Solar collector field and thermal energy storage for auxiliary component in organic rankine cycle for bottoming unit to utilize exhaust steam from back pressure turbine of Ulumbu geothermal power plant

Z Y Berian¹, A T Wijaya¹, and T M Iqbal¹

¹ Department of Mechanical Engineering, Faculty of Mechanical and Aerospace Engineering (FTMD), Bandung Institute of Technology, Indonesia, Jl. Ganesha. No.10, Bandung, West Java, Indonesia, 40132

*Email: zaghyyakana@gmail.com

Abstract. Flores has a vast geothermal energy sources, one of the fields, Ulumbu, producing electricity from the geothermal power plant (PLTP). PLTP Ulumbu has two back pressure turbines generating 2.5 MW electricity. Besides geothermal power, Flores has potential solar energy sources that are recorded as much as 5 to 6 kWh/m²/day. The exhaust steam from the backpressure turbine adds the binary cycle as the bottoming unit. It is able to generate electricity continuously to supply the baseload. Currently, the peak load of electricity usage in Flores is supplied by the diesel power plant (PLTD). This paper studies the addition of solar collectors and thermal energy storage (TES) as auxiliary components in the binary cycle and simulates the system by using the Aspen Plus. The solar energy is stored in TES during the day and then released to the heat exchanger in the evening. This idea reduces the dependency on diesel power plants in the evening. To increase power as much as 500 kW, the collector area needed to supply the system is 1.1 ha using synthetic oil as HTF. The volume of TES to stored the energy by using nitrate salts as HTF is 1,120 m³. As a result, the efficiencies of the non-hybrid system and hybrid systems are 11.38% and 11.31% subsequently.

1. Introduction
As a blessed with abundant geothermal energy potential, Flores Island, East Nusa Tenggara (NTT), Indonesia, has been declared as a geothermal island according to the Ministry of Energy and Mineral Resources of the Republic of Indonesia through Ministrial Regulation No. 2268/K/30/MEM/2017. Ulumbu is one of the existing geothermal power plants (PLTP) in Flores, NTT, Indonesia, owned by the state electricity company (PLN). Ulumbu has been producing a total capacity of 10 MW. The geothermal working area (WKP) of Ulumbu has 103 MW [1].

PLTP Ulumbu has four units generators, each has 2.5 MW capacity. Two of them are using a backpressure turbines and the rest are using the condensing units [2]. A backpressure turbine is mostly used it has lower capital cost, simple construction, less complicated machinery, and requires less support system. Meanwhile, the condensing turbine is selected because it requires less steam [3]. Regarding the waste of heat from the backpressure turbine, this research proposed the binary cycle as the bottoming unit in PLTP Ulumbu, generating 4 MWe with isopentane as the working fluid [2].

Besides huge geothermal potentials, Flores is also blessed with great solar energy potential about 5-6 kWh/m²/day, according to RETScreenExpert software. This research proposed an alternative design for PLTP Ulumbu, utilizing a solar-geothermal hybrid system that can generate 5,450 MWh/year [4].

This study aims to design a solar collector field using a parabolic trough concentrator (PTC) and thermal energy storage (TES) as auxiliary items in a binary cycle. The objective of the proposed items is to generate more power in the evening. The performance of the designed system was assessed and
compared to the previous design. Aspen Plus was utilized to simulate the designed system with isopentane as the working fluid.

2. Literature Study

2.1. Solar and Geothermal Hybrid System
The geothermal power plant can be classified depending upon the circumstances. It can be a standalone system, a combined system, and a hybrid system. In the geothermal power plant, the combined system consist of solar PVs and geothermal without interacting in any thermodynamics aspect. The hybrid system, on the other hand, consists of solar thermal and geothermal that are interacting in thermodynamics aspect.

2.2. Backpressure turbine Unit of PLTP Ulumbu
There are two units of backpressure turbines in PLTP Ulumbu. These turbines operate at steam inlet pressure of 10 bar.a and outlet pressure of 0.98 bar.a. Both units are working exceeding 31.2 ton/hour with an average temperature of 99°C [2].

2.3. Previous research in Ulumbu
There are two research in Ulumbu that supported this study. The first one is the bottoming unit binary cycle and the other is solar energy potential in Ulumbu.

2.4. Bottoming unit binary cycle
The bottoming units are to utilize exhaust steam from backpressure turbines of PLTP Ulumbu. This research proposed a binary cycle using the organic working fluid with essential components such as preheater, evaporator, expander, condenser, and pump as shown in Figure 1. The bottoming unit is able to generate up to 2.25 MW with thermal efficiency about 10%. Meanwhile, the backpressure turbine is able to generate 2.50 MW.

![Figure 1. Binary cycle [2]](image)

2.5. The solar-geothermal hybrid system in Ulumbu.
This research proposed an alternative design to develop Ulumbu 3 & 4. The research provides the solar data in Ulumbu by using the empirical formula in the specific location. The research evaluated two-alignments of parabolic trough concentrators (PTCs), which are E–W alignment and N–S alignment
with Eurotrough-150 (ET-150). The daily average value of irradiance in E–W alignment and N–S alignment are 581 W/m²/day and 758 W/m²/day subsequently. The average daily beam irradiation for duration 4, 6, and 8 hours are shown in Table 1. The suitable alignment to collect highest daily energy is the N–S alignment.

| LPM [hours] | Beam Irradiation Flux E–W [kWh/m²] | Beam Irradiation Flux N–S [kWh/m²] |
|-------------|------------------------------------|------------------------------------|
| 4           | 2.3 ± 0.3                           | 3.0 ± 0.5                           |
| 6           | 3.5 ± 0.5                           | 4.5 ± 0.7                           |
| 8           | 4.6 ± 0.6                           | 6.0 ± 0.6                           |

2.6. Thermodynamic assessment
The thermodynamic assessment in this study elaborates turbine, heat exchanger, and pump for the formulas and functions.

2.6.1 Turbine
In order to determine the work produced by the turbine, Equation (1) is employed. The turbine is coupled to the generator to generate electricity. The formula does not consider the kinetic energy as it is assumed working in the adiabatic condition, and operating in steady condition.

\[ W_t = \dot{m}(h_1 \text{-} h_2) \] (1)

The parameters of \( \dot{W} \), \( \eta_t \), \( \dot{m} \), \( h \) are work produced by a turbine (kW), turbine efficiency, the mass of steam (kg/s), and enthalpy (kJ/kg). It is commonly assumed that the turbine has a certain amount of isentropic efficiency (\( \eta_{st} \)), which is the ratio of the generated work for an actual process and the ratio of the generated work for the isentropic process as shown in Equation (2).

\[ \eta_{st} = \frac{h_1 \text{-} h_2}{h_1 \text{-} h_{2s}} \] (2)

2.6.2 Pump
The pump works similarly to the turbine, but in the opposite way. The pump compresses the fluid in a liquid phase, and thus needs power to do so. The power needed by the pump (\( W_p \)) is approximated by using Equation (3).

\[ W_p = \dot{m}(h_4 \text{-} h_3) \] (3)

It is also commonly assumed that the pump has a certain amount of isentropic efficiency (\( \eta_{sp} \)), which is the ratio of the needed work for an isentropic process and the needed work for the actual process as approximated by Equation (4).

\[ \eta_{sp} = \frac{h_{4s} \text{-} h_3}{h_4 \text{-} h_3} \] (4)

2.6.3 Heat exchanger
The heat exchanger calculation involves several components, such as preheater, evaporator, and condenser. A heat exchanger is a component used to transfer heat from one fluid to another fluid. It can be used to heat the fluid such as in a preheater and evaporator. The other function is to remove heat from a fluid, such as in a condenser. The Equation used to calculate the heat exchange is shown in Equation (5).

\[ \dot{Q} = \dot{m}_h(h_{hi} \text{-} h_{ho}) = \dot{m}_c(h_{vi} \text{-} h_{vo}) \] (5)
Variables of \( \dot{m}_h \) and \( \dot{m}_c \) represent the mass flow rate of the hot and the cold fluid respectively. \( h_h \) and \( h_c \) respectively represent the hot fluid and the cold fluid enthalpy, while index i and o represent inlet and outlet. In the preheater and evaporator, there is a minimum temperature difference between heating fluid and the working fluid referred to as pinch-point (T_{pp}).

The solar geothermal hybrid system requires a heat exchanger to increase steam quality from the geothermal working fluid. The power requirement in kW is approximated by Equation (6).

\[
\dot{Q} = UA \cdot \text{LMTD} = \dot{m} c_p (T_h - T_c)
\]

Where \( \dot{m}, c_p, UA, \text{LMTD}, T_h, T_c \) are mass flow rate (kg/s), specific heat (kJ/kgK), logarithmic mean temperature difference (°C), higher temperature (°C), and lower temperature (°C).

2.6.4 Efficiency

Efficiency calculation is crucial to examine the performance of the power plant. Generally, there are two types of the evaluated efficiencies: energy efficiency and exergy efficiency. The value of those efficiencies can never reach unity (100%).

Thermal efficiency \( (\eta) \) is defined as the ratio between the amount of the generated useful work and the energy absorbed by the system [5]. In a power cycle, the amount of useful work \( (W_{\text{net}}) \) is the subtraction of the work generated by the turbine and the work needed by the pump. The efficiency is the ratio of useful work and the energy added to the system \( (Q_m) \) as plotted in Equation (7).

\[
\eta = \frac{W_{\text{net}}}{Q_m}
\]

Exergy efficiency \( (\varepsilon) \), or second law efficiency, is used to assess energy resource utilization [5]. Exergy efficiency is related to the irreversibilities present in the system. It is defined as the ratio between the amount of useful exergy \( (E_u) \) and the amount of exergy from the system's source \( (E_s) \), shown in Equation (8).

\[
\varepsilon = \frac{E_u}{E_s}
\]

2.7. Thermal system.

To find the collector area \( (m^2) \) that needed in the system, it can be found by using Equation (9).

\[
A_c = \frac{\dot{Q}}{G \eta_c}
\]

The \( \eta_c \) is the collector efficiency 0.6, for ET-150, the efficiency of 0.6 [6] and \( G \) is beam irradiance (W/m²). To find the volume of storage \( (m^3) \) and mass of the HTF (kg) as Equation (10) and Equation (11) [7].

\[
V_{TES} = \frac{Q_{\text{avg-day}}}{h_{pc} \cdot \rho}
\]

\[
m = \frac{Q_{\text{avg-day}}}{c_{pc} \cdot f \cdot \Delta T}
\]

Where \( h_{pc} \) is specific enthalpy (J/kg); \( Q_{\text{avg-day}} \) is collector thermal energy (J); \( f \) is factor utilize as much as 0.85; \( \Delta T \) is difference temperature (°C), and \( \rho \) is density (kg/m³).
3. Methodology

3.1. Thinking framework
Thinking framework is shown in Figure 2. PTC collects the solar radiation with N–S alignment using heat transfer fluid (HTF). During the day, the generated energy is stored in TES. The system operates in two modes: baseload and peak load. In baseload mode, the exhaust steam as heating fluid is connected to a binary cycle as the heating fluid. In peak load, the heat exchanger heats up the working fluid. TES releases energy to supply the heat exchanger.

![Figure 2. Thinking framework.](image)

3.2. ASPEN Plus Model Simulation
The first step of the modeling process in ASPEN Plus is to list all the components involved in the model. The next step as the most crucial step is to model the component in a thermodynamic model accurately. The present work of PENG-ROB method is selected as a thermodynamic model to derive the equation of states by exploring the isopentane properties as the working fluid. The next step is to model all components into equivalent blocks in ASPEN Plus and connect them with streams. Finally, all the inputs and specifications of each component and stream are defined.

ASPEN Plus modeling is based on taking process. ASPEN Plus breaks taking process down into more simple components, called as blocks. For example, Rankine Cycle could be decomposed into a turbine block, two heat exchanger blocks as condenser and evaporator, and a pump block. Figure 3a shows the model of the Organic Rankine Cycle (ORC) with a preheater. The ORC with the preheater model is modified by adding a heat exchanger before the turbine to utilize solar energy used in the evening, called as Hybrid ORC. Figure 3b shows the modeled ORC with a hybrid system.

![Figure 3. (a) ORC with preheater and (b) ORC with preheater and hybrid system.](image)
In ASPEN Plus modeling, it is necessary to model a breakpoint to give input to the model [8]. For both ORC with preheater and Hybrid ORC, the break was inserted at the exit of the condenser, state 3OUT, and the inlet of pump, state 3IN. If the two streams give the same results, it is indicated that the problem is well formulated. The breaks in state 3 allow for inputs to be given for pump inlet, which is the low side pressure, zero vapor quality, and the mass flow rate. The inputs for streams and blocks are shown in Table 2. Some assumptions that were used in this present work are:

- Negligible pressure drops in heat exchanger and pipelines
- Geothermal fluid modeled as water in the same pressure, temperature, and vapor fraction
- Turbine efficiency is 90% for ORC with preheater and 92% for Hybrid ORC
- Pump efficiency is 80% [9]
- Generator efficiency is 96%
- The minimum temperature approach of each heat exchanger is 5°C [10]
- The vapor quality of the exit condenser is zero

| Stream/Blocks | Inputs |
|---------------|--------|
| Stream        |        |
| 3IN           | Pressure, vapor fraction, and mass flow rate |
| A             | Pressure, vapor fraction, and mass flow rate |
| Blocks        |        |
| Condenser     | Pressure and vapor fraction |
| Evaporator    | Cold stream outlet vapor fraction |
| Hybrid        | Pressure and heat duty |
| Preheater     | Cold stream outlet vapor fraction |
| Pump          | Discharge pressure and efficiency |
| Turbine       | Discharge pressure and efficiency |

### 3.3. Thermal System

After the power needed to the heat exchanger is acquired from Aspen Plus, it has to calculate the PTC’s power. The first step is to set the inlet and outlet temperature for each condition. This system chooses the counterflow heat exchanger. There are two heat exchangers: the heat exchanger for the steam power plant (HE1) and the other is a heat exchanger for TES (HE2).

![Figure 4. Thermal system.](image)

The mass flow rate of synthetic oil is calculated before the calculation of power required for HE2. The power calculation for HE2 is to obtain the flow rate for nitrate salt to stored energy in TES. Power calculation is also required for PTC and the collector area. Finally, the TES volume for a 2-tank
system is determined and oversized by 30% to add some nanoparticle solid adjustment for increasing HTF performance.

3.4. Condenser
Condenser calculation is to determine the mass flow rate of the air and to estimate the power needed for the condenser’s fan. Since isopentane is the hot fluid and air is the cold fluid, the condenser uses ambient temperature of 27.45°C. The fan power is 0.15 kW/kg/s of air [11]. The air temperature is assumed to increase up to 35.1°C.

3.5. Efficiency Analysis
Work or heat transfer occurs in a component (turbine, pump, or heat exchanger) can be represented as the enthalpy (h) difference at the inlet and outlet. The thermal efficiency of the non-hybrid system is represented by Equation (12) and the hybrid system is represented by Equation (13).

$$\eta = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \tag{12}$$

$$\eta = \frac{(h_{1B} - h_2) - (h_4 - h_3)}{h_{1B} - h_4} \tag{13}$$

To calculate the exergy efficiency, the source of exergy and the useful exergy must be determined first. The useful exergy is the net power generated by the system. Meanwhile, the source of exergy is the exergy that is carried by the heat transfer in the hybrid system ($Q_{solar}$) added by the sum of exergy flow entering ($\Sigma E_{in}$) and leaving ($\Sigma E_{out}$) the system. Two fluids are entering and leaving the system: geothermal brine and ambient air. So, the sum of exergy flow is shown by Equation (14). Finally, the exergy efficiency equation is shown in Equation (15).

$$\Sigma E_{in} - \Sigma E_{out} = \dot{m}_{brine} \left( h_A - h_C - T_0 (s_A - s_C) \right) + \dot{m}_{air} \left( h_{in} - h_{out} - T_0 (s_{in} - s_{out}) \right) \tag{14}$$

$$\varepsilon = \frac{W_{net}}{(1 - T_0 / T_b) Q_{solar} + \Sigma E_{in} - \Sigma E_{out}} \tag{15}$$

$T_0$ denotes ambient temperature (27.45°C), $s$ denotes entropy at respective stations, and $T_b$ denotes the temperature of the surface at which heat transfer occurs.

4. Results and Discussion

4.1. Model Conceptual
This system operates as a standalone geothermal to supply baseload and operates hybridly to supply peak load. Energy from solar radiation is stored in TES. Based on LPM from BMKG, the range of LPM in Ulumbu is 4–8 hours. The system sets the average LPM of 6 hours with 202 days in a year to operate the hybrid system from 18.00–24.00.
4.2. Aspen Plus Result

To make sure that our model was formulated well, it was necessary to include a break cycle point to provide inputs. In both ORC with preheater and Hybrid ORC model, the break was inserted at stream 3. A well-formulated model will result in identical overall and component mass flow rates on either side of the break. In this model, the stream leaving the condenser is stated as 3OUT and the stream entering the pump is stated as 3IN. Tables 3 and 4 show the ORC results with the preheater and Hybrid ORC model, respectively, and demonstrate this model's accuracy.

### Table 3. ORC with preheater.

| Parameter      | Stream 3IN | Stream 3OUT |
|----------------|------------|-------------|
| Temperature (℃)| 37.755     | 37.755      |
| Pressure (bar) | 1.4        | 1.4         |
| Mass flow rate (kg/s) | 90         | 90          |

### Table 4. ORC with a preheater and hybrid system.

| Parameter      | Stream 3IN | Stream 3OUT |
|----------------|------------|-------------|
| Temperature (℃)| 37.755     | 37.755      |
| Pressure (bar) | 1.4        | 1.4         |
| Mass flow rate (kg/s) | 90         | 90          |

A typical set of states point generated by ASPEN Plus for ORC with preheater, and Hybrid ORC is given in Table 5 and 6. The numbered streams correspond to various state points of interest shown in Figures 6 and 7.
Figure 6. T-s diagram ORC with preheater.

Figure 7. ORC with preheater and hybrid systems.

Table 5. State point for ORC with preheater.

| State Point | Temperature (°C) | Pressure (bar) | Vapor fraction | Mass flow rate (kg/s) |
|-------------|------------------|----------------|----------------|-----------------------|
| 1           | 93.64            | 6.34           | 1              | 90                    |
| 2           | 59.91            | 1.4            | 1              | 90                    |
| 3           | 37.75            | 1.4            | 0              | 90                    |
| 4           | 38.04            | 6.34           | 0              | 90                    |
| 5           | 93.64            | 6.34           | 0              | 90                    |
| A           | 101.54           | 0.98           | 0.93           | 17.35                 |
| B           | 101.54           | 0.98           | 0.30           | 17.35                 |
| C           | 95.98            | 0.98           | 0              | 17.35                 |
Table 6. State point for ORC with preheater and hybrid systems

| State Point | Temperature (°C) | Pressure (bar) | Vapor fraction | Mass flow rate (kg/s) |
|------------|------------------|----------------|----------------|-----------------------|
| 1A         | 93.64            | 6.34           | 1              | 90                    |
| 1B         | 119.59           | 6.34           | 1              | 90                    |
| 2          | 85.92            | 1.4            | 1              | 90                    |
| 3          | 37.75            | 1.4            | 0              | 90                    |
| 4          | 38.04            | 6.34           | 0              | 90                    |
| 5          | 93.64            | 6.34           | 0              | 90                    |
| A          | 101.54           | 0.98           | 0.93           | 17.35                 |
| B          | 101.54           | 0.98           | 0.30           | 17.35                 |
| C          | 95.98            | 0.98           | 0              | 17.35                 |

The result of the energy balance for both cycles are shown in Table 7.

Table 7. Energy result for both ORC with preheater and Hybrid ORC

| Parameter | ORC with preheater | Hybrid ORC |
|-----------|--------------------|------------|
| $Q_{in}$ (MW) | 38.73            | 38.73      |
| $Q_{cond}$ (MW) | 34.33            | 38.79      |
| $W_{net}$ (MW)  | 4.41             | 4.95       |

4.3. Thermal System

In this study's thermal system using N–S alignment, the collector area needed regarding LPM in Ulumbu shown in Table 8.

Table 8. Characteristic of the collector system.

| Parameter | 4 hours | 6 hours | 8 hours |
|-----------|---------|---------|---------|
| Aperture area PTC N–S [ha] | 1.7 ~ 2.1 | 1.1.0 ~ 1.4 | 0.8 ~ 1.0 |

This study selects the 6 hours to represent average LPM in Ulumbu, the collector area that needed is 1.1 ha to fulfilled thermal energy from PTC required with a value of 32 MWh.

Table 9. Characteristic of the storage system

| Parameter | 4 hours | 6 hours | 8 hours |
|-----------|---------|---------|---------|
| PTC N–S  |         |         |         |
| Mass of HTF in PTC [ton] | 1,300 ~ 1,600 | 2,000 ~ 2,300 | 2,700 ~ 3,100 |
| Mass of HTF in TES [ton] | 1,100 ~ 1,300 | 1,700 ~ 2,000 | 2,200 ~ 2,600 |
| The volume of TES [m$^3$] | 790 ~ 900 | 1,120 ~ 1,400 | 1,570 ~ 1,840 |

4.4. Condenser

Condenser calculation results, using equation, can be seen in Table 10. It can be seen that the increase of heat input to the turbine will increase the temperature of the fluid entering the condenser, and therefore increase the heat output in the condenser. As a result, the mass flow rate of air and fan power required will increase too.
**Table 10.** Condenser calculation results.

| Parameter | Non Hybrid | Hybrid |
|-----------|------------|--------|
| $T_2$ ($^\circ$C) | 59.9 | 85.9 |
| $h_2$ (kJ/kg) | -2,074 | -2,024 |
| $T_3$ ($^\circ$C) | 37.8 | 37.8 |
| $h_3$ (kJ/kg) | -2,455 | -2,455 |
| $Q$ (kW) | 34,327.1 | 38,785 |
| $m_{air}$ (kg/s) | 4472 | 5053 |
| $W_{fan}$ (kW) | 670.77 | 757.90 |

4.5. **Efficiency Results**

Efficiency calculation results, using equation, can be seen in Table 11. The use of the hybrid system turns out will cause a slight decrease in thermal efficiency. The slight decrease in thermal efficiency is due to the increase of heat input, increasing the fluid's temperature entering the condenser and, therefore, increasing the heat output. However, the use of the hybrid system will cause a slight increase in exergy efficiency. This increase shows that the hybrid system's use will allow the system to convert the available energy into usable energy.

**Table 11.** Efficiency calculation results.

| Parameter     | Non-Hybrid | Hybrid |
|---------------|------------|--------|
| $\eta$        | 11.38%     | 11.31% |
| $\Delta E_{brine}$ (kW) | 9,065 | 9,065 |
| $\Delta E_{air}$ (kW) | 917 | 1,036 |
| $Q_{solar}$ (kW) | 0 | 1,042 |
| $\epsilon$    | 44.14%     | 44.40% |

5. **Conclusions**

The proposed design is able to increase power about 500 kW using isopentane as working fluid. System operates from 18.00–24.00 to supply evening peak demand. The increased power reduce diesel work to supply evening peak demand. The collector area that is needed to supply the system is 1.1 ha by using synthetic oil as HTF. The volume of TES to store energy by using nitrate salts as HTF is 1,120 m$^3$. The use of a hybrid system has slightly decreased thermal efficiency from 11.38% to 11.31%. However, hybrid system has slightly increased the exergy efficiency, from 44.14% to 44.40%.

**Recommendations**

Further research can be optimized by considering the points as to conduct economic analysis and to use data of solar energy by experiment.

**References**

[1] Kurniawan I, Sutopo S, Pratama H B and Adiprana R. 2019. A natural state model and resource assessment of Ulumbu Geothermal field IOP Conf. Ser. Earth Environ. Sci. 254

[2] Nandaliarasyad N, Maulana D T and Darmanto P S. 2020. Study of Development Scenarios for Bottoming Unit Binary Cycle to Utilize Exhaust Steam from Back Pressure Turbine Geothermal Power Plant IOP Conf. Ser. Earth Environ. Sci. 417

[3] Stewart M. 2019. Overview of commonly used drivers
[4] Zagy B, A System Design of a Solar and Geothermal Hybrid Power Plant for Flores Island, Thesis, Program Studi Teknik Mesin ITB, Bandung, 2020

[5] M.J. Moran, H.W. Shapiro, D.S. Boettner and M.B. Bailey, Fundamentals Engineering of Thermodynamics, 8th Ed., John Wiley & Sons, New York, 2014.

[6] V. E. Dudley, G. J. Kolb, M. Sloan, and D. Kearney. (1994). Sand94-1884, Test Results: SEGS LS-2 Solar Collector, Albuquerque, NM, Sandia National Laboratories

[7] J. Freeman, I. Guarracino, S.A. Kalogirou, C.N. Markides, A small-scale solar organic Rankine cycle combined heat and power system with integrated thermal energy storage, Applied Thermal Engineering, Volume 127, 2017, Pages 1543-1554, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2017.07.163.

[8] C. Somers et al., Modelling Water/Lithium Bromide Absorption Chillers in ASPEN Plus, Applied Energy 88: 4197-4205, 2011.

[9] Astolfi M, La Diega L N, Romano M C, Merlo U, Filippini S and Macchi E 2019 Technoeconomic optimization of a geothermal ORC with novel ‘Emeritus’ heat rejection units in hot climates Renew. Energy

[10] Baragbah F H and Darmanto P S 2015 Effect of Organic Rankine Cycle modification to system 37th New Zeal. Geotherm. Work.

[11] Toffolo A, Lazzaretto A, Manente G and Paci M 2014 A multi-criteria approach for the optimal selection of working fluid and design parameters in organic Rankine cycle systems Appl. Energy 121 219–232