THERMODYNAMIC AND ECONOMIC ANALYSIS OF GEOTHERMAL ENERGY POWERED KALINA CYCLE

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(Geliş Tarihi: 19.03.2020, Kabul Tarihi: 08.10.2020)

Abstract: In this study, thermodynamic and economic analysis have been carried out to the determination of optimum design parameters of Kalina Cycle. The optimization of four key parameters (turbine inlet pressure, geothermal water outlet temperature at evaporator, condenser pressure and ammonia mass fraction) is also conducted. The thermodynamic properties of the medium temperature geothermal resource in the Simav region are used in the system designs. The energy efficiency and exergy efficiency of the system are evaluated through the thermodynamic analysis. Also, the system has been investigated economically with the net present value method. As a result of the exergy analysis, it is determined that the maximum exergy destruction occurs in the evaporator within the total exergy destruction of the system. In the system design with 90 % ammonia mass fraction, the exergy destruction in the evaporator constitutes 66.5 % of the total exergy destruction in the system. The geothermal water outlet temperature at evaporator, ammonia mass fraction, turbine inlet pressure and condenser pressure of the most effective geothermal energy powered Kalina Cycle are determined as 353.15 K, 90 %, 4808 kPa and 700 kPa, respectively. The energy efficiency and exergy efficiency of this system are calculated as 13.04 % and 51.81 %, respectively. Also, the net present value of this system is calculated as 119.377 Million US$ and it is seen that it is suitable for investment in economic terms.

Keywords: Kalina cycle, Geothermal energy, Net present value, Energy, Exergy.

JEOTERMAL ENERJİ KAYNAKLı KALİNA ÇEVİRİMİNİN TERMODİNAMİK VE EKONOMİK ANALİZİ

Özet: Bu çalışmada, jeotermal enerjisiyle çalışan Kalina Çevrime'nin optimum tasarım parametrelerinin belirlenmesi için termodinamik ve ekonominik analizler yapılmıştır. Türbin giriş basancı, evaporatördede jeotermal akışkan çıkış sıcaklığı, kondanser basınç ve amonyak kütle oranı sistemde değişken parametrelerdir. Simav bölgesindeki orta sıcaklıklı jeotermal kaynagın termodinamik özellikleri sistem tasarımında kullanılmıştır. Sistemin enerji ve ekserji verimleri termodinamik analizler ile değerlendirilmiştir. Ayrıca, sistem net bugünkü değer yöntemi ile ekonomik olarak incelenmiştir. Ekserji analizi sonucunda, sistemin toplam exerji yıkımı içerisinde maksimum exerji yıkımının evaporatörde meydana geldiği tespit edilmiştir. Kütle % 90 amonyak bileşenli sistem tasarımında, evaporatördekide enerji yıkımı, sistemdeki toplam enerji yükünü % 66.5'ini oluşturur. Ekserji analizleri sonucunda en yüksek ekserji yükünün evaporatörde oluştuğu ve % 90 amonyak bileşenli sistem tasarımında, evaporatördekide enerji yükü, toplam sistemdeki enerji yükünün % 66.5'ini oluşturur. En etkin sistem tasarımının enerji verimliliği ve ekserji verimliliği sırasıyla % 13.04 ve % 51.81 olarak belirlenmiştir. Optimum sisteme ait evaporatördeki jeotermal su çıkış sıcaklığı, amonyak kütle oranı, türbin giriş basancı ve kondanser basınç sırasıyla 353.15 K, % 90, 4808 kPa ve 700 kPa olarak belirlenmiştir. Ayrıca bu sistemin net bugünküsü değeri 119.377 Milyon ABDS olarak hesaplanmış ve ekonomik açıdan yatırıma uygun olduğu görülmüştür.

Keywords: Kalina çevrimi, Jeotermal enerji, Net bugünkücü değer, Enerji, Ekserji.

NOMENCLATURE

| Symbol | Variable | Unit |
|--------|----------|------|
| C      | Cost     | $    |
| c      | Specific heat | [kJ/ kg·K] |
| Dtubeb | Diameter of the inlet pipe | [m] |
| e0    | Molar exergy | [kJ/mol] |
| Ex    | Exergy    | [kW] |
| F     | Cost Factor |       |
| h     | Specific enthalpy | [kJ/kg] |
| m     | Mass flow | [kg/s] |
| M     | Molar mass | [kg/mol] |
| NPV   | Net present value |       |
| Q     | Heat energy | [kW] |
| Qs    | Volume flow rate of separator | [m³/s] |
| T     | Temperature | [K] |
| U     | Heat transfer coefficient | [W/m²·K] |

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https://doi.org/10.47480/issbted.817063
u_s: Terminal velocity of separator [m/ s]
V: Total volume of separator [m^3]
\dot{W}: Power [kW]
\rho: Density [m^3/ kg]
\varepsilon: Exergy efficiency [%]
\alpha: Ammonia mass fraction [%]
\psi: Specific exergy [kJ/kg]
\eta: Energy efficiency [%]

Subscripts
b: Benefit
BM: Bare module
c: Condenser
ch: Chemical
cw: Cooling water
elec: Electricity
eva: Evaporator
g: Generator
gf: Geothermal fluid
i: Interest rate
ic: Investment cost
j: Discount rate
l: Liquid
M: Material
m, i: Inlet mass flow
m, o: Outlet mass flow
moc: Maintenance and operating
ol: Life time of system
ncf: Net cash flow
p: Pump
P: Pressure
ph: Physical
r: Recuperator
sc: Salvage cost
sep: Separator
sys: System
v: Vapor
wf: Working fluid
t: Time (year)
tr: Turbine
0: Dead state

Abbreviations
CEPCI: Chemical Engineering Plant Cost Index
GEPKC: Geothermal Energy Powered Kalina Cycle
OFC: Organic Flash Cycle
ORC: Organic Rankine Cycle

INTRODUCTION

Due to the increase in energy demand and the decrease in fossil fuel reserves and pollution of the environment, research on power generation from renewable energy sources and increasing the energy efficiency of these systems gained importance (Arslan, 2010; Arslan, 2011; Deepak et al., 2014; Yari et al., 2015; Zare and Moalemian, 2017; Acar and Arslan, 2019). Geothermal energy is one of the most preferred renewable energy sources in terms of sustainability. It is also important because of the absence of environmental pollutants in the Kalina cycle due to the re-injection of geothermal fluid. Also, developing countries tend to use renewable energy sources to ensure energy diversity and energy security. Kalina Cycle (KC) is one of the low-temperature power cycles in which is using ammonia and water mixture as working fluid (Arslan, 2010; Arslan, 2011; Kalina, 1984; Saffari et al., 2016; Igobo and Davies, 2016).

In literature, different configurations of KC and operating parameters of cycles were investigated according to the thermodynamic and economic analysis. In the studies on KC, the ammonia mass fraction of ammonia-water mixture was changed between 50-90 % (Singh and Kaushik, 2013; Modi and Haglind, 2015; Sadeghi et al., 2015). Arslan (2010) investigated optimum geothermal water outlet temperature at evaporator and ammonia mass fraction of Kalina cycle system (KCS-34) according to the exergoeconomic analysis. The results show that energy efficiency of 14.9 % and exergy efficiency of 36.2 % can be achieved for optimum system design with an ammonia mass fraction of 90 %. Arslan (2011) used artificial neural network for optimization of geothermal energy powered KC and determined that the optimum ammonia mass fraction ranges from 80 % to 90 % for KCS-34. Sun et al. (2014) investigated the performance and optimum parameters of solar driven KC. They determined that the maximum annual power generation of system was 553520 kWh and the energy and exergy efficiencies of the system were 64.8 % and 35.6 %, respectively. Zare and Ashouri et al. (2015) compared the performance of fuel and solar powered KC. They found that the levelized cost of electricity of solar KC higher than fuel driven KC. Yari et al. (2015) compared the low-grade heat source powered trilateral Rankine Cycle, Organic Rankine Cycle (ORC) and KC from the viewpoint of exergoeconomic. Moalemian (2017) analyzed the parabolic solar collector integrated KC in the point view of energy, exergy and economic analysis. Saffari et al. (2016) used artificial bee colony algorithm according to thermodynamical analysis of Husavik power plant to determine the optimum operating conditions of KC. The optimum value of exergy and thermal efficiencies were determined as 20.26 % and 48.18 %, respectively.

Wang and Yu (2016) investigated the performance of a composition-adjustable KC. They indicated that the thermal efficiency of the composition-adjustable KC higher than conventional KC. Wang et al. (2017) investigated the efficiency improvement of a KC by sliding condensation pressure theoretically and numerically. They mentioned that the condenser pressure has significant effects on the system performance. The maximum energy and exergy efficiencies of the optimum design were determined as 10.48 % and 48.10 %, respectively. Rodríguez et al. (2013) compared the geothermal energy powered ORC and KC according to thermodynamical and economic analysis. They mentioned that R-290 for ORC and the ammonia-water mixture with 84 % ammonia mass fraction for KC are the most effective working fluids in the point view of
economic analysis. Varma and Srinivas (2017) compared the performance of ORC, Organic flash cycle (OFC) and KC for low temperature heat recovery. They stated that OFC had been generated maximum power. Zhang et al. (2012) indicated that KC more effective than ORC in the point view of thermodynamic analysis in their review research. Li et al. (2013) compared the performance of KC and e-KC in which the ejector was replaced with the throttle valve. They mentioned that the e-KC more effective than KC. Eller et al. (2017) analyzed the pressure, heat exchanger capacity and power output of ORC and KC for 473.25 K, 573.25 K and 473.25 K heat source temperature. They mentioned that the grassroot cost of ammonia-water KC was the lowest in the investigated source temperature range. The grassroot costs of KC were changed between 1203.4 €/kW and 619.4 €/kW. Cao et al. (2018) optimized the Kalina-Flash cycles by genetic algorithm. They determined that the Kalina-Flash cycles more effective than KC from thermodynamic and economic point of view. Nasruddin et al. (2009) reported that the optimum ammonia mass fraction was 78 % for geothermal powered KC according to thermodynamical analysis. Mergner and Weimer (2015) mentioned that the KSG-1 was more effective than the KCS-34 for geothermal power generation. He et al. (2014) investigated the performance of two modified KCS-11 cycle and KCS-11. Also, they determined the effects of two different key parameters (ammonia mass fraction and cooling water temperature) on the performance of the systems. The energy efficiency, power output and exergy efficiency of the systems were increased with the decrease of the cooling water temperature. According to the results, the maximum cycle efficiency of KCS-11 yielded with at ammonia mass fraction of 92 % for 3000 kPa turbine inlet pressure of working fluid. The energy and exergy efficiencies of this system were determined as 10.2 % and 50.6 %, respectively. Singh and Kaushik (2013) parametrically examined the performance of exhaust gas powered KC. They reported that increasing the turbine inlet pressure increase the maximum cycle efficiency further corresponding to a much higher ammonia mass fraction and the best cycle performance is determined with the 80 % of ammonia mass fraction for the turbine inlet pressure of around 4000 kPa. Prananto et al. (2018) investigated the performance of KC which generates electrical power from the brine discharged from the geothermal fluid at the Wayang Windu geothermal power plant. They determined the power generation and energy efficiency of the system as 1600 kW and 13.2 %, respectively.

In this study, thermodynamic and economic analysis have been carried out to determine the optimum design parameters of the Kalina Cycle operating with a medium temperature geothermal resource. KSG-1 (Siemens’ Kalina cycle system) Kalina Cycle is used for modeling of the system. The outlet temperature of the geothermal fluid from the evaporator, ammonia mass fraction, turbine inlet pressure and condenser pressure are variable parameters of the system. Geothermal energy Powered Kalina Cycle (GEPKC) is evaluated by using net present value (NPV) method from the economical point of view for different system designs. The investment costs of equipment of GEPKC are calculated by using Module Costing Technique. Also, the energy and exergy efficiencies of the GEPKC designs are calculated. The exergy destructions and the investment costs of the system equipment are determined.

GEOTHERMAL ENERGY POWERED KALINA CYCLE

The geothermal energy heat source temperature, cooling water temperature and thermodynamical properties of ammonia-water mixture are the design parameters of the GEPKC. In the GEPKC designs, the thermodynamical properties of the geothermal source in the Simav region are used (Arslan, 2008; Arslan, 2010; Arslan, 2011). The geothermal fluid is supplied from 9 wells. The temperature of the geothermal fluid is 406.65 K and its mass flow rate is 462 kg/s (Arslan, 2008; Arslan, 2010; Arslan, 2011). The vapor fraction of the geothermal fluid has been assumed as 10 %. GEPKC flow diagram is given in Figure 1.

![Figure 1. GEPKC flow diagram](image)
As seen in Figure 1, the saturated liquid phased working fluid enters the pump (1) and outlets the pump (2) at evaporator pressure. The high-pressured working fluid enters the recuperator (3) and takes the heat of the mixer and enters the recuperator (9). The geothermal fluid (2a) is transferred the heat to the preheated working fluid (4) at the evaporator and re-injected to the well (2b). The working fluid is separated into two different flows at the separator. One is high ammonia concentration (strong solution) and vapor phased stream (5) and the other is low ammonia concentration (weak solution) and liquid phased stream (6). Superheated strong solution flow is expanded at the turbine (point 7) to produce power. At the same time, weak solution flows pass through the valve to expand the condenser pressure (8). And both weak solution and strong solution flows are mixed in the mixer and enters the recuperator (9). And it gives the heat to the high-pressured working fluid (2-3). The low-pressured ammonia-water mixture enters to the air-cooled condenser (10) and leaves as saturated liquid (1). The properties of the GEPKC equipment and parameters are given in Table 1.

| Component       | Parameter                  | Values   |
|-----------------|----------------------------|----------|
| Geothermal fluid| Inlet temperature $T_{2a}$ | 406.65 K |
|                 | Outlet temperature $T_{2b}$| 353.15 K, 363.15 K, 373.15 K |
|                 | Mass flow rate $\dot{m}_{2a}$ | 462 kg/s |
| Evaporator      | Outlet temperature of mixture $T_{e}$ | 398.15 K |
| Condenser       | Inlet temperature $T_{3a}$ | 288.15 K |
|                 | Outlet temperature $T_{3b}$ | 298.15 K |
|                 | Pressure $P_{r}$ | 4 kPa |
|                 | Efficiency $\eta_{c}$ | 0.85 |
| Generator       | Efficiency $\eta_{g}$ | 0.95 |
| Pump            | Isentropic efficiency $\eta_{p}$ | 0.8 |
| Recuperator     | Efficiency $\eta_{r}$ | 0.85 |
| Turbine         | Isentropic efficiency $\eta_{t}$ | 0.85 |

The GEPKC is designed for 90, 85, 80, 75, 70 % of ammonia mass fraction ($a$). The thermodynamic properties of the ammonia-water mixture are determined by Reference Fluid Thermodynamic and Transport Properties Database 10.1 (REFPROP) (Lemmon et al., 2018).

**THERMODYNAMIC ANALYSIS**

The governing energy equations of the GEPKC were obtained as follows. The heat transfer in the evaporator is calculated as;

$$\dot{Q}_{eva} = \dot{Q}_{gf} = \dot{m}_{gf} \cdot (h_{2a} - h_{2b}) = \left(\dot{m}_{4} \cdot (h_{4} - h_{3})\right) / \eta_{eva} \quad (1)$$

The energy and mass equations of separator are given as;

$$\dot{m}_{4} \cdot a_{4} = \dot{m}_{5} \cdot a_{5} + \dot{m}_{6} \cdot a_{6} \quad (2)$$

$$\dot{m}_{4} = \dot{m}_{5} + \dot{m}_{6} \quad (3)$$

$$\dot{m}_{4} \cdot h_{4} = \dot{m}_{5} \cdot h_{5} + \dot{m}_{6} \cdot h_{6} \quad (4)$$

The turbine power output can be calculated as;

$$\dot{W}_{tr} = \dot{m}_{5} \cdot (h_{5} - h_{7}) = \dot{m}_{5} \cdot (h_{5} - h_{7s}) \cdot \eta_{tr} \quad (5)$$

Here, $s$ indicates the isentropic process. The generator electric power can be calculated as;

$$\dot{W}_{g} = \eta_{g} \cdot \dot{W}_{tr} \quad (6)$$

The expansion through the control valve is an isenthalpic process.

$$h_{6} = h_{8} \quad (7)$$

The energy and mass equations of mixer are given as;

$$\dot{m}_{9} \cdot a_{9} = \dot{m}_{7} \cdot a_{7} + \dot{m}_{8} \cdot a_{8} \quad (8)$$

$$\dot{m}_{9} = \dot{m}_{7} + \dot{m}_{8} \quad (9)$$

$$\dot{m}_{9} \cdot h_{9} = \dot{m}_{7} \cdot h_{7} + \dot{m}_{8} \cdot h_{8} \quad (10)$$

The heat transfer in recuperator is given as;

$$\dot{Q}_{r} = \dot{m}_{9} \cdot (h_{9} - h_{10}) = \left(\dot{m}_{2} \cdot (h_{2} - h_{1})\right) / \eta_{r} \quad (11)$$

The heat transfer in the condenser can be calculated by as;

$$\dot{Q}_{c} = \dot{m}_{10} \cdot (h_{10} - h_{3}) = \left(\dot{m}_{cw} \cdot (h_{3b} - h_{3a})\right) / \eta_{c} \quad (12)$$

The pump power input can be calculated as;

$$W_{p} = \dot{m}_{3} \cdot (h_{3} - h_{2}) = \dot{m}_{3} \cdot (h_{3s} - h_{2}) / \eta_{p} \quad (13)$$

The net power output of the GEPKC is;

$$\dot{W}_{net} = \dot{W}_{g} - W_{p} \quad (14)$$

The energy efficiency of the GEPKC is calculated by;
\[ \eta = \frac{\dot{W}_{\text{net}}}{Q_{\text{gf}}} \]  \hspace{1cm} (15)

The exergy balance equation for steady systems is:

\[ \dot{E}_{x,\text{heat}} - \dot{E}_{x,\text{work}} + \dot{E}_{x,m,i} - \dot{E}_{x,m,o} = \dot{E}_{x,\text{dest}} \]  \hspace{1cm} (16)

The exergy term of heat is calculated with:

\[ \dot{E}_{x,\text{heat}} = \sum \left( 1 - \frac{T_s}{T_i} \right) \cdot \dot{Q}_c \]  \hspace{1cm} (17)

The exergy term of work is given as:

\[ \dot{E}_{x,\text{work}} = \dot{W} \]  \hspace{1cm} (18)

The mass flow exergy terms are given as:

\[ \dot{E}_{x,m,i} = \sum m_i \cdot \psi_i \]  \hspace{1cm} (19)

\[ \dot{E}_{x,m,out} = \sum m_{out} \cdot \psi_{out} \]  \hspace{1cm} (20)

where \( \psi \) is the physical and chemical exergy terms.

\[ \psi = \psi_{ph} + \psi_{ch} \]  \hspace{1cm} (21)

The physical exergy term is given as:

\[ \psi_{ph} = (h - h_0) - T_0 \cdot (s - s_0) \]  \hspace{1cm} (22)

where \( h \) is enthalpy, \( s \) is entropy, and the subscript zero indicates properties of fluids at the dead state. The reference state is 298.15 K and 101.325 kPa. The chemical exergy term given as:

\[ \psi_{ch} = \frac{a}{M_{\text{NH}_3}} \cdot e_{\text{ch,NH}_3}^0 - \frac{(1-a)}{M_{H_2O}} \cdot e_{\text{ch,H}_2O}^0 \]  \hspace{1cm} (23)

where \( e_{\text{ch,NH}_3} \) and \( e_{\text{ch,H}_2O} \) are molar exergy of the pure component at dead state conditions (kJ/mol), \( M \) is the molar mass (kg/mol), \( a \) is the mass ratio of ammonia in the mixture (Bejan et al., 1996).

The exergetic efficiency of system is then calculated by the following equation:

\[ \varepsilon = 1 - \frac{\dot{E}_{x,total}}{\dot{E}_{x,m,i}} = \frac{\dot{W}_{\text{net}}}{(m_{gf}(\psi_{2a} - \psi_{2b}))} \]  \hspace{1cm} (24)

**ECONOMIC ANALYSIS**

The life cycle cost (\( C_{\text{sys}} \)) of GEPKC can be determined as:

\[ C_{\text{sys}} = C_b - (C_{ic} + C_{sc} + C_{moc}) \]  \hspace{1cm} (25)

where, \( C_{ic} \); the investment costs ($), \( C_{sc} \); salvage cost ($), \( C_{moc} \); maintenance and operating costs ($) and \( C_b \); benefit ($) of the GEPKC. The \( C_{sc} \) of GEPKC was taken as 10% of the \( C_{ic} \) (Acar and Arslan, 2017).

\[ C_{sc} = C_{ic} \cdot 0.10 \]  \hspace{1cm} (26)

The \( C_{moc} \) of GEPKC system was taken as 6% of the \( C_{ic} \) of the GEPKC (Ashouri et al., 2015).

\[ C_{moc} = C_{ic} \cdot 0.06 \]  \hspace{1cm} (27)

The benefit of GEPKC includes electricity earning.

\[ C_b = \dot{W}_{\text{net}} \cdot C_{\text{elec}} \cdot t_o \]  \hspace{1cm} (28)

where \( C_{\text{elec}} \); the unit price of electricity ($/kWh) and \( t_o \); operating time of plant is 8400 h per annum (Arslan, 2010). \( C_{\text{elec}} \) is calculated by:

\[ C_{\text{elec}} = \frac{CEPCI_{2018}}{CEPCI_{2014}} \cdot C_{\text{elec,2014}} \]  \hspace{1cm} (29)

where \( C_{\text{elec,2014}} \); the unit price of electricity in 2014 is 0.06 $/kWh (Aminyavari et al., 2014), \( CEPCI_{2018} \); Chemical Engineering Plant Cost Index in 2018 is 603.1 (CEPCI, 2018) and \( CEPCI_{2014} \); Chemical Engineering Plant Cost Index in 2014 is 576.1 (Cao et al., 2018). The net cash flow;

\[ C_{\text{ncf}} = (C_b - C_{moc}) \cdot (1 + i)^{t-1} \]  \hspace{1cm} (30)

here, \( i \); the interest rate and \( t \); the related year time of cash flow. The \( NPV \) of GEPKC;

\[ NPV = (C_{sc} - C_{ic}) + \sum_{t=1}^{\text{lif}} \frac{C_{\text{ncf}}}{(1+i)^{t}} \]  \hspace{1cm} (31)

where \( \text{lif} \); the lifetime of GEPKC, \( j \); the discount rate. In this study, the lifetime of GEPKC system was added to calculations as 20 years. The discount and interest rates were taken as 18.5 % and 19.5 %, respectively (CBRT, 2019).

The investment cost of GEPKC is calculated by using Module Costing Technique (Cao et al., 2018; Turton et al., 2018). The data used to calculate the purchase costs of equipment were obtained from the literature in 2001 (Turton et al., 2018). The equipment costs were modified for the year 2018 by CEPCI. The equipment costs in 2018 can be calculated by;

\[ C_{eq} = \frac{CEPCI_{2018}}{CEPCI_{2001}} \cdot F_{BM} \cdot C^0 \]  \hspace{1cm} (32)

here, \( F_{BM} \); the bare module cost factor, \( C^0 \); the purchase cost of equipment and \( CEPCI_{2001,\text{Chemical Engineering Plant Cost Index in 2001 is 397}} \) (Cao et al., 2018). The purchase cost of equipment is calculated by (Turton et al., 2018);

\[ \log C^0 = K_1 + K_2 \cdot \log X + K_3 \cdot (\log X)^2 \]  \hspace{1cm} (33)

here, \( K \); constants are determined depending on the equipment and \( X \); the parameter which is related to the equipment. These parameters are the total volume for separator, total heat transfer area for the evaporator,
condenser and recuperator, power consumption for pump and power output of the turbine.

The bare module cost factors can be calculated by (Turton et al., 2018):

$$ F_{BM} = B_1 + B_2 \cdot F_M \cdot F_p $$  \hspace{1cm} (34)

here, $B_i$; the constants based on equipment types, $F_M$; material factor and $F_p$; pressure factor. The pressure factor of the pump can be calculated by (Turton et al., 2018);

$$ logP_p = C_1 + C_2 \cdot logP_p + C_3 \cdot (logP_p)^2 $$  \hspace{1cm} (35)

here, $P_p$ is the design pressure of the pump. The constants of the cost equations according to the equipment’s are given in Table 2.

The heat transfer areas of the heat exchangers are calculated by using the logarithmic mean temperature difference (LMTD) method (Ashouri et al., 2015). Heat transfer coefficient ($U$) values of the equipment are given in Table 3.

Table 2. The constants of the cost equations (Turton et al., 2018).

| Equipment     | $K_1$ | $K_2$ | $K_3$ | $B_1$ | $B_2$ | $F_M$ | $F_{BM}$ | $F_p$ | $c_1$ | $c_2$ | $c_3$ |
|---------------|-------|-------|-------|-------|-------|-------|----------|-------|-------|-------|-------|
| Evaporator    | 4.6656| -0.1557| 0.1547| 0.9600| 1.210 | 2.450 | -        | 1.0000| 0.0000| 0.0000| 0.0000|
| Separator     | 3.4974| 0.4483 | 0.1074| 2.2500| 1.820 | 3.200 | -        | 0.0000| 0.0000| 0.0000| 0.0000|
| Recuperator   | 4.6656| -0.1557| 0.1547| 0.9600| 1.210 | 2.450 | -        | 1.0000| 0.0000| 0.0000| 0.0000|
| Condenser     | 4.6420| 0.3698 | 0.0025| -      | -     | 3.000 | 0.0000   | 0.0000| 0.0000| 0.0000| 0.0000|
| Turbine       | 2.6259| 1.4398 | -0.1776| -     | -     | 11.600| -        | -     | -     | -     | -     |
| Pump          | 3.3892| 0.0536 | 0.1538| 1.8900| 1.350 | 2.200 | -        | -0.3935| 0.3957| -0.0023|       |

The investment cost of each component is considered negligible.

Table 3. Heat transfer coefficient ($U$) values of the equipment (Ashouri et al., 2015; Rodríguez et al., 2013).

| Equipment | $U$ (W/m²·K) |
|-----------|--------------|
| Recuperator | 1000         |
| Evaporator   | 900          |
| Condenser     | 1100         |

The total volume of the separator can be calculated with (Cao et al., 2018; Zarrouka and Purnanto, 2015);

$$ V_{sep} = \frac{\pi(3\cdot D_{tube})^2}{4} \cdot (7 \cdot D_{tube} + 4 \cdot D_{tube}) $$  \hspace{1cm} (36)

here $D_{tube}$; the diameter of the inlet pipe (m) and it is given by (Cao et al., 2018);

$$ D_{tube} = \left(\frac{Q_{VS}/u_t}{\pi}\right)^{0.5} $$  \hspace{1cm} (37)

here $Q_{VS}$; the volume flow rate of inlet flow of separator (m³/s) and $u_t$; the terminal velocity of separator (m/s). The terminal velocity of vertical cyclone type separator is given by (Cao et al., 2018; Zarrouka and Purnanto, 2015);

$$ u_t = Z \cdot \left(\frac{\rho_s - \rho_v}{\rho_v}\right)^{0.5} $$  \hspace{1cm} (38)

Here $\rho_s$; liquid density (kg/m³), $\rho_v$; vapor density (kg/m³) and $Z$ is 0.069 (Cao et al., 2018; Zarrouka and Purnanto, 2015).

The variation of energy efficiency, $\eta$ of GEPKC versus $\alpha$ and $P_{in}$.

Figure 2. The variation of energy efficiency, $\eta$ of GEPKC versus $\alpha$ and $P_{in}$.

According to Fig. 2, the $\eta$ values of the GEPKC increase by the decrease of $P_{in}$ and the increase of the ammonia mass fraction. The $\eta$ values of the proposed system range between 5.57 % and 13.04 %. The highest $\eta$ value of GEPKC system is obtained for 700 kPa of turbine outlet pressure and 90 % of ammonia mass fraction at 4808 kPa of turbine inlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator. The $\eta$ values of the system increase with the increase of the ammonia mass fraction and decrease of the condenser pressure. The change of exergy efficiency values of GEPKC system for $P_{in}$=4808 kPa and $T_{in}$=353.15 K
According to different ammonia mass fraction (α) and turbine outlet pressure \( (P_{10}) \) are given in Fig. 3.

**Figure 3.** The variation of exergy efficiency, \( \varepsilon \) of GEPKC versus α and \( P_{10} \).

According to Fig. 3, the \( \varepsilon \) values of the GEPKC increase by the decrease of \( P_{10} \) and the increase of the ammonia mass fraction. The \( \varepsilon \) values of the proposed system range between 8.79 % and 51.81 %. The highest \( \varepsilon \) value of GEPKC system is obtained for 700 kPa of turbine outlet pressure and 90 % of ammonia mass fraction at 4808 kPa of turbine inlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator.

**Figure 4.** The variation of \( NPV \) of GEPKC versus α and \( P_{10} \).

Fig. 4 shows that the \( NPV \) of the GEPKC decrease by the increase of \( P_{10} \) and the decrease of the ammonia mass fraction. The \( NPV \) of the proposed system range between 2.196 Million US$ and 121.446 Million US$. The maximum \( NPV \) of GEPKC system was obtained for 800 kPa of turbine outlet pressure and 90 % of ammonia mass fraction at 4808 kPa of turbine inlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator.

**Figure 5.** The variation of the net power output of GEPKC versus α and \( P_{10} \).

According to Fig. 5, the net power output values of the GEPKC increase by the decrease of \( P_{10} \) and the increase of the ammonia mass fraction. The net power output values of the proposed system range between 11372.672 kW and 26633.930 kW. The highest net power output value of GEPKC system is obtained for 700 kPa of turbine outlet pressure and 90 % of ammonia mass fraction at 4808 kPa of turbine inlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator.

**Figure 6.** The variation of energy efficiency, \( \eta \) of GEPKC versus α and \( P_{4} \).

According to Fig. 6, the \( \eta \) values of the proposed system range between 6.47 % and 13.04 %. The minimum \( \eta \) value of GEPKC system at the same ammonia mass fraction is obtained for the maximum value of turbine inlet pressure. Furthermore, the variation of the \( \eta \) values with the turbine inlet pressure for different ammonia mass fractions show different trends. The highest \( \eta \) value of GEPKC system is obtained for 4808 kPa of turbine inlet pressure and 90 % of ammonia mass fraction at 4808 kPa of turbine inlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator.
inlet pressure and 90 % of ammonia mass fraction at 700 kPa of turbine outlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator. The change of $\epsilon$ values of GEPKC system for $P_{10}=700$ kPa and $T_2=353.15$ K according to different ammonia mass fraction ($\alpha$) and turbine inlet pressure ($P_4$) are given in Fig. 7.

According to Fig. 7, the $\epsilon$ values of the proposed system range between 21.01 % and 51.81 %. The maximum $\epsilon$ value of GE PKC system is obtained for 4808 kPa of turbine inlet pressure and 90 % of ammonia mass fraction at 700 kPa of turbine outlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator. The change of $NPV$ of GE PKC system for $P_{10}=800$ kPa and $T_2=353.15$ K according to different ammonia mass fraction ($\alpha$) and turbine inlet pressure ($P_4$) is given in Fig. 8.

Fig. 8 shows that the $NPV$ of the proposed system ranges between 23.656 Million US$ and 122.529 Million US$. The maximum $NPV$ of GE PKC system is obtained for 800 kPa of turbine outlet pressure and 5308 kPa of turbine inlet pressure. For the same system, the energy and exergy efficiencies are calculated as 12.38 % and 46.71 %, respectively. The change of net power output values of GE PKC system for $P_{10}=700$ kPa and $T_2=353.15$ K according to different ammonia mass fraction ($\alpha$) and turbine inlet pressure ($P_4$) are given in Fig. 9.

According to Fig. 9, the variation of the net power output values of the GE PKC with the turbine inlet pressure for different ammonia mass fractions show the same trends. The net power output values of the proposed system range between 13218.33 kW and 26633.93 kW. The highest net power output value of GE PKC system is obtained as 26633.93 kW for 4808 kPa of turbine inlet pressure and 90 % of ammonia mass fraction at 700 kPa of turbine outlet pressure and 353.15 K of outlet temperature of the geothermal fluid from the evaporator. Handling the operating parameters as $P_{10}=700$ kPa and $P_4=4808$ kPa, the variation of $\epsilon$ of the GE PKC system with different outlet temperature of the geothermal fluid from the evaporator ($T_{2b}$) and ammonia mass fraction ($\alpha$) obtained as seen in Fig. 10.

Fig. 10 shows that the $\epsilon$ values of the GE PKC increase by the decrease of outlet temperature of the geothermal fluid from the evaporator and the increase of the ammonia mass fraction. The $\epsilon$ values of the proposed system range between 19.88 % and 51.81 %. The highest exergy efficiency value of GE PKC system was obtained for 353.15 K of outlet temperature of the geothermal fluid from the evaporator. The $\epsilon$ values of the system increase with the increase of the ammonia mass fraction and outlet temperature of the geothermal fluid from the evaporator.
temperature of the geothermal fluid from the evaporator. Handling the operating parameters as \( P_4 = 4808 \) kPa and \( P_{10} = 700 \) kPa, the variation of \( NPV \) of the GEPKC system with different outlet temperature of the geothermal fluid from the evaporator \( (T_{2b}) \) and ammonia mass fraction \( (\alpha) \) obtained as seen in Fig. 11.

Fig. 11 shows that the \( NPV \) of the GEPKC increase by the decrease of \( T_{2b} \) accepts 70 % of the ammonia mass fraction. The \( NPV \) of the proposed system ranges between 28.142 Million US$ and 119.377 Million US$. The maximum \( NPV \) of GEPKC was obtained for 353.15 K of the outlet temperature of the geothermal fluid from the evaporator and 90 % of the ammonia mass fraction. The change of net power output values of GEPKC for \( P_4 = 4808 \) kPa and \( P_{10} = 700 \) kPa according to different ammonia mass fraction \( (\alpha) \) and turbine outlet pressure \( (T_{2b}) \) are given in Fig. 12.

According to Fig. 12, the net power output values of the GEPKC increase by the decrease of \( T_{2b} \) and the increase of the ammonia mass fraction. The net power output values of the proposed system range between 10702.51 kW and 26633.93 kW. The net power output values of the system increase with the increase of the ammonia mass fraction and decrease of the outlet temperature of the geothermal fluid from the evaporator. Handling the operating parameters as \( P_4 = 4808 \) kPa, \( P_{10} = 700 \) kPa and \( T_{2b} = 353.15 \) K, the variation of exergy destructions of the system equipment with ammonia mass fraction are given in Fig. 13.

As seen in Fig. 13, the highest exergy destruction is determined as 16461.22 kW for the evaporator. The minimum exergy destruction in the evaporator is determined as 15010.84 kW at 75 % of ammonia mass fraction. The exergy destruction of separator and condenser decreases with the increase of the ammonia mass fraction. The exergy destruction of the turbine decreases with the decrease of the ammonia mass fraction. In the system design with 90 % ammonia mass fraction, the exergy destruction in the evaporator accounts for 66.5 % of the total exergy destruction in the system. The maximum exergy destruction occurs in the evaporator within the total exergy destruction of the system. Handling the operating parameters as \( P_4 = 4808 \) kPa, \( P_{10} = 700 \) kPa and \( T_{2b} = 353.15 \) K, the variation of investment costs of the system equipment with ammonia mass fraction are given in Fig. 14.

As seen in Fig. 14, the highest investment cost is determined as 27.9927 Million US$ for the evaporator. The investment cost of the turbine, separator and condenser increase with the increase of the ammonia mass fraction. The investment costs of the evaporator and recuperator decrease with the increase of the ammonia mass fraction. According to the energy efficiency and
Table 4. The optimum GEPKC design parameters.

|   | T (K) | P (kPa) | h (kJ/kg) | s (kJ/kg K) | m (kg/s) | α (%) | E_{xch} (kW) | Ψ (kJ/kg) |
|---|-------|---------|-----------|-------------|----------|-------|--------------|----------|
| 1 | 290.60| 700     | 358.04    | 1.6851      | 151.94   | 90    | 2714097.31   | 196.66   |
| 2 | 291.72| 4808    | 365.91    | 1.6905      | 151.94   | 90    | 2714097.31   | 202.92   |
| 3 | 309.93| 4808    | 449.89    | 1.9697      | 151.94   | 90    | 2714097.31   | 203.65   |
| 4 | 398.15| 4808    | 1592.1    | 5.0858      | 151.94   | 90    | 2714097.31   | 416.80   |
| 5 | 398.15| 4808    | 1789.1    | 5.6081      | 123.68   | 96.41 | 2366346.95   | 458.08   |
| 6 | 398.15| 4808    | 729.96    | 2.7998      | 28.25    | 61.91 | 347650.36    | 178.31   |
| 7 | 308.37| 700     | 1561.81   | 5.7391      | 123.68   | 96.41 | 2366346.95   | 191.73   |
| 8 | 323.04| 700     | 729.96    | 2.9558      | 28.25    | 61.91 | 347650.36    | 189.72   |
| 9 | 314.93| 700     | 1407.1    | 5.2243      | 151.93   | 90    | 2714097.31   | 190.53   |
| 10| 307.21| 700     | 1308.32   | 4.9065      | 151.93   | 90    | 2714097.31   | 186.48   |
| 2a| 406.65| 300     | 777.13    | 2.2030      | 462.00   | -     | -             | 461.33   |
| 2b| 353.15| 300     | 335.21    | 1.0754      | 462.00   | -     | -             | 355.61   |
| 3a| 288.15| 4       | 62.98     | 0.2244      | 3450.31  | -     | -             | 728.59   |
| 3b| 298.15| 4       | 104.83    | 0.3672      | 3450.31  | -     | -             | 727.87   |

\( W_{\text{net}} \) 26633.93 kW

\( \eta \) 13.04 %

\( \varepsilon \) 51.81 %

NPV 119.377 Million US$

exergy efficiency values of GEPKC, the optimum plant design parameters are given in Table 4.

As seen in Table 4, the NPV, energy and exergy efficiency of the most effective GEPKC are determined as 119.377 Million US$, 13.04 % and 51.81 %, respectively. The geothermal water outlet temperature at the evaporator, ammonia mass fraction, turbine inlet pressure and condenser pressure of this system are determined as 353.15 K, 90 %, 4808 kPa and 700 kPa, respectively.

The thermodynamic model of GEPKC is validated comparing the results obtained with published literature results (Arslan, 2010; Wang et al., 2017), as shown in Table 5 and Table 6. The data obtained from Arslan (2010) are used to validate the thermodynamic model according to the thermodynamic properties of geothermal source and cooling water.

As seen in Table 5, the energy efficiency of the KCS-34 system, which has the same geothermal resource characteristics, is 2.5% higher than that of the KCS-1 system. When both systems are evaluated in terms of exergy efficiency, it is seen that the KCS-1 system is 12.6% more efficient than the KCS-34 system. The thermodynamic properties of the key state points well coincide with each other.

Table 5. Model validation with the results of Arslan (2017).

| Parameter                                         | Arslan (2010) | Present study |
|--------------------------------------------------|--------------|--------------|
| Cycle type                                       | KCS-34       | KCS-1        |
| Mass flow rate of geothermal fluid               | 462 kg/s     | 462 kg/s     |
| Geothermal fluid inlet temperature (evaporator)  | 406.65 K     | 406.65 K     |
| Geothermal fluid outlet temperature (evaporator) | 353.15 K     | 353.15 K     |
| Working fluid ammonia mass fraction              | 90 %         | 90 %         |
| Turbine inlet ammonia mass fraction of working fluid | 93 %         | 93.70 %      |
| Turbine inlet temperature of working fluid       | 403 K        | 398.15 K     |
| Turbine inlet pressure of working fluid          | 3000 kPa     | 3308 kPa     |
| Condensing temperature                           | 311.85 K     | 312.43 K     |
| Condensing pressure                              | 714 kPa      | 700 kPa      |
| Cooling water inlet temperature (condenser)      | 288.15 K     | 288.15 K     |
| Cooling water outlet temperature (condenser)     | 298.15 K     | 298.15 K     |
| Mass flow rate of cooling water                  | 3463.6       | 3110 kgs     |
| \( \eta \)                                       | 14.8 %       | 12.3 %       |
| \( \varepsilon \)                                | 36.2 %       | 48.8 %       |
Table 6. Model validation with the results of Wang et al. (2017).

| State | Present study | Wang et al. (2017) |
|-------|---------------|-------------------|
|       | $T$ (K) | $P$ (kPa) | $m$ (kg/s) | $\alpha$ (%) | $T$ (K) | $P$ (kPa) | $m$ (kg/s) | $\alpha$ (%) |
| 1     | 298.79 | 700 | 177.06 | 0.750 | 282.35 | 435 | 32.673 | 0.760 |
| 2     | 299.28 | 2808 | 177.06 | 0.750 | 282.71 | 2392 | 32.673 | 0.760 |
| 3     | 338.20 | 2808 | 177.06 | 0.750 | 321.77 | 2325 | 32.673 | 0.760 |
| 4     | 398.15 | 2808 | 177.06 | 0.750 | 380.45 | 2280 | 32.67 | 0.760 |
| 5     | 398.15 | 2808 | 113.39 | 0.921 | 380.45 | 2280 | 19.914 | 0.964 |
| 6     | 398.15 | 2808 | 58.77 | 0.405 | 380.45 | 2280 | 12.759 | 0.441 |
| 7     | 341.11 | 700 | 113.39 | 0.921 | 312.95 | 447 | 19.914 | 0.964 |
| 8     | 345.36 | 700 | 58.77 | 0.405 | 334.76 | 447 | 12.759 | 0.441 |
| 9     | 343.20 | 700 | 177.06 | 0.750 | 326.77 | 447 | 32.673 | 0.760 |
| 10    | 327.31 | 700 | 177.06 | 0.750 | 308.29 | 443 | 32.673 | 0.760 |
| 2a    | 406.65 | 300 | 462 | $gf$ | 393.15 | 2000 | 142 | $gf$ |
| 2b    | 353.15 | 300 | 462 | $gf$ | 341.78 | 1961 | 142 | $gf$ |
| 3a    | 288.15 | 4 | 3507 | water | 271.85 | 118 | 2259 | air |
| 3b    | 298.15 | 4 | 3507 | water | 283.60 | 113 | 2259 | air |
| $\eta$ | 10.30 % | 10.48 % |
| $\varepsilon$ | 37.22 % | 48.10 % |

As seen in Table 6, the energy efficiency value in Reference (Wang et al., 2017) is 0.18 % higher than the present work. The exergy efficiency of the system in Reference (Wang et al., 2017) is 18 % higher than the exergy efficiency of the GEPKC. The reason of this difference is that the temperature and mass flow rate of the resource is used in this study are higher than the Reference (Wang et al., 2017). Considering the properties of the geothermal resources used in the studies, it is seen that the thermodynamic properties of the state points coincide with each other. It is observed that the pressure values and the ammonia mass fractions of the state points are compatible with the Reference (Wang et al., 2017).

CONCLUSION

In this study, the GEPKC is designed for different geothermal water outlet temperature at evaporator, ammonia mass fraction, turbine inlet pressure and condenser pressure. The net power output, NPV, energy and exergy efficiencies of the GEPKC are determined. The most effective system parameters determined as $P_{\text{ev}}=700$ kPa, $P_{\text{t}}=4808$ kPa, $T_{\text{gf}}=353.15$ K and $\alpha=90$ % according to the energy and exergy efficiencies. The energy and exergy efficiencies of this system are determined as 13.04 % and 51.81 %, respectively. In addition, the NPV value of this system is calculated as 119.377 Million US$ and it is seen that it is suitable for investment in economic terms. The net power output of this system is calculated as 26633.93 kW. Also, the following results were obtained:

- The net power output, energy and exergy efficiencies of the GEPKC increase with the decrease of the condenser pressure.
- The investment cost of the evaporator is found to have a significant effect on the NPV value. The investment cost of the evaporator decreases with the increase of the ammonia mass fraction.
- The maximum exergy destruction is realized in the evaporator.
- The exergy efficiency values of the GEPKC increase by the decrease of outlet temperature of the geothermal fluid from the evaporator, turbine outlet pressure and the increase of the ammonia mass fraction.
- The NPV of the GEPKC increases with the increase of ammonia mass fraction ratio.

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