

# Experimental Design of Wellbore Heat Exchanger in Binary Optimization for Low - Medium Enthalpy to Utilize Non-Self Discharge Wells in Indonesia

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## ABSTRACT

Indonesia has a large potential in geothermal energy resources. However, the development of the geothermal industry in Indonesia is still focused on high-temperature or high-enthalpy fields. There are 246 geothermal areas in Indonesia which classified into low to medium enthalpy, which is potentially able to be developed with binary system. In this paper, an experimental design was studied by modifying the Organic Rankine Cycle at the wellhead generating unit. This design utilizes non-self-discharge wells to be used as "heat exchanger" where the working fluid is pumped into the well through a U-pipe and heated by the geothermal fluid. The outflow of the working fluid from the U-pipe is expected to be steam-phased and directly flowed into the turbine. Several working fluids have been studied to determine maximum power generation at a certain reservoir temperature. In this study, pressure and temperature profile of well XX-02 is used for case study. The working fluids evaluated were Isobutane, Propane, Isopentane, and Butane. The maximum generation output was found in Isobutane and Butane with generation output 294 kW and 241 kW respectively, and flow rate needed was 5 kg/s. This un-significant generation give flexibility in working fluid selection. Sensitivity analysis for temperature decline conducted to evaluated the feasibility of the design for 30 years project lifetime. Economics analysis was also have been done to study the feasibility of the design and electric price produced. ORC (Organic Rankine Cycle) price used in economic analysis is the average price of ORC (3,000 USD/kWh), resulting electricity price produced is 7.76 USD cents/kWh. This results conclude that Wellbore Heat Exchanger design is feasible to improve non-self dischare well to be economically developed.

## 1. INTRODUCTION

Geothermal well drilling activities, especially at the exploration stage, have a low success ratio (SR) of 60% to 70%. The failure of this drilling activity cannot be separated from the lack of data owned (uncertainties factor). Wells that are unable to produce geothermal fluids from the reservoir are called non-self-discharging wells. In non-self-discharge wells, some stimulation efforts are expected to produce geothermal fluids. However, well stimulations are not always able to change the productivity of the wells to stimulate geothermal fluid from the reservoir flows. One of the contributing factors to the unsuccessful of well stimulation is that the temperature of the brine fluid in the well is not high enough and/or the height of the water column in the well is quite thick.

Productive wells that produce medium temperature fluids (less than 225°C) can be utilized using binary cycles. However, if the well is not able to naturally flows the fluid (non-self-discharge) it will usually be abandoned because it has no economic value, even if suction pump have been used to extract the fluids. Drilling of geothermal wells on average requires a very high cost, about 5 to 9 million USD (65 to 117 billion rupiahs). So the purpose of this study was conducted is to analyse the feasibility of increasing the value of non-self-discharge wells by modifying binary cycles. The modification design is expected to produce small-scale power generation and become an experimental design for medium temperature geothermal fields in other areas for future plant.

The key point that will be discussed in this paper are study of convection and conduction heat transfer from brine to working fluid within U-pipe, fluid mechanism to determine pressure and temperature profile of working fluid, selection of several types and flowrate of working fluid for optimum generation, and pros-cons and financial analysis of WHE design.

## 2. BASIC THEORY

### 2.1 Conduction and Convection Heat Transfer on U-pipe

In this design, conduction heat transfer that would be considered is the heat ransfer of the fluid when going through the U-pipe. The heat transfer follow the equation

$$Q = U_{cond} \cdot A \cdot \Delta T \quad (1)$$

with  $Q$  is heat flux (Watt),  $U_{cond}$  is overall conduction heat transfer coefficient ( $J/m^2.K$ ),  $A$  is cross-section area in ( $m^2$ ), and  $\Delta T$  is temperature difference between inner side and outer side of U-pipe (K). The conduction heat transfer coefficient with the cross-section area  $A$  of U-pipe follow the equation

$$\frac{1}{U_{cond} \cdot A} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi \cdot k \cdot L} \quad (2)$$

with  $r_2$  and  $r_1$  is the outer radius and inner radius of the U-pipe, respectively.  $k$  is heat conductivity of the U-pipe material, and  $L$  is the length of the U-pipe.

Convection heat transfer occurs in the working fluid inside the U-pipe. The convection heat transfer equation will be used to find heat transfer from the inner part of U-pipe that will be transferred to the working fluid. The convection heat transfer in the working fluid follow the equation

$$Q = U_{conv} \cdot A \cdot \Delta T \quad (3)$$

with with  $Q$  is heat flux (Watt),  $U_{conv}$  is overall convection heat transfer coefficient ( $J/m^2.K$ ),  $A$  is cross-section area in ( $m^2$ ), and  $\Delta T$  is temperature difference between inner side of the U-pipe and the working fluid entering the U-pipe (K). The convection heat transfer coefficient of working fluid with the cross-section area  $A$  of U-pipe follow the equation

$$\frac{1}{U_{conv} \cdot A} = \frac{1}{h \cdot (2\pi \cdot r_1^2 \cdot L)} \quad (4)$$

with  $h$  is convection heat transfer coefficient of working fluid ( $W/m^2.K$ ),  $r_1$  is inner radius of U-pipe, and  $L$  is the length of U-pipe. convection heat transfer coefficient of working fluid follow the equation

$$h = \frac{N_u \cdot k}{D} \quad (5)$$

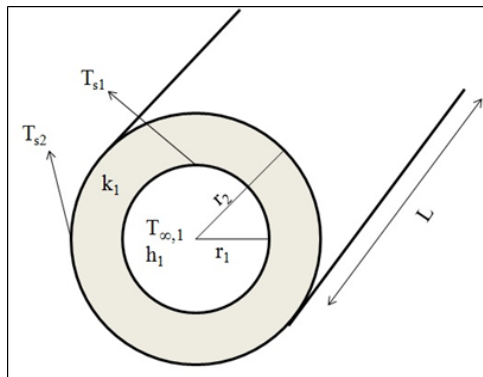
with  $Nu$  is Nusselt number,  $k$  is heat conductivity of working fluid ( $W/m.K$ ),  $D$  is the inner diameter of the U-pipe (m). Using the Dittus-Boelter equation,  $Nu$  could be written as

$$N_u = 0.023 \cdot Re^{4/5} \cdot Pr^n \quad (6)$$

with  $Re$  is Reynold number,  $Pr$  is Prantdl number, and  $n$  is a constant. Prantdl number value can be determined using thermodynamics table of working fluid. The value of  $n$  is 0.4 if  $Re > 10^5$  and 0.3 if  $Re < 10^5$ . Reynold number for a cylindrical pipe follow the equation

$$Re = \frac{\rho \cdot v \cdot D}{\mu} \quad (7)$$

with  $\rho$  is the density of working fluid ( $kg/m^3$ ),  $v$  is fluid velocity (m/s),  $D$  is inner diameter of U-pipe (m), and  $\mu$  is fluid viscosity (Pa.s). The heat transferred from brine to working fluid can be determined by finding the value of heat transfer coefficient from: brine to U-pipe (free convection), outer part to inner part of U-pipe (conduction), inner part of U-pipe to working fluid (forced convection). The schematic cross-section of the pipe is shown in figure 1.



**Figure 1: The schematic cross-section of U-pipe where  $T_{s2}$  is the outer temperature of U-pipe,  $T_{s1}$  is inner temperatur of U-pipe, and  $T_{\infty,1}$  is the working fluid temperature flowing into the U-pipe.  $k_1$  is heat conductivity of U-pipe,  $r_2$ ,  $r_1$ , and  $L$  is the outer radius, inner raidus, and length of U-pipe segment, respectively.**

In this design, the convection heat transfer from the brine is considered to be very high, so the brine and the outer part of the U-pipe is assumed to have the same temperature. Thus, the heat transferred in an U-pipe segment can be written as

$$Q = U \cdot A \cdot \Delta T \quad (8)$$

where  $U$  is the total heat transfer coefficient from the outer part of U-pipe into the working fluid.  $U$  consisted of conduction heat transfer coefficient from outer to inner part of U-pipe and convection of working fluid, that can be written as

$$U \cdot A = U_{cond} \cdot A + U_{conv} \cdot A \quad (9)$$

The heat transferred to the flowing working fluid follow the equation

$$Q = \dot{m} \cdot (h_2 - h_1) \quad (10)$$

where  $Q$  is the heat flux (kW),  $\dot{m}$  is the working fluid mass flow rate (kg/s),  $h_2$  and  $h_1$  is working fluid enthalpy after and before receiving heat transferred (kJ/kg). The working fluid enthalpy before receiving heat transferred can be determined with the initial pressure and temperature. If the mass flowrate of the working fluid and heat flux (from equation 8) has been found, we could determine the enthalpy of the working fluid out of the U-pipe segment (after receiving heat) to find the temperature of the working fluid leaving the U-pipe. In this design, the heat transfer in the U-turn of the U-pipe is neglected.

## 2.2 Bernoulli Law

In this design, the pressure difference of the working fluid flowing down to the well and up from the well is determined using Bernoulli Law. The equation is

$$P_1 + \rho_1 \cdot g \cdot h_1 + \frac{1}{2} \rho_1 \cdot v_1^2 = P_2 + \rho_2 \cdot g \cdot h_2 + \frac{1}{2} \rho_2 \cdot v_2^2 \quad (11)$$

where  $P_1$  and  $P_2$  is the working fluid pressure in the beginning and end point of U-pipe segment (Pa),  $\rho_1$  and  $\rho_2$  is the working fluid density in the beginning and end point of U-pipe segment ( $\text{kg/m}^3$ ),  $v_1$  and  $v_2$  is the working fluid velocity in the beginning and end point of U-pipe segment (m/s),  $h_1$  and  $h_2$  is the depth of well in the beginning and end point of U-pipe segment, and  $g$  is gravitational acceleration ( $\text{m/s}^2$ ).

## 2.3 Organic Rankine Cycle (ORC)

This design is a modification of the basic Organic Rankine Cycle. The basic Organic Rankine Cycle scheme that is shown in figure 2.

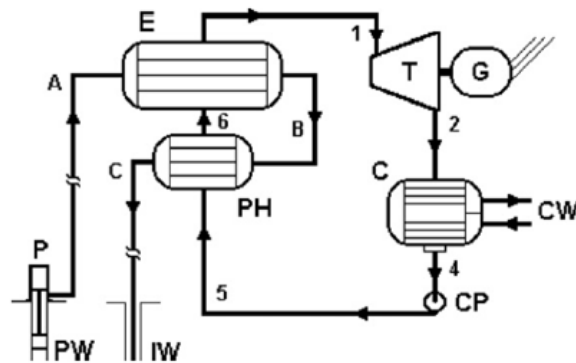
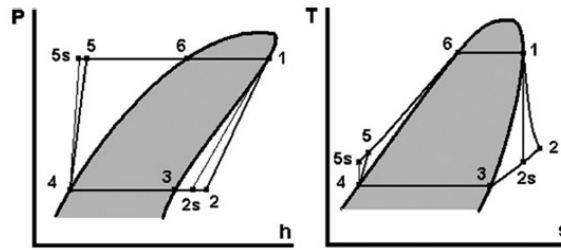


Figure 2: Basic Organic Rankine Cycle scheme (DiPippo, 2012).

The components of this system are:

- Pre-Heater, heats the working fluid using brine fluid.
- Evaporator, changes working fluids phase into gas using brine fluid.
- Turbine, produces electricity from moving turbine blades.
- Condenser, condenses and cools working fluid out from turbine using cooling water to increase enthalpy difference between turbine inlet and condenser, resulting higher electricity produced.
- Cooling Tower, cools the cooling water.
- Pump, pumps cooled working fluid back to pre-Heater.

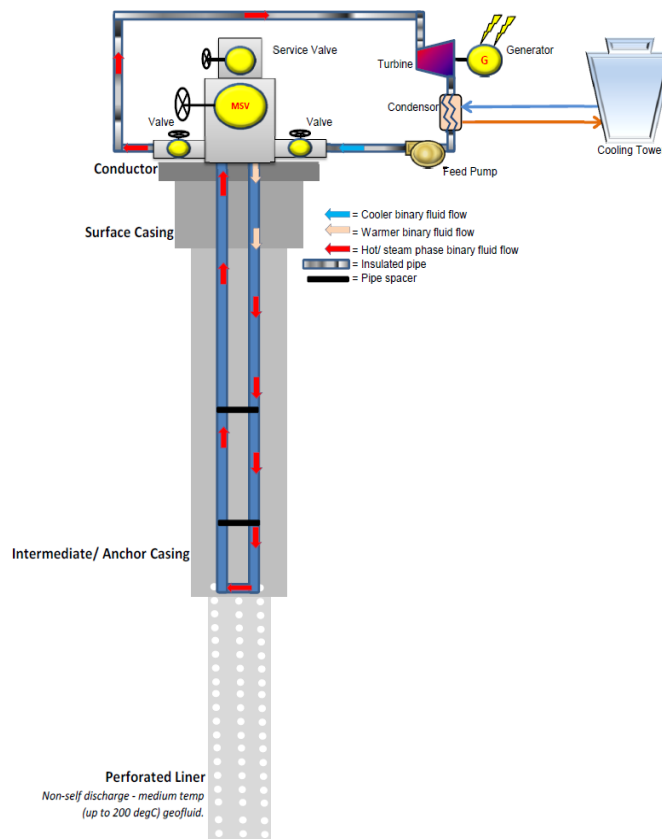
The pressure versus enthalpy (P-h) graph and temperature versus entropy (T-s) graph from the figure 2 cycle is shown in the figure 3.



**Figure 3: Pressure versus enthalpy (P-h) graph and temperature versus entropy (T-s) graph from the figure 2 cycle (DiPippo, 2012).**

### 3. ORC MODIFICATION

WHE (wellbore heat exchanger) is a modification of binary cycle by using the well itself as a heat exchanger. U-pipe is placed in the wellbore and working fluid pumped from feedpump to 2<sup>nd</sup> U-pipe inlet. As the working fluid travel inside the U-pipe, it is heated by brine outside of the U-pipe. Working fluid has lower boiling point and expected to be steam-phased while exiting the U-pipe outlet. This steam is then directly flowed and actuated the turbine which is coupled with the generator and generate electricity. Steam out from the turbine is condensed with non-direct contact tube by cold water from cooling tower and then pumped again by feedpump into the U-pipe. The WHE design shown in figure 4.



**Figure 4: The experimental design of WHE (Wellbore Heat Exchanger) from modification of binary cycle.**

### 3.1 Positive and Negative Impact of the Design

#### 3.1.1 Positive Impact

- a. As the design is only extract the heat but not the mass from the brine, then the pressure of the well is relatively constant. This constant pressure can avoid:
  - Cold water intrusion from marginal of the reservoir and lower the reservoirs temperature.
  - Changing in fluid characteristic due to temperature changing that lead to scaling in formation pore.

- Possibility of subsidence.
- b. There is no chemical contamination or thermal pollution in surface.
- c. Suit for medium enthalpy geothermal area with moderate demand of electricity
- d. Maximize the economic value of non-self discharge wells.
- e. Easier and quicker in installation, compact facilities and require small space. Fewer unit/instruments lower the operational and maintenance cost (O&M).
- f. Small scale power generation shortening the time to get the revenue.
- g. Not require injection wells.

### 3.1.2 Negative Impact

This WHE design is unsuitable for high seismic activity and magnitude. This condition can potentially damage the casing and U-pipe.

Challenges that are still need to be studied in the future such is that this design is only capable of generating small electricity and monitoring activities need to be done especially in handling and storage of the working fluids.

## 4. CASE STUDY

### 4.1 Working Fluid Selection

This research was conducted by using PT profile data from well XX-02 (non-self discharge well) to 2,100 m depth. The U-pipe will be placed in the wellbore to 1,050 m depth where the highest temperature zone is recorded (154.7 °C), as shown in figure 5.

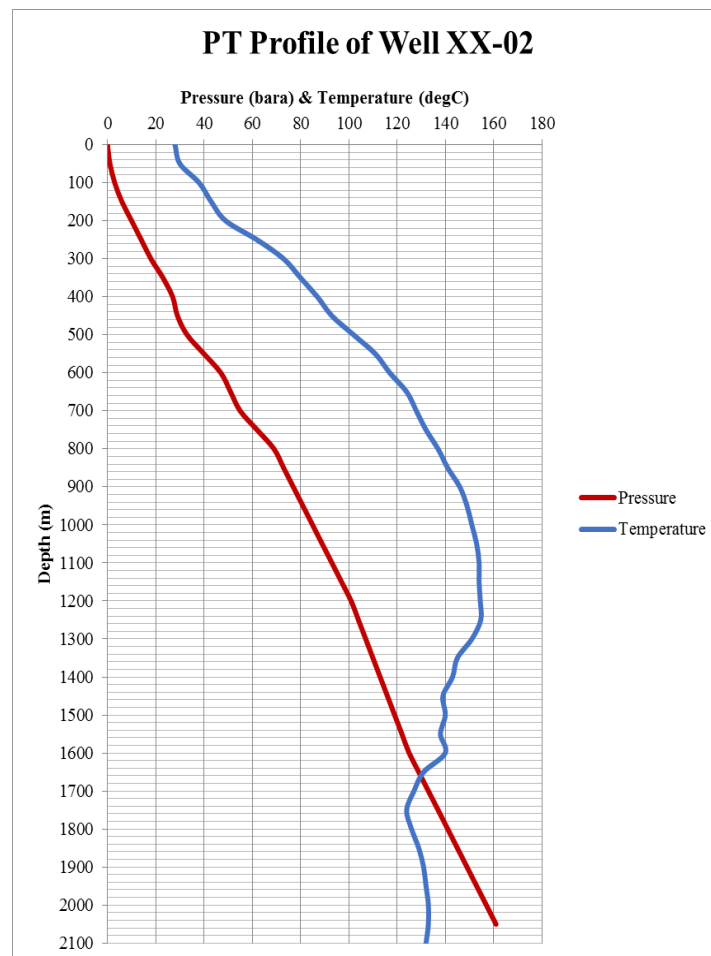


Figure 5: Pressure and Temperature (PT) Profile of Well XX-02.

The heat transfer from the brine fluid to the working fluid inside the 2" steel U-pipe is calculated for every 50 m intervals both conduction and convection in a 9-5/8" casing filled with full brine fluid. Assuming the effect of heat transfer between the U-pipe inlet (down) and outlet (up) can be neglected, and the working fluid pumped with 3 kg/s, 4 kg/s and 5 kg/s rate. For each type of working fluid used, we can determine the optimum fluid type and flowrate which can give the highest power generation. The working fluids to

be compared are Isobutane, Propane, Isopentane, and Butane. The simulation of heat transfer along the U-Pipe down and the U-Pipe up shown in table 1.

**Table 1: Power generation for each working fluids type.**

Working Fluid	Flowrate (kg/s)	Feed Pump Pressure (bara)	T.out U-pipe (degC)	TIP (bara)	Fluid phase	T.condenser (degC)	Q.turbine (kW)
Isobutane	3	3	75,95	10,22	Superheated vapor	25,53	169,62
		3,5	76,37	10,47	Superheated vapor	25,76	168,50
	4	3,5	74,51	8,16	Superheated vapor	25,17	250,75
		<b>5</b>	<b>3,5</b>	<b>71,21</b>	<b>7,56</b>	<b>Superheated vapor</b>	<b>25,76</b>
Propane	3	9,5	77,75	27,49	Superheated vapor	25,66	102,57
	4	9,5	76,10	24,85	Superheated vapor	25,14	163,64
	5	9,5	73,61	23,15	Superheated vapor	25,30	211,50
Isopentane	3	3	81,97	3,24	Superheated vapor	63,80	53,41
	4	3	81,72	2,93	Superheated vapor	64,19	80,08
	5	3	80,71	2,67	Superheated vapor	63,95	104,15
Butane	3	3	78,51	7,60	Superheated vapor	32,28	167,72
	4	3	77,54	7,09	Superheated vapor	32,86	229,60
	5	3	74,48	5,15	Superheated vapor	32,58	241,00

**4.2 Economical Analysis**

Economical study has been done to analyze if this design is economical and determine the feasible electricity price with a certain value of hurdle rate (RRR, rate of return) for the geothermal field time contract in Indonesia (30 years) . The power generation in this study will be using the output produced by the working fluid that has the optimum electricity produced, that is Iso-Butane with 5 kg/s mass flowrate. The total cost of this design is shown in table 2. The WHE components cost is using the basic ORC cost by 3,000 USD/kWh. The economical study to determine the feasible electricity price will use economical parameters shown in table 3. Note that some of this parameters (in bold) is assumptions from average parameters of geothermal field economical analysis, while others use the actual parameters in Indonesia.

**Table 2: The total cost of WHE design.**

No.	Elements	Value	Price / Unit \$000		
			Base	Total	
1	U-pipe	2100 m	343	377	
2	Non capital	1%	22	25	
3	WHE components	0.294 MW	3,000	882	
<b>Total (USD 000)</b>				1,284	

**Tabel 3: Economical parameters in Indonesia.**

No.	Parameter	Value
1	Capacity Factor	80%
2	Corporate Income Tax Rate	10%
3	Investment Tax Credit	5%
4	Value Added Tax	10%
<b>5</b>	<b>Wells Intangible Portion</b>	<b>80%</b>
6	Depreciation Life, Years	8
7	Depreciation rate	25%
8	Operation Cost , USD/kWh	0.01
9	Debt Ratio	70%
10	Interest Rate	12%
<b>11</b>	<b>Tenor, Years</b>	<b>15</b>
13	R R R (discount rate), %	12%

In well XX-02, with RRR 12%, the electricity price is 7.76 USD cents/kWh with IRR 15%, NPV 65,000 USD, and pay back period in 6 years after commercial. The national production cost average for electricity in 2017 is 12 USD cents/kWh. The electricity price for WHE is lower than the national electricity production cost average, thus, it can be safely said that this design is applicable in Indonesia or otherwise this design can be used as company CSR (Corporate Social Responsibility).

## 5. CONCLUSION

Conclusions of this study are:

1. This WHE design can optimize the productivity of non-self discharge wells.
2. In case study of well XX-02, WHE using Isobutane with 5 kg/s mass flowrate produce the optimum generation of 294 kWe.
3. Economic analysis concludes that WHE design is applicable in Indonesia. The WHE economical analysis in well XX-02 resulting in electricity price of 7.76 USD cents/kWh, which is lower than the national electricity production cost average. This design can also considered to be used as company CSR.

For further study, the hole geometry should be considered, because the maximum U-pipe size and spacing will be affected by it. The smaller the hole geometry, the smaller the maximum size of U-pipe that can be used. To get more accurate approach, the thermodynamics calculation should be done in a closer intervals, and the heat transfer thermodynamics in the U-turn of the U-pipe should also be considered. Further study of the reservoir and reservoir simulation near the area of the well should also be conducted to ensure that the heat flux from the geothermal system into the well is higher than the heat flux extracted for WHE generation.

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APPENDIX

