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Effect of Different Binary Working Fluids on Performance of Combined Flash Binary Cycle

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ABSTRACT

Due to limited fossil fuels supply and their negative impact on environment, utilization of renewable energy resources attracts more attention among researchers every day. Geothermal power generation is the most reliable renewable energy resource. Geothermal power plants have the highest capacity factor among other renewable power generation technologies. Unlike solar radiation and wind energy, it is not fluctuating during each day and different times of the year. Introducing technologies with higher energy conversion efficiency, or improving existing power generation stations would make geothermal power plant more competitive with conventional fossil fired power plants for base load power supply.

Consider the relative simplicity and reliability of single flash plants; they are often the first type of plants installed in a newly developed field [1]. But, hot brine coming out of the separator (that is re-injected to the reservoir) has high working potential and can be used for further power generation. Using combined flash-binary cycles is an effective way of capturing this available energy and increasing power output of the system. One of the most important design parameters in a combined flash-binary power system is selection of secondary working fluid. Because each geothermal field is unique and different from others, selection of secondary fluid must be done with respect to incoming brine from the field.

In this paper we will analyze the combined flash-binary power cycles from an energy and exergy point of view. For different brine temperatures, optimum performance and the most appropriate working fluid will be presented. Binary working fluids R-32, R-125, HC 270 and R-22 (as wet fluids), R-21, R-142b, R-134a (as isentropic fluids) and R-601, R-600 and FC-4-1-12 (as dry fluids) will be considered as secondary fluid selections for each case.

1. INTRODUCTION

Due to the limited supply of fossil fuels and negative impacts of burning conventional fuels to produce power, utilization of renewable energy resources gets more attention every day. Among renewable energy power generation technologies, geothermal power plants have an outstanding position. It is claimed that they can reach the highest capacity factor—even higher than conventional thermal plants [1]. Unlike solar radiation and wind energy, the thermal energy stored below earth's surface could be available without daily or annual fluctuations. Higher capacity factors of geothermal power plants make them suitable to supply base load in electricity grids, which is a privilege compared with other renewable power stations.

Bertani [2] reported the installed capacity of geothermal power plants worldwide equal 10.9 GW with electrical energy generated equal 67 TWh/year as of 2010. Approximately 1.8 GW (18%) of the installed capacity has been constructed between 2005 and 2010. Also it is predicted the total installed power generation capacity would reach to 19.8 GW (With 160 TWh/year energy production) by year 2015. This indicates that power generation from thermal energy stored below earth surface has been accelerating and is predicted to continue this growth in future years.

Table 1: Operating combined flash-binary geothermal power plants [8]

Plant	Location	Installed binary capacity [MW]
Miravalles 5	Costa Rica	18
Leyte	Philippines	61
Mak-Ban	Philippines	15.7
Mokai	New Zealand	18
Wairakei	New Zealand	15
Puna	Hawaii (USA)	30

The location of geothermal fields with available high temperature geofluid are very limited worldwide, and most of these fields have been tapped for electricity production [3]. Hence, introducing technologies with higher energy conversion efficiency or improving existing power generation stations is crucial to maintain the increasing trend of geothermal power capacity worldwide. Additionally, like all other renewable power generation systems, geothermal power stations require high capital cost. High initial costs of these power systems make them less competitive with fossil fueled power plants. The costs can be reduced by improving the performance of the system.

Using the combined flash-binary cycle is an effective way to increase the efficiency of existing and planned flash steam geothermal power plants. Table 1 presents some of the combined flash-binary plants installed in different locations.

The aim of this paper is to present a thermodynamic analysis and optimization of the flash-binary geothermal power cycle. An energy and exergy study has been conducted for each cycle component. Also for different geofluid temperatures, maximum work output for different types of secondary working fluids is presented.

2 COMBINED FLASH-BINARY CYCLE

Flash steam cycles are the most common power generation units, used to produce electricity from liquid dominated geothermal fields [4]. In 2010, flash steam cycles counted for 61% of total installed geothermal power capacity (Table 2).

Table 2: Installed geothermal power plants by 2010 [2]

Type of the plant	Installed capacity	Number of units	Electricity produced
Dry steam	27 [%]	12 [%]	24 [%]
Single flash	41 [%]	27 [%]	42 [%]
Double flash	20 [%]	11 [%]	21 [%]
Binary	11 [%]	45 [%]	9 [%]
Back pressure	1 [%]	5 [%]	4 [%]
Total	10.9 GW	536	67 TWh

Flashing is the process of transforming the pressurized liquid into a mixture of liquid and vapor by lowering the fluid pressure below the saturation pressure corresponding to the geofluid temperature. When the production wells produce a mixture of steam and liquid, an efficient way to produce power is to separate the steam portion of the flow and use it to drive a steam turbine. For high temperature geofluids (higher than 150 C), flashing is a viable process to reduce the pressure of the flow more in order to increase the steam flow rate. The liquid portion of the flow leaving the separator usually has high temperature and flow rate. In a single flash cycle this part of the geofluid is reinjected into the reservoir to maintain the productivity of the extraction wells. Pambudi et al. [5] conducted an exergy analysis on Diang geothermal power plant in Indonesia. They reported that there is a considerable amount of exergy wasted (counting 18% of the total available exergy) by geofluid reinjection at high temperatures to the reservoir.

In a combined flash-binary cycle an Organic Rankine Cycle (ORC) is added to the single flash plant as shown in Fig 1, to tap into reinjection pipeline. By extracting heat from the high temperature liquid and reducing its temperature it is possible to increase the overall efficiency. Also, because there is no additional expenses needed to produce the hot fluid, adding the ORC to a flash steam plant would result to cheap additional electricity production.

Binary geothermal power plants are not a new concept. In 1961, Tabor and Bronicki [6] developed a procedure to use an organic fluid with low boiling temperature as working medium to be used in a power cycle. The first geothermal binary ORC was installed a Paratunka – Russia in 1967 [7]. In 2010 the sum of all binary geothermal power units was only 1.1 GW, even though 44% of all existing geothermal power units were of binary type. Binary cycles are used for low grade heat sources. Due to the low temperature of the heat source, the power conversion efficiency of these plants is low. In a Binary cycle, hot geofluid is passed through a heat exchanger to evaporate a secondary fluid. The saturated or superheated vapor which is usually an organic fluid with low boiling temperature and high vapor pressure is used to run an expander and produce mechanical power. Binary power plants have no emissions and because geofluid is not in contact with cycle components (except the evaporator) face no problems regarding scaling and corrosion.

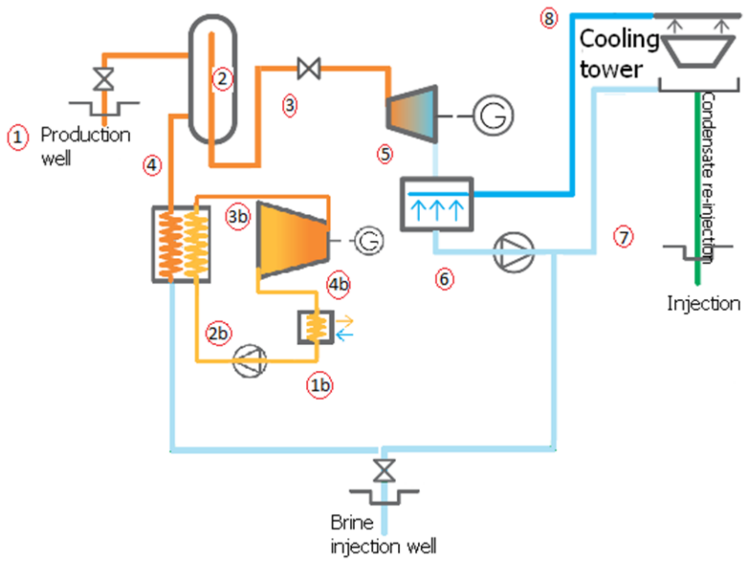

Figure 1: Schematic of a flash - binary combined cycle

3 MODELING AND ANALYSIS

The combined flash-binary cycle consists of two separate cycles. Although there are no common components in two cycles, operation conditions of the top cycle (flash steam cycle) would affect the performance of the bottoming cycle (binary cycle). Fig 2 shows the T-s Diagram of a typical flash steam cycle. Geofluid from production wells (point 1) enters the separator after a constant enthalpy flashing process as a saturated steam-liquid mixture. Inside the separator (point 2), steam and liquid phases are separated. The saturated steam (point 3) which is a small fraction of total flow is conducted toward the steam turbine. After expanding inside the turbine and producing mechanical work the exhaust flow of turbine (point 5) is mixed with the cold water from the cooling tower (point 8) in a direct contact condenser. The condenser outlet would be saturated liquid (point 6). The majority of effluent from this point is pumped to the cooling tower (point 7), and the rest is reinjected into the ground. The fluid inside the cooling tower transfers heat to the environment and enters the condenser (Point 8) to mix with turbine outlet. The liquid portion of the geofluid from the separator (point 4) enters the binary cycle evaporator to heat up the ORC working fluid. Inside the evaporator the ORC working fluid transforms into saturated or super-heated steam (point 3b) by absorbing heat from hot liquid geofluid. Vapor from the evaporator enters into the ORC turbine and produces mechanical work. The turbine exhaust (point 4b) is entered closed condenser and transfers into saturated liquid (point 1b) by loosing heat to the cooling medium. After flowing out of condenser the working fluid is conducted through the pump to get to the evaporator pressure (point 2b). Sub cooled working fluid enters the evaporator and the cycle is completed. Fig 3 depicts the T-s diagram of a typical ORC for different types of working fluids.

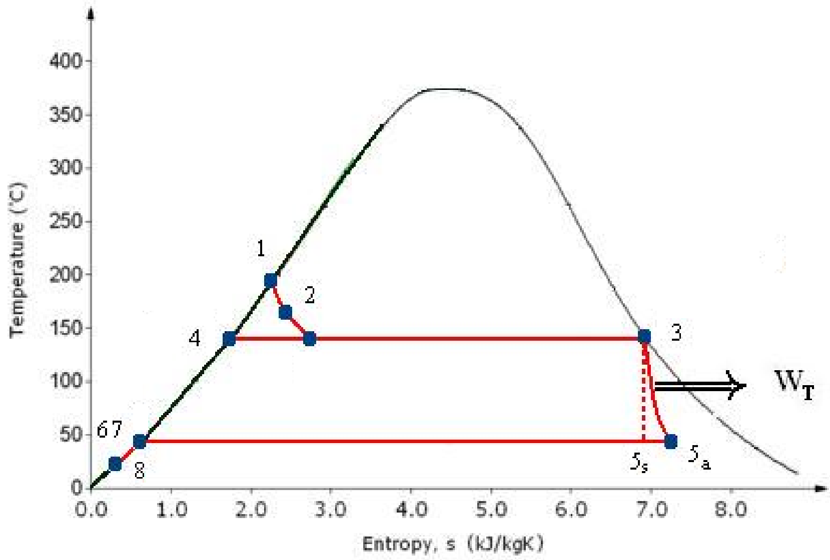

Figure 2: Flash steam cycle T-s Diagram

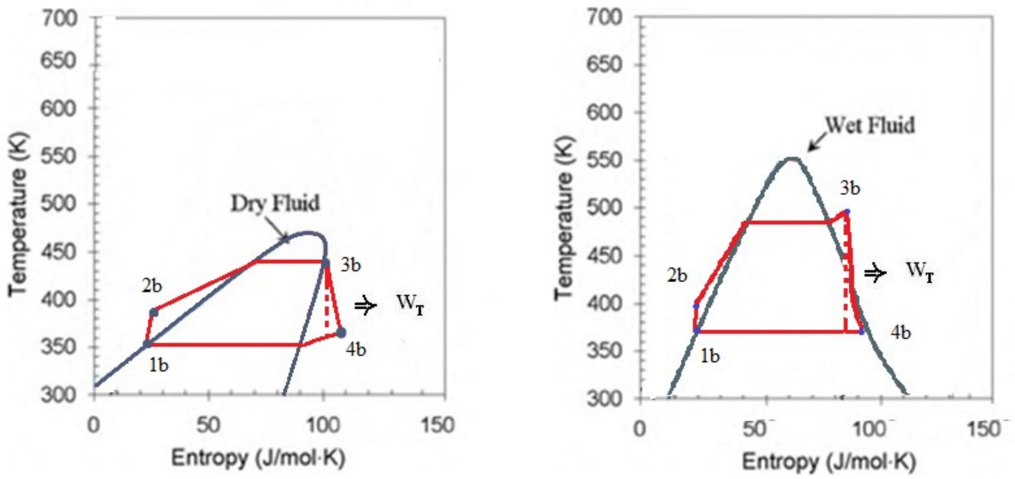

Figure 3: ORC T-s Diagram for (a) Dry working fluids (b) Wet working fluids

The working fluid's condition evaluation methods at different parts of each cycle are straightforward and described in detail in reference [4], hence are not repeated in this paper.

The following assumptions are considered in this study:

- Combined flash-binary cycle operates under steady state conditions.
- Potential and kinetic energy and changes in potential, kinetic and chemical exergies are neglected.
- Heat and pressure losses in all components are neglected.
- Thermophysical properties of geofluid are considered as pure water.
- Geofluid is assumed to be free of any chemical substances, and non-condensable gases.

3.1 Energy and Exergy Balances

Mass, energy and exergy balances for each component of the cycle can be expressed by:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} + \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \quad (2)$$

$$\dot{E}_Q + \dot{W} = \sum \dot{E}_{out} - \sum \dot{E}_{in} + \dot{I} \quad (3)$$

Where $\dot{m}$ is the mass flow rate, $\dot{Q}$ and $\dot{W}$ are the net heat and work input respectively, h is the enthalpy, $\dot{E}$ is the exergy or availability and $\dot{I}$ is the rate of exergy destruction. Subscripts in and out depict the input and exit states and subscript 0 denotes the dead state.

Table 3: mass, energy and exergy equations for combined flash-binary cycle components

Component	Mass Balance	Energy Balance	Exergy Balance
Steam-Separator	$\dot{m}_1 = \dot{m}_3 + \dot{m}_4$	$\dot{m}_1 h_1 = \dot{m}_3 h_3 + \dot{m}_4 h_4$	$\dot{I}_{sep} = \dot{m}_1 e_1 - \dot{m}_3 e_3 - \dot{m}_4 e_4$
Steam -Turbine	$\dot{m}_3 = \dot{m}_5$	$\dot{W}_{T\_s} = \dot{m}_3 h_3 - \dot{m}_5 h_5$	$\dot{I}_{T\_s} = \dot{m}_3 e_3 - \dot{m}_5 e_5 - \dot{W}_{T\_s}$
Steam -Condenser	$\dot{m}_6 = \dot{m}_8 + \dot{m}_5$	$\dot{m}_6 h_6 = \dot{m}_5 h_5 + \dot{m}_8 h_8$	$\dot{I}_{C\_s} = \dot{m}_5 e_5 + \dot{m}_8 e_8 - \dot{m}_6 e_6$
Steam-Cooling tower *	$\dot{m}_7 = \dot{m}_8$	$\dot{Q}_{CT\_s} = \dot{m}_7 h_7 - \dot{m}_8 h_8$	$\dot{I}_{CT\_s} = \dot{m}_7 e_7 - \dot{m}_8 e_8$
Binary-Evaporator	$\dot{m}_4 = \dot{m}_9$ & $\dot{m}_{2b} = \dot{m}_{3b}$	$\dot{m}_4 h_4 + \dot{m}_{2b} h_{2b} = \dot{m}_9 h_9 + \dot{m}_{3b} h_{3b}$	$\dot{I}_{E\_b} = \dot{m}_{2b} e_{2b} + \dot{m}_4 e_4 - \dot{m}_9 e_9 - \dot{m}_{3b} e_{3b}$
Binary -Turbine	$\dot{m}_{3b} = \dot{m}_{4b}$	$\dot{W}_{T\_b} = \dot{m}_{3b} h_{3b} - \dot{m}_{4b} h_{4b}$	$\dot{I}_{T\_b} = \dot{m}_{3b} e_{3b} - \dot{m}_{4b} e_{4b} - \dot{W}_{T\_b}$
Binary –Condenser *	$\dot{m}_{1b} = \dot{m}_{4b}$	$\dot{Q}_{C\_b} = \dot{m}_{4b} h_{4b} + \dot{m}_{1b} h_{1b}$	$\dot{I}_{C\_b} = \dot{m}_{4b} e_{4b} - \dot{m}_{1b} e_{1b}$
Binary-Pump	$\dot{m}_{1b} = \dot{m}_{2b}$	$\dot{W}_{P\_b} = \dot{m}_{2b} h_{2b} - \dot{m}_{1b} h_{1b}$	$\dot{I}_{P\_b} = \dot{m}_{1b} e_{1b} - \dot{m}_{2b} e_{2b} + \dot{W}_{P\_b}$
* the exergy transfer to the cold sink is considered as waste or destructed exergy			

Net exergy transfer by heat transfer at temperature T, is calculated by:

$$\dot{E}_Q = \sum \dot{Q}(1 - \frac{T_0}{T}) \tag{4}$$

And specific flow exergy is given by:

$$e = h - h_0 - T_0(s - s_0) \tag{5}$$

Table 3 presents the mass, energy balance and exergy destruction equations for each component of the cycle.

3.2 Energy and Exergy Efficiencies

Combined flash binary cycle is not a closed cycle process, and the working fluid in the steam cycle does not return to its initial condition at the end of the cycle. Although, by considering the highest available heat from the working fluid with respect to the dead state, it is possible to define the first law thermal efficiency of the cycle. The energy efficiency (first law efficiency) of a geothermal power plant may be expressed as [6 and7]:

$$\eta = \frac{\dot{W}_{T\_b} + \dot{W}_{T\_s} - \dot{W}_{P\_b}}{\dot{m}_1(h_1 - h_0)} \tag{6}$$

The expression in the denominator is the maximum energy input to the power cycle with respect to the environmental state. For the exergy effectiveness (second law efficiency), the exergy of the geofluid with respect to environment dead state is considered the input to the cycle.

$$\varepsilon = \frac{\dot{W}_{T\_b} + \dot{W}_{T\_s} - \dot{W}_{P\_b}}{\dot{E}_1} \tag{7}$$

While first law analysis considers the quantity of energy interactions in different parts of the cycle as well as the whole system, second law analysis evaluates each component's quality of energy interaction. By exergy analysis of the cycle, it is possible to highlight the processes with highest exergy destruction.

4 DESIGN PARAMETERS

4.1 Pinch Point

The place in the heat exchanger where hot and cold fluids experience the minimum temperature difference is called the pinch point. Fig 4 shows the possible pinch point locations in typical binary evaporator heat exchangers. In a typical heat exchanger design, the process starts from the pinch temperature difference (T_{pp}) and using heat transfer equations, the required contact area of the heat exchanger is calculated. A small T_{pp} value would result in high heat exchanger effectiveness, but would require a very large heat exchanger area, which is not economically viable. As shown in Fig 4, if the working fluid is super-heated (wet fluids) there are two points where the pinch may occur, and both of these should be checked for design purposes.

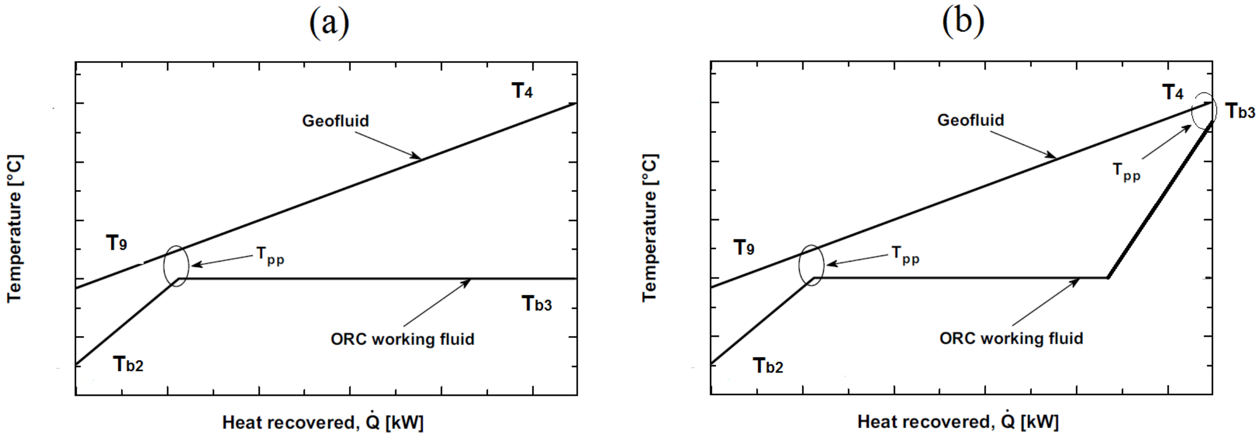

Figure 4: Temperature changes inside the evaporator (a) saturated vapor output (b) superheated vapor output

4.2 Separator Pressure

Flashing pressure (separator pressure) is an important design parameter of the cycle. Higher separator pressure results in lower steam glow rate into the steam turbine, but the steam would be at a higher temperature and possess higher workability. By lowering the separator pressure, the steam mass flow rate to the turbine would increase but the flow would have lower workability. In a single flash cycle the optimum separator temperature (saturated temperature corresponding to saturated pressure) to produce the maximum power is approximately the average of saturated input liquid and condenser temperatures [3 and 4]:

$$T_{sep} = \frac{T_{geo\_in} + T_{C\_s}}{2} \tag{8}$$

But in the combined flash-binary cycle, changes in separator pressure would also alter the temperature and flow rate of the geofluid into the binary evaporator. Hence for each case, the separator pressure value should be selected in a manner that gives the highest total work output of the cycle.

4.3 ORC Working Fluid

Selection of the ORC working fluid is a key parameter in a binary cycle design. Design of the cycle equipment, operation condition, efficiency and environmental impact are determined to a large degree by the selected working fluid characteristics. Many studies of the effect of working fluid selection on ORC efficiency are reported in the literature [6, 7, 14-16]. By increasing the awareness of the negative effects of some refrigerants on the environment and putting worldwide limitation on specific substances, these studies are increasingly concentrated on environmentally friendly working fluids in recent years.

Table 4 presents a list of ORC working fluid candidates with their considered critical temperature and pressure for this study. These fluids have been used as preselected options in this study. Refprop 9.0 database has been used as reference for thermo physical properties of working fluids in this study.

Table 4: ORC working fluid options

Name	Tcr [C]	Pcr [MPa]	Type
Propyen	129.23	5.62	Wet
R-152a	113.26	4.52	Wet
R-32	78	5.78	Wet
HC-270	125.15	5.58	Wet
R-134a	101.06	4.06	Wet
R-142b	137.42	4.06	Isentropic
R-21	178.33	5.18	Isentropic
Isobutane	134.66	3.63	Dry
Butane	151.89	3.8	Dry
FC-4-1-12	147.41	2.05	Dry

Although some of the options are flammable, as long as adequate precautions are taken this is not a problem. Also, due to low working temperatures there is no concern for self-ignition.

Historically, the most popular working fluids for ORC systems were refrigerants. CFC refrigerants like; R11, R12, R113 and R114 have been used widely for many years in industry. The Montreal protocol makes restrictions on using and producing substances from CFC, and HCFCs [11]. Refrigerants containing chlorine and/or bromine atoms lead to ozone layer depletion. These components are extremely stable, and when released to the atmosphere, they ultimately reach the upper atmosphere. There they break down and the chlorine reacts with existing ozone, depleting ozone concentration [12]. Therefore, working fluids with chlorine are phased out of the option list. Environmental impacts of using different working fluids should also be taken into consideration. The main parameters that should be considered are Ozone Depletion Potential (ODP), Global Warming Potential (GWP) and Atmospheric Lifetime (ALT).

Depending on the slope of saturated vapor line in the T-s diagram as shown in Fig 5, fluids can be classified as dry, isentropic and wet (with positive, infinite and negative slopes respectively).

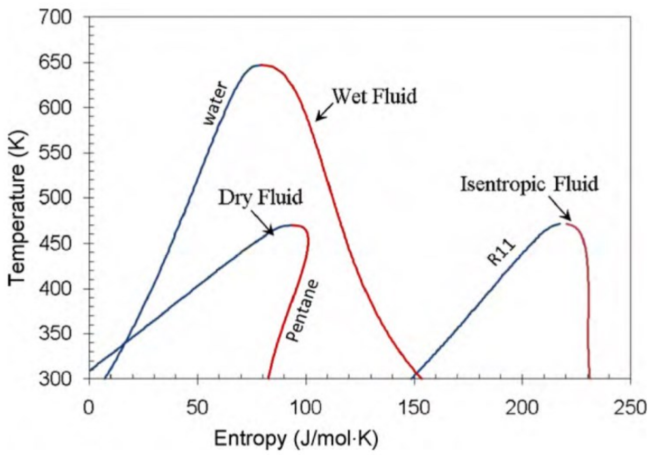

Figure 5: Three classes of working fluids; Dry, Isentropic and wet [16]

Dry and isentropic working fluids are more favorable for ORC systems, because the working fluid is super-heated after isentropic expansion in the turbine. Therefore, problems occurring from formation of liquid droplets at the final stages of the turbine are eliminated.

The maximum acceptable temperature of the working fluid is one of the screening parameters. At temperatures close to the critical point, working fluid conditions are unstable. Any small change in temperature can cause large variations in fluid condition. Hence, the maximum temperature that the working fluid experiences should be reasonably lower than its critical temperature. There are different methods to determine the maximum allowable working fluid temperature. In this paper, for wet and isentropic fluids the T_{max} is considered 10 C lower than critical temperature, which was proposed by Garcia-Rodeiguez [13]. For dry fluids, the method proposed by Rayegan and Tao [14] is used. In this method, the temperature for which the slope of the saturated vapor line in T-s diagram is infinity is calculated (point A Fig 6). To find T_{max}, from this point the temperature is increased until the quality of the working fluid in the expander falls lower than 99%.

The condensing temperature is another factor in working fluid selection. For the ORC, the condensing temperature of 30 C is considered in this study. The minimum acceptable pressure inside the condenser is reported as 5kPa in different references [14]. Therefore, fluids with corresponding saturated pressure lower than 5 kPa at 30 C are removed the potential working fluids.

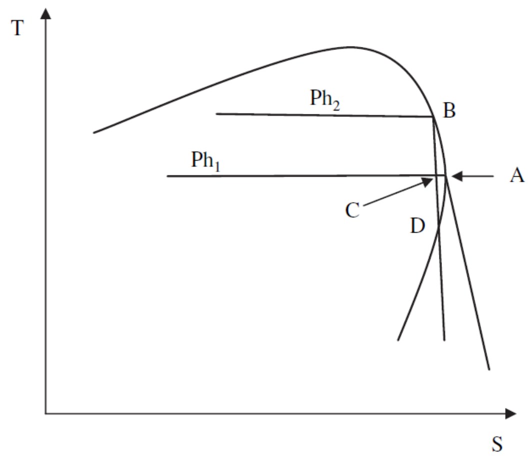

Figure 6: Max allowable temperature of dry fluids [14]

Table 5 represents the final ORC working fluid candidates considered in this study, with their higher and lower temperature limitations in the evaporator and condenser of the ORC. To make the comparison between different types of working fluids (Dry, Isentropic and Wet), only fluids with similar T_{max} are considered to reduce the effect of other parameters.

Table 5: Practical temperature limits for the ORC working fluid candidates

Name	T_{ev} [C]	T_{co} [C] *
Propyen	119.23	25
R-142b	127.42	25
Isobutane	120.32	25

5 RESULTS AND DISCUSSION

Different parameters may be considered for comparison of working fluid effects on system performance. The total work output of the system, which is the main factor in both energy efficiency (η) and exergy effectiveness (ε) of the combined cycle, is selected as the performance indicator in this study. Also, temperature profiles of hot and cold fluids in the binary evaporator are checked to ensure the pinch point limit is not violated.

Table 6: Design parameters of the combined flash-binary cycle.

Steam turbine efficiency [%]	80
Steam cycle condenser pressure [kPa]	13.5
Steam turbine minimum quality in [%]	90
Separator pressure minimum separator pressure [kPa]	100
Minimum geofluid temperature [C]	70
ORC turbine efficiency [%]	80
ORC pump efficiency [%]	85
ORC evaporator T_{pp} [C]	5
ORC condenser temperature [C]	30

In general, superheating in ORC increases the thermal efficiency of the cycle with a very low slope, but decreases the exergy efficiency of the cycle [15]. Hence, super-heated cycles are not recommended. In the present study, saturated Rankine cycle is investigated in the case of dry and isentropic fluids. In the case of the wet fluids, the turbine exhaust quality is the limiting factor. Minimum turbine quality is set to 90%. If the quality at this point falls below the lower limit, fluid should be superheated at the evaporator. Also, a higher pressure ratio in ORC leads to higher thermal efficiency. Therefore it is preferred to expand higher (evaporator pressure) and lower (condenser pressure) limits of the cycle, but there are some practical restrictions. In the evaporator, the temperature of the working fluid cannot go higher than the maximum available temperature (T_{max}); additionally, the minimum pinch point temperature difference should be satisfied in all sections of the evaporator. Condenser temperature is considered 30 C in this study for the ORC, which is 5 C higher than environmental temperature.

To prevent air from being sucked inside the separator, the minimum separator pressure limit is set to 100 kPa. The minimum condenser pressure in the steam cycle is set to 13.5 kPa, as mentioned in reference [4]. Also, to prevent the scaling problems in the piping system and evaporator, heat exchanger minimum geofluid temperature is set to 70 C.

Calculations have been performed for different geofluid temperatures at the production well. To make the comparison meaningful, some parameters have been selected as fixed values. In particular, in regards to the mass flow rate of geofluid ($\dot{m}_1$), the chosen value is equal to 1kg/s and the fluid is at saturated liquid conditions for all cases. Sensitivity analysis has shown that $\dot{m}_1$ has only a scale effect on the extensive variables, while it does not have any effect on intensive variables (i.e. temperature, pressure). Hence, the obtained results can be considered general for other mass flow rates too. Other parameters are considered fixed as presented in Table 6.

Figure 7 and Figure 8 show the effect of separator pressure change on the flash steam cycle and three different types of working fluids, for a specific geofluid temperature. As expected, maximum work output for the flash steam cycle occurs at a pressure corresponding to saturated temperature, close to the average of hot liquid and condenser temperatures. Moreover, for the binary part of the system, higher flash separator pressure results in an increase of both mass flow rate and geofluid temperature to the evaporator. Therefore, higher separator pressure always results in higher work output in the binary part of the combined cycle. By the way, the increase in rate of work output with respect to separator pressure increase is not constant. As the binary working fluid reaches its maximum allowable temperature (T_{max}), the rate of work output line declines. The reason for this is the higher pinch temperature in the evaporator, which increases exergy losses in the system. Fig 8 shows the total work output for the flash-binary cycle as a function of separator pressure. Adding the binary cycle to the flash steam cycle has resulted around 62% higher work out put in the system.

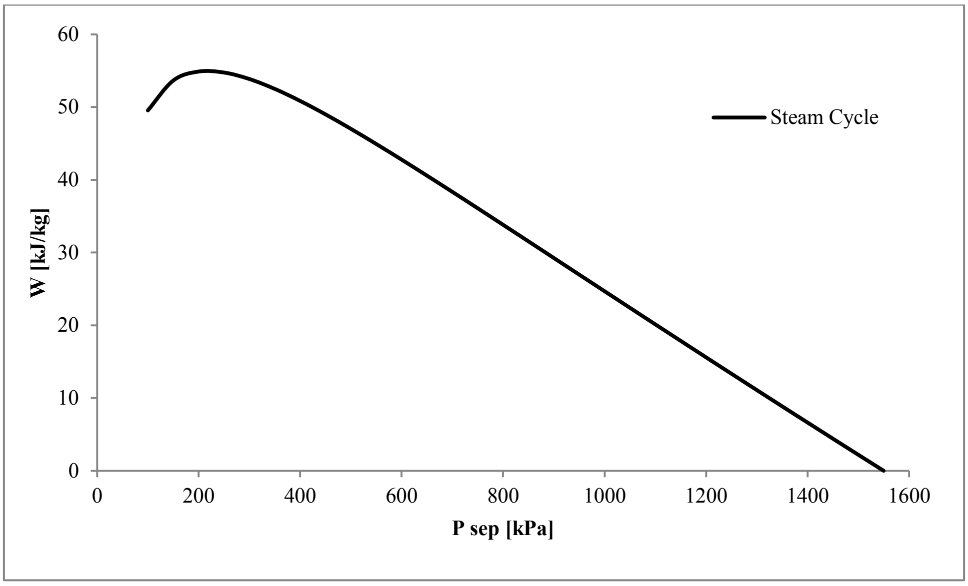

Figure 7: Steam cycle work out put for m_1=1 kg/s geofluid flow rate at T_1=200 C as a function of separator pressure

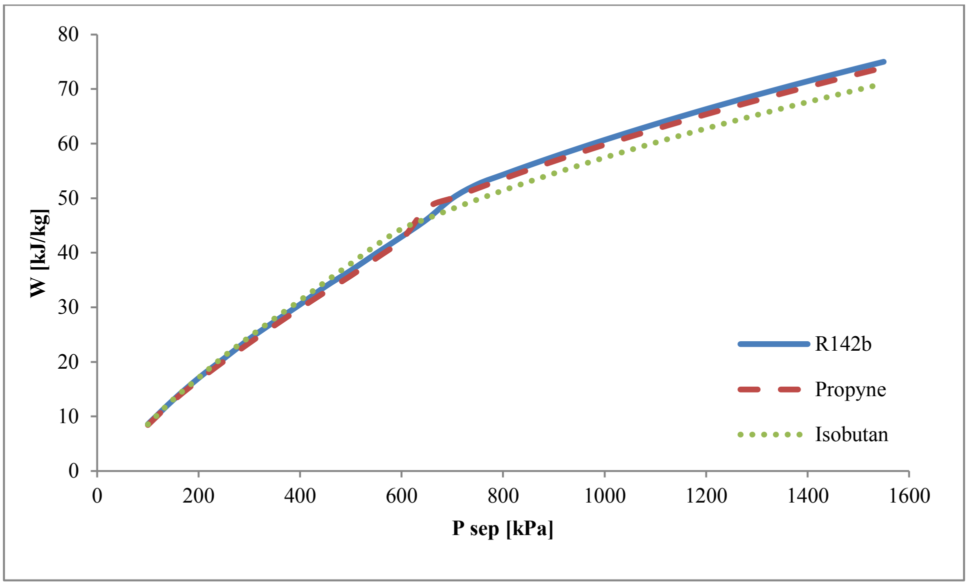

Figure 8: Binary cycle work out put for m_1=1 kg/s geofluid flow rate at T_1=200 C as a function of separator pressure

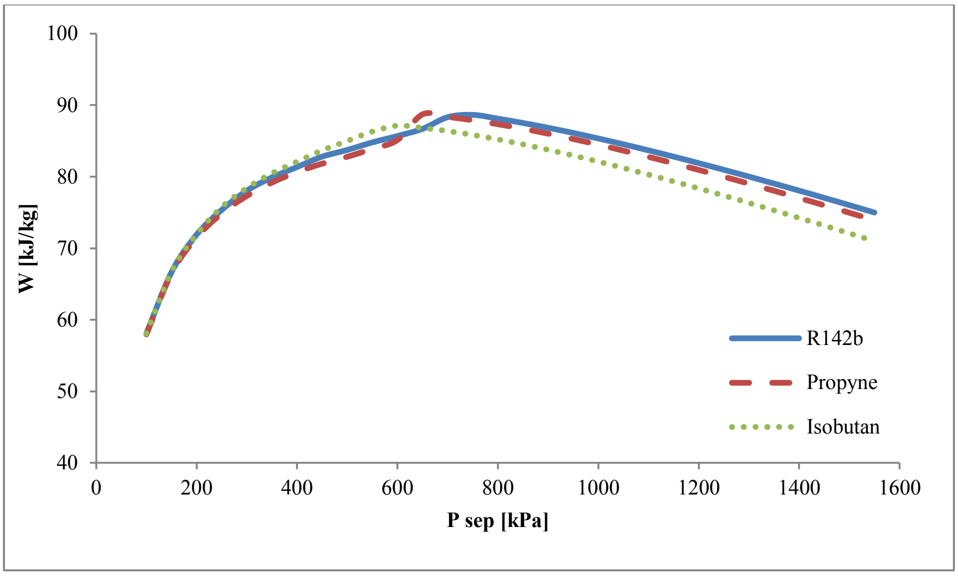

Figure 9: Combined flash-binary cycle total work for m_1=1 kg/s and T_1=200 C as a function of separator pressure

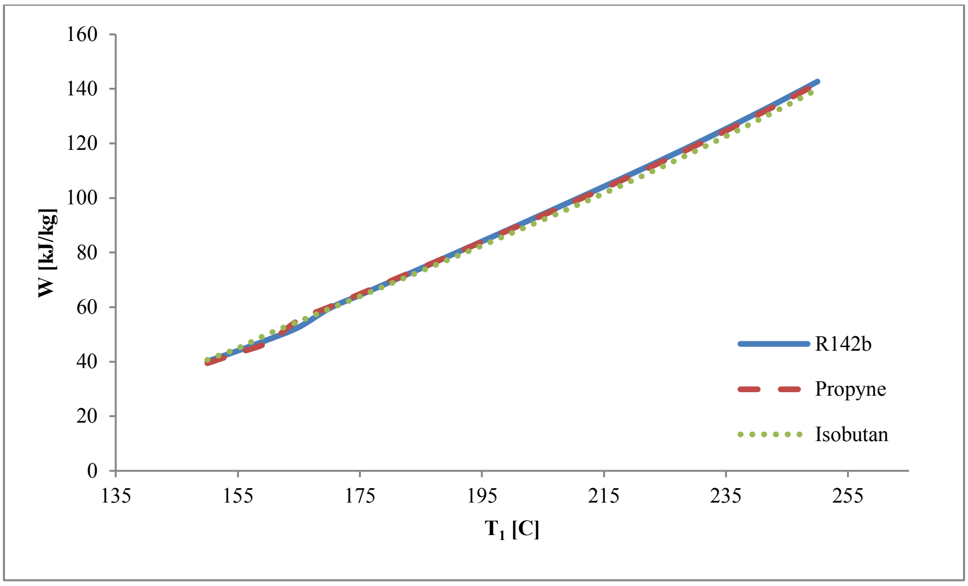

Figure 10: Combined flash-binary cycle work out put for m_1=1 kg/s geofluid flow rate at different extraction temperatures

Geothermal power generation stations are considered costly. However, increasing the power output of the system by 62% without adding to exploration and drilling expenses makes the combined flash binary cycle a viable option in suitable fields. As shown in Fig. 9, the highest total work output is when Propyne is utilized as the working fluid. Propyne is a wet fluid and this shows that it is not reasonable to eliminate wet fluids from potential working fluids for binary cycles as done by some references [14].

Figure 10 shows the maximum theoretical work output of the combined flash-binary cycle for different geofluid temperatures. It is shown that the total work output of the cycle is a linear function of the input temperature. Also there are small differences between wet, isentropic and dry working fluids. The reason is that the working fluids are selected because their T_{max} are close to each other.

Figure 11 is a graphical representation of exergy balance of the optimized combined flash-binary cycle for T_1=200 C. All the losses in steam cycle and binary cycle subsystems are accumulated in one entry. Table 7 presents the distribution of the initial exergy in different parts of the cycle for T_1=200 C. The highest source of exergy destruction in the system is the binary cycle evaporator with 12-13 percent of total exergy wasted in the process.

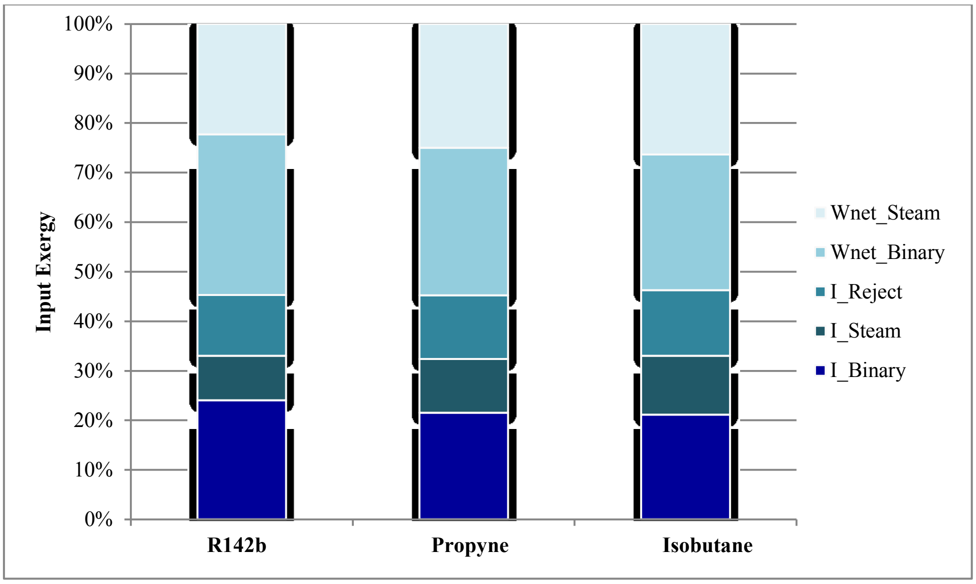

Figure 11: Destination of the initial hot geofluid exergy content for T_1=200 C

Table 7: Exergy destination of hot geofluid in the cycle components for T_1=200 C

	R142b	Propyne	Isobutane
E_in [kW]	162.02	162.02	162.02
I_reinjected [kW]	19.84	20.86	21.41
W_steam [kW]	36.08	40.57	42.77
W_net-Binary [kW]	52.59	48.15	44.36
Binary sub-cycle			
I_cond [kW]	5.41	5.05	5.63
I_eva [kW]	18.14	15.29	16.7
I_pump [kW]	1.11	1.2	0.91
I_tur [kW]	14.3	13.36	11.52
Flash steam sub-cycle			
I_cond [kW]	4.54	5.33	5.75
I_cool [kW]	0.21	0.24	0.26
I_sep [kW]	3.95	5.41	6.32
I_tur [kW]	5.84	6.57	6.92

6 CONCLUSIONS

A thermodynamic analysis, using energy and exergy concepts, is presented for a combined flash-binary geothermal power generation cycle. Separator pressure is found to be the main design parameter of the system. Changing the pressure in the flash separator will change the amount of work output of each sub system of the combined cycle. There is an optimum separator pressure for each geofluid condition at extraction wells.

Also presented is a comparison among three types of working fluids for binary cycle in a combined flash-binary geothermal power cycle. Although Propyne (as a wet fluid) gives the highest efficiency, the difference between the three different types of working fluids is less than 2% at various geofluid temperatures at the extraction well. This is explained by the close values of T_{max} for all three candidates. Therefore, eliminating wet fluids from the options is not suggested, especially if there is a cost privilege to this option.

Applying the exergy analysis to the cycle components shows that the highest place of exergy destruction in the system is the evaporator of the binary cycle. Therefore, one main place to improve the system efficiency is the evaporator. Using mixture working fluids or super critical ORC is one way to fit hot and cold fluid temperature profiles in the evaporator and reduce the exergy destruction.

It is worth mentioning that the calculations presented in this paper are based on the assumption of saturated liquid extraction from production wells. Normally, in a hydrothermal geofield the condition of the geofluid at the production well is a mixture of steam and liquid. Therefore, in existing flash binary geothermal power plants, the power production share of the steam cycle is higher with respect to our calculations. The ratio of the steam cycle power to the generated ORC power depends on the temperature and enthalpy of the geofluid entering the separator.

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