

## Design and Testing of n-Pentane Turbine for 2 kW Model of Binary Cycle Power Plant

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**Keywords:** geothermal, binary cycle, n-pentane, thermal waste, brine, turbine

### ABSTRACT

Design, manufacture and testing of a 2 kW n-pentane turbine were carried out. This project is dedicated to support the study of the 2 kW model of binary cycle power plant using geothermal brine as the energy source. Brine defined in this study is separated hot water at a separated steam geothermal plant. Currently, 100% of brine is injected into injection wells. Injected brine generally has a temperature higher than 100° C and mass flow rate of hundreds of tons/hour. A quick estimation showed the content of rejected energy is around several hundred million watts.

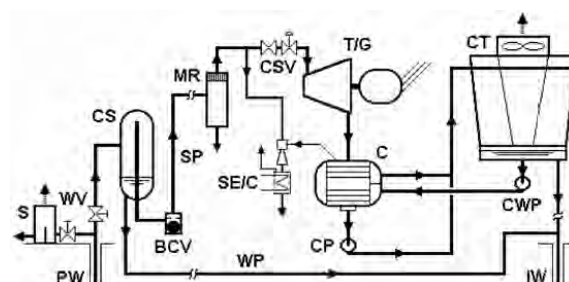
A binary cycle plant is an advantageous technology for recovering this energy waste. In this plant, thermal energy from brine is transferred via a heat exchanger to a working fluid for using in a fairly conventional Rankine cycle. The aim of the whole study is to recover the unused energy in order to optimize the efficiency of a conventional geothermal power plant.

From a design analysis, the turbine used is a single-stage impulse turbine with five convergent nozzles, 382.3 mm rotor diameter, 60 blades, 16.5 mm height of blade, and 10 mm width of blade. This turbine is designed to operate at 3000 rpm and 65.6% of turbine efficiency. The turbine test was conducted using BPPT's 2 kW system of research-scale-binary-cycle power plant. The experimental result shows that the turbine produced power of 1.2 kW successfully. It is 8.5% lower than the efficiency of the original design. Therefore it is reasonable to conclude that the turbine is well designed.

### 1. INTRODUCTION

In the conventional geothermal power plant as in Figure 1, almost 100% of the brine from a separator is re-injected into the earth in hot condition through a reinjection well. A brine of around 150° C with a mass flow rate at hundreds of tons per hour has the potential of producing an estimated several hundred MW of electric power.

Similarly, a survey conducted in industries such as chemical and petrochemical industries, energy industries and so on, indicates that potentially hundreds of MW of hot waste can still be extracted for other purposes. Table 1 gives an approximate estimate on the potential utilization of hot waste from several types of industries.



**Legend:** S – Silencer, PW – Production Well, WV – Wellhead valve, CS – Cyclone Separator, BCV – Ball check valve, SP – Steam Piping, MR – Moisture Removal, CSV – Control and Stop Valve, SE/C – Steam Ejector/Condenser, T/G – Turbine/Generator, C – Condenser, CP – Condensate Pump, CT – Cooling Tower, CWP – Cooling Water Pump, WP – Water Piping, IW – Injection Well

**Figure 1: Schematic of a Direct Steam Geothermal Power Plant (Thomas C. Elliot et al., 1989)**

Temperature conditions of hot waste fluid vary from 60° C until 400° C. Furthermore, there are also numerous types of industries that produce hot waste fluid. Therefore, technology choices for utilizing hot waste energy must be appropriately chosen in accordance with condition and requirement of the selected industry.

**Table 1. Estimation of the potential utilization of hot waste industry (Source: BTMP-BPPT)**

Type of Industry	Hot Waste (MW)	Temp. of Fluid (°C)
Plastic industry	17 (hot waste)	200
Chemical industry	12.5 (hot waste), utility & process	170 - 400
Steam electricity power plant 400 MW	4.3 (CBD*), 83.5 (hot waste)	355, 225
Geothermal electricity power plant 1x110 MW	12.8 (brine)	180

\*) CBD= Continuous Blowdown

### 1.1 Utilization of Hot Waste (Brine) from Conventional Geothermal Power Plant

This paper focuses on design, manufacture and testing of a n-pentane turbine for the 2 kW binary cycle power plant. This power plant is intended for a utilization model of hot waste brine from a separated steam geothermal power plant.

Indonesia is a country with the biggest geothermal resources in the world, with potential of around 27,000 MW, or equivalent to 40% of world reserves in geothermal potential. Currently, Indonesia focuses on developing high enthalpy geothermal wells. However, there are only a total of 1600 MW of geothermal power plants that are up and running. All of these operational geothermal power plants are utilizing separated steam technology, as in Figure 1, where 100% of brine from separator is reinjected back to earth through the reinjection well. The brine from separation process normally still contains thermal energy of around several hundred MW.

As an example, a separated steam geothermal power plant of 1x110 MW in West Java, produces brine with flow rate of 270 tons/hour at 180° C and 11 bar<sub>a</sub>. If the brine temperature is assumed reducible to 140° C to avoid scaling in the system, then thermal energy that can be utilized is around 12.8 MW, or about 10% of electric power being produced by the plant. Using the assumption above, Table 2 shows estimation on hot waste brine potentials with respect to the conventional geothermal power plants operating in Indonesia.

**Table 2. Estimation on hot waste brine (assumed reduced temperature at 140°C), [source: BPPT]**

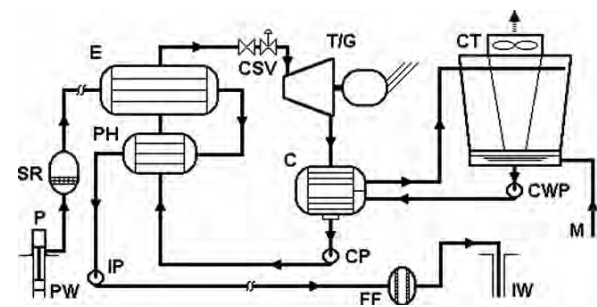
Location	Power (MW <sub>e</sub> )	Brine			
		Flow rate (ton/hr)	T.(°C)	P. (bar <sub>a</sub> )	Heat (MW <sub>t</sub> )
Sibayak	1 x 2	471	180	11	22,7
Wayang Windu	1x110	270	180	11	26,5
Dieng	1 x 60	640	190	13	13,01
Lahendong	2 x 20	550	180	11	38,70
Awibengkok - Gn. Salak, unit 1, 2, 3	380	6966	173,4	8,5	279,64
Awibengkok - Gn. Salak, unit 4, 5, 6	365				

During today's energy crisis, the hot waste becomes very important to be utilized as an alternative energy source in order to increase total system efficiency. Moreover, Indonesia has tremendous low enthalpy geothermal energy resources, which have not been optimally utilized.

A binary cycle geothermal power plant is an appropriate technology for taking advantages of hot waste or others thermal energy that has not yet been utilized, especially low temperature geo fluid (150° C). In the binary cycle power plant, thermal energy from brine or geo fluid is moved through a heat exchanger to the working fluid used in a Rankine cycle. The flow chart for a typical binary cycle geothermal power plant is shown in Figure 2 [Thomas C. Elliot et al. 1989].

Energy cycle components in Figure 2 consist of pre-heater, evaporator, a set of control valves, turbine, condenser and filling pump. In addition, there is a water cycle that is used

as condenser cooler. If a wet cooling system is employed, water supply is necessary on location to provide source for makeup water.

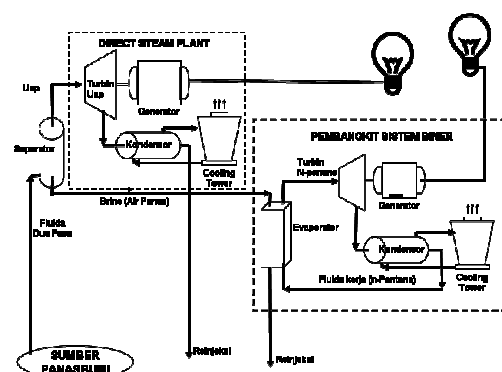


**Legend:** PW Production well, P (Well) Pump, SR Sand Remover, E Evaporator, PH Pre-heater, IP Injection Pump, FF Final Filter, IW Injection Well, CSV Control and Stop Valve, T/G Turbine/Generator, C Condenser, CP Condensate Pump, CT Cooling Tower, CWP Cooling Water Pump, M Make-up Water

**Figure 2: Schematic of the binary cycle geothermal power plant (Thomas C. Elliot et al.1989)**

In preparing the plant design, a selection of working fluid for the system must be considered. The main focus on the working fluid is having a good thermodynamic property, a compatibility with geofluid characteristic used, especially in temperature. Other important considerations are critical temperature and pressure, characteristics in the saturated condition (flammable, toxicity, ozone depleting potential, heat transfer coefficient, sonic speed on exhaust turbine and price). Hydrocarbons such as isobutane, isopentane and propane are good working fluids, in addition to some refrigerants. An appropriate selection of working fluid will result in a high efficiency system, as well as safe and economical in operation.

BPPT developed the 2 kW binary cycle plant with a heat source from brine, for an implementation study of geothermal power plant utilizing a binary cycle technology. Schematic of the model is shown in the Figure 3 [Bambang T. Prasetyo et al., 2008]. The power cycle from this system consists of an evaporator, a turbine-generator, a condenser and filling pump. The other equipment is a cooling tower to cool down working fluid in the condenser.



**Figure 3: Diagram of geothermal power plant binary cycle with brine as energy source**

In this model n-pentane is used as a working fluid. The study focused on plant design and manufacture, as well as system turbine performance test. The pilot plant testing is conducted at Wayang Windu geothermal Field, utilizing the brine from the plant at the reinjection point.

## 2. METHODOLOGY

### 2.1 System Design

Considering limitations, especially on the research budget, design of the 2 kW binary cycle geothermal power plant model has been fixed as in Figure 4 [Bambang T. Prasetyo et al., 2008].

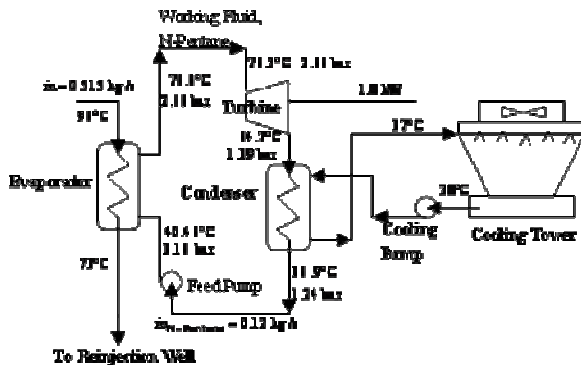


Figure 4; Diagram of binary cycle geothermal power plant model and design parameter

Heat exchangers in the 2 kW binary cycle plant model include an evaporator, a condenser and a cooling tower. The wet cooling tower is chosen to cool down n-pentane as condenser coolant. The chosen evaporator and condenser are shell and tube types. These types are preferred since they are simpler in terms of design and easily manufactured locally in Indonesia.

The shell and tube type heat exchanger consists of a bundle of tubes which are positioned inside the cylindrical shell. The tubes are mounted on tube sheets at both ends, which separate fluids within the shell from that flowing inside the tubes.

The evaporator and condenser thermal designs have been done using the temperature-enthalpy diagram (T-h) method. This method is selected since both heat exchangers involve phase change of the respected working fluids, so that conventional approach is not applicable. The method is started by dividing fluid temperature-enthalpy profile to a number of segments along the heat exchanger. Then, heat transfer coefficient is evaluated at each segment to determine heat transfer surface area. In other words, apply a LMTD method for every segment. Brackenbury has developed a software to design heat exchangers applying this method [Brackenbury, et al., 1993].

A mechanical design calculation has been done accordingly with TEMA [TEMA, 1998] and other standards. The evaporator and condenser designs are shown in Figure 5 [Bambang T. Prasetyo et al., 2006].

BFM type shell and tube evaporator lets the brine flows within the tubes inside the shell containing the n-pentane working fluid to be evaporated. Evaporator shell inner diameter is 339.8 mm, while the tube length is 1500 mm, with total thermal power of 56.31 kW. The condenser is chosen to be of BXM type shell and tube, due to its functionality, where the cooling water flows within the tube inside the shell containing the n-pentane working fluid to be condensed. The calculated condenser shell diameter is 340 mm, while the tube length is 1500 mm, with thermal power of 53.68 kW.

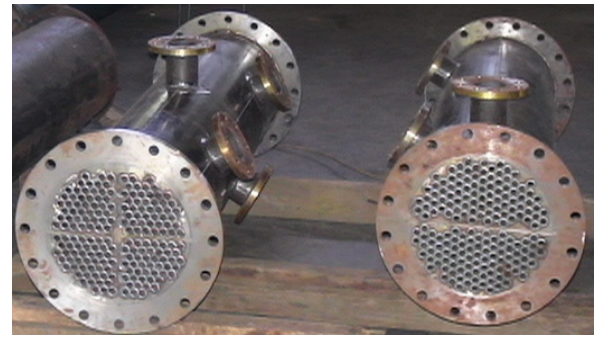
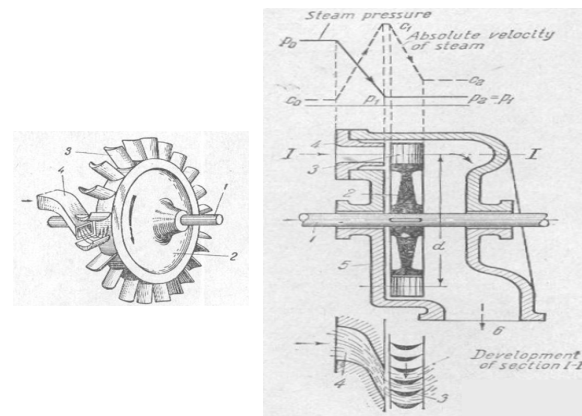


Figure 5: Evaporator and condenser installed

### 2.2. n-Pentane Turbine Design

Turbine design for the 2 kW binary cycle plant is based on a simple turbine impulse calculation for one stage of velocity and pressure. The schematic of the turbine design is shown on Figure 6. The main components of turbine are: shaft, disc, blade, nozzle, stator, and exhaust pipe. Considering physical properties and n-pentane vapor thermodynamics, a convergent nozzle is adopted allowing for a maximum velocity to be equivalent with a critical velocity.

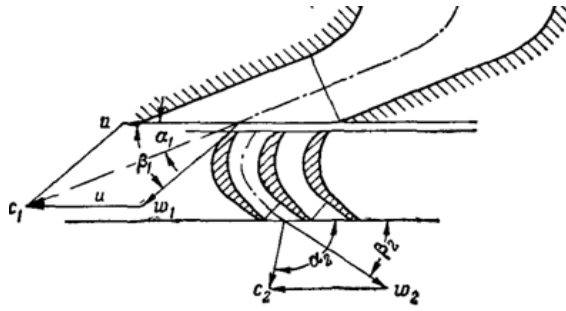


- |                  |                 |
|------------------|-----------------|
| 1. Shaft         | 4. Nozzle       |
| 2. Disc          | 5. Stator       |
| 3. Moving blades | 6. Exhaust pipe |

Figure 6: One stage turbine impulse diagram [P. Shlyakhin (s.a)]

According to this configuration, the transformation of energy on the rotating blades is shown in the Figure 7, where  $c_1$  is the absolute input velocity of working fluid to row of blades,  $\alpha_1$  is the angle of entrance,  $u$  is the peripheral velocity of blade wheel,  $w_1$  is the relative velocity of working fluid leaving row of blades with an angle  $\beta_1$ ,  $w_2$  is the relative velocity of working fluid leaving the blade at an angle  $\beta_2$ , and  $c_2$  is the absolute velocity of working fluid leaving blade at an angle  $\alpha_2$ . Because of that, if  $\dot{m}$  is the mass flow rate of working fluid, the thermal power of working fluid which is transferred to the blades is stated as below:

$$P = \dot{m} u (c_1 \cos \alpha_1 - c_2 \cos \alpha_2) \quad (1)$$



**Figure 7: Fluid velocity variation on moving blades of turbine impulse [P. Shlyakhin (s.a)]**

The turbine design as in [P. Shlyakhin (s.a)] is done in the following order:

• **Enthalpy drop in theoretical nozzle ( $\Delta h_t$ )**

Schematic of the flow in nozzle is shown in Figure 8. Enthalpy drop of vapor flow in nozzle is as follows:

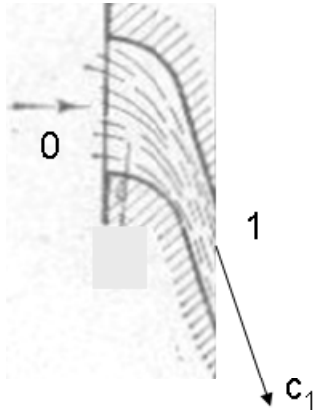
$$\Delta h_t = h_0 - h_1 \quad (2)$$

where,  $h_0$  is a specific enthalpy of vapor entering the nozzle, and  $h_1$  is a specific enthalpy of vapor leaving the nozzle.

2.2.1. **Theoretical nozzle output vapor velocity ( $c_{1t}$ )**

By ignoring potential energy, the relationship between kinetic energy of the flow entering turbine, and losses of flow inside the nozzle, the nozzle output vapor velocity is described below:

$$c_{1t} = \sqrt{2 \Delta h_t} \quad (3)$$



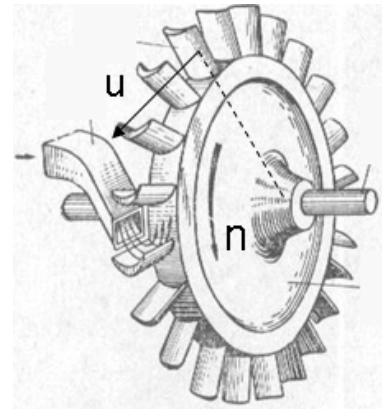
**Figure 8: Flow inside nozzle**

2.2.2. **Actual nozzle output vapor velocity ( $c_1$ )**

By considering losses of the flow when entering and leaving also along the nozzle, the nozzle output vapor actual velocity can be described as follows:

$$c_1 = F_1 \sqrt{2 \Delta h_t} \quad (4)$$

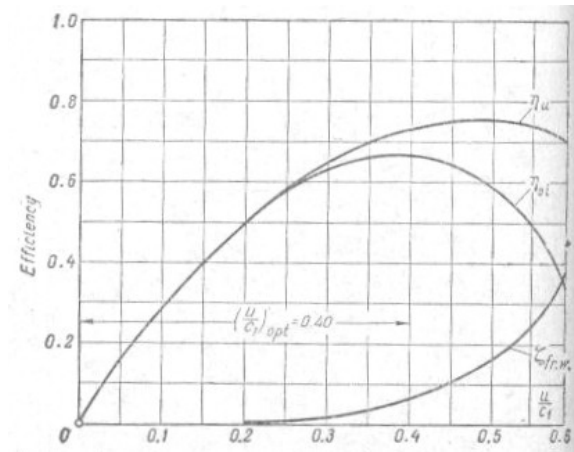
where  $F_1$  is a velocity coefficient which can be taken around 0.92.



**Figure 9: Blade Wheel**

2.2.3. **Peripheral velocity of blade wheel ( $u$ )**

The schematic of blade wheel is shown in Figure 9, whereas the relation between efficiency and value of  $u/c_1$  for the single stage impulse turbine is shown in Figure 10, where  $u/c_1$  optimum is around 0.4. In this design, the peripheral velocity of the blade wheel is determined by  $u = 0,4 c_1$ .



**Figure 10: Efficiency of single stage impulse turbine with one velocity stage**

2.2.4. **Diameter of blade wheel ( $d$ )**

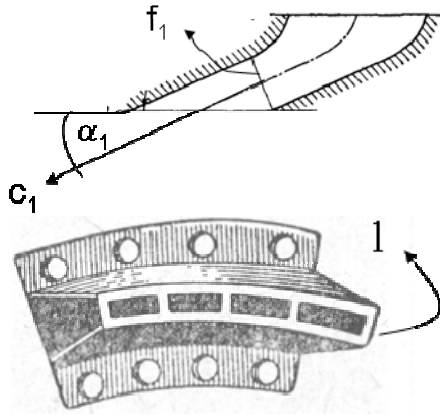
To determine diameter of blade wheel, the following equation is employed:

$$d = \frac{60u}{\pi n} \quad (5)$$

where  $n$  is rotation of blade wheel which is fixed at 3000 rpm for this design.

2.2.5. **Nozzle Parameters**

The main parameters of a nozzle is the nozzle outside surface  $f_1$ , the angle  $\alpha_1$ , the nozzle height  $l$ , the ratio of the peripheral fraction which is covered by the nozzle  $\varepsilon$ , (see Figure 11)



**Figure 11: Nozzle convergent**

The nozzle outside surface  $f_l$  is determined by the relation below:

$$f_1 = \frac{m v_{1a}}{c_1} \quad (6)$$

where  $m$  is the vapor mass flow rate and  $v_{1a}$  is the specific volume of vapor at actual condition leaving the nozzle. The angle  $\alpha_1$  is around  $14^\circ - 20^\circ$  which in this design is chosen to be  $14^\circ$ . the nozzle height  $l$  is preferable  $\geq 10$  mm and for this design  $l = 10$  mm.

The ratio of peripheral fraction which is covered by the nozzle  $\varepsilon$  is determined as follows:

$$\varepsilon = \frac{m v_{1a}}{\pi d l c_1 \sin \alpha_1} \quad (7)$$

#### 2.2.6. Blade Parameters

The main parameter of the blade is the number of blades  $z_1$ , width of each nozzle  $a_1$ , the blade height  $l'_1$ , the relative angle of entrance  $\beta_1$ , relative exit angle  $\beta_2$ , the blade height on exit side  $l''_1$  and the radius of blade curve  $r$ .

The number of blades  $z_1$  is determined by:

$$z_1 = \frac{\pi d \varepsilon}{t_1} \quad (8)$$

where  $t_1$  is distance between blades. Thereby, the width of each nozzle  $a_1$  can be determined as follows:

$$a_1 = \frac{f_1}{l z_1} \quad (9)$$

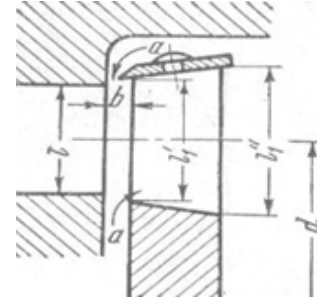
The blade height  $l'_1$  (Figure 12) is determined as  $l'_1 - 1 \approx 2$  to 4 mm, and for this design it is determined that  $l'_1 = 1+2$  mm.

The relative angle of entrance  $\beta_1$  is determined by:

$$\beta_1 = \arcsin \left( \frac{c_1 \sin \alpha_1}{w_1} \right) \quad (10)$$

where  $w_1$  is relative velocity of vapor entering the blades, which according to Figure 7 is determined by the following:

$$w_1 = \sqrt{c_1^2 + u^2 - 2uc_1 \cos \alpha_1} \quad (11)$$



**Figure 12: Blade measurement**

Relative exit angle  $\beta_2$  is determined as  $\beta_1 - \beta_2 \approx 2^\circ$  to  $10^\circ$ , where for this design it is determined as  $\beta_2 = \beta_1 - 3^\circ$ . Thereby, the blade height on exit side  $l''_1$  can be determined as below:

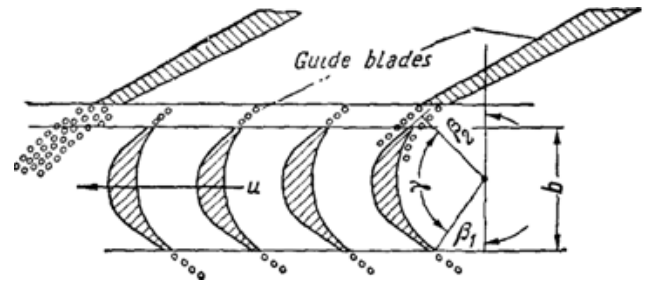
$$l''_1 = \frac{m v_2}{\pi d \varepsilon \sin \beta_2} \quad (12)$$

where  $v_2$  is the specific volume of vapor at actual condition exiting the nozzle.

The radius of blade curve  $r$  is determined with the following relation:

$$r = 2b (\sin \beta_1 + \sin \beta_2) \quad (13)$$

where  $b$  is the blade width as shown in Figure 13, which for this design is around  $1/3$ .



**Figure 13: Blade width**

N-pentane is a flammable fluid. Therefore, the seal is one of the important parts in this turbine design. Considering the simplicity of design and manufacture, a stuffing box is used as a seal between shaft and stator for the current design.

## 3. RESULT AND DISCUSSION

### 3.1. Result of n-Pentane Turbine Design

By referring to design parameters shown in Figure 4, the n-pentane turbine was designed with enthalpy drop of 12.5 kJ/kg and mass flow rate of 0.139 kg/s.

The turbine thermal power is calculated to be 1.73 kW with rotating speed determined at 3000 rpm. The result of design calculation is shown in Figure 14, while the detail dimension is shown in Table 3 [Bambang T. Prasetyo et al., 2006].



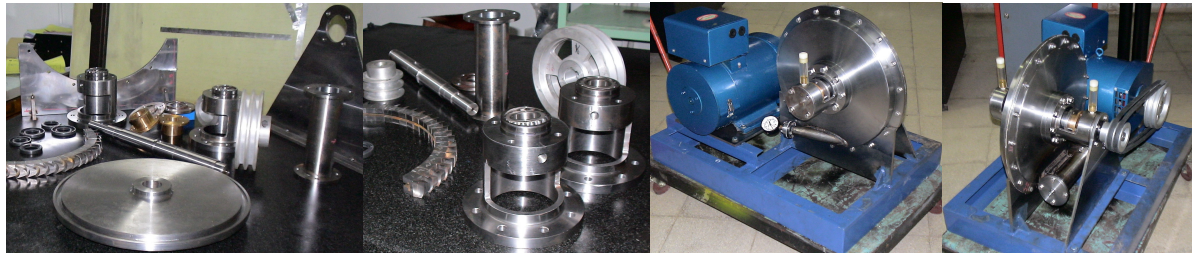


Figure 14: Picture of parts and installed unit of the 2 kW n-pentane turbine

### 3.2. Experimental Results

Performance test of the 2 kW binary cycle geothermal power plant was conducted using facilities at Wayang Windu geothermal power plant. To run the 2 kW model plant, the heat source is the tapped brine from a reinjection well pipeline. The installation of all equipment and parts on site is shown in Figure 14 and Figure 15.

To analyze performance of the model plant, some measurements are taken at points shown in Figure 16, whereas the experimental results are summarized in Table 4.



Figure 15: Installation of the binary cycle geothermal power plant model

Table 3. Dimension of n-pentane turbine parts

Parts	Dimension		
Nozzle	total		5.0
	high	mm	10.0
	width	mm	5.0
	angle $\alpha_1(o)$	(o)	14.5
Rotor	pitch diameter	mm	382.3
	shaft diameter	mm	33.0
Blades	total		60.0
	high	mm	12/16.6
	width	mm	10.0
	angle $\beta_1$	(o)	23.8
	angle $\beta_2$	(o)	16.8
	radius	mm	13.8
	curvature R		
	blade distance	mm	20

Table 4. Series of measurements at one of the tests

Parameter/Location	Unit	Design	Test
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Brine Temp. - in Evaporator	°C	95	110
Brine Temp. - out Evaporator	°C	78,5	84
N-pentane Debit- in Evaporator	l/min.	13,7	14
N-pentane Pressure- in Evaporator	Bar a	3,5	3,8
N-pentane Temp. - in Evaporator	°C	40,46	38
N-pentane Pressure- out Evaporator	bar <sub>a</sub>	3,48	3,8
N-pentane Temp. - out Evaporator	°C	74	84
N-pentane Pressure - in Condenser	bar <sub>a</sub>	1,36	1,6
N-pentane Temp. - in Condenser	°C	62,3	69
N-pentane Pressure - out Condenser	bar <sub>a</sub>	1,35	1,4
N-pentane Temp. - out Condenser	°C	42	48
Cooling water Temp. - in Condenser	°C	31	32
Cooling water Temp. - out Condenser	°C	37,5	44
Turbine rotation	rpm	3000	3124
Generator Voltage	V	220	210
Generator Power	W	1730	1200

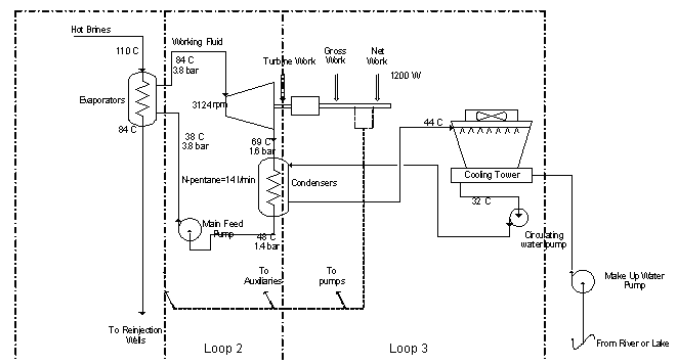


Figure 16: Location of parameter measurement at binary cycle 2 kW at Wayang Windu

### 4. CONCLUSIONS

Design and performance testing of a model 2 kW binary cycle geothermal power plant were performed. Considering experimental results shown in Table 4, some important conclusions are taken as in the following:

- Binary cycle power plant can be applied utilizing geothermal resources with low enthalpy as energy sources.

- Methods for designing binary cycle plant were tested and proven to be accurate.
- This model can be employed for utilizing other waste heat as a alternative energy source.

#### ACKNOWLEDGEMENT

We would like to express gratitude to all those providing help and supports during the performance test, especially to Star Energy Ltd. and Pertamina for providing test facilities in Wayang Windu Geothermal field.

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