) | 2,807,700 |

**Table 5**
Chemical exergy of flue gas components [13].

| Gas | $\bar{\varepsilon}_{o,i}$ |
|---|---|
| $CO_2$ | 14.20 |
| $O_2$ | 97.3 |
| $N_2$ | 0.72 |
| $H_2O$ | 11.71 |
| Ar | 11.69 |

**Table 6**
mol fraction of flue gas components.

| Gas | Mole fraction |
|---|---|
| $CO_2$ | 23 |
| $O_2$ | 2.5 |
| $N_2$ | 68 |
| $H_2O$ | 6.5 |

**Table 7**
Reference ambient model of air [15].

| Air components | Molar fraction (%) |
|---|---|
| $N_2$ | 76.62 |
| $O_2$ | 20.85 |
| $H_2O$ | 1.88 |
| $CO_2$ | 0.03 |
| Others | 0.92 |

**Table 8**
Turbines information.

| Turbine | $P$ (MW) | $\dot{I}_{des}$(MW) | $\eta_2$ | $\eta_1$ |
|---|---|---|---|---|
| HPC | 62.9 | 8.79 | 78.28 | 87.67 |
| IPC | 102.51 | 10.01 | 87.34 | 91.08 |
| LPC | 34.59 | 7.31 | 80.62 | 82.62 |

total lost heat through cooling tower cycle water is directed outside of the cycle, lost heat from cooling tower and condenser is considered the same in current calculations. Respective equations are as follows [11]:

Energy balance in the condenser:

$$Ql_c = \dot{m}_8(h_8 - h_9) + \dot{m}_{21}(h_{21} - h_9) - \dot{m}_{10}(h_{10} - h_{12}) \tag{19}$$

Exergy efficiency:

$$\eta_{2,c} = \frac{\dot{m}_{10}(\psi_{10}) + \dot{m}_9(\psi_9)}{\dot{m}_8(\psi_8) + \dot{m}_{21}(\psi_{21}) + \dot{m}_{12}(\psi_{12})} \tag{20}$$

It should be noted that some processes with variable and relatively small flow rates have been ignored in calculations for simplification. Among these processes are the steam entering and the condensate leaving the ejector, gland cooler, and gland condenser. There are many sub-processes which have been also ignored, such as the steam leaked from glands of turbines and all valves, boiler continues blow down and vents which flow consistently in the cycle.

### 2.7. Closed feed water heater

Closed feed water heater is a heat exchanger in which the processes are not mixed with each other. In the present cycle, ejector, gland cooler, gland condenser, low pressure and high pressure feed water preheating heaters belong to this group. Here, calculations related to low pressure heater No. 1 are presented [11]:

Energy balance and energy efficiency:

$$Ql_{\mathrm{LPH}-1} = \dot{m}_{19}(h_{19} - h_{20}) - \dot{m}_{17}(h_{18} - h_{17}) \tag{21}$$

$$\eta_{1,\mathrm{LPH}-1} = \frac{\dot{m}_{19}(h_{19} - h_{20})}{\dot{m}_{17}(h_{18} - h_{17})} \tag{22}$$

Exergy balance and exergy efficiency:

$$\dot{I}_{\mathrm{LPH}-1} = T_0\left[\dot{m}_{19}(\psi_{19} - \psi_{20}) - \dot{m}_{17}(\psi_{18} - \psi_{17})\right] \tag{23}$$

$$\eta_{2,\mathrm{LPH}-1} = \frac{\dot{m}_{17}(\psi_{18} - \psi_{17})}{\dot{m}_{19}(\psi_{19} - \psi_{20})} \tag{24}$$

For equipment like ejector, gland cooler, and gland condenser whose steam and condensate process has been ignored, the increase in feed water temperature is balanced with inlet heat which is the lost heat with a negative sign.

### 2.8. Open feed water heater

In open feed water heater, inlet water is mixed with heating steam and we have only one exit process. Deaerator in the present cycle is considered an open feed water heater and its respective equations are as follows [11]:

Energy balance and thermal efficiency:

$$Ql_{\mathrm{De}} = \dot{m}_{36}h_{36} + \dot{m}_{34}h_{34} + \dot{m}_{41}h_{41} - \dot{m}_{37}h_{37} \tag{25}$$

$$\eta_{1,\mathrm{De}} = \frac{\dot{m}_{37}h_{37}}{\dot{m}_{36}h_{36} + \dot{m}_{34}h_{34} + \dot{m}_{41}h_{41}} \tag{26}$$

Exergy balance and exergy efficiency:

$$\dot{I}_{\mathrm{De}} = \dot{m}_{36}\psi_{36} + \dot{m}_{34}\psi_{34} + \dot{m}_{41}\psi_{41} - \dot{m}_{37}\psi_{37} \tag{27}$$

$$\eta_{2,\mathrm{De}} = \frac{\dot{m}_{37}\psi_{37}}{\dot{m}_{36}\psi_{36} + \dot{m}_{34}\psi_{34} + \dot{m}_{41}\psi_{41}} \tag{28}$$

Results from exergy analysis of all feed water heaters are observed in Table 9.

### 2.9. Pump

In the present cycle, there are two circulating water pumps, one first-stage condensate pump, two second-stage condensate pumps, one drip pump and two working boiler feed pumps. Calculations for all pumps are identical. For the boiler feed pump, we have [11]:

Energy balance and heat efficiency:

$$W_{\mathrm{BFP}} = \dot{m}_{37}(h_{38} - h_{37}) + Ql_{\mathrm{BFP}} \tag{29}$$

$$\eta_{1,\mathrm{BFP}} = \frac{\dot{m}_{37}(h_{38} - h_{37})}{W_{\mathrm{BFP}}} \tag{30}$$

Exergy balance and exergy efficiency:

$$W_{\mathrm{BFP}} = \dot{m}_{37}(\psi_{38} - \psi_{37}) + \dot{I}_{\mathrm{BFP}} \tag{31}$$

$$\eta_{2,\mathrm{BFP}} = \frac{\dot{m}_{37}(\psi_{38} - \psi_{37})}{W_{\mathrm{BFP}}} \tag{32}$$

**Table 9**
Exergy balance of the feed water heaters.

| Heater | $\dot{I}_{des}$ (MW) | $\eta_2$ |
|---|---|---|
| LPH-1 | 0.43 | 72.91 |
| LPH-2 | 0.87 | 78.02 |
| LPH-3 | 0.63 | 81.18 |
| LPH-4 | 1.52 | 82.39 |
| Deaerator | 0.02 | 90.91 |
| HPH-5 | 0.53 | 82.88 |
| HPH-6 | 2.13 | 87.29 |
| HPH-7 | 0.88 | 92.16 |

## 3. Cycle analysis

Steam cycle of Mohammad Montazeri Power Plant of Isfahan was analyzed in terms of energy and exergy. For exergy analysis, environment base condition was considered with a pressure of 101.3 kpa and temperature of 298.15 K. Given this assumption, specific exergy and flow exergy values for all cycle points (Fig. 1) are as in Table 1 obtained by EES software. In Table 10, results of energy analysis for the cycle equipment are observed. Energy efficiency obtained based on lower heating value of the consumed fuel is 32%. In this table, the extent of heat loss as well as energy efficiency and heat drop percentage are observed for each equipment. The condenser with 296.8 MW energy loss covers 69.8% of the total lost energy in the cycle, while the boiler with 42.9 MW energy loss has only 10.16% of the total lost energy. In Table 11, results from exergy analysis of the cycle and equipment are observed. The boiler with 315.39 MW inlet fuel exergy loss covers 85.66% of the total lost exergy, while the condenser with 5.63 MW lost exergy has only 1.53% of the total lost exergy. According to the first law of thermodynamics analysis, the condenser covers 69.8% of the total lost energy of the cycle, while the second law of thermodynamic concerns energy quality and measures the temperature difference of fluid relative to environment base temperature in its analysis and knows the lost energy in the condenser very small in spite of small temperature difference of condenser water with environment. In the boiler, given high temperature of the flame and relatively high temperature difference between generated steam and gases from combustion, a large exergy is lost. Combustion process is also another major factor of exergy loss in the boiler. Exergy efficiency calculated for the cycle is 35.2% which is a relatively appropriate efficiency for the steam cycle. This is due to designing proper number of feed water preheating heaters and combustion air preheating. Given the results from cycle exergy analysis, the boiler has the highest potential for optimization. To reduce heat transfer exergy losses in the boiler, temperature difference should be reduced as much as possible and heat transfer area should be increased. Given the results from turbines analysis, it is observed that exergy loss in turbines is 26.1 MW. Among the factors causing these exergy losses can refer to throttling in governing valves, heat loss from turbine body, steam leakage from glands of turbines and valves on the way of steams and internal irreversibilities. Given the relationship flow exergy has with base temperature and assuming constant environment temperature, as base temperature increases, flows exergy decreases; but what is obvious is that the effect of environment temperature changes on condenser efficiency is proportionally more than that on the boiler efficiency. Generally, the more the temperature difference with environment, the less exergetic efficiency is affected by the environment temperature. This can be observed in Figs. 2 and 3.

**Table 10**
Energy balance of the power plant components and percent ratio to total energy loss.

| Component | $Ql$ (MW) | Percent ratio | $\eta_1$ |
|---|---|---|---|
| Boiler | 42.9 | 10.16 | 90.55 |
| HPC | 17.383 | 4.09 | 78.28 |
| IPC | 14.85 | 3.49 | 87.34 |
| LPC | 8.314 | 1.96 | 80.62 |
| Condenser | 296.8 | 69.8 | – |
| LP and HP heaters | 7.32 | 1.72 | – |
| BFP | 1.76 | 0.414 | 68.1 |
| CWP | 0.93 | 0.25 | 69 |
| Generator | 4.1 | 0.96 | 0.98 |
| Piping | 30.643 | 7.21 | – |
| Cycle | 425 | 100 | 32 |

**Table 11**
Exergy balance of the power plant components and percent ratio to total exergy loss.

| Component | $\dot{I}_{des}$ (MW) | Percent ratio | $\eta_2$ |
|---|---|---|---|
| Boiler | 315.39 | 85.66 | 44.5 |
| HPC | 8.79 | 2.38 | 87.67 |
| IPC | 10.01 | 2.72 | 91.08 |
| LPC | 7.31 | 1.98 | 82.62 |
| Condenser | 5.63 | 1.53 | – |
| LP and HP heaters | 6.51 | 1.77 | – |
| BFP | 0.524 | 0.14 | 90.5 |
| CWP | 0.51 | 0.13 | 83 |
| Generator | 4.1 | 1.11 | 0.98 |
| Piping | 9.406 | 2.55 | – |
| Cycle | 368.18 | 100 | 35.2 |

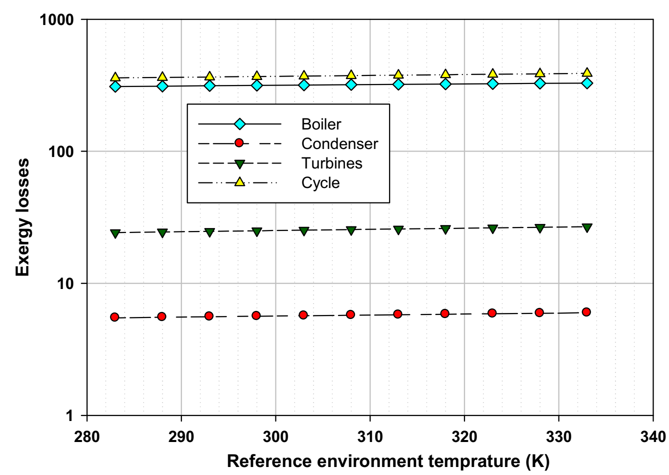


**Fig. 2.** Effect of reference environment temperature on total exergy destruction rate in major plant component.

As mentioned before the energy and exergy losses vary for power equipments. Also the energy and exergy efficiency defined for these equipments are also different. There are detailed discussions in this regard in previous sections and these cases can also be observed in Tables 10 and 11. To better understand the energy and exergy losses of each device are shown in Fig. 4. Also the energy and exergy efficiency are shown in Fig. 5 simultaneously.

In Figs. 6 and 7, the unit load changes following the inlet steam temperature changes into HP and IP turbines are visible. By reducing the steam temperature the produced load is reduced in both situations. Fig. 8 presents the impact of the inlet steam pressure to HP turbine on the unit load. As you can see the inlet steam pressure drop into the turbine leads to reduced load. This is because of the steam enthalpy drop following the pressure drop.

The passage of steam through the reheat superheaters is followed by pressure drop. This pressure drop is followed by reduction in load production. Assuming a constant outlet pressure from HP turbine and changes in the inlet pressure into the IP turbine changes occur at the unit load. This is shown in Fig. 9.

In powerhouse in order to provide some requirements to specific pressure of steam, specific temperature to heat various sectors and the need for auxiliary equipments the outlet steam from the HP turbine is used. By increasing the rate of surplus outlet steam flow described in this section, unit load is reduced. Fig. 10 presents the effect of flow changes on the load unit.

## 4. Conclusions and recommendations

In this paper, steam cycle of Shahid Mohammad Montazeri Power Plant and all its equipment were analyzed in terms of

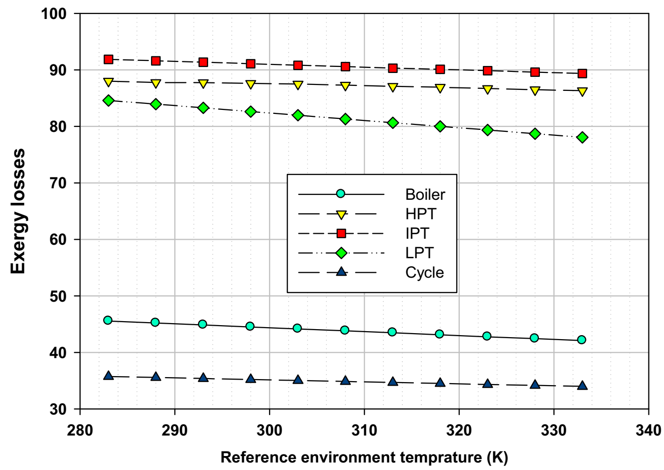


**Fig. 3.** Effect of reference environment temperature on the exergy efficiency of major plant component.

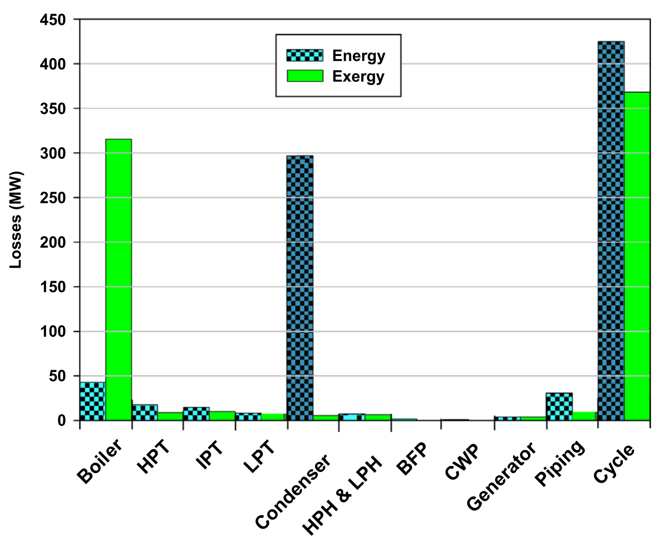


**Fig. 4.** Amount of energy and exergy losses in power plant and main equipments.

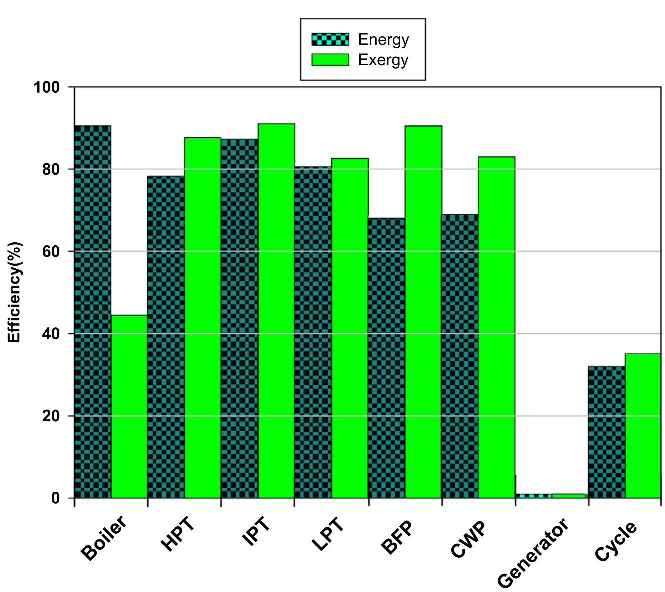


**Fig. 5.** The energetic and exergetic efficiencies of power plant and main equipments.

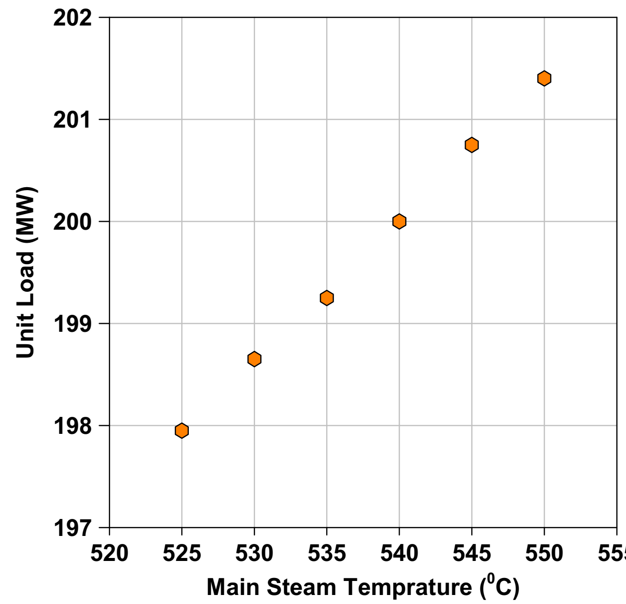


**Fig. 6.** Effect of unit load changes against the HP turbine inlet temperature.

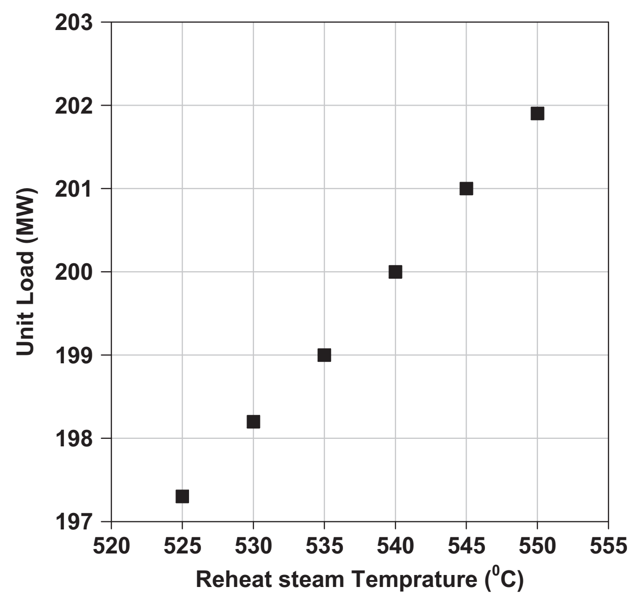


**Fig. 7.** Effect of unit load changes against the IP turbine inlet temperature.

energy and exergy. Values of heat loss, exergy loss, energy efficiency and exergy efficiency were calculated for many equipment. In addition, the effect of environment temperature changes on exergy loss and exergy efficiency were presented for boiler, condenser and turbine. In the performed analysis, the condenser has the largest heat loss in the cycle as 69.8% of the total lost heat which has a high potential for optimization and output increase according to first law of thermodynamics. While in exergy analysis, the condenser has only 1.53% of the total lost exergy which has almost no potential for optimization. Exergy analysis introduces the boiler as the main equipment destroying the exergy. The boiler has only 10.16% of the total lost heat but 85.66% of the total lost exergy. Combustion process optimization through enough combustion air preheating, creating suitable fuel and air mixing operation, reducing temperature difference of combustion products and steam in every stage of heat transfer process, proper isolating of boiler body, controlling the amount of combustion excess air at optimal level, and using energy of the exit gases can

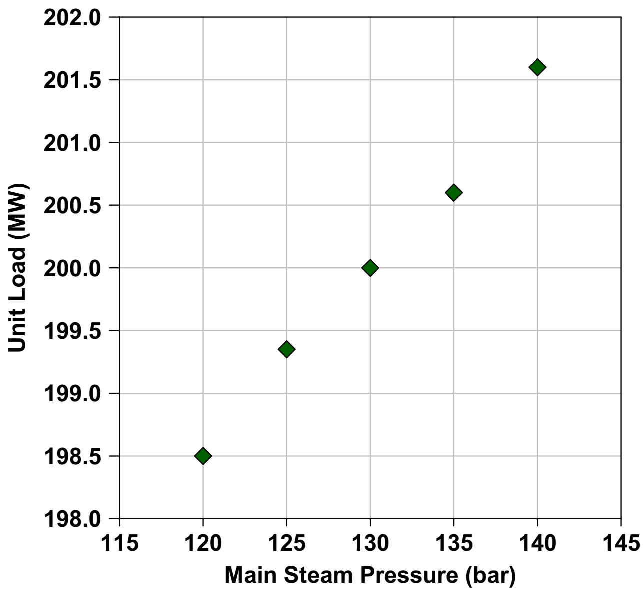


**Fig. 8.** Effect of unit load changes against the HP turbine inlet pressure.

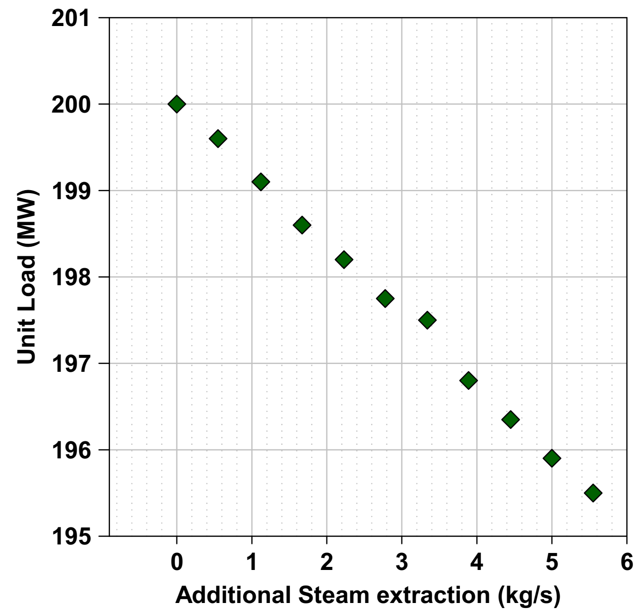


**Fig. 10.** Variations of the unit load against additional steam extraction flow rate.

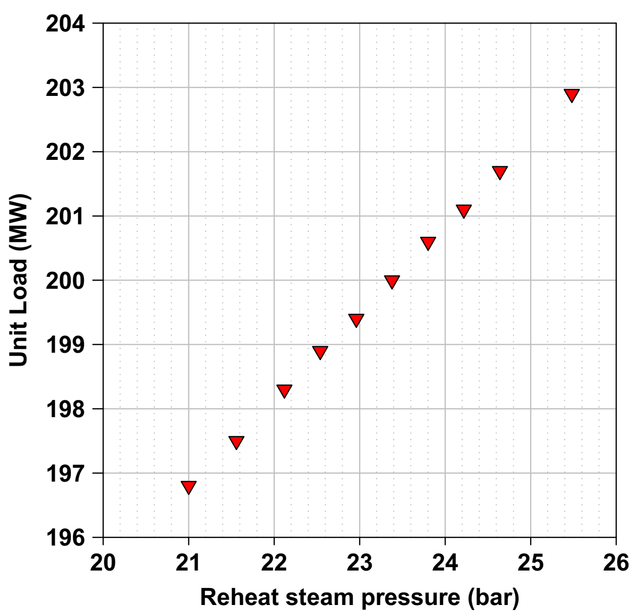


**Fig. 9.** Effect of unit load changes against the IP turbine inlet pressure.

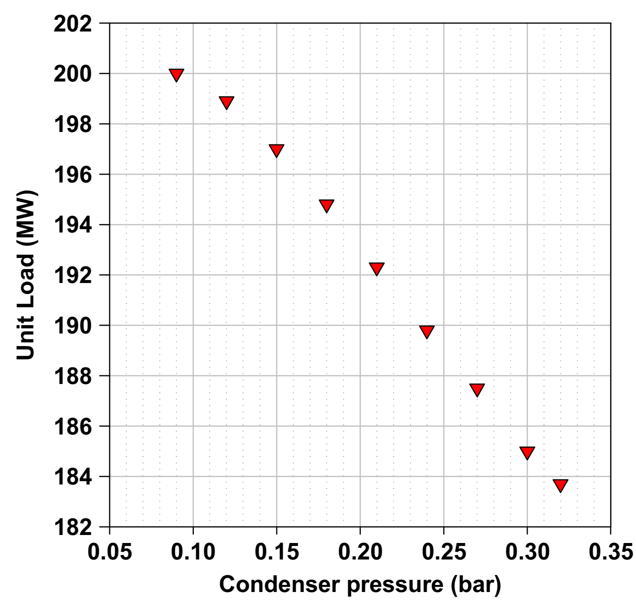


**Fig. 11.** variations of the unit load against pressure variation in the condenser.

be effective in controlling exergy loss. In energy analysis, efficiency of the first law of thermodynamics was calculated as 32%. Exergy analysis shows that although the energy lost through the condenser comprises a major part of the lost energy, it is not significant due to its low quality. Exergetic efficiency obtained is 35.2%. As it is higher than the energy efficiency, heat transfer processes in the cycle and especially the boiler are in a good range. This shows proper design of the cycle equipment. In exergy analysis, turbine is introduced as the second equipment destroying the exergy with 7.08% of the total lost exergy. By changes in environment temperature, percentages of exergy loss and exergy efficiency change for equipment, while the boiler covers yet the major part of exergy loss. To increase the cycle efficiency, several suggestions are introduced each with positive effects on improving power plant efficiency and saving energy consumption. Fig. 11 shows the variations of the load against pressure variation in the condenser. As can be seen condenser pressure has a significant impact on the load produced by the turbine. In this figure it is seen that for each 0.01 bar increase in condenser pressure approximately 0.7 MW power generation is reduced. As it is specified by increasing the condenser pressure from 0.09 bar to 0.32 bar the load unit reduced from 200 MW to 183.7 MW. In Fig. 12 the changes in thermal efficiency of the cycle following the condenser pressure changes can be observed. As condenser pressure has a large effect on turbines efficiency and finally on production increase, every step toward reducing the condenser pressure will be effective. Units' cooling towers are Heler tower type. Removing unevenness around the cooling towers, cutting trees near the towers which have caused reduction of exhaust airflow from radiators, installing walls directing the airflow and using mirrors reflecting sun light for heating upper part of the phase-1 cooling towers of the power plant are among strategies for increasing tower efficiency and in turn increasing condenser vacuum. As the shell of phase-1 towers is made of metal and metal sheets have been also used for tower building, if we can heat the tower body

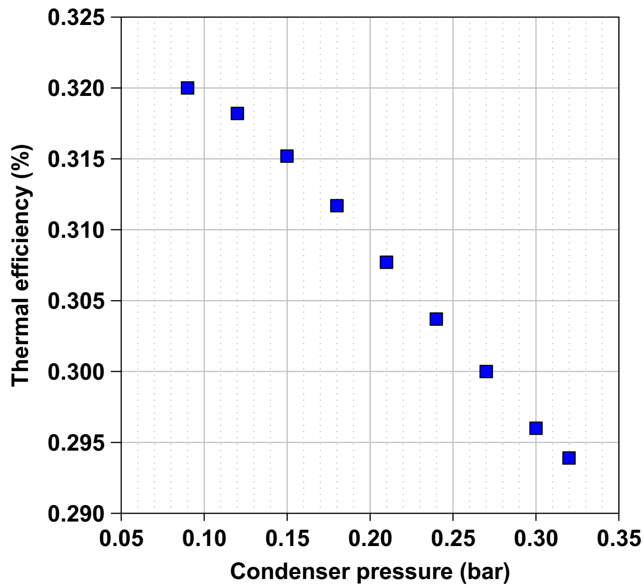


**Fig. 12.** variations of the power plant efficiency against the condenser pressure.

using fixed mirrors installed in a proper distance from the tower, heat transfer to the internal air and finally floating power of the internal airflow increase. This increases the speed of air suction from radiators pores and cooling power. Pumps and exhaust fans, blowers and circulation fans cover the largest part of internal power consumption. Using hydro coupling and drivers changing electromotor rotation can decrease the consumed power of these equipment. As two circulating water pumps are used in normal states, we can use one pump in cold seasons while the condenser vacuum remains at its optimal level. Repowering of steam cycle of the power plant can also be an effective and efficient suggestion for increasing production capacity and total efficiency of the units [19]. Repowering refers to adding gas turbine units to the existing steam units. Repowering itself consists of several methods and partial repowering is recommended for this power plant. Partial repowering itself also consists of the methods of hot wind box repowering, feed water heating repowering, and supplied boiler repowering. Choosing each of these methods depends on the parameters like original investment, the time along which the unit is out of the circuit, the level of increase in productive power of the power plant, and the level of increase in efficiency.

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