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Thermoeconomic Analysis and Parametric Study of Geothermal and Biomass Source ORC, Dual Fluid and Hybrid Power Plants

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Abstract: In this paper, we have continued doing a research project which had done previously. Three types of power plants include an Organic Rankine Cycle (ORC), a Dual-Fluid-Hybrid (DFH) and a single-fluid hybrid-fueled (HYB); were re-modeled. After model validation, thermodynamic studies and exergy analysis were extended for the defined cases using the first and second laws of thermodynamics. Then thermoeconomic analysis by three various parametric capital cost functions was conducted to select the best model accurate. Based on new results, from thermodynamics viewpoint, the HYB plant with cyclohexane is the best option, as it had been concluded already, but the new results show that from the economical view, the DFH plant with R245fa and R236fa are suitable choices. Based on the results of the first step, the DFH plant has better overall performance; thus, in the next step, more and variant organic fluids were evaluated as an operating fluid for its low pressure cycle. It is observed that, in the thermodynamic view, toluene is the best option, and,in the economical view,R236faand toluene are suitable for DFH plants. Through a comprehensive survey, R717 is selected as the best operating fluid for DFH plants.

Keywords: Organic Rankine Cycle (ORC), Dual Fluid Hybrid power plant (DFH), Thermodynamics, Exergy, Economic, Thermoeconomics.

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1. Introduction

The Organic Rankine Cycle (ORC) power plant is one of the low temperature plants, and it has four main parts that contained pump, evaporator, turbine and condenser. The energy source of these plants can be provided from the renewable energies. There are many researches about the ORC plants based on renewable energies that some of them are presented in the following.

Liu et al [1] studied a two stage Rankine cycle for power generation. The system was constructed with a water steam Rankine as well as an Organic Rankine bottoming Cycles. They concluded that using an organic working fluid with higher density than water was possible to reduce the installation size and use an air-cooled condenser. They tested nine potential candidates from four different organic fluid families and ammonia in order to search suitable working fluids for their application. They evaluated the performances of the two stages Rankine cycle with different working fluids. Astolfi et al [2] have analyzed a combined concentrating solar power system and a geothermal binary plant based on an organic Rankine cycle. They designed a supercritical ORC for the optimal utilization of an intermediate enthalpy geothermal source. The plant also included a solar parabolic trough field, introducing an additional high temperature heat source for the cycle and increasing power production. They performed a differential economic analysis to determine the cost of the additional electricity generated by the solar source. Tempesti et al [3] analyzed micro combined heat and power plants operating through an organic Rankine cycle using renewable energy. Their reference system was designed to produce 50 kWe. The heat sources of the system were considered to be geothermal energy at low temperature (80–100 ^{o}C) and solar energy. In their study, two different system layouts, a single and a double stage arrangement, were presented, and different working fluids (e.g. R134a, R236fa, and R245fa) were considered. The results of their simulation in terms of efficiencies, heat and electricity production, and the main characteristics of the system were discussed. Zhou et al [4] studied the hybrid solar–geothermal power plants as a means of boosting the power output, and where possible moderating the impact of diurnal temperature change. Their ultimate goal was to explore the potential benefits from the synergies between the solar and geothermal energy sources. So, the performances of the hybrid systems in terms of power output and the cost of electricity were compared with stand-alone solar and geothermal plants. Prando et al [5] in an experimental work assessed the energy performance of a biomass boiler coupled with an organic Rankine cycle generator in order to district heating under real operating conditions and identified its potential improvements. The analysis of the plant showed that the ORC pump, the flue gases extractor, the thermal oil pump and the condensation section fan are the main causes of the electric self-consumption. Calise et al [6] presented a dynamic simulation model of a novel prototype of a 6 kWe solar power plant based on organic Rankine cycle. The model was used to evaluate the energy and economic performance of the solar CHP system under analysis, in different climatic conditions. A sensitivity analysis was also performed, in order to determine the combination of system/design parameters able to maximize the thermo-economic performance of the system. They found that the system may be economically feasible for the majority of locations in the Mediterranean area, whereas the profitability was unsatisfactory for Central-Europe sites. Habka et al [7] analyzed the performance characteristics of the series Combined Heat and Power (CHP) system based on organic Rankine cycle with R134a as a working flow, under influence of the heating plant parameters without considering the chemistry of the geothermal water considered as heat source. The results showed that increasing the heat demand or the return temperature and only the high supply temperatures led to destruct the net power generated by the ORC–CHP system. While, the influence of the last parameters on the total exergy efficiency and losses was different; whereas raising the heat demands optimized these exergetic indicators, the variation of the supply temperature has led to an optimum condition for these performances. Since increasing the return temperature has purely negative impacts on all exergetic and energetic criteria, the latter could be improved by reducing this temperature with attention to the heat transfer capacities. Also Habka et al [8] studied the performances of some zeotropic mixtures in an organic Rankine cycle for evaluating the potential of utilizing the low-temperature and caloric geothermal water. The possible optimization of these applications when using the mixtures opposite pure fluids was the main objective and has been discussed. The results showed that in case of stand-alone ORC, the mixtures: R438A, R422A and R22M were more efficient than the advised pure fluids, and could enhance the power productivity and geothermal water utilization at the sources' temperatures 80, 100 and 120 C, respectively. Eyidogan et al [9] studied technical and economic analysis of organic Rankine cycle systems in Turkey and their application areas were examined in detail. An application in a biomass based plant, with 1 MW of installed capacity, has been given as an example. According to the feasibility analysis, the investment payback period of the ORC application has been calculated as 2.7 years.F iliz Tumen Ozdil et al [10] done an exergoeconomic analysis of an organic Rankine cycle for a local power plant, located in the southern part of Turkey. The capital investment cost, operating and maintenance costs, and total investment cost of ORC steam plant calculated as 7.43 $\$.h^{-1}$, 6.69 $\$.h^{-1}$ and 14.12 $\$.h^{-1}$, respectively. The unit exergy cost and exergy cost of the electricity produced by the turbine found as 11.05 $\$.GJ^{-1}$ and 14.96 $\$.h^{-1}$, respectively. The highest exergoeconomic factor was observed in the pump because of the lowest exergy destruction rate and low total investment cost, while the lowest exergoeconomic factor was observed in the evaporator due to the highest exergy destruction rate in the evaporator. Moreover, payback period assessment has calculated as 3.27 years for the ORC power plant. Eyerer et al [11] analyzed the applicability of the new fluid as drop-in replacement for R245fa in existing systems, and compared system parameters such as cycle efficiency and power output. The test rig used an electric heater as a heat source and a scroll compressor as an expander. It was concluded that R1233zd-E can be used as a substitute for R245fa in the existing ORC systems. Comparing the highest achieved thermal efficiency, R1233zd-E performed 6.92% better than R245fa. However, comparing the maximal gross power output, R245fa performed 12.17% better than R1233zd-E.Song et al [12] proposed a one-dimensional analysis method for an organic Rankine cycle system. The net power output of the ORC system was 534 kW, and the thermal efficiency reached to 13.5%. The results showed that the inlet temperatures of the heat source and the cooling water had a significant influence on the system. With the increment of the heat source inlet temperature, the mass flow rate of the working fluid, the net power output and the heat utilization ratio of the ORC system increased. While, the system thermal efficiency has decreased with increasing cooling water inlet temperature. Kumar et al [13] analyzed the performance of an organic Rankine cycle with benzene working fluid to improve efficiency and achieve better economy. They also described that in order to produce 9 kW of power with the same variation of mass flow rate as well as the Reynolds number, the efficiency of the ORC system will have to vary from 32.87% to 54.98% and that is possible only when the temperature at the outlet of turbine vary from 259.53°C to 127.22°C respectively.

Borsukiewicz-Gozdur [14] in his research assessed three variants of power plants, and he searched for the best use of the energy contained in a stream of 80–120 °C geothermal water. The power plants were an Organic Rankine Cycle (ORC), a dual fluid hybrid (DFH) and a single fluid hybrid fueled (HYB) that all were fueled or co-fueled by geothermal water. He concluded that hybrid and dual-fluid-hybrid power plants led to a proper exploitation of the energy in the geothermal waters, and if the aim were to optimize the utilization of the geothermal resources using the least share of energy from other sources, the best option was a dual-fluid-hybrid power plant with R236fa as a working fluid. In continuation of this work (Ref. [14]), in this paper, the power plants (ORC, DFH and HYB) are re-modeled and also are simulated in order to data validation. The results of data validation are acceptable, and they have convenient adaptation with the reference model. Since something else could be studied in this way, we continued the research to more it. Therefore, in this paper, we studied the first and second laws of thermodynamics in the defined cases in detail. Thermoeconomic analysis also is performed by using three various capital cost parametric functions. The total capital cost of plants and the unit cost of generated power in each power plant are estimated in order to compare the cases with each other. For generally assessment of models based on the important options, the level of the mis determined, and the rate of each model is specified. Because of DFH plant good economic situation, the effects of various organic working fluids on their operation are studied, and the fluid levels according to their performances are determined too.

2. Modeling of power plants

As previously mentioned, three variants of power plants are considered. The ORC power plant is shown in Fig. 1. The energy source of this case is just a stream of geothermal water with80–120 °C temperature[14].

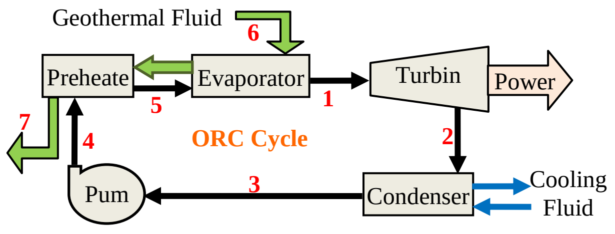


Fig. 1. Schematic diagram of Organic RankineCycle power plant (ORC)

The single fluid HYB power plant is illustrated in the Fig. 2. In this case, in addition to the geothermal stream, a biomass boiler is the high temperature energy source of system[14].

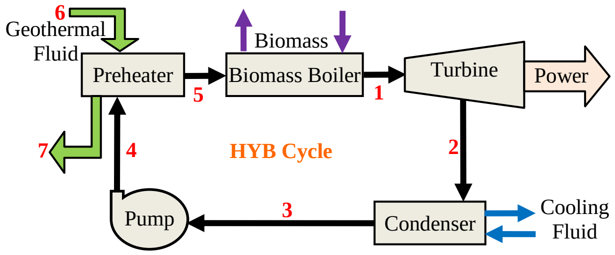


Fig. 2. Schematic diagram of hybrid power plant (HYB)

The cycle of dual fluid hybrid power plant is revealed in Fig.3. Its energy source is like the HYB cycle [14]but it has different arrangement.

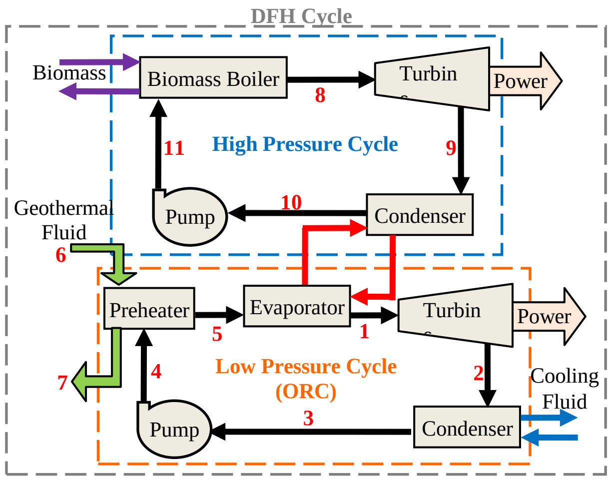


Fig. 3. Schematic diagram of a dual-fluid-hybrid power plant (DFH)

As seen in Fig. 3, in an organic Rankin cycle, which is the low pressure cycle, the fluid is preheated by the geo-fluid, and then its temperature rises to the operation point using the rejected energy from the condenser of the high pressure cycle. The high pressure cycle generates power using biomass energy, too. The operating fluid in the high pressure cycle is water. There are some assumptions in the simulation as presented in the Table 1 [14].

Table 1. Assumption of models

Item	Value
Pinch point between geo fluid and operating fluid in the HTC	5 (°C)
Temperature difference between HP and LP	5(°C)
Mass flow rate of geo fluid	30 (kg.s^{-1})
Outlet temperature offluid from condenser	30 (°C)
Inlet temperature of geofluid	120 (°C)
Outlet temperature of geofluid	35(°C)
Temperature of inlet fluid to the HP turbine in DFH cycle (T_8)	230(°C)
Quality of water at the outlet of fluid from HP turbine in DFH cycle (x_9)	0.89
Ambient temperature	25 (°C)
Ambient pressure	100 (kPa)

3. First-law analysis

Since the dual fluid hybrid power plant somehow cover the relations of the other cycles, the equations of first law of thermodynamics for dual fluid hybrid power plant are presented in this section [15-21]. Based on Fig. 3, for the high pressure cycle, entered heat from the biomass boiler is,

$$\dot{Q}_{in,HP} = \dot{m}_{HP}(h_8 - h_{11}) \tag{1}$$

The mass flow rate of biomass obtain from,

$$\dot{m}_{Biomass} = \eta_{BiomassBoiler} \times \left(\frac{\dot{Q}_{in,HP}}{\Delta h_{Biomass}} \right) \tag{2}$$

The rejected heat from the condenser of high pressure cycle is,

$$\dot{Q}_{out,HP} = \dot{m}_{HP}(h_9 - h_{10}) \tag{3}$$

The work of high pressure turbine is,

$$\dot{W}_{turb,HP} = \dot{m}_{HP}(h_8 - h_9) \tag{4}$$

The work of high pressure pump is,

$$\dot{W}_{pump,HP} = \dot{m}_{HP}(h_{11} - h_{10}) \tag{5}$$

Net work of high pressure cycle is,

$$\dot{W}_{net,HP} = \dot{W}_{turb,HP} - \dot{W}_{pump,HP} \tag{6}$$

The thermal efficiency of high pressure cycle is,

$$\eta_{th,HP} = \frac{\dot{W}_{net,HP}}{\dot{Q}_{in,HP}} \tag{7}$$

For the low pressure cycle the relations are presented in the following. Amount of the exited heat from the geofluid calculates as,

$$\dot{Q}_{geo,LP} = \dot{m}_{geo} C_p (T_6 - T_7) \tag{8}$$

Amount of the absorbed heat by organic fluid from the geofluid is,

$$\dot{Q}_{prh,LP} = \dot{m}_{LP}(h_5 - h_4) \tag{9}$$

With considering 100% efficiency for the geothermal heat exchanger, we have,

$$\dot{Q}_{geo,LP} = \dot{Q}_{prh,LP} \tag{10}$$

Amount of the absorbed heat by the evaporator in low pressure cycle is,

$$\dot{Q}_{eva,LP} = \dot{m}_{LP}(h_1 - h_5) \tag{11}$$

This is equal with the rejected heat from the high pressure condenser, and,

$$\dot{Q}_{eva,LP} = \dot{Q}_{out,HP} \tag{12}$$

Therefore, amount of the total absorbed heat by the low pressure cycle is calculated as,

$$\dot{Q}_{in,LP} = \dot{Q}_{eva,LP} + \dot{Q}_{geo,LP} \tag{13}$$

The rejected heat from the low pressure cycle condenser is,

$$\dot{Q}_{out,LP} = \dot{m}_{LP}(h_2 - h_3) \tag{14}$$

The work of low pressure turbine is,

$$\dot{W}_{turb,LP} = \dot{m}_{LP}(h_1 - h_2) \tag{15}$$

The work of low pressure pump is,

$$\dot{W}_{pump,LP} = \dot{m}_{LP}(h_4 - h_3) \tag{16}$$

Net work of low pressure cycle is,

$$\dot{W}_{net,LP} = \dot{W}_{turb,LP} - \dot{W}_{pump,LP} \tag{17}$$

The thermal efficiency of low pressure cycle is,

$$\eta_{th,LTC} = \frac{\dot{W}_{net,LP}}{\dot{Q}_{in,LP}} \tag{18}$$

At the end, total thermal efficiency of the dual fluid hybrid power plant is,

$$\eta_{th,tot} = \frac{\dot{W}_{net,tot}}{\dot{Q}_{in,tot}} \tag{19}$$

Where,

$$\dot{W}_{net,tot} = \dot{W}_{net,HP} + \dot{W}_{net,LP} \tag{20}$$

And,

$$\dot{Q}_{in,tot} = \dot{Q}_{in,HP} + \dot{Q}_{geo,LP} \tag{21}$$

4. Second-law analysis

In order to exergy analysis [22-25], we have the following equations,

$$ex_i = (h_i - h_0) - T_0(s_i - s_0) \tag{22}$$

In a determined mass flow,

$$\dot{E}x_i = \dot{m}\, ex_i \tag{23}$$

Equation (23)is used, for all components of the cycle. The amount of entered exergy to the high pressure cycle is,

$$\dot{E}x_{in,HP} = \dot{E}x_{eva,HP} + \dot{W}_{pump,HP} \tag{24}$$

The amount of entered exergy to the low pressure cycle is sum of the exergy difference in the preheater and the evaporator and the pumping work. So,

$$\dot{E}x_{in,LP} = \dot{E}x_{prh,LP} + \dot{E}x_{eva,LP} + \dot{W}_{pump,LP} \tag{25}$$

Also, we have,

$$\dot{E}x_{cond,HP} = \dot{E}x_{eva,LP} \tag{26}$$

When the control volume exceeds to the whole of cycle the internal exergy flows would be deleted; such entered exergy to the low pressure cycle by the low pressure evaporator. So, the amount of total entered exergy to the DFH cycle is,

$$\dot{E}x_{in,tot} = \dot{E}x_{prh,LP} + \dot{E}x_{eva,HP} + \dot{W}_{pump,LP} + \dot{W}_{pump,HP} \tag{27}$$

The amount of exited exergy from the high pressure cycle is,

$$\dot{E}x_{out,HP} = \dot{E}x_{cond,HP} + \dot{W}_{turb,HP} \tag{28}$$

The amount of exited exergy from the low pressure cycle is,

$$\dot{E}x_{out,LP} = \dot{E}x_{cond,LP} + \dot{W}_{turb,LP} \tag{29}$$

The outlet exergy flow of high pressure condenser is entered to the low pressure cycle through the low pressure evaporator; therefore, by deleting the high pressure condenser in the control volume of whole of cycle, the amount of total exited exergy from the DFH cycleis,

$$\dot{E}x_{out,tot} = \dot{E}x_{cond,LP} + \dot{W}_{turb,LP} + \dot{W}_{turb,HP} \tag{30}$$

Total exergy difference of the cycle is,

$$\Delta\dot{E}x_{tot} = \dot{E}x_{in,tot} - \dot{E}x_{out,tot} \tag{31}$$

And it is equal to,

$$\Delta\dot{E}x_{tot} = \left(\dot{E}x_{prh,LP} + \dot{E}x_{eva,HP} + \dot{W}_{pump,LP} + \dot{W}_{pump,HP}\right) - \left(\dot{E}x_{cond,LP} + \dot{W}_{turb,LP} + \dot{W}_{turb,HP}\right) \tag{32}$$

The exergy efficiency of the high pressure cycle is,

$$\eta_{Ex,HP} = \frac{\dot{W}_{net,HP}}{\dot{E}x_{in,HP}} \tag{33}$$

The exergy efficiency of the low pressure cycle is:

$$\eta_{Ex,LP} = \frac{\dot{W}_{net,LP}}{\dot{E}x_{in,LP}} \tag{34}$$

Therefore, total exergy efficiency of the dual fluid hybrid power plant calculates as:

$$\eta_{Ex,tot} = \frac{\dot{W}_{net,tot}}{\dot{E}x_{in,tot}} \tag{35}$$

5. Simulation and data validation

The results of model validation are presented in the next figures. The markers show our simulation findings, and the lines refer to the reference results. In Fig.4, the data of temperature of injected geothermal water as a function of initial temperature have been validated for different power plants.

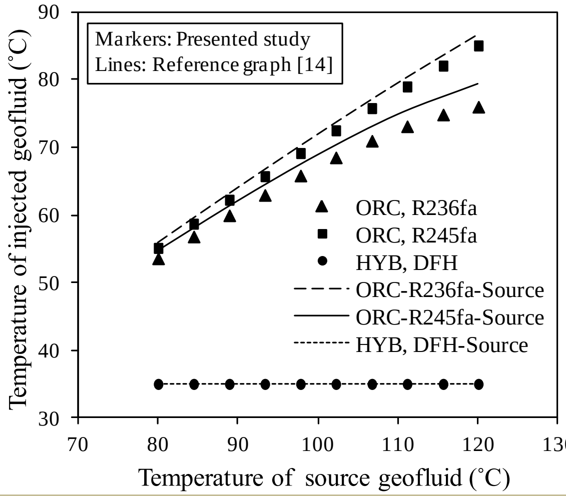


Fig. 4. Results of data validation for temperature of injected geothermal water as a function of initial temperature for different power plants

In Fig.5, we have validated the data of thermal efficiency as a function of initial temperature for the power plants.

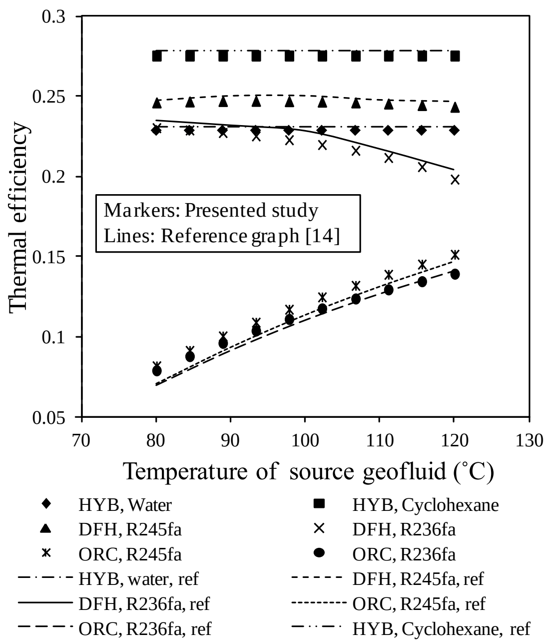


Fig. 5.Results of data validation for thermal efficiency of power plants variants as a function of initial geothermal water temperature

As seen, the results of models validation have good agreement with the reference models.

6. Cost modeling

Power plants main costs consist of the capital cost of equipment, energy costs, operating and maintenance costs. The energy costs contain the cost of thermal and electrical sources (fuels, electricity …) or operating costs; for example the fluids pumping and circulation cost. Generally, the economic assessments for various sizes are not completely exact, because there are multiple different parameters that influence on the final cost of equipment such as the materials, the manufacturing process and technologies, the manufacturing equipment and etc. So, the parametric capital cost functions of equipment and economic models just can estimate the economic amounts approximately. In this paper, we estimated the costs of the models; although the costs are approximate but they are for all models the same and we use cost functions in order to compare the models and operating fluids based on generated unit power costs and capital costs of the power plants. In this manner, we have used three various parametric capital cost functions to estimate the equipment costs [26-28]. The capital cost models are presented in Table 8 in the appendix. We need to have some assumption for biomass source and they are presented in Table 2 [29-31].

Table 2. Assumption for biomass in thermoeconomic analysis of models

Item	Type or Value
Biomass material	Ricehusk
Enthalpy of biomass	14000(kJ/kg)
Biomass Price	25 ($/Ton)

7. Thermoeconomic analysis

In the thermoeconomic analysis, the relations of economic and thermodynamic analysis are combined by applying the economic properties in the exergy parameters. The model can be written for the whole of power plant or major control volumes such as turbine, pump, evaporator and condenser [32-37]. The cost balance for the whole of power plant is [25]:

$$\sum_{i=1}^{n}\dot{C}_{i,in}+\dot{Z}_{k}=\sum_{j=1}^{m}\dot{C}_{j,out} \tag{36}$$

$C_{i,in}$ is entering and $C_{j,out}$ is exiting cost rate associated with an exergy stream, and these are the energy costs. $\dot{Z}_k$ is sum of annual capital investment for the equipment and also the operating and maintenance cost.

$$\dot{Z}_{k}=\dot{Z}_{c}+\dot{Z}_{om} \tag{37}$$

If c_j denote average costs per unit of exergy, then we have,

$$\dot{C}_{j}=c_{j}\cdot\dot{Ex}_{j} \tag{38}$$

The exergy transfer rate is defined as,

$$\dot{Ex}_{j}=\dot{m}_{j}.ex_{j} \tag{39}$$

These equations are applied to the defined power plants. So, by applying these equations to the three defined models, we have three following sub-sections.

7.1. ORC power plant

Cost balance for the ORC power plant is,

$$\begin{aligned}&\dot{C}_{geo,in}+\dot{C}_{pump}+\dot{C}_{cooling,in}+\dot{Z}_{k,ORC}\\&=\dot{C}_{geo,out}+\dot{C}_{turb}+\dot{C}_{cooling,out}\end{aligned} \tag{40}$$

By applying eq.(38) in theeq. (40)we have,

$$\begin{aligned}&c_{geo,in}.\dot{Ex}_{geo,in}+c_{pumping}.\dot{W}_{pump}+c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,ORC}\\&=c_{geo,out}.\dot{Ex}_{geo,out}+c_{power}.\dot{W}_{turb}+c_{cooling,out}.\dot{Ex}_{cooling,out}\end{aligned} \tag{41}$$

So, we obtain the cost per exergy unit of the generated power,

$$\begin{aligned}c_{power}=&\frac{c_{geo,in}.\dot{Ex}_{geo,in}+c_{pumping}.\dot{W}_{pump}+c_{cooling,in}.\dot{Ex}_{cooling,in}}{\dot{W}_{turb}}\\&+\frac{\dot{Z}_{k,ORC}-c_{geo,out}.\dot{Ex}_{geo,out}-c_{cooling,out}.\dot{Ex}_{cooling,out}}{\dot{W}_{turb}}\end{aligned} \tag{42}$$

The capital cost of the ORC power plant is defined as,

$$\dot{Z}_{c,ORC}=\dot{Z}_{prh}+\dot{Z}_{eva}+\dot{Z}_{turb}+\dot{Z}_{cond}+\dot{Z}_{pump} \tag{43}$$

The Z_k for the ORC cycle is,

$$\dot{Z}_{k,ORC}=\dot{Z}_{c,ORC}+\dot{Z}_{om,ORC} \tag{44}$$

7.2. HYB power plant

Cost balance for the HYB power plant is,

$$\dot{C}_{geo,in}+\dot{C}_{bio,in}+\dot{C}_{pump}+\dot{C}_{cooling,in}+\dot{Z}_{k,HYB} \\ =\dot{C}_{geo,out}+\dot{C}_{bio,out}+\dot{C}_{turb}+\dot{C}_{cooling,out} \tag{45}$$

By applying eq.(38) in theeq. (45)we have,

$$c_{geo,in}.\dot{Ex}_{geo,in}+c_{bio,in}.\dot{Ex}_{bio,in}+c_{pumping}.\dot{W}_{pump} \\ +c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,HYB} \\ =c_{geo,out}.\dot{Ex}_{geo,out}+c_{bio,out}.\dot{Ex}_{bio,out} \\ +c_{power}.\dot{W}_{turb}+c_{cooling,out}.\dot{Ex}_{cooling,out} \tag{46}$$

So, we obtain the cost per exergy unit of the generated power,

$$c_{power}=\frac{c_{geo,in}.\dot{Ex}_{geo,in}+c_{bio,in}.\dot{Ex}_{bio,in}+c_{pumping}.\dot{W}_{pump}}{\dot{W}_{turb}} \\ +\frac{c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,HYB}-c_{geo,out}.\dot{Ex}_{geo,out}}{\dot{W}_{turb}} \\ -\frac{c_{bio,out}.\dot{Ex}_{bio,out}+c_{cooling,out}.\dot{Ex}_{cooling,out}}{\dot{W}_{turb}} \tag{47}$$

The capital cost of the HYB power plant is defined as,

$$\dot{Z}_{c,HYB}=\dot{Z}_{prh}+\dot{Z}_{bio.boiler}+\dot{Z}_{turb}+\dot{Z}_{cond}+\dot{Z}_{pump} \tag{48}$$

The Z_k for the HYB cycle is,

$$\dot{Z}_{k,HYB}=\dot{Z}_{c,HYB}+\dot{Z}_{om,HYB} \tag{49}$$

7.3. DFH power plant

Cost balance for the DFH power plant is,

$$\dot{C}_{geo,in}+\dot{C}_{bio,in}+\dot{C}_{pump,HP}+\dot{C}_{pump,LP}+\dot{C}_{cooling,in}+\dot{Z}_{k,DFH} \\ =\dot{C}_{geo,out}+\dot{C}_{bio,out}+\dot{C}_{turb,HP}+\dot{C}_{turb,LP}+\dot{C}_{cooling,out} \tag{50}$$

By applying eq. (38) in the eq. (50)we have,

$$c_{geo,in}.\dot{Ex}_{geo,in}+c_{bio,in}.\dot{Ex}_{bio,in}+c_{pumping\,HP}.\dot{W}_{pump,HP} \\ +c_{pumping\,LP}.\dot{W}_{pump,LP}+c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,DFH}= \\ c_{geo,out}.\dot{Ex}_{geo,out}+c_{bio,out}.\dot{Ex}_{bio,out}+c_{power,HP}.\dot{W}_{turb,HP} \\ +c_{power,LP}.\dot{W}_{turb,LP}+c_{cooling,out}.\dot{Ex}_{cooling,out} \tag{51}$$

And,

$$c_{power,HP}.\dot{W}_{turb,HP}+c_{power,LP}.\dot{W}_{turb,LP}= \\ c_{geo,in}.\dot{Ex}_{geo,in}+c_{bio,in}.\dot{Ex}_{bio,in}+c_{pumping\,HP}.\dot{W}_{pump,HP} \\ +c_{pumping\,LP}.\dot{W}_{pump,LP}+c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,DFH} \\ -c_{geo,out}.\dot{Ex}_{geo,out}-c_{bio,out}.\dot{Ex}_{bio,out}-c_{cooling,out}.\dot{Ex}_{cooling,out} \tag{52}$$

It can be considered that,

$$c_{power,HP}=c_{power,LP} \tag{53}$$

So, we obtain the cost per exergy unit of the generated power,

$$c_{power}=\frac{c_{geo,in}.\dot{Ex}_{geo,in}+c_{bio,in}.\dot{E}x_{bio,in}+c_{pumping\,HP}.\dot{W}_{pump,HP}}{\dot{W}_{turb,HP}+\dot{W}_{turb,LP}} \\ +\frac{c_{pumping\,LP}.\dot{W}_{pump,LP}+c_{cooling,in}.\dot{Ex}_{cooling,in}+\dot{Z}_{k,DFH}}{\dot{W}_{turb,HP}+\dot{W}_{turb,LP}} \\ -\frac{\left(c_{geo,out}.\dot{Ex}_{geo,out}+c_{bio,out}.\dot{Ex}_{bio,out}+c_{cooling,out}.\dot{Ex}_{cooling,out}\right)}{\dot{W}_{turb,HP}+\dot{W}_{turb,LP}} \tag{54}$$

The capital cost of the DFH power plant is defined as,

$$\dot{Z}_{c,DFH}=\dot{Z}_{c,HP}+\dot{Z}_{c,LP} \tag{55}$$

And,

$$\dot{Z}_{c,DFH}=(\dot{Z}_{bio.boiler,HP}+\dot{Z}_{turb,HP}+\dot{Z}_{cond,HP}+\dot{Z}_{pump,HP}) \\ +(\dot{Z}_{prh,LP}+\dot{Z}_{eva,LP}+\dot{Z}_{turb,LP}+\dot{Z}_{cond,LP}+\dot{Z}_{pump,LP}) \tag{56}$$

The Z_k for the DFH cycle is,

$$\dot{Z}_{k,DFH}=\dot{Z}_{c,DFH}+\dot{Z}_{om,DFH} \tag{57}$$

By applying the equations in the defined models, the economic analysis is performed. There are some common costs such as land cost, tax, labor cost, carrying cost as a constant charge that are not considered in this economic assessment[25,32]. It was considered that the lifetime of cycles, annual full load working days and daily work time were 30 years, 300 days and 24 hours, respectively. Annual operating and maintenance costs for all cases assumed to be 0.02 of the capital costs [18,38].

8. Organic fluids in ORC of DFH plant

Various organic fluids can use in the low pressure cycle of the dual fluid hybrid ORC power plant. Some of organic fluids are selected in this manner. Each of these fluids has special operation and so they will have different behavior and different effect on the important parameters of cycles. Critical temperature and pressure of these selected organic fluids are presented in Table 3 [1,38].

Table 3. Critical temperature and pressure of organic fluid for the DFH cycle

Fluid	T_{cr} (°C)	P_{cr} (kPa)
Cyclohexane	280.5	4075
Isopentane	187.2	3370
n-Pentane	196.5	3364
R600	152	3796
R717	132.3	11333
Toluene	318.6	4126
R245fa	154	3651
R236fa	124.9	3198

9. Results and discussion

In this research, thermodynamic and economic parametric studies of mentioned models are done. Generally, mass flow rate is one of the important variables, and it affects on the size of equipment, capital costs, energy costs and etc. For example, if the mass flow rate of the operating fluid of a cycle increase, the size of the pump and the amount of pumping work or pressure drop increase too. Also, this cause to increase the capital cost of pump and the energy cost of pumping. At the otherwise, by increasing mass flow rate of the cycle, required area of heat exchangers would be decreased, and so the capital cost of heat exchangers would be decreased too. Mass flow rate of working fluid in power plants varies as a function of initial temperature of geothermal water, and it is shown in the Fig.6. As seen, the low pressure cycles of DFHs need tomore mass flow rateof working fluid comparedwith the others. The high pressure cycles of DFHs have the least mass flow rates of working fluid. HYB cycles have a moderate situation.

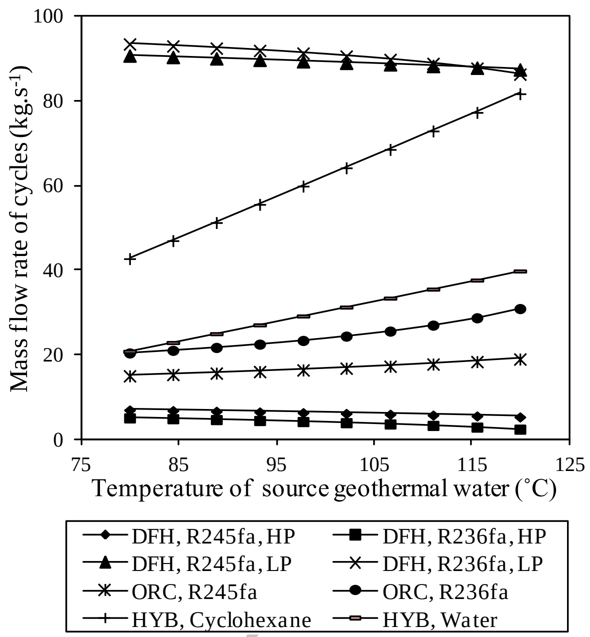


Fig. 6.Cycle's working fluidmass flow rates as a function of initial geothermal water temperature

Exergy efficiency of power plants varies as a function of initial geothermal water temperature as are shown in the Fig.7. At the lower temperatures of geothermal water, the HYB cycles are efficient from exergy viewpoint; but by increasing the temperature of the heat source, the ORCswill be more efficient. DFH cycles have a moderate situation.

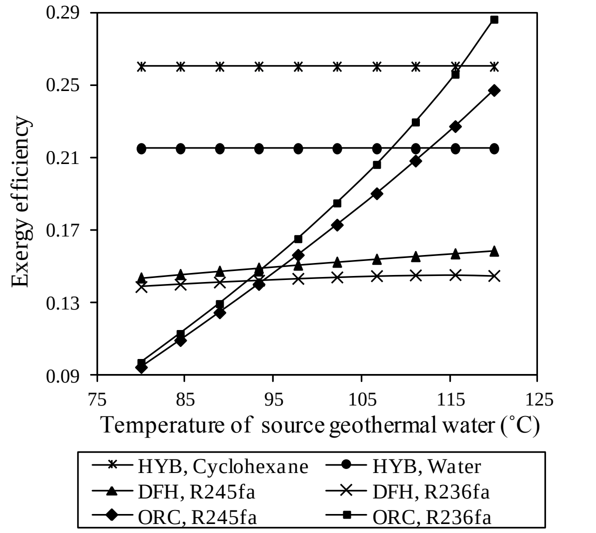


Fig. 7. Exergy efficiency of power plants variants as a function of initial geothermal water temperature

Destroyed exergy variations asa function of initial geothermal water temperature are shown in Fig. 8. As seen, the destroyed exergy of HYB cycles increase by increasing the temperature of heat source, and it has a reversetrend in the DFH cycles. Variation of destroyed exergy in the ORC cycles is very slight, and it has negative gradient like that DFH cycles.

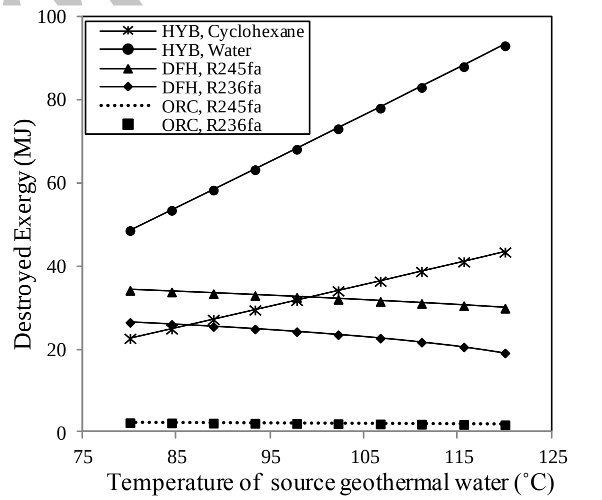


Fig. 8. Destroyed exergy of power plants variants as a function of initial geothermal water temperature

Unit generated power costs of power plants as a function of initial geothermal water temperature, based on three capital cost models are shown in Fig. 9.Generally, it can be seen that by increasing the temperature of heat source, the cost of unit generated power is decreased in the ORC and DFH plants, and it is increased in the HYB plants. The DFH cycle with R236fa has the least generated power price, and then the DFH cycle with R245fahas the lowest price. As seen, after the DFH cases, in low temperature source condition the HYB cycles with water and cyclohexane as their fluid have the least prices for unit power; but in high temperature source condition the ORC with R236fa and R245fa has the lowest prices.

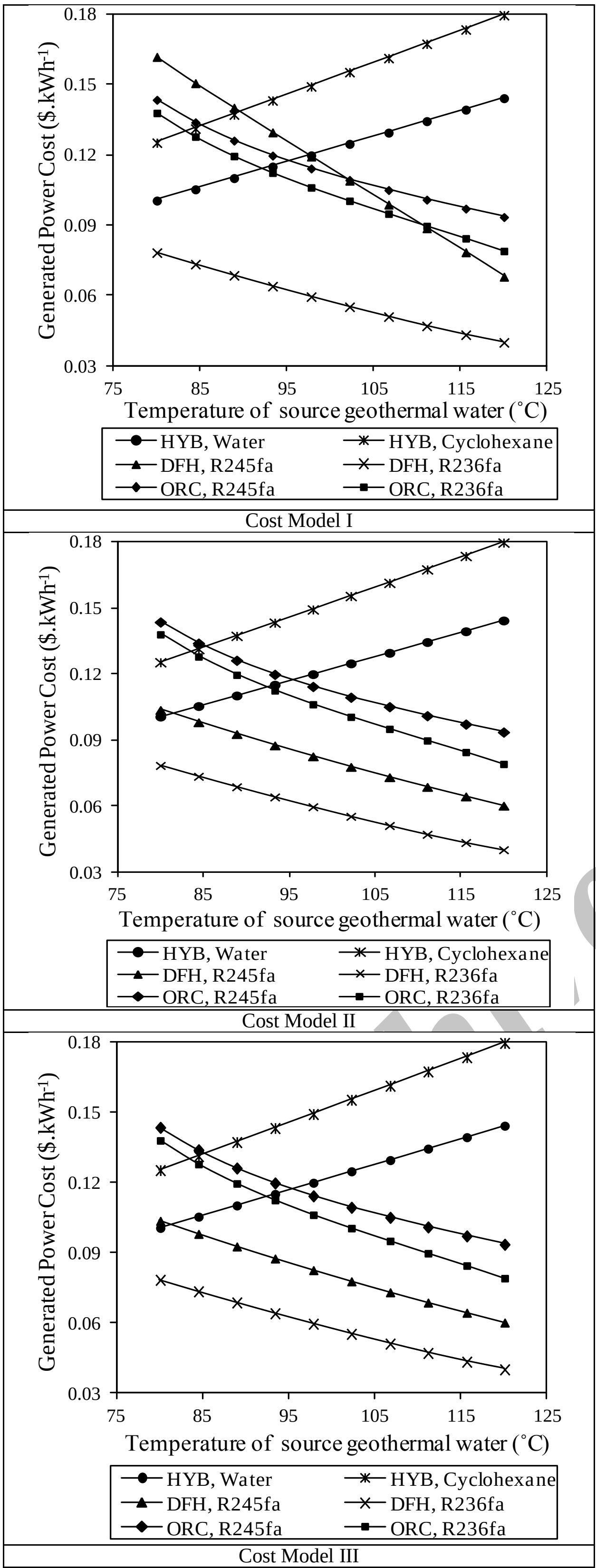


Fig. 9. Unit power cost of power plants variants as a function of initial geothermal water temperature

By approximating the initial investment for the cases, the capital costs of power plants per unit capacity as a function of initial geothermal water temperature are shown in the Fig. 10. As seen, based on capital cost models II and III,the initial investment per kilowatt in the ORC cycles is clearly higher than the other cycles.

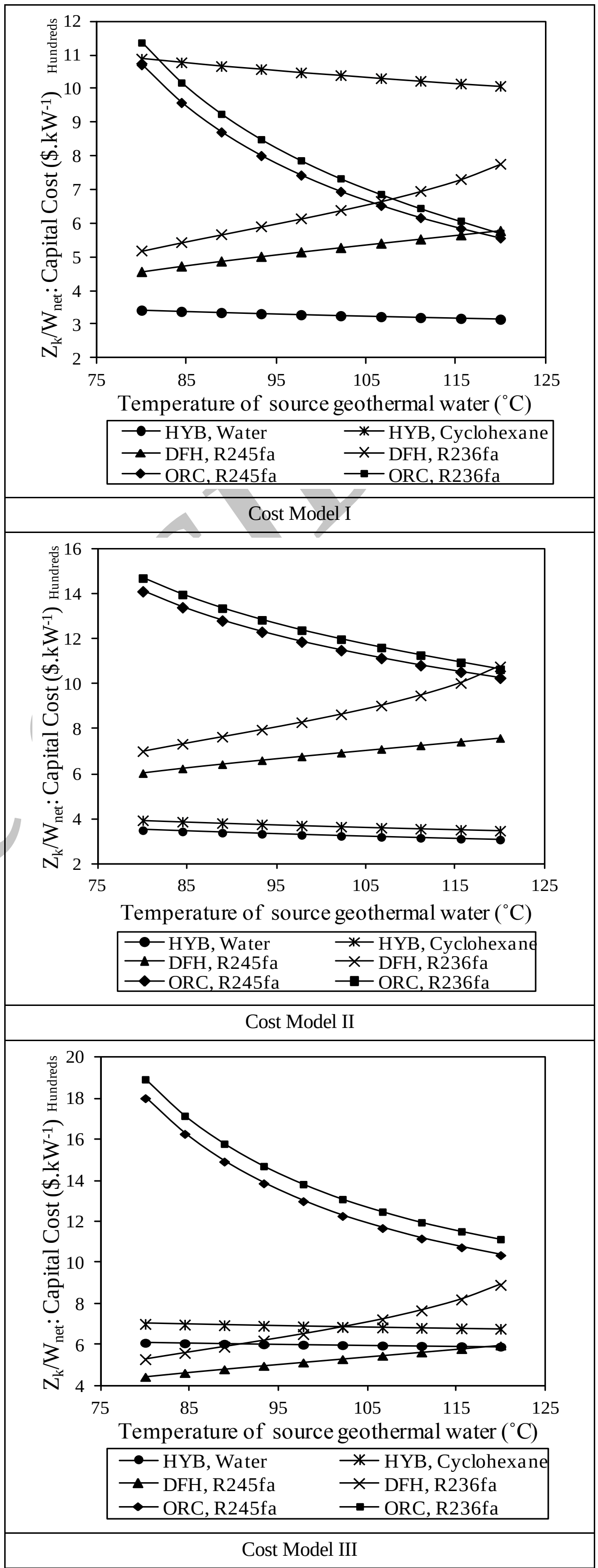


Fig. 10. Capital costs of power plants variants as a function of initial geothermal water temperature

Generally, it can be seen that, by increasing the temperature of heat source in the ORC plants, the capital costs of plant have decreased. The HYB cycles have low initial investments (capital costs) per kilowatt, and the DFH has a moderate behavior. The DFH with R236fa has a lower capital cost comparing with the DFH with R245fa.

As seen, there are various items in order to assessment the models. So, here five options have been collected, and the best choice for each one are presented, respectively. The items are thermal efficiency, exergy efficiency, destroyed exergy, generated power unit cost and the average capital costs. For the thermal and exergy efficiencies, the best is which that has the most amount, and for the destroyed exergy, generated power unit cost and average capital costs the best is which has the least amount.

In Table 4, assessment of options is presented at the columns, and level of each model in every option is determind by level of the rows.

Table 4. Best option for various items

	Thermal efficiency	Exergy efficiency	Destroyed exergy	Generated electricity cost	Capital costs
Weight factor	1	1	1	1	1
6	HYB, Cyclohexane	HYB, Cyclohexane	ORC, R236fa	DFH, R236fa	DFH, R245fa
5	DFH, R245fa	HYB, Water	ORC, R245fa	DFH, R245fa	HYB, Water
4	HYB, Water	ORC, R236fa	DFH, R236fa	ORC, R236fa	DFH, R236fa
3	DFH, R236fa	ORC, R245fa	DFH, R245fa	ORC, R245fa	HYB, Cyclohexane
2	ORC, R245fa	DFH, R245fa	HYB, Cyclohexane	HYB, Water	ORC, R245fa
1	ORC, R236fa	DFH, R236fa	HYB, Water	HYB, Cyclohexane	ORC, R236fa

Table 5. Score of various items

Power plant	DFH, R245fa	DFH, R236fa	HYB, Cyclohexane	HYB, Water	ORC, R236fa	ORC, R245fa
Score	21	18	18	17	16	15

Many other options can beadded in this table or some weight factors can be applied to the collected options based on the desired goals. For example, if the economic issues are important, for limited initial investment the average capital cost column should has a weight factor greater than the others, or for low price for consumer of power the generated power unit cost column should has a weight factor greater than the others. Here, with a moderate assessment and without weight factor for any option, Table 5 has been achieved. In this survey, first the DFH plant with R245fa as working fluid has a good situation, and then the DFH plant with R236fa as operating fluid is good option. It is seen that, in the thermodynamic view,the HYB plant with cyclohexane is the best option, and in the economic view, the DFH plant with R245fais suitable choice.

In the most analysis, the generated power unit cost has a determinant role, and DFH plants have better situation in this manner because of their low costs. However, a lot of organic fluids with suitable critical temperature and pressure for the defined conditions can be applied and they will have different acts. So, in the following, DFH plants with various operating fluids are studied.

Net work, thermal efficiency, exergy efficiency and destroyed exergy of DFH cycles with different organic fluids as functions of initial geothermal water temperature are shown in the Figs. 11, 12, 13 and 14, respectively.

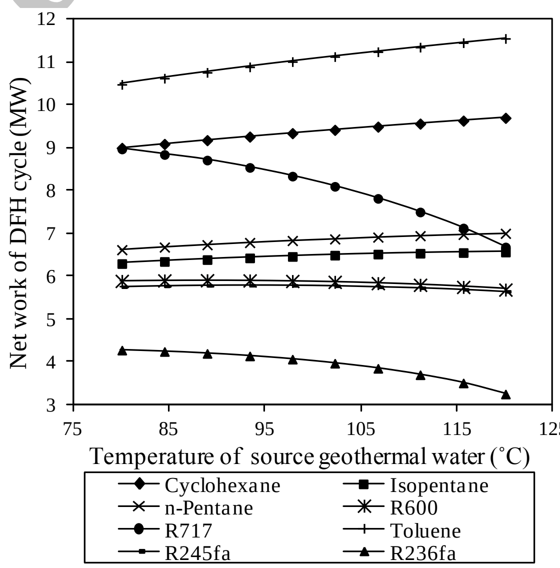


Fig. 11. Net work of DFH with organic fluids variants as function of initial geothermal water temperature

As seen in the all cases, toluene has the most net work, thermal efficiency, exergy efficiency and destroyed exergy in defined heat source ranges and the amounts increase by increasing the temperature of heat source. Inall cases, after toluene, cyclohexane and R717 have most amounts and R236fa has the least.

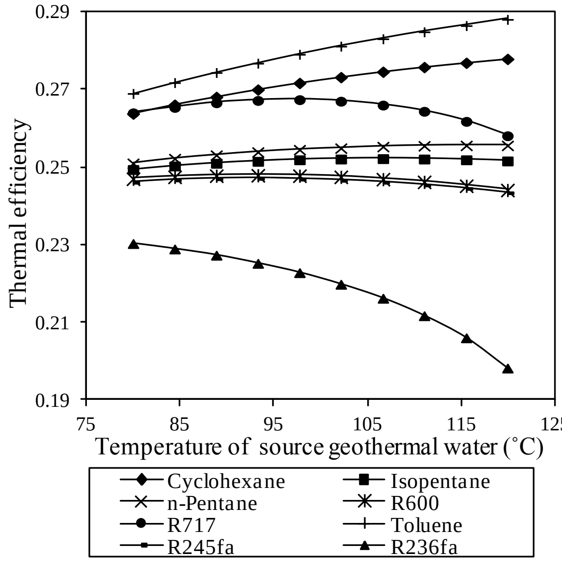


Fig. 12. Thermal efficiency of DFH with organic fluids variants as function of initial geothermal water temperature

Generally, the changes of net work, thermal efficiency, exergy efficiency and destroyed exergy are harmonious with each other.

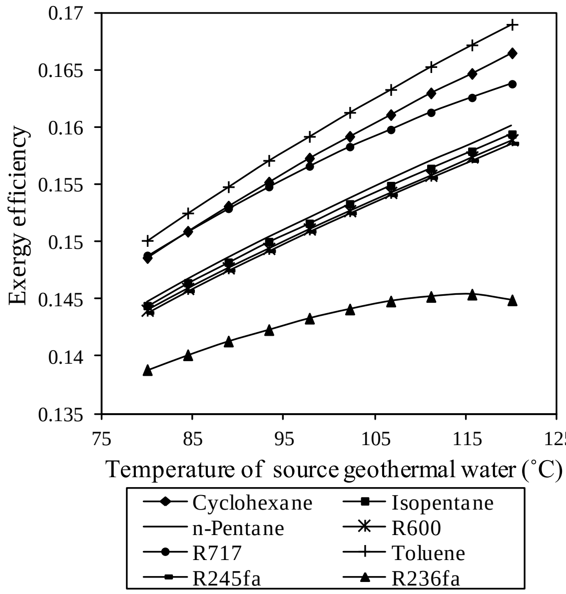


Fig. 13. Exergy efficiency of DFH with organic fluids variants as function of initial geothermal water temperature

Generally, it can be seen that, by increasing the temperature of heat source the exergy efficiency of cycles are increased and the destroyed exergy of cycles are decreased.

Working fluid mass flow rates of low pressure cycles as functions of initial geothermal water temperature are shown in Fig. 15.The gradients of mass flow rates are small and each fluid has a special level of rate. R236fa and R245fa have the most flow rates and R717 has the least amount.

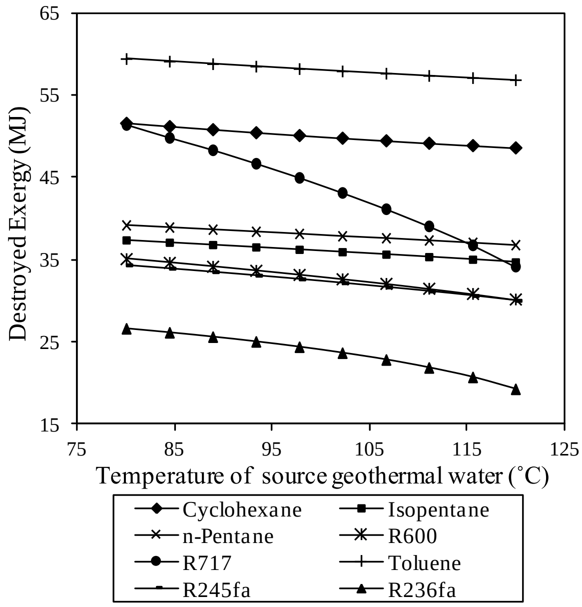


Fig. 14. Destroyed exergy of DFH with organic fluids variants as function of initial geothermal water temperature

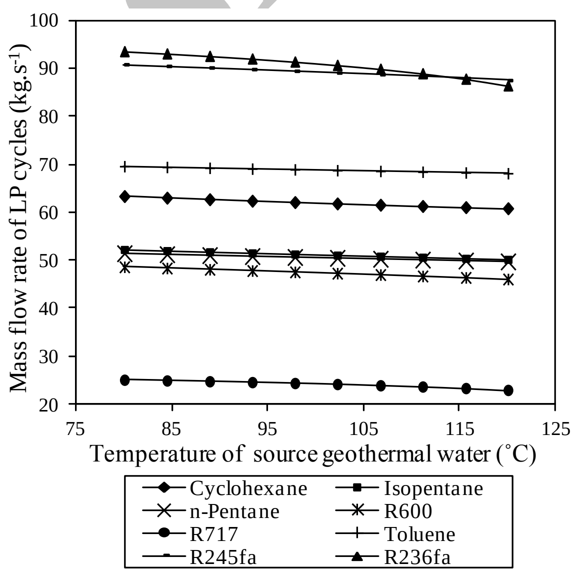


Fig. 15. Mass flow rates of LP cycles working fluid variants as function of initial geothermal water temperature

Variations of working fluid mass flow rates in LP cycles effect on HP cycle working fluid mass flow rate. Working fluid mass flow rates of HP cycles variants as functions of initial geothermal water temperature are shown in Fig. 16. Water in HP cycle of DFH has the most flow rate when toluene is the fluid of LP cycle, and it has the least amount of flow rate when R236fa is the fluid of LP cycle.R717 has the most gradient. Generally, that by increasing the temperature of heat source the mass flow rates of HP cycles are decreased.

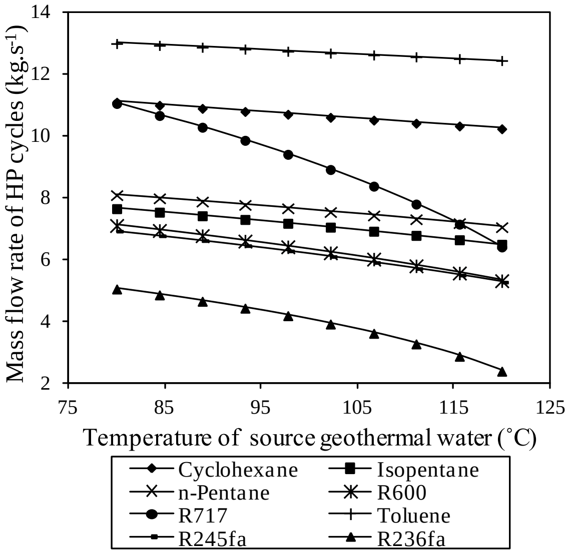


Fig. 16.Working fluid mass flow rates of HP cycles variants as function of initial geothermal water temperature

The unit costs of generated power in the power plants as functions of initial geothermal water temperature, based on three capital cost models are shown in Fig. 17. The models have similar behaviors, and generally, that by increasing the temperature of heat source the unit price of generated power is decreased in all types of DFH plants. Based on all three models, first the DFH cycle with R236faand then the DFH cycle with R245fahave the least generated power prices, respectively. The DFH with toluene has the most generated power prices. Generally, that by increasing the temperature of heat source, the unit price of generated power is decreased.

By approximating the initial investment of the cases, the capital costs of power plants per unit capacity as a function of initial geothermal water temperature are shown in the Fig. 18. About the capital costs, the models have similar behavior, and based on them it can be said that the DFH with toluene and cyclohexane has the least capital cost. R236fa has the highest cost. Generally, that by increasing the temperature of heat source the capital cost of the plant is increased.

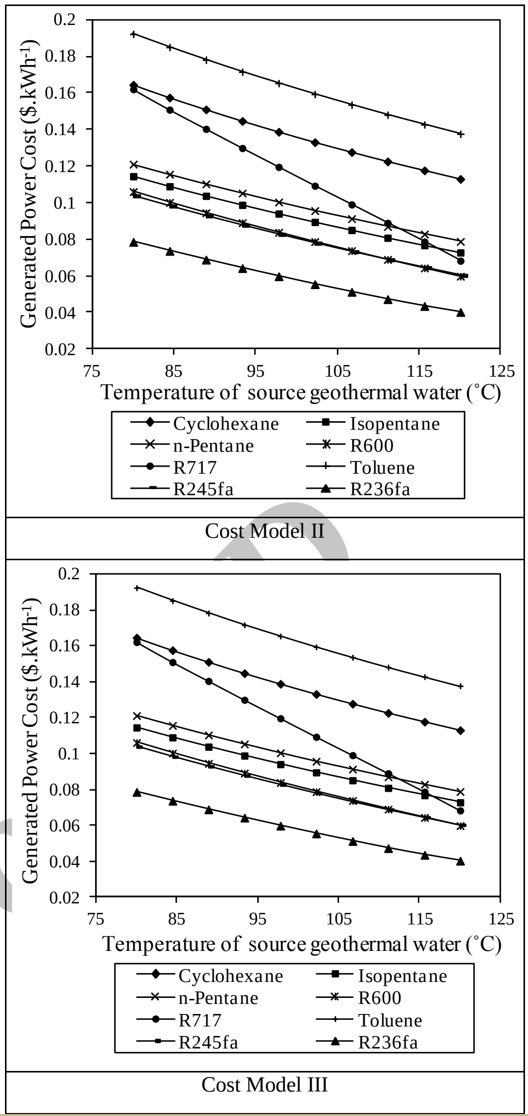
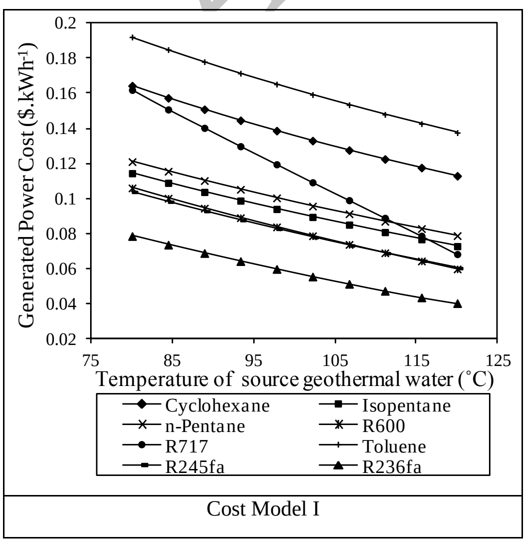
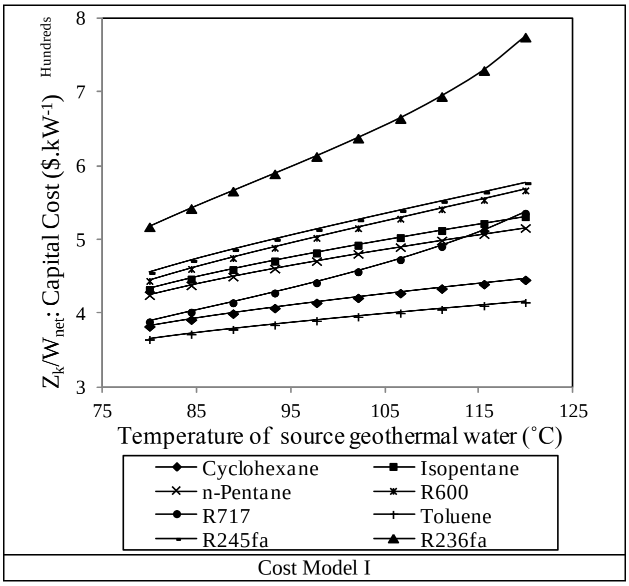
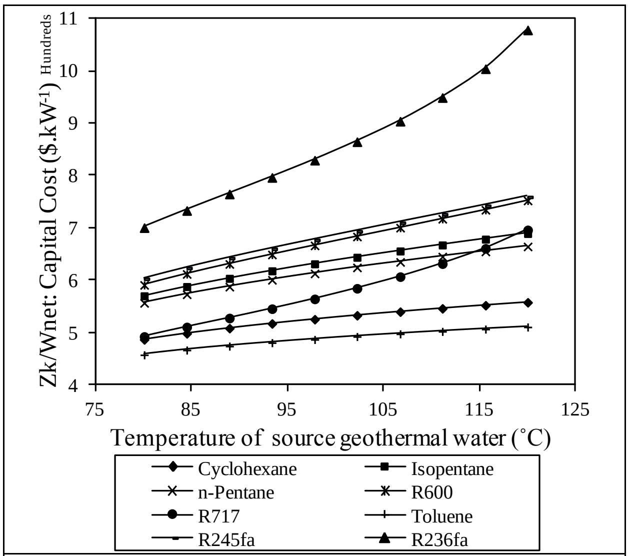


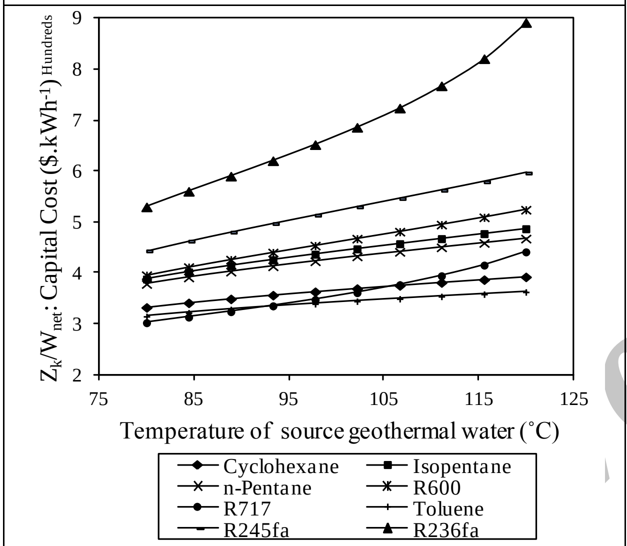
Fig. 17. Power cost of DFH with organic fluids variants as function of initial geothermal water temperature







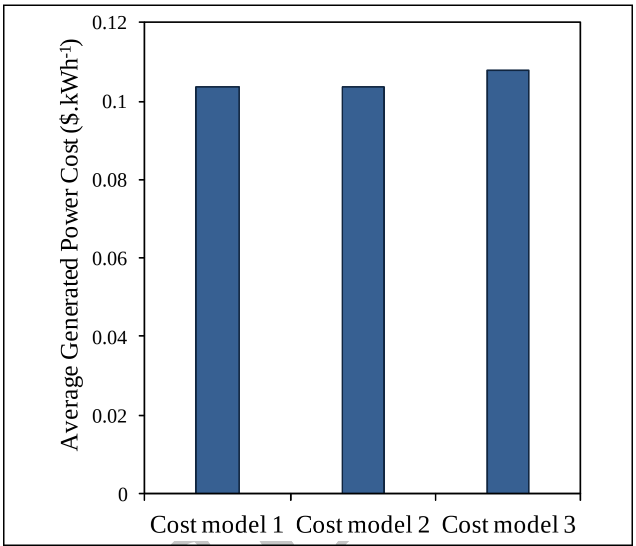
Cost Model II



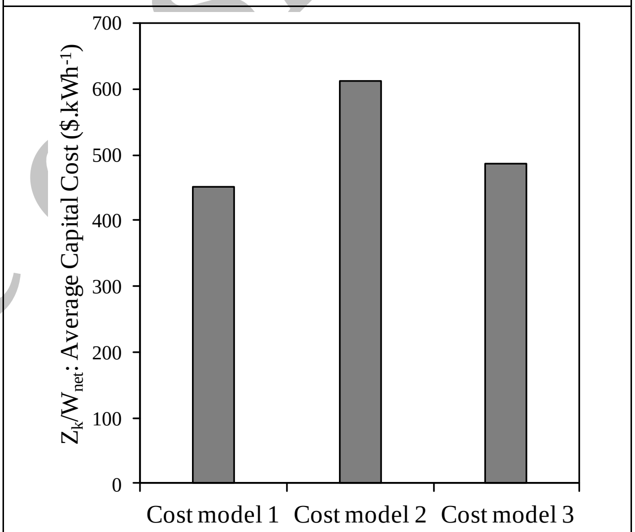
Cost Model III

Fig. 18. Capital cost of DFH with organic fluids variants as a function of initial geothermal water temperature

In order to assessing the three capital cost models the averages of results are compared,and the results are presented inFig. 19. It is observed that,for the generated power cost, the models I and II are close together, and the model III has the most amount. Also for the average capital cost,the models I and III are close together, and the model II has the most amounts.The rate of changes in the average generated power cost is 1.7% and in the average capital cost is 12%.Generally, we can say that, all of three models have satisfactory results; since the results procedure are similar and just partial differencesare observed in the results.



a. Average generated power cost by three costmodels



b. Average capital costs of three cost models

Fig. 19. Average results of three capital costs

Similar to the section 8, five options are collected, and the best fluid for each option is presented, respectively. In Table 6,the level of each fluid for the optionsis determined by applying the weight factor from 8 to 1. The weight factor of the all five options is considered one, and this means that any option has not preference against the others.

Table 6. Best option for various items, respectively

	Thermal efficiency	Exergy efficiency	Destroyed exergy	Generated electricity cost	Capital costs
Weight factor	1	1	1	1	1
8	Toluene	Toluene	R236fa	R236fa	Toluene
7	Cyclohexane	Cyclohexane	R600	R245fa	Cyclohexane
6	R717	R717	Isopentane	R600	R717
5	n-Pentane	n-Pentane	n-Pentane	Isopentane	n-Pentane
4	Isopentane	Isopentane	R717	n-Pentane	Isopentane
3	R600	R600	R245fa	R717	R600
2	R245fa	R245fa	Cyclohexane	Cyclohexane	R245fa
1	R236fa	R236fa	Toluene	Toluene	R236fa

Table 7. Score of various items								
Operating fluid	Toluene	Cyclohexane	R717	n-Pentane	Isopentane	R600	R236fa	R245fa
Score	26	25	25	24	23	22	19	16

By a moderate assessment and without weight factor for any option, Table 7 is achieved. In this survey, first toluene and then cyclohexane have good situationsas operating fluid.

10. Conclusions

Thermodynamic and exergy studies of three different types of power plants are done. each model has its preferences, and the selecting of power plants can be done according to the main aims. However, as the thermodynamic approach, the HYB plant with cyclohexane is the best option, and as the economic approach, the DFH plant with R245fa and R236fa are suitable choices. If both of thermodynamic and economic problems would be important, first the HYB plant with cyclohexane, and then the DFH plant with R245fa have better situations. According to the importance of economic topics, and good situations of DFH plants in this manner, variant organic fluids are evaluated for the ORC cycle of dual fluid hybrid power plant (LP) as working fluids. It is concluded that, in the dual fluid hybrid power plant, each model has its preferences, nevertheless as the thermodynamic approach, toluene is the best option, and as the economicapproach, R245fa and toluene are suitable. By an all-around survey, first toluene and then cyclohexane as theoperating fluid have good situations in the DFH. Generally, it is seen that, in the DFH plants, by increasing the temperature of the heat source, the exergy efficiency and the capital cost of plants are increased and destroyed exergy of cycles, mass flow rates of operating fluids in HP and LP cycles as well as generated power unit cost are decreased. So, in order to have a good selection, the conditions should be defined by setting goals and priorities. Also, according to the unit generated power costs and their changes based on three capital cost models, it is observed that the effect of equipment capital cost on generated power unit cost is insignificant, and the operating conditions in the period of a lifetime are more influential.

Nomenclature

Abbreviations

HYB	Hybrid power plant
DFH	Dual fluid hybrid power plant
ORC	Organic Rankin cycle power plant

Subscripts

PP	Pinch point
HP	High pressure cycle
LP	Low pressure cycle
prh	Preheater
eva	Evaporator
cond	Condenser
turb	Turbine
th	Thermal
tot	Total
c	Capital cost
om	Operating and maintenance cost
cr	Critical

Parametes

c_p	Specific heat of geothermal water ($kJ.kg^{-1}.K^{-1}$)
h	Specific enthalpy ($kJ.kg^{-1}$)
$\dot{m}$	Mass flow rate ($kg.s^{-1}$)
P	Pressure (kPa)
S	Entropy ($kJ.kg^{-1}.K^{-1}$)
T	Temperature (˚C)
W	Work (kW)
Q	Heat (kW)
x	vapor mass fraction
η	Efficiency
ex	Exergy in mass unit ($kJ.kg^{-1}$)
Ex	Exergy (kW)
C	Cost ($)
c	Average costs per unit of exergy ($\$.kW^{-1}$)
Z	Investment costs ($)

Appendix

The three capital cost models are presented in Table 8.

Table 8. Three capital cost models
Capital cost model (I) [26]
$\dot{Z}_{eva} = 208582\dot{m}_{eva}^{0.8} \times \left(\exp\left(\frac{P_{eva}-28}{150}\right)^{0.78}\right) \times \left(1+5\exp\left(\frac{T_{eva}-593}{10.42}\right)\right) \times \left(1+\left(\frac{1-0.9}{1-\eta_{eva}}\right)^{7}\right)$
$\dot{Z}_{turb} = 3880.5P_{turb}^{0.7}\left(1+\left(\frac{0.05}{1-\eta_{turb}}\right)^{3}\right) \times \left(1+5\exp\left(\frac{T_{turb}-866}{10.42}\right)\right)$
$\dot{Z}_{pump} = 705.48P_{pump}^{0.71}\left(1+\left(\frac{0.2}{1-\eta_{pump}}\right)\right)$
$\dot{Z}_{cond} = 280.74\left(\frac{\dot{Q}_{cond}}{k.\Delta T_{in}}\right)+746\dot{m}_{cooling}+\left[70.5\dot{Q}_{cond}\times\left(-0.6936Ln\left(T_{cooling}-T_{a}\right)+2.1898\right)\right]$
Capital cost model (II) [27]
$\dot{Z}_{eva} = 130\left(\frac{A_{eva}}{0.093}\right)^{0.78}$
$\dot{Z}_{turb} = 3644.3\left(\dot{W}_{turb}\right)^{0.7} - 61.3\left(\dot{W}_{turb}\right)^{0.95}$

$\dot{Z}_{cond} = 248A_{cond} + 659\dot{m}_{cooling}$
$\dot{Z}_{Pump} = 442\left(\dot{W}_{pump}\right)^{0.71}1.41 f_{\eta}$ and $f_{\eta} = 1 + \left(\frac{1-0.8}{1-\eta_{pump}}\right)$
Capital cost model (III) [28]
$\dot{Z}_{eva} = 3650\left[\left(\frac{Q_{eva}}{\Delta T_{LMeva}}\right)^{0.8} + \left(\frac{Q_{eco}}{\Delta T_{LMeco}}\right)^{0.8}\right] + 11820m_{eva} + 658\left(m_{hx}\right)^{1.2}$
$\dot{Z}_{prh} = 2290\left(A_{prh}\right)^{0.6}$
$\dot{Z}_{turb} = \frac{266.3}{0.92-\eta_{turb}} \times \ln\left(\frac{P_{in}}{P_{out}}\right) \times \left[1 + \exp(0.036T_{in} - 54.4)\dot{m}_{turb}\right]$
$\dot{Z}_{Pump} = \frac{39.5}{0.9-\eta_{pump}}\frac{P2}{P1}\left(\dot{m}_{pump}\right)$

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