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Geothermal Binary Demonstration Power Plant Pangolombian-Lahendong, Indonesia

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ABSTRACT

In order to successfully demonstrate geothermal binary power plant technology at an Indonesian site, a technical plant concept has been developed that complies with different requirements. The binary plant should use a quite small temperature difference in order to avoid scaling and preserve the possibility of hot reinjection. It should be possible to integrate a fully automatic binary plant at an already existing, predominantly manually operated geothermal field. And furthermore components from both, German and Indonesian suppliers should be used. Within this paper the design concept is explained and operational data and experiences are shared.

1. INTRODUCTION

Indonesia is known for its tremendous geothermal potential of around 29 GW_e which is dominated by wet steam fields. The currently installed capacity amounts to 2 GW from 15 areas (Darma 2016, Richter 2018a) still leaving huge part of the geothermal potential untapped. The prevailing plant type in Indonesia is currently the single-flash which directly uses the vapor part from the produced vapor-liquid-mixture to drive the turbine. Binary plants which transfer the geothermal heat to a separate working fluid are not yet an established technology at Indonesian sites. The first commercial binary units have been commissioned at Sarulla field in North Sumatra just in 2017 (Wolf & Gabbay 2015, Richter 2018b). Due to their adaptability they could however be implemented at much more sites and help to increase the geothermal capacity in Indonesia.

In order to successfully demonstrate geothermal binary power plant technology at an Indonesian site and to intensify the know-how transfer in this technology field a German-Indonesian collaboration project has been initiated in 2013 involving GFZ Potsdam (Germany), the Agency for the Assessment and Application of Technology in Indonesia (BPPT) and PT Pertamina Geothermal Energy (PGE).

The demonstration plant is located in the Lahendong geothermal area close to the village Pangolombian in the northern part of the island Sulawesi. The on-site construction phase of the demonstration plant, rated with a capacity of 500 kW, started in 2015 (Frick et al. 2015). Technical concept, component specification, coordination and supervision of detail planning, construction and commissioning were executed by the project consortium under guidance of GFZ. The operational phase commenced in September 2017. In January 2019 the demonstration plant was handed over to the Indonesian consortium.

This paper describes the technical concept and summarizes the first operational experiences of the demonstration plant that has been integrated at an already existing geothermal site.

2. STATE OF KNOWLEDGE

Figure 1 shows different types of integration for typical geothermal binary plants. About 20 % of today's binary plants complement the direct steam use at a geothermal high enthalpy site (Figure 1, left). In case of produced wet steam, the liquid phase from the separator can be used as heat source. But it is also possible to utilize the waste heat from the direct-steam turbine for driving a binary cycle. About 35 % of the installed geothermal binary plants use the complete fluid flow from a geothermal high- or medium-enthalpy field (Figure 1, middle). In most cases, steam and liquid phase are separated and used for evaporation and preheating, respectively. The remaining 45 % of the geothermal binary plants use hydrothermal systems (Figure 1, right).

When realizing binary power plants, different technical decisions have to be made. Important topics are working fluid selection, process design, turbine and heat exchanger specification (e.g. Lakew & Bolland 2010, Gao et al. 2012, Maraver et al. 2014, Toffolo et al. 2014, Walraven et al. 2014). Many authors thereby agree that a thermodynamic analysis is not sufficient and that a techno-economic evaluation based on different criteria is necessary. Different approaches for a techno-economic design point evaluation are presented and discussed by Hettiarachchi et al. 2007, Quoilin 2011, Shengjun et al. 2011, Cataldo et al. 2014, Li et al. 2014 and Astolfi et al. 2014.

A specific design constraint for designing geothermal binary power plants is the limitation for the cooling of the geothermal fluid in order to avoid scaling and material aspects regarding corrosion (Franco 2011, Bronicki 2013, Wendt & Mines 2010) or in order to realize hot fluid reinjection close to the production wells (Noorolahi & Itoi 2011). Another important aspect is the consideration of changing operating conditions. Sanyal et al. 2005, Gabbrielli 2012 and Budisulistyo et al. 2017 discuss possible design strategies to deal with the degradation of the geothermal resource over the lifetime of the plant. Astolfi et al. 2011, Manente et al. 2011 and Usman et al. 2017 address the effect of variable ambient when using direct air-cooled condensers (ACC). Astolfi et al. 2019 investigate in this context the implementation of an intermediate cooling water cycle for being able to use both, a compact water-cooled condenser and a flexibly operable dry cooler with modular and adiabatic pre-cooling mode.

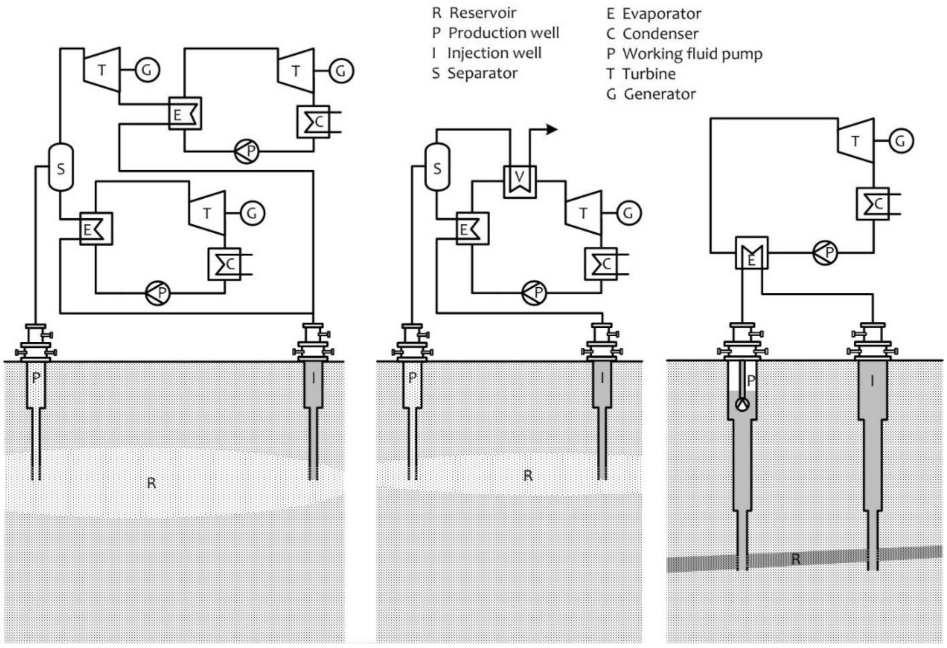

Figure 1: Site integration of geothermal power plants

3. TECHNICAL CONCEPT AND DESIGN DATA

The demonstration plant has been integrated at the Lahendong geothermal field close to the village Pangolombian where geothermal brine with a temperature of about 170°C corresponding to a separator pressure of 7,9 bar$_a$ was available. The cool down of the brine should be limited to 140°C in order to have the possibility for hot brine reinjection close to the production well. Based on the available project budget the electrical capacity was rated with 500 kW so that the necessary brine flow rate was estimated with 30 to 35 kg/s. Groundwater or Cooling water was not available.

For being able to handle a broad range of brine compositions and operate the binary plant with variable capacity without changing the brine supply as well as to meet also project specific constraints (e.g. pretesting of the ORC-prototype in Germany), it was decided to integrate the power conversion cycle by using intermediary closed water cycles for heat supply and heat removal. Aiming for high reliability a subcritical, single-stage ORC with internal heat recovery was chosen as conversion cycle. N-Pentane, a well-known working fluid suitable for the heat source temperature at the demonstration site, was selected as working fluid. The process diagram is shown in Figure 3. Solid lines indicate normal operation. Dotted lines indicate operation during start-up or shut-down of the demonstration plant.

During normal operation, the heat of the brine is transferred to the hot water cycle in the primary heat exchanger. The hot water is then used to heat up and evaporate the working fluid in the ORC preheater and evaporator. The hot water is continuously circulated and the pressure in the hot water cycle is maintained by means of an expansion vessel with nitrogen cushion.

In the ORC-unit the working fluid vapor drives the turbine-generator-unit. After the turbine, the superheated working fluid vapor is flowing through a recuperator register before getting in touch with the water-cooled condenser tubes. The heat removal in the cooling water cycle is realized with a dry cooler consisting of 6 units.

Utilizing intermediary water cycles, the net power output is decreased due to additional heat resistance and additional power consumption of the pumps. The net power loss due to the hot water cycle has been estimated with about 13 % and the loss due to the cooling water cycle with about 11 %, both compared to a plant without intermediary cycle. However, in this project the intermediary water cycles were realized due to design, installation and operational advantages. Therefore it was possible to transport and install a completely preassembled and pretested ORC-unit which was also advantageous since the ORC unit was a prototype as well.

Using the hot water cycle, it was further possible to operate, shut-down and start-up the binary plant without changing the existing operational regime of the brine supply. The used prototype ORC-unit can only be started at low supply temperatures since large temperature differences between hot water and working fluid can lead to steam hammer in the evaporator. Therefore, an additional dry cooler is necessary to cool down the hot water before restart (e.g. after shut down due to grid black out) and the controlled valves are used to realize a defined temperature ramp. The hot water by-pass pump is implemented to realize modest temperature differences around the primary heat exchanger during restart and to reduce the risk of steam hammer on the brine side during brine line start up.

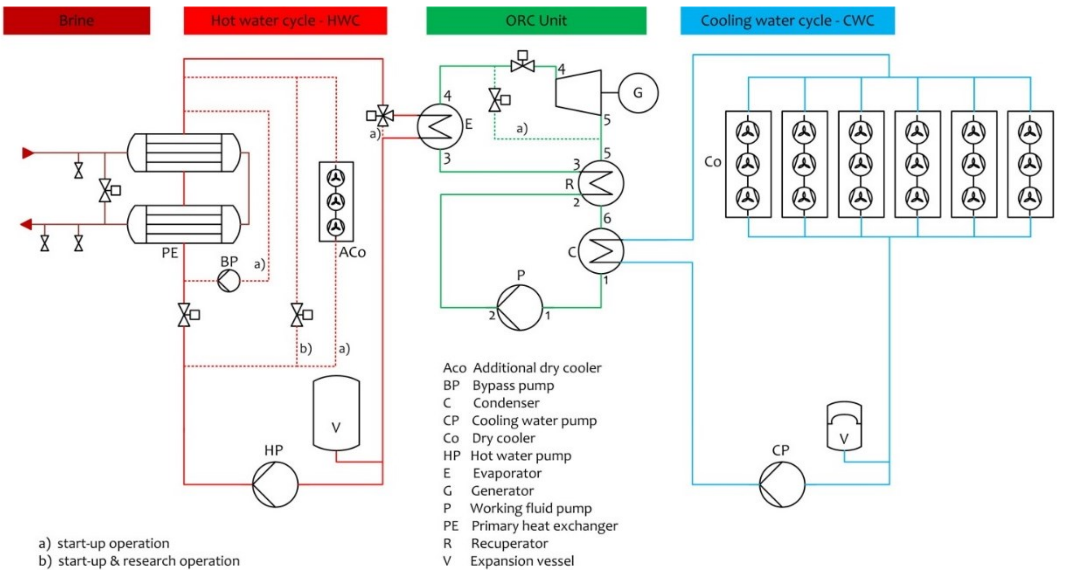

Figure 2: Flow diagram of the demonstration power plant

Another advantage of the hot water cycle is the different design priorities for the heat exchanger that can be realized. For the primary heat exchanger transferring the heat from the brine to the hot water, the accessibility of the tubes for cleaning procedures and corrosion resistant materials have a high priority. The evaporator design can focus on heat transfer since there are less constructive restrictions. An advantage of implementing the cooling water cycle was that the design and operation of the dry cooler becomes more reliable due to the well-known single phase heat-transfer of water. Furthermore, less working fluid was needed to fill the ORC-cycle and the risk of n-pentane leakage could be decreased. The practical advantages of an intermediary closed water cycle are also addressed by Astolfi et al. 2019. The 3D-layout illustrated the component positioning and is shown in Figure 3

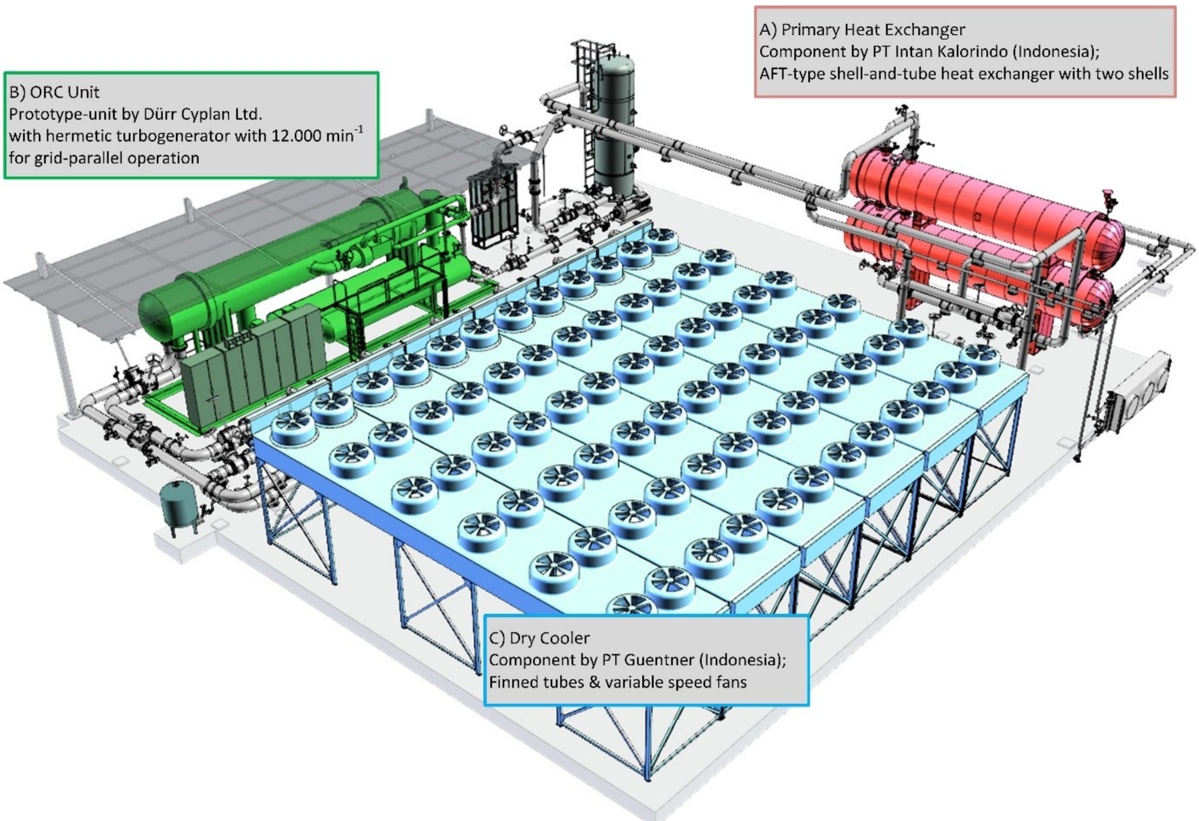

Figure 3: 3D-Plant layout

The main process design data are listed in Table 1. For the geothermal fluid supply, a design mass flow $\dot{m}_{geo}$ of 32 kg/s was considered. Supply and return temperature $T_{geo,i}$ and $T_{geo,o}$ of the geothermal liquid were assigned with 172°C and 142°C, respectively. The heat capacity rate of the hot water was adapted to the heat capacity rate of the geothermal fluid in order to minimize the exergy destruction in the primary heat exchanger. Due to the fact that the heat transfer area increases significantly with lower pinch points, the minimum temperature difference between geothermal fluid and hot water was set with 7 K.

Using a relatively small temperature spread at the heat supply, the process design of the ORC-unit had to be adapted. From Figure 4 it can be seen that the evaporation temperature delivering the maximum power output would lead to a hot water outlet temperature of app. 120°C which results in a brine outlet temperature much lower than 140°C. Therefore the outlet temperature and not the power output was decisive for the selection of the evaporation temperature.

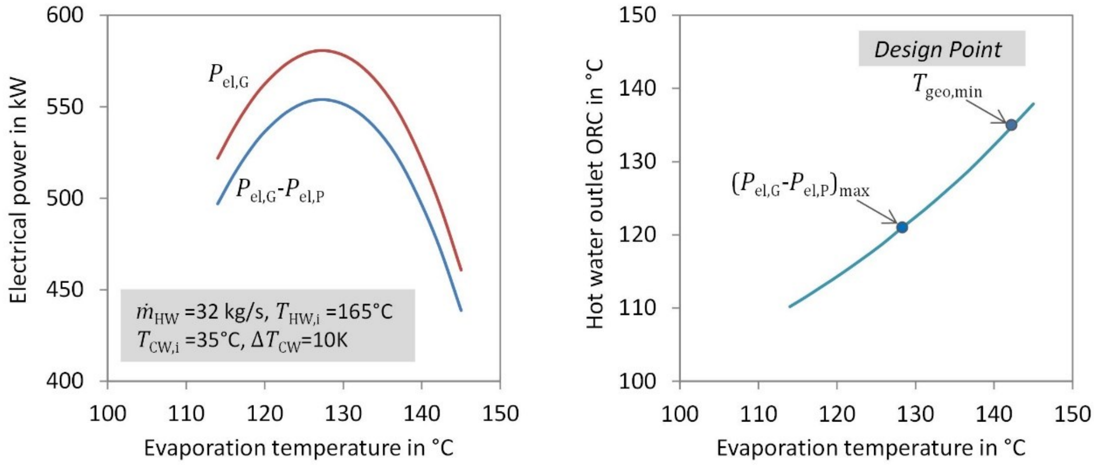

Figure 4: Plant design criteria: design evaporation temperature considering the limitation of brine cooling

The condensation temperature of the ORC-unit was selected based on the maximum net power output considering the power consumption of all components, also the cooling water pump and the dry cooler fans. Since a lower condensation temperature leads to a higher gross power output but also higher fan power consumption, an optimum condensation temperature exists yielding the maximum net power output for a given dry cooler size. Based on the evaluation of different quotations with different dry cooler sizes, the dry cooler cross-flow section was defined with 120 m^2.

Table 1: Design data of the demonstration plant

	Temperatures		Mass flow rates		Electrical power	
Brine supply	$T_{geo,i}$ °C	172	$\dot{m}_{geo}$ in kg/s	32		
	$T_{geo,o}$ °C	142				
Hot water cycle	$T_{HW,i}$ in °C	165	$\dot{m}_{HW}$ in kg/s	32	$P_{el,HP}$ in kW	8.9
	$T_{HW,o}$ in °C	135				
Cooling water cycle	$T_{a,i}$ in °C	25.0	$\dot{m}_a$ in kg/s	221.4	$P_{el,F}$ in kW	25.4
	$T_{a,o}$ in °C	40.7				
	$T_{CW,i}$ in °C	34.0	$\dot{m}_{CW}$ in kg/s	86.1	$P_{el,CP}$ in kW	16.2
	$T_{CW,o}$ in °C	44.0				
ORC-unit	T_{ev} in °C	142.6	$\dot{m}_{WF}$ in kg/s	9.5	$P_{el,G}$ in kW	499.4
	T_{Cd} in °C	49.4			$P_{el,P}$ in kW	23.4
Net power					$P_{el,Net}$ in kW	425.5

4. EXPERIENCES AND PLANT OPERATION

At the moment, the plant is operating with reduced power which is sufficient to supply the injection pumps nearby and feed in the surplus electricity in the local grid. The operating power is thereby manipulated by controlling the hot water inlet temperature to the ORC unit with the automatic valves in the hot water cycle. The main reason for the power reduction is to reduce the wear of the turbogenerator bearings which are most stressed during start-up and shut-down in case of failure in the electrical grid.

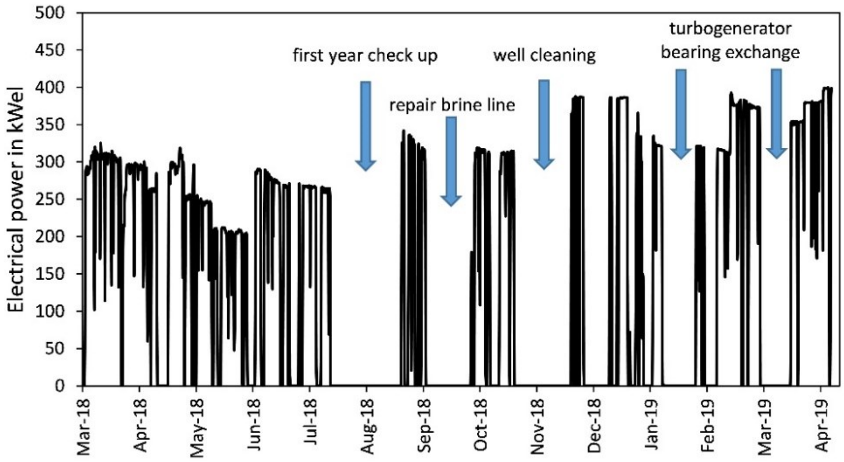

Figure 5: Plant operation (power production) from March 2018 until April 2019

Figure 5 shows the power production between March 2018 and April 2019. It can be seen that the plant operation was very discontinues with 112 shut down and start up. Besides some maintenance work, most stops where caused by electrical grid failure. Figure 6 left shows a 3-day-period in April 2019. It can be seen that gross power and net power show a daily fluctuation which is related to changing ambient temperatures (see Figure 6 right). The cooling water supply temperature can be kept constant until an ambient temperature of about 27°C by means of variable speed fan operation. For higher ambient temperatures the fans are operated at full speed and the cooling water supply temperature follows the course of the ambient temperature. The stronger fluctuation of the net power is therefore caused by the changing ambient temperature and the variable fan power consumption.

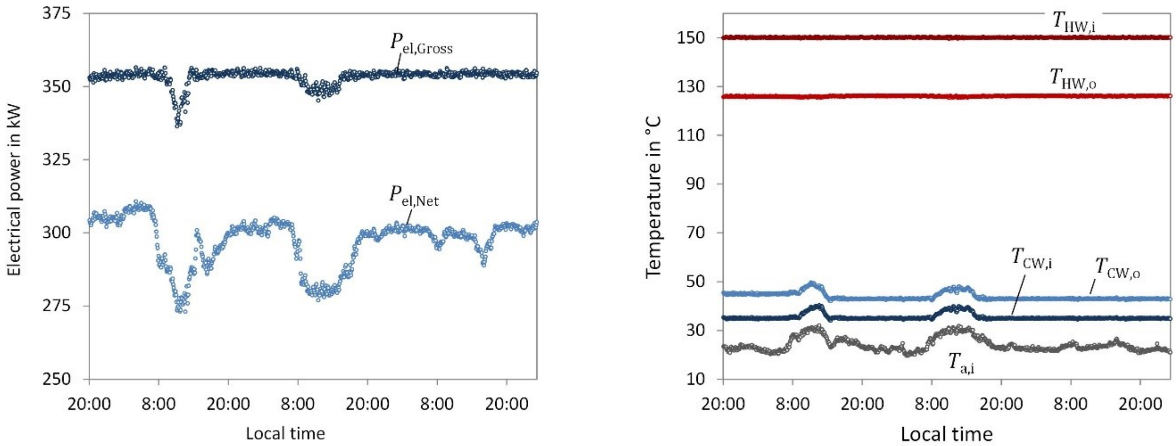

Figure 6: Plant operation (left) during 3 days in April 2019 showing daily fluctuations due to ambient temperature fluctuation (right)

In Figure 7 the temperatures and pressures of the brine supply are shown in relation to the saturation pressure of water. It can be seen that the supply pressure is very close to the saturation pressure and that there is only little cooling of the brine, both facts indicating two-phase flow into and liquid flow out of the primary heat exchanger.

Due to the operation at off-design conditions, the maximum proven gross power until now is 400 kW. In order to evaluate the maximum power output of the demonstration plant, a numerical plant model has been developed which will be described shortly in the next chapter.

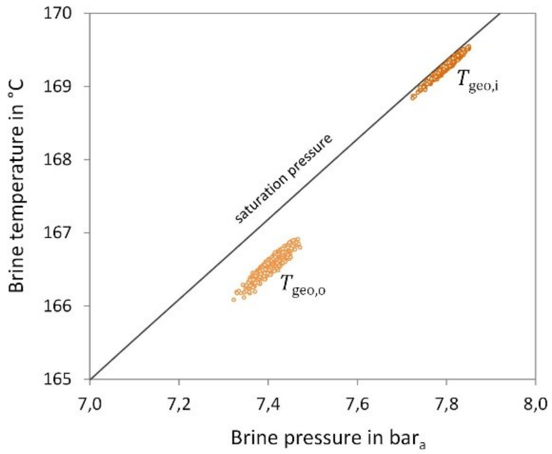

Figure 7: The temperatures and pressures of the brine supply

5. MODEL BASED POWER PREDICTION

In order to evaluate the maximum power output of the demonstration plant, a numerical plant model has been developed. The plant model has been developed with the software environment Engineering Equation Solver (EES)[1] and comprises the hot water cycle, the ORC-unit and the cooling water cycle. For modeling of the heat exchangers, heat transfer and pressure loss correlations from the VDI heat atlas (VDI Gesellschaft 2010) have been used. The turbogenerator is represented as a turbine wheel with upstream laval-nozzle stage and the isentropic turbine efficiency has been adapted based on operational data.

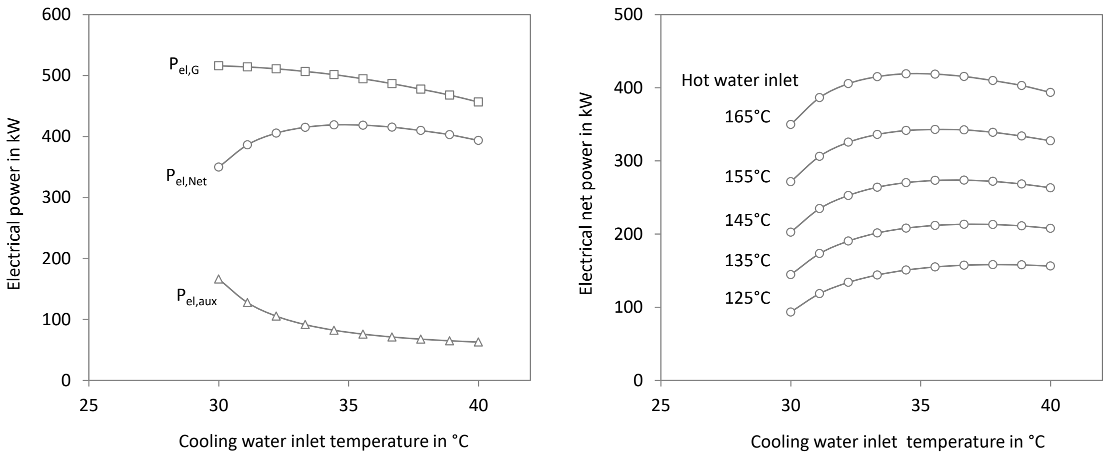

Figure 8: Estimated plant performance depending on the cooling water temperature for different hot water inlet temperatures ($T_{a,i}$=25°C)

With the model, the power output of the binary demonstration plant can be calculated for varying hot water and cooling water temperatures. In Figure 8 (left) the gross and net power output for a saturated vapor cycle depending on cooling water temperature with a hot water inlet temperature of 165 °C is shown. The net power reaches a maximum depending on the cooling water inlet temperature and increases with higher hot water temperatures. The net power reaches a maximum depending on the cooling water inlet temperature and increases with higher hot water temperatures. In Figure 8 (right) the power increase with increasing hot water inlet temperature is shown.

6. CONCLUSION AND OUTLOOK

The presented technical concept was developed to produce power from a quite small brine temperature change in order to avoid scaling and preserve the possibility of hot reinjection and to integrate a fully automatic binary plant at an already existing geothermal site. Within this paper the feasibility of this technical concept could be shown.

The presented technical concept integrates an Organic Rankine Cycle unit using an intermediary hot water and cooling water cycle. Even though the net power output is decreased by using intermediary cycles, the technical and organizational advantages explained in the paper prevailed. The operational phase has even shown that the demonstration plant could not have been operated without intermediate hot water cycle. Due to changing well (operating) conditions the binary plant is supplied with two-phase flow instead of pure liquid as previously designed. It can be concluded that intermediate cycles, especially the hot water cycle, are interesting

[1] http://www.fchart.com/ees/

technical solutions also for other projects. However, cost reduction potentials (e.g. decreasing expansion vessel size or using thermal oil instead of water) should be evaluated.

Besides high ambient humidity, the availability of the electrical grid is the largest technical challenge for the operation of the demonstration plant. The binary plant has been designed for grid-parallel operation and is not capable for electrical island mode operation. As a result of power outages, and phase failure, the plant has experienced over 150 plant stops and starts until now. Due to technical modifications an automatic re-start is now possible for various situations. For the future it is planned to change the grid connection point in order to connect to a more stable electrical grid. A modification of the ORC-unit for grid-parallel and island operation would be technically possible but is much more costly.

The plant is currently operated with limited power by controlling the hot water supply temperature to the ORC-unit in order to meet the consumption of the injection pumps on site. Real plant data for typical operation where shown. Due to the technical concept it is possible to adapt the plant operation in order to realize also low auxiliary power consumption at different operating conditions. Due to the operation at off-design conditions, the maximum proven gross power until now was 400 kW.

In January 2019, the demonstration plant was handed over to the Indonesian consortium that is now operating and maintaining the plant. Besides commercial operation, the plant will also be used for demonstration activities and training. It is also planned to make technical improvements regarding the connection to the electrical grid in order to increase the plant availability.

The experiences and results from this demonstration project will be helpful for other projects especially for upgrading existing flash power plants by additional binary power plants. The experience shows that intermediate cycles, especially the hot water cycle without which the demonstration would not be operable, are interesting technical solutions. However, cost reduction potential should be evaluated.

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LIST OF ABBREVIATIONS

$\dot{m}_a$	Air mass flow rate in kg/s
$\dot{m}_{geo}$	Brine mass flow rate in kg/s
$\dot{m}_{CW}$	Cooling water mass flow rate in kg/s
$\dot{m}_{HW}$	Hot water mass flow rate to ORC-unit in kg/s
$\dot{m}_{WF}$	Working fluid mass flow rate in kg/s
$P_{el,aux}$	Electrical auxiliary power in kW
$P_{el,CP}$	Electrical power cooling water pump in kW
$P_{el,HP}$	Electrical power hot water pump in kW
$P_{el,F}$	Electrical power dry cooler fans in kW
$P_{el,G}$	Electrical power generator in kW
$P_{el,Gross}$	Electrical power feed-in unit (gross power) in kW
$P_{el,Net}$	Electrical net power in kW
$P_{el,P}$	Electrical power working fluid pump in kW
p_{ev}	Evaporation pressure working fluid in bar
$T_{a,i}$	Air inlet temperature to dry cooler in °C
$T_{a,o}$	Air outlet temperature from dry cooler in °C
T_{cd}	Condensation temperature working fluid in °C
$T_{CW,i}$	Cooling water inlet temperature to ORC-unit in °C
$T_{CW,o}$	Cooling water outlet temperature from ORC-unit in °C
T_{ev}	Evaporation temperature working fluid in °C
$T_{HW,i}$	Hot water inlet temperature to ORC-unit in °C
$T_{HW,o}$	Hot water outlet temperature from ORC-unit in °C
$T_{goe,i}$	Brine inlet temperature to primary heat exchanger in °C
$T_{goe,o}$	Brine outlet temperature from primary heat exchanger in °C

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