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Performance evaluations of a geothermal power plant

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ABSTRACT

Thermodynamic analysis of an operational 7.5 MWe binary geothermal power plant in Tuzla-Turkey is performed, through energy and exergy, using actual plant data to assess its energetic and exergetic performances. Eight performance-related parameters, namely total exergy destruction ratio, component exergy destruction ratio, dimensionless exergy destruction, energetic renewability ratio, exergetic renewability ratio, energetic reinjection ratio, exergetic reinjection ratio and improvement potential are investigated. Energy and exergy losses/destructions for the plant and its units are determined and illustrated using energy and exergy flow diagrams. The largest energy and exergy losses occur in brine reinjection unit. The variation of the plant energy efficiency is found between 6% and 12%. Exergy efficiency values change between 35 and 49%. The annual average energy and exergy efficiencies are found as 9.47% and 45.2%, respectively.

1. Introduction

Geothermal energy appears to be an attractive energy source due to fluctuating oil prices and increasing environmental pollution concerns, including global warming. Since the price of oil has reached its peak and efforts are necessary to find alternative energy resources, the use of geothermal energy is found to be more competitive in comparison to the conventional fossil fuel systems and the direct use of geothermal energy has increased approximately twofold in the last five years [1,2].

Three major types of power plants are widely operated: dry-steam plants, flash-steam plants and binary-cycle plants where the binary and combined flash/binary plants are relatively recent designs. Geothermal energy is used to generate electricity and it finds direct use in areas such as space heating and cooling, industrial processes, and greenhouse heating. High-temperature geothermal resources above 150 °C are generally used for power generation. Geothermal resources that possess moderate temperatures (between 90 and 150 °C) and lower-temperatures (below 90 °C) are best suited for direct use applications [2,3]. Researchers mainly focus on two research areas regarding this issue, which can be expressed as follows: (a) economic evaluation of geothermal power generation [4] and (b) optimum design criteria and suitable working fluids for power cycles [5–13].

Geothermal energy based electricity production for Turkey achieves 100 MWe with six running plants as the first half of 2010. Geothermal power production capacity has increased four-fold within the past four years [14,15]. Table 1 shows existing power plants in Turkey [14,15]. In literature, there have only been a few studies on energetic and exergetic performance parameters for geothermal systems. Specific exergy index, fuel depletion ratio, relative irreversibility, productivity lack and exergetic factor have been introduced [16–19] and applied to geothermal district heating [16,18,20–24]. Lee [17] has proposed the parameter of specific exergy index for some degree of classification and evaluation of geothermal resources using their exergy. Fuel depletion ratio, relative irreversibility, productivity lack and exergetic factor are defined by Xiang et al. [18] for the thermodynamic analysis of some systems. Coskun et al. [22,25] have introduced some system related renewable energy and exergy parameters, namely energetic renewability ratio, exergetic renewability ratio, energetic reinjection ratio, and exergetic reinjection ratio, total exergy destruction ratio, component exergy destruction ratio and dimensionless exergy destruction parameter for geothermal systems. In this study, thermodynamic analysis and performance investigation of Tuzla binary geothermal power plant located in Canakkale, Turkey, are performed using actual plant data. The originality of this study is the first study on investigation of eight energetic-exergetic performance parameters namely; total exergy destruction ratio, component exergy destruction ratio, dimensionless exergy destruction, energetic renewability ratio, exergetic renewability ratio, energetic reinjection ratio, exergetic reinjection ratio and

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improvement potential for a geothermal power plant. Also, an application of considering the various outdoor temperature distributions in the exergy calculations is presented for the first time in this study.

2. System description

The geothermal power plant analyzed in this study is a binary designed plant that generates a capacity of 7.5 MWe gross power. The full power production began after the tests in the month of February 2010. The plant operates in a closed loop with no environmental discharge (100% reinjection). The power plant operates on a liquid dominated resource at 175 °C. It utilizes dry-air condensers to condense the working fluid. The geothermal field includes two production wells (T-9, T-16) and two reinjection wells (T-10, T-15). The plant uses isopentane as the working fluid, circulates in a closed cycle. The schematic demonstration and picture of the plant are given in Figs. 1 and 2.

3. Thermodynamic analysis

3.1. Assumptions

The effects of salts and non-condensable gases in the geothermal brine are neglected for calculations. A thermal and physical property of the geothermal water is considered as water in the analyses. EES (Engineering Equation Solver) software program is utilized for the determination of the thermodynamic properties of the geothermal water and isopentane. This program is commonly utilized for the determination of many material thermodynamic properties.

3.2. Analysis

The mass balance for any control volume at steady state can be expressed by

$$\sum_{i=1}^{n} \dot{m}_{in.} = \sum_{i=1}^{n} \dot{m}_{out.} \tag{1}$$

where $\dot{m}$ indicates the mass flow rate. The subscripts in and out indicate the inlet and outlet.

3.2.1. Energy analysis

The energy rate can be expressed by

$$\dot{E}_i = \dot{m}_i \cdot (h_i - h_0) \tag{2}$$

where h indicates enthalpy. Net plant energy efficiency can be described by using the given equation

$$\eta_{sys.} = \frac{\dot{E}_{net}}{\dot{E}_{in}} \tag{3}$$

where, $\dot{E}_{in}$ and $\dot{E}_{net}$ are input and net output energy rates and can be found through

$$\dot{E}_{\text{net}} = \dot{W}_{\text{Turb.}} - \dot{W}_{\text{parasitic load}} \tag{4}$$

$$\dot{E}_{in} = \dot{E}_1 + \dot{E}_6 \tag{5}$$

Here, $\dot{W}_{\text{Turb.}}$ is the electricity production from isopentane cycle. $\dot{W}_{\text{parasitc load.}}$ represents parasitic load. In all plants, there are electrical loads such as pumps fans and controls which are necessary to operate the facility. Often these loads are referred to as "parasitic loads". Air-cooled condenser unit has a great effect on parasitic load and occurs about 60–75% of parasitic loads for investigated system.

The energy efficiency of the isopentane cycle can be described as

$$\eta_{iso.\ cyc.} = \frac{\dot{W}_{\text{Turb.}}}{(\dot{E}_{11} + \dot{E}_{12}) - \dot{E}_{14}} \tag{6}$$

The energy loss for preheater-I and II can be found by

$$\dot{E}_{loss,\ \text{Pre-I}} = (\dot{E}_{18} + \dot{E}_{21}) - (\dot{E}_{20} + \dot{E}_{17}) \tag{7}$$

$$\dot{E}_{loss,\ \text{Pre-II}} = (\dot{E}_{13} + \dot{E}_{17}) - (\dot{E}_{14} + \dot{E}_{16}) \tag{8}$$

The energy efficiency of the pre-heaters can be written as

$$\eta_{\text{Pre-I}} = \frac{\dot{E}_{17} - \dot{E}_{18}}{\dot{E}_{21} - \dot{E}_{20}} \tag{9}$$

$$\eta_{\text{Pre-II}} = \frac{\dot{E}_{16} - \dot{E}_{17}}{\dot{E}_{13} - \dot{E}_{14}} \tag{10}$$

The energy loss and energy efficiency for the turbine become

$$\dot{E}_{\text{loss, Turb.}} = (\dot{E}_{22} - \dot{E}_{21}) - \dot{W}_{\text{Turb.}} \tag{11}$$

$$\eta_{\text{Turb.}} = \frac{\dot{W}_{\text{Turb.}}}{\dot{E}_{22} - \dot{E}_{21}} \tag{12}$$

The energy loss and energy efficiency for the vaporizer are:

$$\dot{E}_{dl.,\ Vap.} = (\dot{E}_{11} + \dot{E}_{12} + \dot{E}_{16}) - (\dot{E}_{13} + \dot{E}_{22}) \tag{13}$$

$$\eta_{Vap.} = \frac{(\dot{E}_{22} - \dot{E}_{16})}{(\dot{E}_{11} + \dot{E}_{12} - \dot{E}_{13})} \tag{14}$$

The energy loss and energy efficiency for two separators (includes expansion valves) are

$$\dot{E}_{lose,\ Sep.} = (\dot{E}_1 + \dot{E}_6) - (\dot{E}_3 + \dot{E}_5 + \dot{E}_8 + \dot{E}_{10}) \tag{15}$$

$$\eta_{Sep.} = \frac{(\dot{E}_3 + \dot{E}_5 + \dot{E}_8 + \dot{E}_{10})}{(\dot{E}_1 + \dot{E}_6)} \tag{16}$$

The energy loss and the energy efficiency for the pumps become

$$\dot{E}_{lose,\ \text{Pump}} = \dot{W}_{\text{Pump}} - (\dot{E}_{out} - \dot{E}_{in}) \tag{17}$$

$$\eta_{\text{Pump}} = \frac{\dot{E}_{out} - \dot{E}_{in}}{\dot{W}_{\text{Pump}}} \tag{18}$$

The energy losses for condenser and the brine reinjection unit are

$$\dot{E}_{lose,\ \text{Cond.}} = \dot{E}_{20} - \dot{E}_{19} \tag{19}$$

$$\dot{E}_{lose,\ \text{Re}\ in.} = \dot{E}_{15} \tag{20}$$

3.2.2. Exergy analysis

The specific flow exergy (ψ) is given by

$$\psi = (h - h_0) - T_0(s - s_0) \tag{21}$$