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An Improved Hybrid and Cogeneration Cycle for Enhanced Geothermal Systems

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ABSTRACT

Geothermal energy is one of alternative energies that should be used as optimally as possible. Conversion of geothermal energy to be electric energy plays important role in saving energy resource due to advantages of utilization of electric energy. Energy state of steam before entering turbine significantly influence in specific output power. Shortcoming of steam quality from geothermal well is one factor causing most geothermal direct expansion power plant produces low thermal efficiency.

A hybrid system is proposed to enhance geothermal system. Modification of a conventional geothermal power plant is conducted by combining with a gas turbine, a refrigeration-heat pump, and organic Rankine cycles. Quality of steam from the geothermal well should be heated before entering the steam turbine. Organic Rankine cycle is included to utilize low thermal energy of exhaust gas to generate power. A refrigeration and heat pump system is introduced to cool inlet air into gas turbine and as an additional heat source of working fluid in an organic Rankine cycle. Most evolved parameters are investigated such as temperature, pressure, working fluid, and pressure ratio. The hybrid system can improve significant efficiency from 19.85% to be 35.83%. Fossil fuel utilization factor of the hybrid system is up to 63.81%. Increment of net output power and efficiency of the direct expansion cycle in the hybrid system are 42.70% and 15.98%, respectively, compared to the conventional geothermal power plant. The best working fluid from thermodynamic aspect for organic Rankine cycle and vapor compression cycle is R-600.

1. INTRODUCTION

Growth of industrial activities and increasing human life style will influence in increasing of energy consumption either electrical energy or fossil fuels. Growth of electric infrastructure lower than electric energy consumption has caused an electric energy crisis in Indonesia. Indonesian electric generation in 2008 has shortcomings due to the power plant backup is inadequate when peak load time. Consequently, this shortcoming enforces us to shut down partial electric utilities. Besides a plan for establishing new power plant until 2010 year around 10,000 MWe, any shorten strategies have also been conducted by Indonesian government such as energy saving program and time usage diversity of industrial energy consumption by shifting working days. Recently, total nationally installed electric generator capacities are more 22,000 MWe. Among of them, 6300 MWe power plant capacity work with base of combine cycle, 6900 MWe power plant capacity works with base of Rankine cycle, 2700 MWe power plant capacity works with Brayton cycle, and 3000 MW power plant capacity works with Diesel cycle. The other power plants are working with bases of geothermal system and hydro system [Statistics Indonesia: Statistics Energy, 2008].

More 80% of the energy input to a typical geothermal generation system is wasted as heat rather than conversed into electricity. Steam from the geothermal well should be heated before expanding in a steam turbine since its superheat degree is very low. This heating way can improve thermal efficiency of the power plant. Bidini et al. (1998) reported their work to enhance performance of conventional geothermal plant. Our previous work (Astina et al. 2008a) presented a similar effort to improve performance of conventional geothermal. Both these research results confirm that improvement possibility of the geothermal system by combining with fossil fuel power plant to solve current shortcomings. More open view to judge of geothermal heat utilization for power generation was discussed either on first and second law thermodynamic analysis in our previous work (Astina 2010).

Polyzakis et al. (2008) concerned on optimal condition design of gas turbine system in a combined cycle. It was pointed out that the optimum gas turbine cycle to operate in a combined cycle power plant came out to be the reheated cycle. This performs the steam cycle efficiency of 36.6% and the overall efficiency of 53.5%. Increase of gas turbine system by air-inlet cooling is interesting to investigate. Punwani et al. (2001) reported his work on implementation of air inlet cooling for Calpine Clear Lake power plant. Technical and economic aspects are discussed in their paper.

A methodology for the economic evaluation of the cooling systems was proposed by Gareta et al. (2004). This method offers more straightforward information that allows the cooling equipment technology to be sized or selected. In accordance with recent economic condition, Astina et al. (2008b) reported thermoeconomic analysis of implementation of air-inlet cooling system for Indonesian power plant. This result confirms that implementation of air-inlet cooling system for Indonesian power plant gives significantly financial benefit.

These all motivate us for developing cooling, heating and power systems in implementation for geothermal power plant. Organic Rankine cycle (ORC) is included in the system to utilize low thermal energy for generating electric power. A refrigeration and heat pump system is introduced to cool inlet air gas turbine and as additional heat source of working fluid in the ORC. Gas turbines are a great machines producing electric power, which their power output increases when ambient air temperature entering compressor is decreased. This gives opportunity to sell more electric power when the inlet air of the gas turbine system is cooled. The electric power generating system is located near a heat load that utilizes the heat rejected from the generation equipment to improve their efficiency. High temperature of exhaust gas from the gas turbine also gives recovery opportunity by using heat exchanger for increasing temperature of steam from the geothermal well and then the heat waste is utilized by the ORC.

2. NEW GEOTHERMAL HYBRID CYCLE

The new geothermal hybrid cycle for improving geothermal power plant is shown in Figure 1. This system consists of geothermal steam direct expansion (STG) cycle, Brayton cycle, vapor compression cycle and ORC. Waste heat from the gas turbine (GT) system is recovered to heat steam from the geothermal well in a steam heater. Remaining heat in a hot gas from the steam heater is recovery again in an organic heater to heat the working fluid of ORC. Vapor compression cycle included in the hybrid system has two functions, i.e. a refrigeration system for air entering compressor, and a heat-pump system for heating the organic working fluid before entering an organic heater.

3. METHODOLOGY

In order to make convenient simulation, software was developed and it is supported by more than 15 fluid types. Most the properties are calculated directly from thermodynamic equation of state in accordance with recent trend of the calculation method for thermophysical properties. The calculation method and program code was adapted from program codes on development of equation of state (Astina, 2003). This method was reported in our previous work (Phommavongsa et al., 2005). Most available equations of state that had been published are included in database.

Cooling load of air in evaporator is considered as moist air so that the psychometric relations are involved in the calculation. Air as dry air is considered in the gas turbine analysis and it is evaluated based on an equation of state (Lemmon et al., 2000). The purposed cycle is established to evaluate performance for various conditions. Any parameter is optimized to find the best performance from thermodynamic aspect.

All computation works are conducted in program codes. The codes were written in C++ and software was developed in friendly interface as shown in Figure 2. Any parameter input is needed to simulate the proposed hybrid cycle. Cycle view menu and analysis result are included in the software.

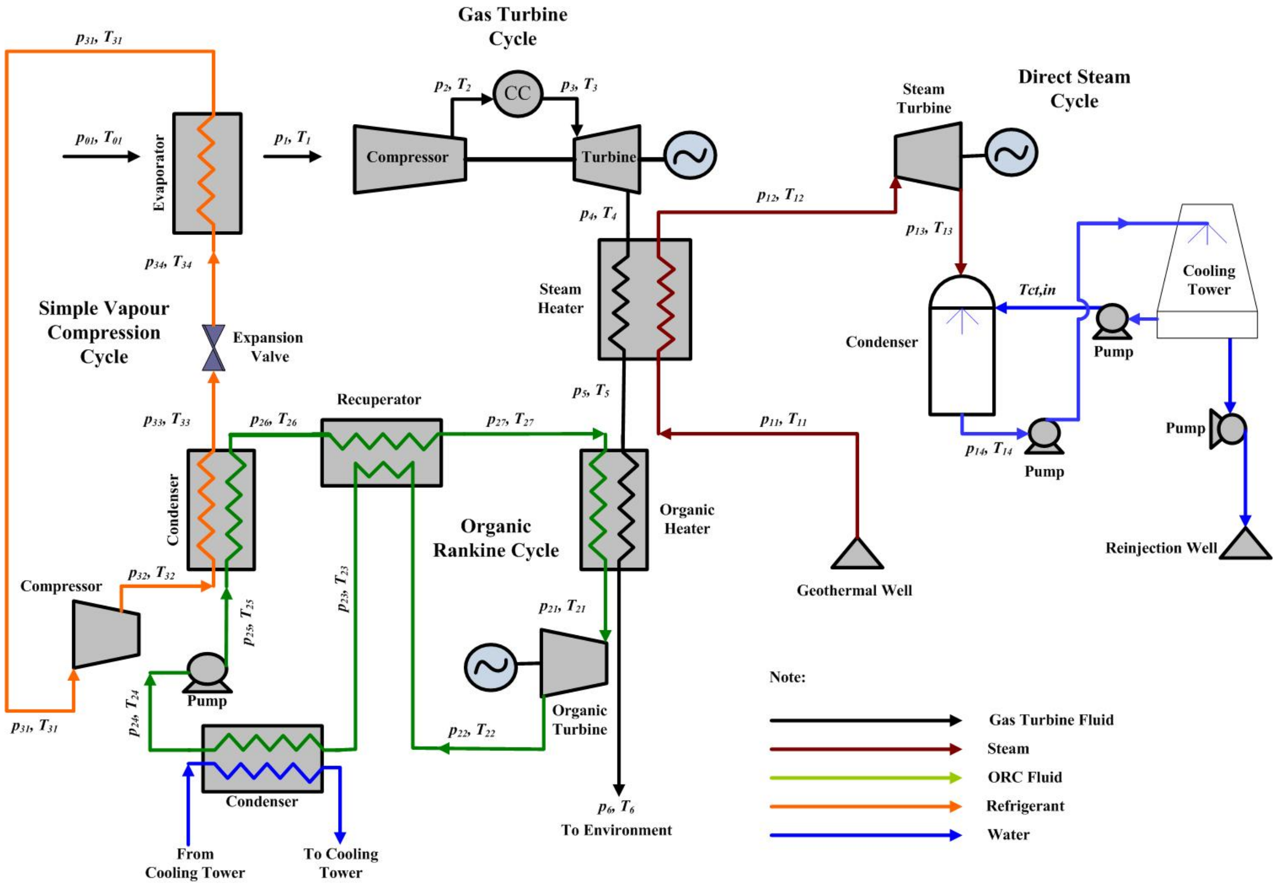

Figure 1: Schematic diagram of an improved hybrid and cogeneration cycle for enhanced geothermal systems

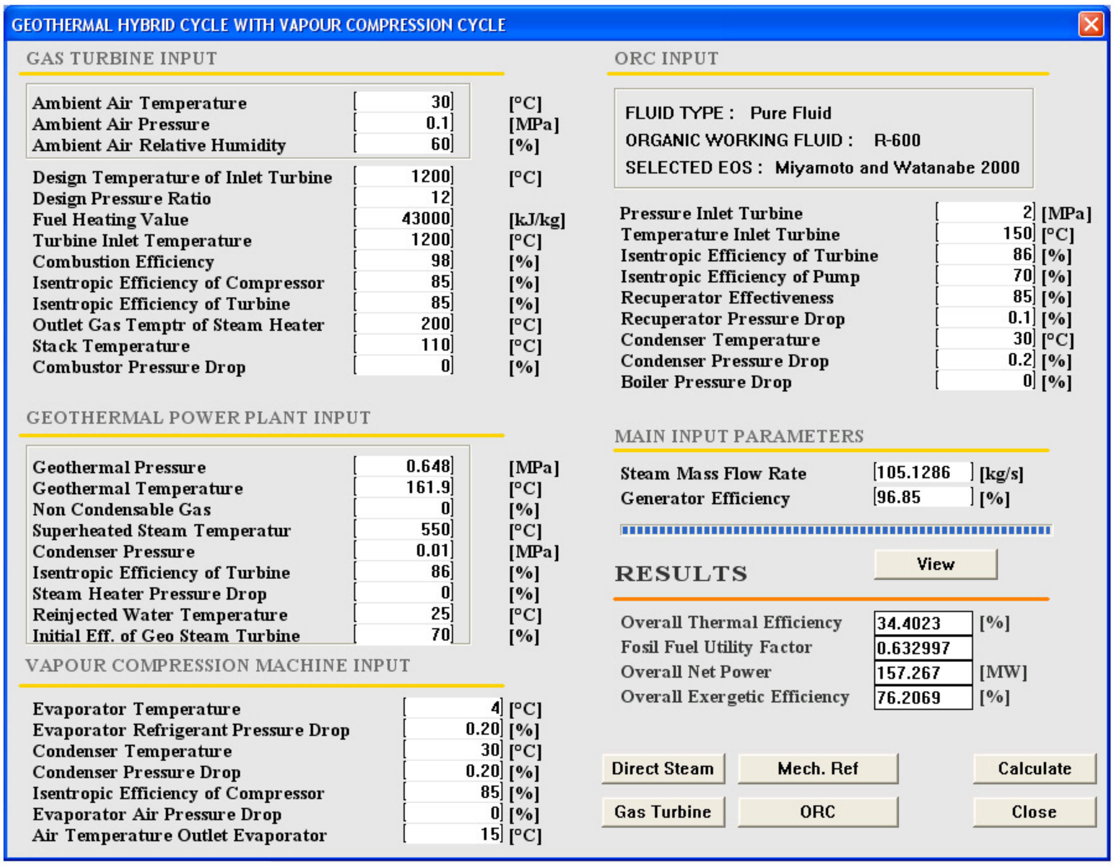

Figure 2: Software interface of an improved hybrid and cogeneration cycle

4. DATA AND ASSUMPTIONS

Any field data and assumption are taken to conduct simulations. Any related literature including one of geothermal well of Kamojang Power plant as a case study object are considered in this study. The data input of gas turbine system covers 900°C of turbine inlet temperature (design), 15°C of air inlet temperature (design), ambient air condition at 30°C and 0.1 MPa, fuel heating value of 43,000 kJ/kg, combustion efficiency of 98%, isentropic compressor efficiency of 85%, isentropic turbine efficiency of 86%, exit hot gas temperature from the steam heater at 200°C, stack temperature of 110°C, and no pressure drop in combustor. The data for geothermal system are geothermal source pressure and temperature at 0.648 MPa and 161.9°C, respectively, condenser pressure at 0.01 MPa, isentropic steam turbine efficiency of 70% for the conventional system (before modification), no pressure drop in the steam heater and water re-injection temperature at 25°C.

Vapor compression cycle has function as refrigeration and heat pump system. Evaporator temperature and pressure drop in refrigerant side are 4°C and 0.20%. Condenser temperature and pressure drop are 30°C and 0.20%. Compressor isentropic efficiency is 85% and no pressure drop of air in the evaporator. The data needed for ORC covers recuperator effectiveness of 85%, no pressure drop in recuperator, condenser temperature at 25°C, no pressure drop of working fluid ORC and steam in organic heater. Other main parameters are 105.1286 kg/s of geothermal steam mass flow rate and 96.85% of generator efficiency referring to the field data.

5. THERMODYNAMIC PARAMETERS

Thermodynamic evaluations for geothermal power plant systems including energy and exergy analyses were exploited in previous paper [Astina 2010]. In order to keep thermodynamic consistencies and agree with a state reference used in IAPWS (International Association of Properties for Water and Steam), the earth is approached as a boiler so that the cycle works as closed cycle. Efficiency of the hybrid cycle is calculated in accordance with equation (1).

$$\eta_{\text{Hybrid}} = \frac{\dot{W}_{\text{Net, GT}} + \dot{W}_{\text{Net, STG}} + \dot{W}_{\text{Net, ORC}} - \dot{W}_{\text{Comp}}}{\dot{Q}_{\text{GT}} + \dot{Q}_{\text{Geo}}} \tag{1}$$

Where $\dot{W}_{\text{Net, GT}}$ is total net output power produced by two gas turbine generators after it is subtracted by the compressor work; $\dot{W}_{\text{Net, STG}}$ is total net output electric power produced by a steam turbine generator after it is subtracted by the pump works; $\dot{W}_{\text{Net, ORC}}$ is total net output power produced by ORC after the turbine generator is subtracted by pump power; $\dot{W}_{\text{Comp}}$ is power needed to drive heat-pump-refrigeration system; $\dot{Q}_{\text{GT}}$ is input heat rate into combustor as combustion result. Energy input from geothermal into Rankine cycle is $\dot{Q}_{\text{Geo}} = \dot{m}_{\text{SG}}(h_g - h_{\text{in}})$.

In order to know the effect of utilization of fuel in the conventional geothermal power plant, fossil fuel utility factor (FFUF) is introduced. The FFUF is defined as ratio of increment power net of the hybrid system respect to the energy input of fuel into the gas turbine. The mathematic relation may be expressed as (2).

$$\mathrm{FFUF} = \frac{\dot{W}_{\mathrm{Net,GT}} + W_{\mathrm{Net,ORC}} + \Delta\dot{W}_{\mathrm{STG}} - \dot{W}_{\mathrm{Comp}}}{\dot{Q}_{\mathrm{GT}}} \quad (2)$$

where $\Delta\dot{W}_{\mathrm{STG}} = \dot{W}'_{\mathrm{STG}} - \dot{W}^{o}_{\mathrm{STG}}$; W'_{STG} is net output power from geothermal power system in the hybrid system; $\dot{W}^{o}_{\mathrm{STG}}$ is net output power from the conventional geothermal power system (direct expansion).

Exergy of input heat rate from the geothermal energy source can be evaluated similarly by taking dead state at p_0 and T_0 and applying the mass and exergy balances on control volume of the earth as boiler. Change of kinetic and potential energies are neglected, the input exergy of heat into a direct expansion cycle is then derived and obtained as equation (3).

$$\begin{aligned}\dot{\mathrm{E}}_{in} &= \dot{m}_e\left((h_e - h_0) - T_0(s_e - s_0)\right) \\ &- \dot{m}_i\left((h_i - h_0) - T_0(s_i - s_o)\right)\end{aligned} \quad (3)$$

Considering a dead state is a state at which is same as water entering into the earth, the equation (3) becomes an equation as given in equation (4).

$$\dot{\mathrm{E}}_{in} = \dot{m}\left((h_e - h_0) - T_0(s_e - s_0)\right) \quad (4)$$

Here is the mass flow rate of steam from the geothermal, h and s are enthalpy and entropy respectively, p and T are pressure and temperature respectively, and subscripts e and i are noticed as exit and inlet terminals. This equation may be used, but it does not mean that the geothermal power plant is analyzed as the open cycle, but it is a closed cycle in keeping thermodynamic consistencies either enthalpy or exergy properties.

The other parameters for each cycle that composes the hybrid cycle are evaluated in accordance with thermodynamic formulas described in the next section. These parameters just want to reveal from each cycle independently. Subscript number in the formulations are refered to the given number in the schematic diagram.

5.1 Direct Steam Parameters

Evaluation parameters in the direct steam cycle consist of turbine work, steam heater heat rate, geothermal heat rate, and the efficiency of geothermal direct steam cycle.

- Turbine Work Rate

$$\dot{W}'_{\mathrm{STG}} = \dot{m}_{\mathrm{SG}}(h_{12} - h_{13}) \quad (5)$$

- Steam Heater Heat Rate

$$\dot{Q}_{\mathrm{SH}} = \dot{m}_{\mathrm{SG}}(h_{12} - h_{11}) = \dot{m}_{GT}(h_4 - h_5) \quad (6)$$

- Geothermal Source Heat Rate

$$\dot{Q}_{\mathrm{Geo}} = \dot{m}_{\mathrm{SG}}(h_{11} - h_{in}) \quad (7)$$

In this case, net output power from all pumps in the direct steam cycle is neglected, because it is very small compared to the steam turbine output power.

- Thermal Efficiency

$$\eta_{\mathrm{STG}} = \frac{\dot{W}'_{\mathrm{STG}}}{\dot{Q}_{\mathrm{Geo}} + \dot{Q}_{\mathrm{SH}}} \times 100\% \quad (7)$$

5.2 Gas Turbine Parameters

Evaluation parameters in the gas turbine cycle consist of turbine work, combustor heat input, compressor work and thermal efficiency of the gas turbine cycle.

- Compressor Work Rate

$$\dot{W}_{\mathrm{C}} = \dot{m}_{\mathrm{GT}}(h_2 - h_1) \quad (8)$$

- Turbine Work Rate

$$\dot{W}_{\mathrm{T}} = \dot{m}_{\mathrm{GT}}(h_3 - h_4) \quad (9)$$

- Net Output Power

$$\dot{W}_{\mathrm{Net,GT}} = \dot{W}_{\mathrm{GT}} - \dot{W}_{\mathrm{C}} \quad (10)$$

- Heat Rate from Combustion

$$\dot{Q}_{\mathrm{GT}} = \dot{m}_{\mathrm{F}} LHV = \dot{m}_{\mathrm{GT}}(h_3 - h_2) \quad (11)$$

In this case, gas mass flow of the gas turbine before and after combustion chamber is assumed constant.

- Thermal Efficiency

$$\eta_{\mathrm{GT}} = \frac{\dot{W}_{\mathrm{Net,GT}}}{\dot{Q}_{\mathrm{GT}}} \times 100\% \quad (12)$$

5.3 Organic Rankine Cycle Parameters

Evaluation parameters in the ORC consist of turbine work, organic heater heat rate, recuperator heat rate, pump work and thermal efficiency of the ORC.

- Turbine Power Output

$$\dot{W}_{\mathrm{ORC}} = \dot{m}_{\mathrm{ORC}}(h_{21} - h_{22}) \quad (13)$$

- Organic Heater Heat Rate

$$\dot{Q}_{\mathrm{OH}} = \dot{m}_{\mathrm{ORC}}(h_{21} - h_{27}) = \dot{m}_{\mathrm{GT}}(h_5 - h_6) \quad (14)$$

- Vapour Compression Condenser Heat Rate

$$\dot{Q}_{\mathrm{Cond}} = \dot{m}_{\mathrm{ORC}}(h_{26} - h_{25}) = \dot{m}_{\mathrm{Ref}}(h_{32} - h_{33}) \quad (15)$$

- Net Power

$$\dot{W}_{\mathrm{Net,ORC}} = \dot{W}_{\mathrm{ORC}} - \dot{W}_{\mathrm{P,ORC}} \quad (16)$$

- Thermal Efficiency

$$\eta_{\mathrm{ORC}} = \frac{\dot{W}_{\mathrm{Net,ORC}}}{\dot{Q}_{\mathrm{OH}} + \dot{Q}_{\mathrm{C}}} \times 100\% \quad (17)$$

5.4 Vapour Compression Machine Parameters

Evaluation parameters in the refrigeration-heat pump system consist of compressor work, coefficient of performance of the cycle.

- Compressor Work Rate

$$\dot{W}_{\mathrm{Comp}} = \dot{m}_{\mathrm{Ref}}(h_{32} - h_{31}) \quad (18)$$

- Evaporator Heat Rate

$$\dot{Q}_{\mathrm{Eva}} = \dot{m}_{\mathrm{Ref}}(h_{31} - h_{34}) = \dot{m}_{\mathrm{GT}}(h_{01} - h_1) \quad (19)$$

- Coefficient of Performance of the Refrigeration

$$COP_{\mathrm{Refrig}} = \frac{\dot{Q}_{\mathrm{Eva}}}{\dot{W}_{\mathrm{Comp}}} \quad (20)$$

- Coefficient of Performance of the Heat Pump

$$COP_{\mathrm{HP}} = \frac{\dot{Q}_{\mathrm{Cond}}}{\dot{W}_{\mathrm{Comp}}} \quad (21)$$

6. RESULT AND ANALYSIS

Simulations were conducted to reveal thermodynamic performance behaviors of the new hybrid system. The developed software was intensively used to simulate most important thermodynamic calculation. The results for assessment of each cycle consisting hybrid cycle are based on gas turbine hot gas temperature at 900^{o}C, and working fluid for ORC and vapor compression cycle is R-134a and 8 of pressure ratio. On the other hand, when analysis of the integrated hybrid system, it is based on condition, i.e. 12 of gas turbine pressure, 1200^{o}C of the hot gas temperature, R-600 as working fluid for ORC and the vapor compression cycle and the same temperature of air entering the compressor. Other conditions are referred to the previous data and assumption.

6.1 Gas Turbine Performance

Figure 3 shows the any power performed in the hybrid system changes as consequence of the gas turbine air inlet cooling temperature. Compressor work relates to volumetric flow of air. Therefore, for the same compressor, increase of mass flow rate of air that caused by air inlet cooling will not significantly affects on performance and work rate of the compressor. Additionally, compressor is mostly designed for 15^{o}C and 60% of RH in accordance with ISO. This agrees with the result in shown in Figure 4 that compressor work is not significantly affected by the inlet air temperature.

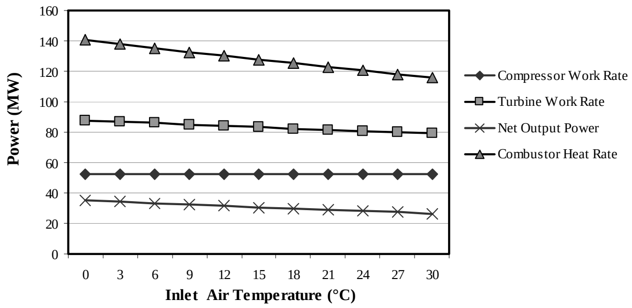

Figure 3: Air inlet cooling affect on gas turbine system

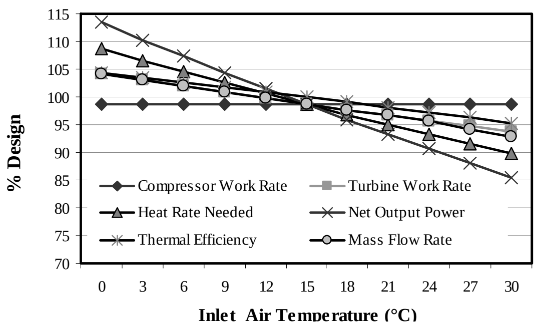

Figure 4: Gas turbine performance respect to cooling

Based on ISO condition of inlet air as a reference state, most performance parameters of the gas turbine system is influenced by the air inlet cooling temperature as shown in Figure 4. Cooling load increases with decreasing the air inlet temperature into the compressor. However, this effort gives benefit outcomes, i.e. higher thermal efficiency of the gas turbine system and net output power of the gas turbine, even though the input heat into a combustor and air mass flow rate increases. One parameter value near constant in this case is that compressor work rate needed since the compressor work rate is proportionally to volumetric rate and pressure ratio. This simulation results are varied from 30^{o}C to 0^{o}C of air inlet temperature, the net output power of the gas turbine increases around 32.67%. The gas turbine efficiency increment can also be achieved around 9.66%.

Other parameters also varied are pressure ratio in a gas turbine system and gas temperature entering the gas turbine. Effects of these parameters are also revealed from performances of the gas turbine system and the total hybrid system. In these simulations, R-290 is used as working fluid for ORC and refrigeration systems. Figures 6 and 7 show the behaviors of any parameters obtained from this simulation.

Behavior of net output power of the gas turbine system as shown Figure 5 is influenced by pressure ratio and highest hot gas temperature in the gas turbine system. This result is obtained for exhaust hot gas temperature from the gas turbine is 450°C and R-290 as working fluid for ORC and refrigeration systems. Increase of pressure ratio will increase net output power until a certain value and then the net output power decreases again. It looks available optimum pressure ratio that will give highest net output power of the gas turbine system. In case of inlet hot gas temperature into the gas turbine at 1200^{o}C, optimal pressure ratio will exist in a value that higher than 30.

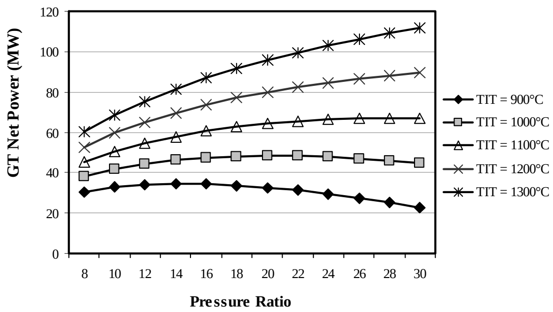

Figure 5: Influence of GT pressure ratio and TIT

Similar result that can be performed for net output power and thermal efficiency of the gas turbine system also has maximum value at a certain value of pressure ratio and at a specified gas temperature entering into the gas turbine as indicated in Figure 6. This result is performed for R-290 as working fluid in ORC and refrigeration system and air inlet temperature into the compressor at 15°C and 15% of RH.

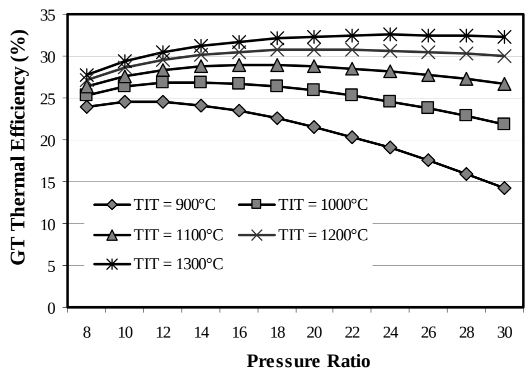

Figure 6: Thermal efficiency of gas turbine system

If pressure ratio is increased then thermal efficiency of gas turbine system will increase until finding a maximum condition at a certain pressure ratio, and then the value will decrease again when the pressure ratio is higher than optimum pressure ratio. Table 1 summarizes optimal results of thermal efficiency and pressure ratio of the gas turbine system for different hot gas temperatures entering the gas turbine system. This result was obtained at ISO condition for air inlet into the compressor.

Table 1: Maximum thermal efficiency of GT system

T max (°C)	η_{GT}, %	Pressure ratio
900	24.49	10
1000	26.86	14
1100	28.92	16
1200	30.81	20
1300	32.49	26

Figure 7 shows influence of the air inlet temperature entering compressor for gas turbine inlet hot gas temperature variation. This result is that R-290 as a working fluid in ORC and refrigeration system, and 450^{o}C of exhaust hot gas temperature, and the pressure ratio referring the optimal result in Table 1. For the same study parameters, influence of both air inlet temperature and hot gas temperature entering gas turbine respect to thermal efficiency of gas turbine system is shown in Figure 8.

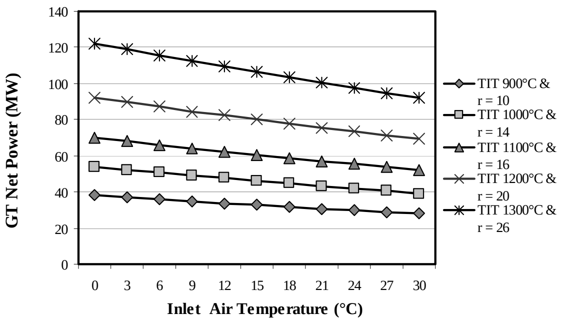

Figure 7: Net output power of gas turbine VS inlet air

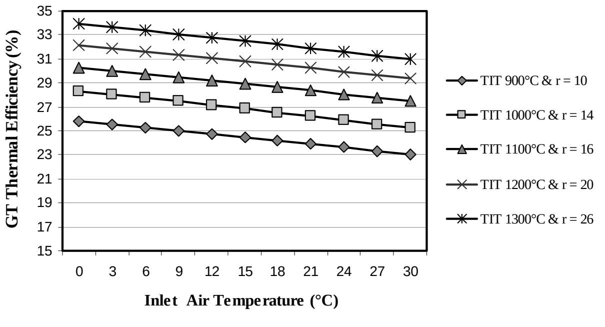

Figure 8: GT Thermal efficiency VS TIT and cooling

Although additional power needed to drive compressor for refrigeration system and increase of input heat in the combustor, those all also increase thermal efficiency of gas turbine system. Main numerical values of this simulation are summarized in Table 2. Significant performance changes respect to hot gas temperature and air inlet temperature cooling.

Table 2: Gas turbine power and efficiency

Condition	ISO Condition		Cooling 0°C	
	η_{GT} (%)	$W_{Net,GT}$ (MW)	η_{GT} (%)	$W_{net,GT}$ (MW)
900°C, r = 10	24.49	32.75	25.82	38.21
1000°C, r = 14	26.86	46.17	28.32	53.95
1100°C, r = 16	28.92	60.53	30.26	69.89
1200°C, r = 20	30.81	79.98	32.16	92.03
1300°C, r = 26	32.49	106.26	33.90	122.28

6.2 Performance of Direct Expansion Cycle

In this simulation, any parameter is varied in accordance with possibility in the object of the case study and the purposed cycle. Important interesting parameter to be presented is the steam temperature entering the steam turbine. As summary in Table 3, increasing the steam temperature as consequence oh heating in steam heater can increase the steam turbine power and thermal efficiency of the direct expansion cycle. In this simulation for any parameter varied, the maximal power and efficiency of the direct expansion cycle are 78.4988 MW and 22.82%, respectively.

Table 3: Power and efficiency of direct expansion cycle

T_{12} (°C)	W_{STG} (MW)	η_{STG} (%)
200	57.2996	19.85
250	60.683	20.57
300	64.5209	20.75
350	68.7902	21.37
400	73.4593	22.07
450	78.4988	22.82

6.3 Performance of ORC

ORC parameters may be varied in the integrated hybrid system, in which is concerned in this research. The parameters are superheated vapor working fluid entering turbine, working fluid type, and maximal pressure in ORC. Heat recovery from the steam heater should consider gas temperature will enter the organic heater. Following results are simulated for the same main data of simulations.

Figures 9 to 10 show that influence of saturated vapor temperature and working fluid types on performances of ORC. Great power can be performed when R-600 is operated as a working fluid. In contrast, smaller output power is yielded when R-143a is operated as a working fluid in ORC. Similarly, result was obtained for thermal efficiency of ORC. R-600 gives highest thermal efficiency and R-143a gives lowest thermal efficiency. Numerical values of the results for maximal efficiency and output power are given Table 4.

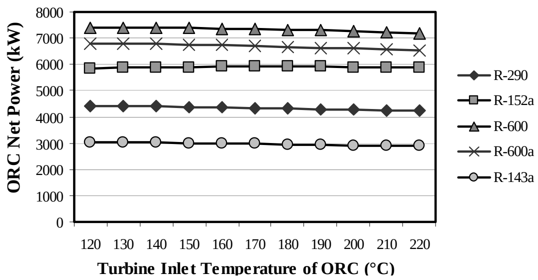

Figure 9: Net power VS highest temperature in ORC

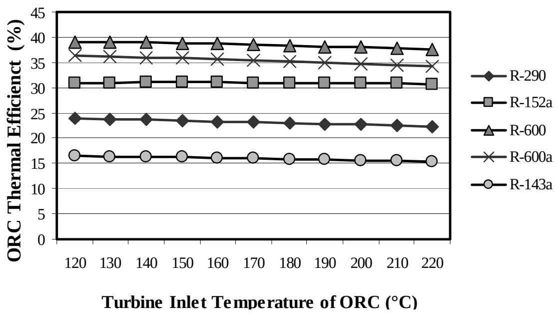

Figure 10: Thermal efficiency of ORC

Table 4: Maximal power and efficiency (GT TIT 900°C)

ORC fluid	W_{net} (kW)	$T_{w\text{-}opt}$ (°C)	η_{ORC} (%)	$T_{ef\text{-}opt}$ (°C)
R-290	4427.44	120	23.89	120
R-152a	5908.82	180	31.02	150
R-600	7384.74	130	39.12	120
R-600a	6789.08	120	36.30	120
R-143a	3030.43	120	16.50	120

Referring to previous result, R-600 is used as working fluid for ORC, simulation with various pressures and temperatures of the working fluid exiting from the organic heater. It was conducted for temperatures from 150^{o}C as consequence of using recuperator in ORC. The recuperator used to recovery heat of condensing process in the condenser. Therefore, the organic working fluid temperature is not lower than the compressor exited refrigerant temperature so that the temperature satisfies for the requirement is temperature from 150^{o}C. The steam temperature is assigned at 450^{o}C as used in the previous simulation result.

Influence of operating pressure and temperature on ORC net output power is shown in Figure 11. Higher operating pressure will give higher the output power. Thorough observing the curve, it can be revealed that significant improvement of efficiency is performed by increasing the pressure. This result confirms that it is better to assign at higher pressure of organic heater. The net output power that can be achieved at 3.5 MPa and 190^{o}C is 8049.08 kW. If the temperature is increased again, then the power will be drop. The ORC efficiency has similar behavior, even though the maximal value occurs at different temperature of saturated vapor organic working fluid. The detail result is illustrated in Figure 12.

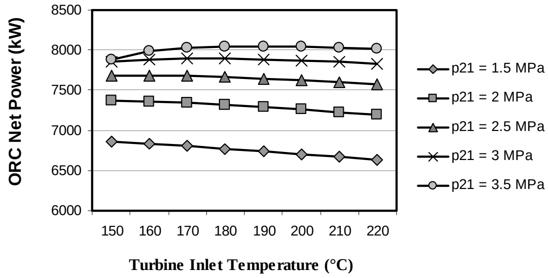

Figure 11: ORC net output power VS *T* max and *p* max

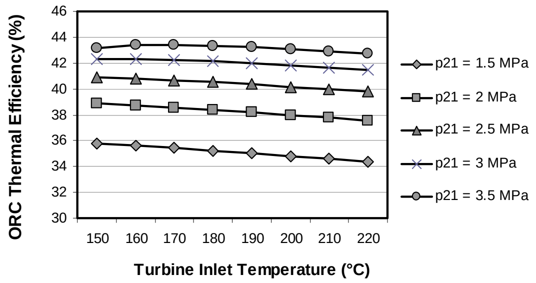

Figure 12: ORC Efficiency for any pressure

6.4 Performance of the Integrated Hybrid System

This section represents total performance of the integrated hybrid system. As shown in Figure 1, mass flow rate in the gas turbine system was obtained from mass and energy balance in the steam heater. The remaining heat energy after heating the steam is then used to heat a working fluid of ORC in the organic heater. Vapor compression cycle works as both refrigeration system and heat pump, in which work for cooling air before entering the compressor and work for heating an organic working fluid before entering the organic heater, respectively.

Any parameter of gas turbine system is varied and it is investigated from performance of the integrated hybrid system. Main simulation data are made as same as the previously simulation. Following is summarized for the different parameters used in the simulation. Hot gas temperature entering gas turbine is varied from 900^{o}C to 1300^{o}C; the pressure ratio is varied from 8 to 34; ambient temperature is assumed at 30^{o}C; steam entering the steam turbine is at 550^{o}C for the same geothermal well pressure; exhaust hot gas from the steam heater is 200^{o}C; working fluid operated in ORC is R-600 with highest operating temperature and pressure at 150^{o}C and 2 MPa; and working fluid for the vapor compression cycle is R-600.

Vapor compression cycle is operated in accordance with cooling load of air before entering the compressor of the gas turbine system. In case of air temperature is assigned at lower, cooling load of refrigeration system increases. Consequently, compressor work of the refrigeration system becomes larger. This result is shown in Figure 13 for several temperature of hot gas entering the gas turbine. This result is for exhaust hot gas from the gas turbine at 550°C and working fluid in the ORC and heat-pump refrigeration system are R-600.Total energy rate for the compressor and cooling load are utilized to preheat the working fluid in the ORC.

Simulation results of effect to additional power for varied TIT hot gas temperature and the exiting gas from gas turbine at 550^{o}C respect to fuel energy into the system is shown in Figure 14. This result also shows that effect of pressure ratio of gas turbine system on FFUF. Lower temperature of air entering the gas turbine system causes the net output power of the hybrid system increases as shown in Figure 15. This result confirms that the air inlet cooling also causes the net power of hybrid system increasing; it is not only for the gas turbine system. The hot gas temperature entering the gas turbine is increased; it will give higher the net output power.

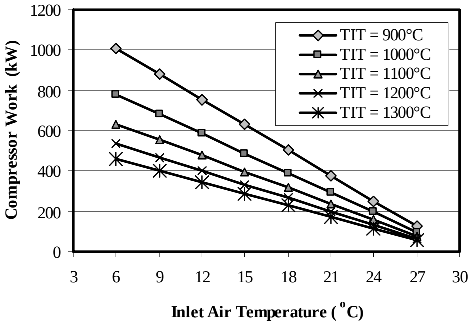

Figure 13: Refrigeration compressor work and cooling air inlet temperatures

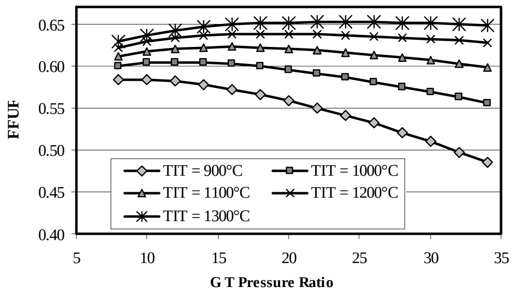

Figure 14: FFUF of hybrid system for varied gas turbine pressure ratio

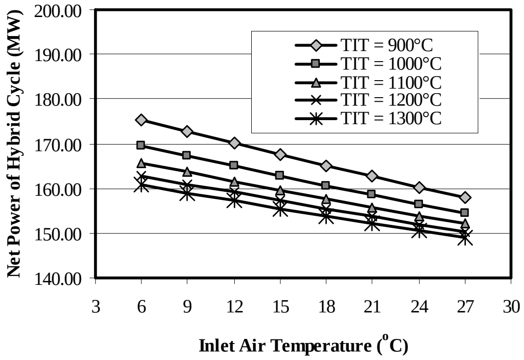

Figure 15: Total net output power of hybrid system

For the same simulation data, thermal efficiency of the hybrid system is also investigated and the result as shown in Figure 16. Higher thermal efficiency is achieved when the air inlet cooling temperature is lower. This result is as an indicator that the hybrid system is prospect from thermodynamic aspect. At the ISO condition of air entering the compressor of the gas turbine system, influence of pressure ratio of the gas turbine in the thermal efficiency of the developed hybrid system is shown in Figure 17. This result also confirms that there is an optimal ratio pressure of the compressor for a certain value of hot gas temperature.

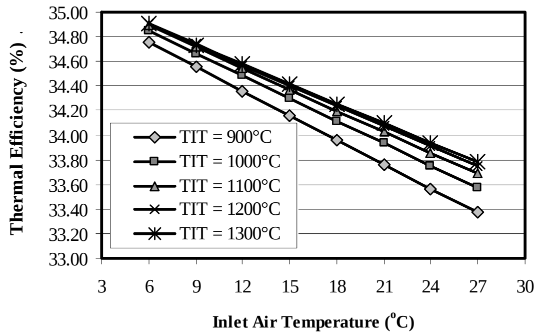

Figure 16: Thermal efficiency of the integrated hybrid system

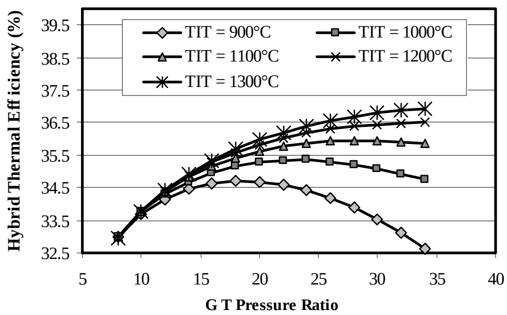

Figure 17: Thermal efficiency of the integrated hybrid system

As comparison of using fossil fuel as conducting in a combined cycle, an indicator FFUF as defined in Section 5 are used to judge the developed hybrid system. Figure 18 shows influence of the air inlet cooling temperature and the hot gas temperature in FFUF. This value range confirms that performance of the integrated hybrid is better alternative than using fossil fuel separated from geothermal system such as the combined cycle.

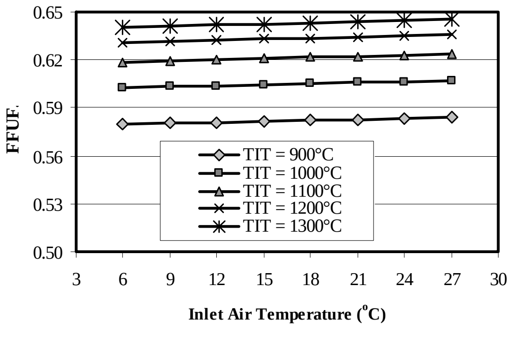

Figure 18: FFUF of hybrid system

Figure 19 shows optimal pressure ratio assessed from FFUF aspect. Two temperature sets of the air inlet cooling are introduced to observe pressure ratio behavior. There is also an optimum value of pressure ratio in gas turbine system.

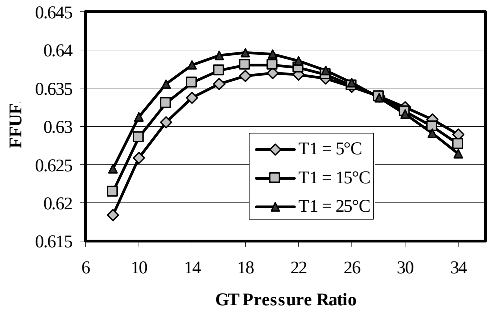

Figure 19: Optimal pressure ratio based on FFUF

The last figure 20 shows second law assessment of the hybrid cycle with variation of hot gas temperature in gas turbine system and air inlet temperature into compressor of the gas turbine system. This result confirms that highest temperature of hot gas of gas turbine can not performs the best second law efficiency. However, lower air temperature entering compressor of the gas turbine system also gives higher exergitic efficiency. These values show that the relative value of the thermal efficiency of the hybrid cycle respect to the ideal Carnot cycle.

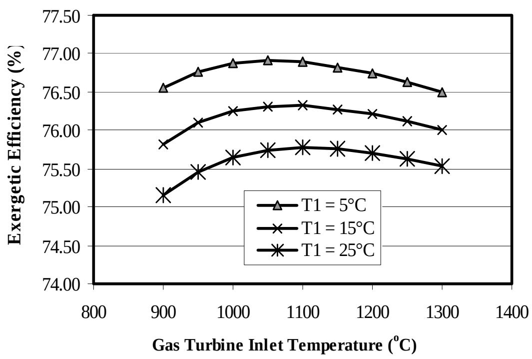

Figure 20: Exergetic efficiency of the hybrid cycle

7. DISCUSSION

As described in the previous result that the proposed hybrid system for enhancing geothermal power plant gives propective result. Overall efficiency of the system performs significant efficiency improvement, i.e. the value from 19.69% up to 35.83%. However, this value can not be revealed in further, since the recent technology fossil fuel power plant can perform 50% of thermal efficiency. Therefore, judgment of which economical alternative systems whether in case of using fuel as single heat source such as be used in combined cycle or integrated with geothermal power as proposed in this study.

Alternative judgment in comparison of both systems, the FFUF can be used as an indicator to judge in consideration of better system. This indicator may reveal the effectiveness of implementation of fuel whether it is used in an integrated system with geothermal or in separated system such as in combined cycle. As shown in the previous result, the proposed hybrid system can perform 63.81% of FFUF for case of the steam temperature before expanded in the turbine at 550^{o}C and the highest temperature in the gas turbine system up to 1200^{o}C as shown in Figure 2. This value is significant higher than thermal efficiency in the combined cycle. This means that it is more effective to combine fossil fuel in the proposed hybrid system.

Higher superheated steam before entering the steam turbine will give opportunity to use high efficiency of the steam turbine. Commonly situation in geothermal steam turbine requires low turbine efficiency in order to avoid the working fluid exhausted from the turbine at low quality. Low steam quality will cause erotion in the steam turbine blades. This technical aspect contributes in advatange of the hybrid system. This opportunity will increase the overall thermal efficiency and also FFUF. Of course, higher structure strength of steam turbine should be required to support higher operation temperature.

Consequence of overall thermal efficiency improvement, it will give increasing of net output power of the hybrid system including net output power of direct expansion of geothermal steam turbine. The increment of the net output power of geothermal steam turbine can be performed up to 42.7%. This value strictly depends on superheat degree of steam heater recovery heat from the exhaust gas of the gas turbine.

ORC operates bases of any constrain of other cycle in the proposed hybrid cycle. It should be noticed that most working fluids applicable from thermodynamic aspect for these temperature ranges have specific enthalpy different in turbine expansion is very small so that recovery heat from exhaust gas from steam heater requres high mass flow rate. This concequence, in order to get large ORC capacity, several ORC should be paralel installed.

8. CONCLUSION

The proposed hybrid system performs higher efficiency than conventional geothermal cycle. The overall efficiency can be formed up to 35.83% and the FFUF up to 63.81%. Increment of net power of geothermal direct expansion steam turbine can be performed up to 42%. Increase in inlet air temperature from 15°C to 30°C increases heat rate, which causes a decrease in fuel efficiency by about 3% for generating the same amount of the net power. The hybrid system can avoid this effect and it can even help to increase fuel efficiency by cooling the inlet air to below 15°C. R-600 is a best working fluid to achieve highest thermodynamic performance in the proposed hybrid system. Technical and thermodynamic aspect considerations of the hybrid system give more benefit than combined cycle or conventional geothermal steam power plant.

NOMENCLATURE

Symbols:

COP: coefficient of performance

FFUF: fossil fuel utilization factor

LHV: lower heating value

P: pressure

Q: heat

RH: relative humidity

r : pressure ratio

T: temperature

TIT: gas inlet temperature into turbine

W: work

Subscripts:
Comp: compressor of a vapor compression cycle

C: compressor of a gas turbine system

F: fuel

Geo: geothermal

GT: gas turbine

max : maximum

Net: net energy rate

ORC: organic Rankine cycle

OH: organic heater

P: pump

SH: steam heater

STG: steam turbine of geothermal

SG: steam from geothermal well

Superscripts:
o : initial condition, without modification

$'$: after modification

$\cdot$: rate with time base

REFERENCES

Astina, I M.: Alternative Systems for Enhancing Geothermal Power Plant System, *CD Proceeding of The 2nd AUN/SEED-Net Regional Conference on New/Renewable Energy*, Chonburi, Thailand, January 21-22, (2010)

Astina, I M.: Development of Fundamental Equations of State for Thermodynamic Properties of HFC Refrigerants, *Ph.D. Dissertation*, Keio University, Japan, (2003).

Astina, I M., and Sato, H.: A Rational Helmholtz, Fundamental Equation of State for Difluoromethane with an Intermolecular Potential Background, *Int. J. Thermophysics*, **24(4)**, (2003) 963-990

Astina, I M., Kurniawan, K.E., and Darmanto, P.S.: Implementasi Siklus Hibrida pada Pembangkit Listrik Tenaga Panas Bumi, *CD Proceeding*, Seminar Nasional VII Rekayasa dan Aplikasi Teknik Mesin di Industri, ITENAS, Bandung (2008a).

Astina, I M. Jasjkur, C., Hendrawan, and Mustafa, M.: Thermo-Economic Study of Cooling, Heating and Power System Implemented for Indonesian Power Plants, *CD Proceedings*, Seminar Nasional Tahunan Teknik Mesin VII, Manado (2008b)

Bidini, G., Desideri, U., Maria, F. D., Baldacci, A., Papale, R., and Sabatelli, F.: Optimization of an Integrated Gas Turbine-Geothermal Power Plant, *Energy Convs. Mgmt*, **39:16-18,** (1998), 1945-1956.

Boonnasa, S., Namprakai, P., and Muangnapoh, T.: Performance improvement of the combined cycle power plant by intake air cooling using an absorption chiller, *Energy*, **31,** (2006) 2036–2046.

Datawarehouse ESDM: Listrik (http://dtwh2.esdm.go.id /dw2007/index.php?mode=el), (access date: 11 Aug. 2008).

Gareta, R., Romeo, L. M., and Gil, A.: Methodology for the economic evaluation of gas turbine air cooling systems in combined cycle applications, *Energy*, **29,** (2004), 1805–1818.

Huttrer, G.W.: The Status of World Geothermal Power Generation 1995-2000, *Geothermic*s, **30,** (2001), 1-27, (http://iga.igg.cnr.it/geoworld/pdf/Huttrer_geothermics.pdf, (access date 6 October 2006).

Lemmon, E.W., Jacobsen, R.T., Penoncello, S.G., and Friend, D.G., Thermodynamic Properties of Air and Mixture of Nitrogen, Argon and Nitrogen from 60 to 2000 K at Pressures to 2000 MPa, J. Phys. Chem. Ref. Data, **29:3**, (2000), 331-385.

Miyamoto, H., and Watanabe, K.: A Thermodynamic Property Model for Fluid-Phase n-Butane, *Int. J. Thermophys.*, **22(2), (**2001), 459–475.

Phommavongsa, D., Astina, I M., Darmanto, P.S., and Sato, H., State of Art on Implementation of Thermodynamic Model for Providing Thermodynamic Property Database, *Proceedings*, Seminar Nasional Tahunan Teknik Mesin, pp. 105-110, Bali, Indonesia, 21-22 November (2005).

Polyzakis, A.L., Koroneos, C., and Xydis, G.: Optimum gas turbine cycle for combined cycle power plant, Energy *Conversion and Management*, **49,** (2008) 551–563.

Punwani, D.V., Pierson, T., Bagley, J.W., and Ryan, W.A.: A Hybrid System for Combustion Turbine Inlet Air Cooling at the Calpine Clear Lake Cogeneration Plant in Pasadena Texas, *ASHRAE winter meeting*, January, Atlanta (2001).

Statistics Indonesia: Energy Statistics Capacity of Electric State Company, 1995 – 2006 (http://www.bps.go.id /sector/energy/table4.shtml), (access date: 14 Aug. 2008)

Wu, D.W., and Wang, R.Z.: Combined Cooling, Heating and Power: A Review, *Progress in Energy and Combustion Science*, **32**, (2006), 459-495.