DEVELOPMENT AND TEST OF A 100KW CLASS ORC POWER-GENERATOR FOR LOW TEMPERATURE GEOTHERMAL APPLICATIONS

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ABSTRACT

The major portion of the heat sources is in the lower end of the temperature spectrum. Therefore, the successful utilization of low-grade heat is very essential to energy. The organic Rankine cycles (ORCs) are one of the appropriate technologies to convert low grade heat to power. Korea Institute of Energy Research (KIER) and Jinsol Turbomachinery have jointly developed an ORC power-generator applicable to very low-temperature heat sources. This paper deals with the design, fabrication and test results of the ORC power generator. The ORC system was designed the maximum electric power output of 100kW utilizing geothermal hot water. The ORC has a simple configuration with an evaporator, an expander, a condenser and a pump. The completely hermetic turbo-generator was developed by Jinsol Turbomachinery and applied to the system. The turbo-generator has twin radial inflow turbines connected with high-speed synchronized generator. Two plate heat exchangers are used for evaporator and condenser. The performance test was conducted and the resulted gross electric power output was 91.22kW with cycle efficiency of 7.14%.

1. INTRODUCTION

An organic Rankine cycle (ORC) is a type of a Rankine cycle, which uses organic compounds as a working fluid instead of water. Thanks to the low evaporation temperature of the organic compounds, ORCs could be applied to lower temperature heat sources compared to the steam Rankine cycle. Low grade waste heat, geothermal hot water, solar thermal heat, and biomass combustion can be the heat source for ORCs (Lecompte et al., 2015). There are a couple of alternative technologies for converting low-temperature heat to power, such as Kalina cycle (Kalina, 1984), Uehara cycle (Uehara et al., 1994), Goswami Cycle (Goswami, 1995), trilateral Flash cycle (TFC) and thermoelectric generator. Despite of the high potential of TFC, a lack of efficient two-phase expander is the main obstacle of the TFC. Many researchers have been compared the ORCs and Kalina cycles (Yari et al., 2015, Bombarada et al., 2009, Zare and Mahmoudi, 2015, Yue et al., 2015 and Victor et al., 2013) from a thermodynamic and/or economical perspective. There is still controversy about the best heat recovery power generation technique. However, simple configuration, low pressure level and high freedom of design (selection of working fluids including mixtures, transcritical/supercritical cycle) make the ORS to the most practical technology.

Designing ORC power-generator is quite complex problem due to excessively high freedom of design. Selection of working fluids, configuring a cycle, choosing an expander and heat exchanger types, and etc. Colonna, P. (2013) pointed out that there are 1.6 million alternatives in designing ORCs. Various ORC power-generators are already developed and commercialized in the market. The power ranges of ORCS are from a few kW to tens of MW. Most of the ORCs adopt turbines as expanders, except very small systems due to relatively high isentropic efficiency of turbine. In spite of the long history and successful commercialization of ORCs (Mario Gaja, 2011), many researchers have been making efforts to develop a new and advanced ORC system. Many researchers introduced novel cycles and/or new optimization design method to enhance the cycle efficiency of ORCs (Xiao et al., 2015, Feng et al., 2015 and Imran et al., 2015) and adopting new working fluids (Lecompte et al., 2014 and Mavrou et al., 2015). Several researchers built their own ORCs with various configurations. Yun et al. (2015) designed parallel-expanders ORC for the application with large heat source variations. . Yamada et al (2015) developed a prototype of 10W ORC with a scroll expander. They showed the possibility of micro-scaled ORC in spite of low cycle efficiency. Fu et al. (2015) designed and constructed a simple cycle 250kW class ORC with turbine expander and achieved 9.5% thermal efficiency using 120°C hot water.
In the present work, KIER and Jinsol Turbomachinery jointly developed a 100 class ORC power generator for the commercialization purpose. A simple cycle was selected to avoid complexities in construction/maintenance. Instead of that, we focused on improving and optimizing the performances of the ORC system. An originally designed turbo-generator and hybrid type plate heat exchangers are developed and integrated to the present system. This paper will describe the design, fabrication procedures and test results of the system.

2. CYCLE DESIGN

2.1 Design conditions

To design an ORC system, the conditions of heat source and heat sink should be defined. In the present work, one of the hot spring wells in Seokmo Island is selected as a heat source. The heat source temperature is 75°C and the well is expected to produce about 1,400ton of geothermal hot water per day. For a heat sink, dry cooling tower was chosen. The power output of ORCs changes with external temperature and the maximum power is achieved during the winter season. When designing an ORC power generator, not only the rated power but also maximum power output should be considered due to the safety reasons. Therefore, the cooling water temperature of 5°C was assumed for the winter season.

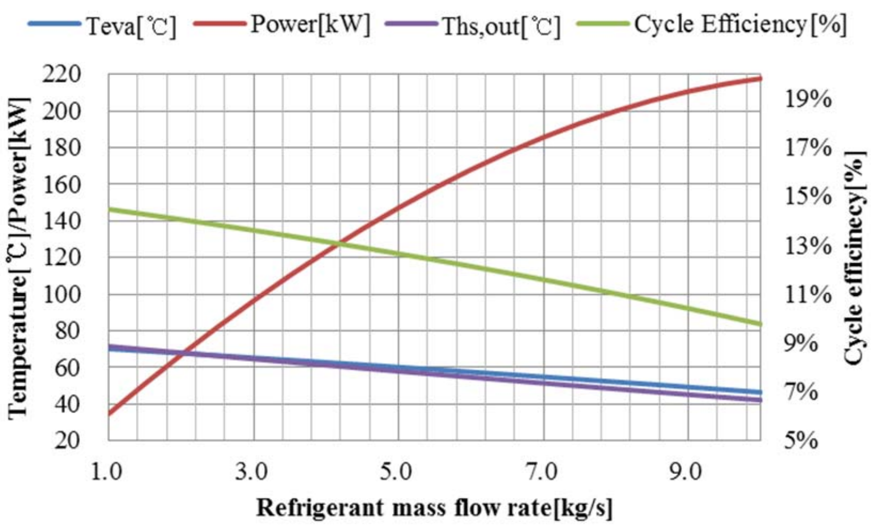

Figure 1: Effects of working fluid flow rate

For the given heat source conditions, the output power of the ORCs increases with the mass flow rate of working fluid, while the cycle efficiency reduces. Therefore, there is no correct answer in deciding a mass flow rate. Hot water in Suokmo Island is planned to be used in hot springs, so the water temperature after the power generation should be high enough for hot springs. Figure 1 shows the effect of mass flow rate of working fluid on evaporation temperature, power output, cycle efficiency and heat source outlet temperature. The working fluid is refrigerant R245fa and the pinch point temperature difference (PPTD) is 2.5°C. For quick estimation, no loss was assumed. Form the calculation, the mass flow rate was determined to 5kg/s with the heat source outlet temperature of 57.9°C.

2.2 System configuration and cycle design

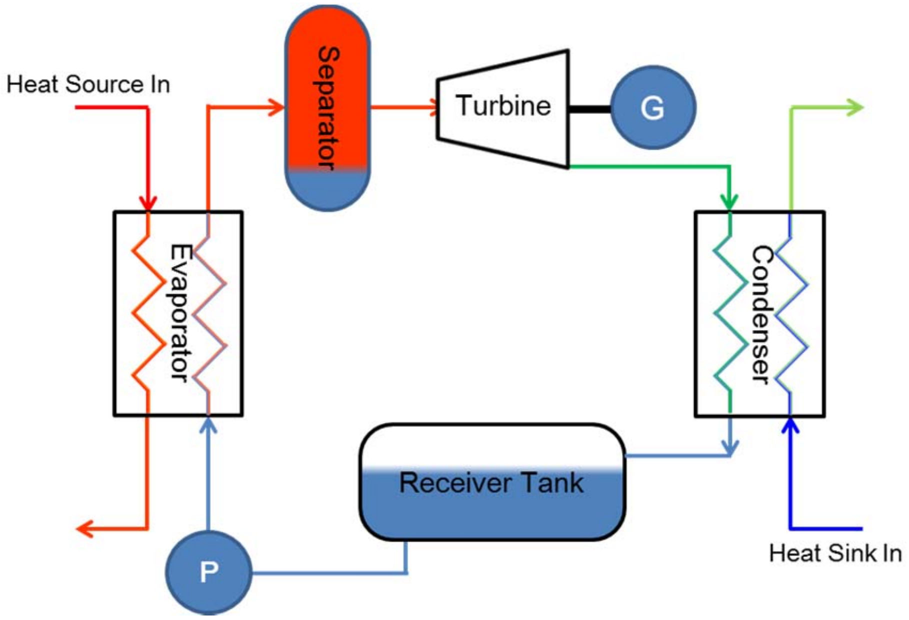

Figure 2: Schematic diagram of ORC power generator

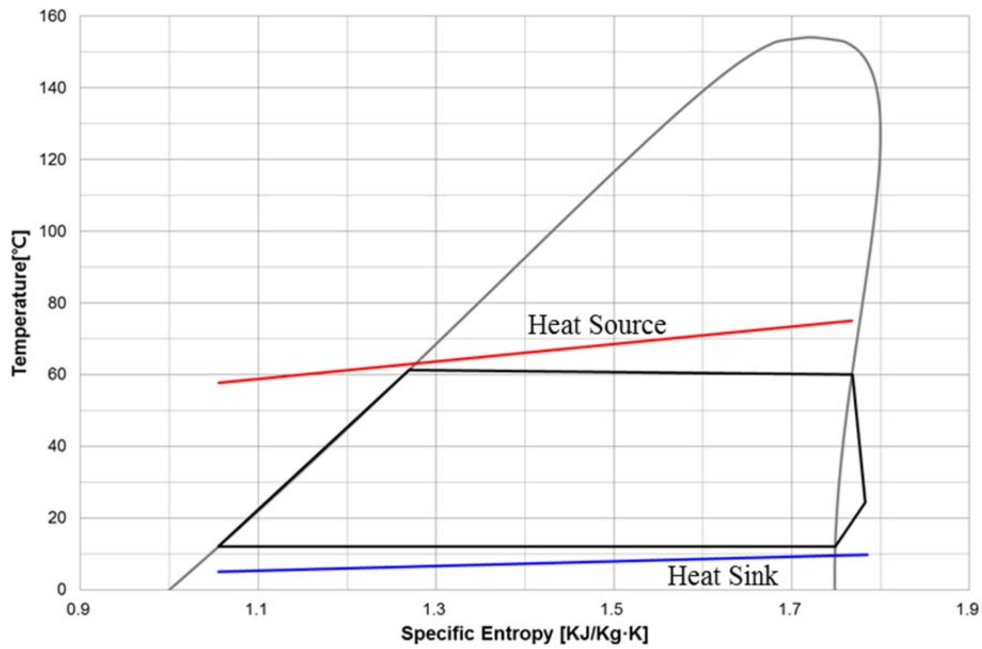

Figure 3: T-s diagram of the present ORC cycle

Figure 2 shows the schematic diagram of the present ORC power generator. The system is consisted of a pump, an evaporator, a separator, turbines, a generator, a condenser and a receiver tank. The R245fa in the receiver tank is assumed an equilibrium state and the temperature and pressure are the same with the condensing condition. For cycle design, efficiencies of the pump, motor, turbine and the generator are assumed to be 0.700, 0.895, 0.850 and 0.950, respectively. And the pressure losses through each component and pipes are considered. The pinch point temperature differences of 2.5°C were applied for both the evaporator and condenser. Required mass flow rate of heat source and heat sink were calculated from the PPTD and were 16.0kg/s and 52.3kg/s, respectively. The *T-s* diagram of the calculated ORC cycle is shown in Fig. 3. The red and blue lines refer the heat source and heat sink, respectively. The generator power output is 113.1kW with 2.268kW of pumping power consumption. The expected gross cycle efficiency is 9.74% (Net: 9.55%).

3. SYSTEM FABRICATION AND TEST

3.1 Component selection and system fabrication

Design requirements for each component are fixed from the cycle design and summarized in Tables 1 and 2.

Table 1: Design requirements of evaporator and condenser

		Evaporator		Condenser	
		Cold side	**Hot side**	**Cold side**	**Hot side**
Working Fluid		R245fa	Water	Water	R245fa
$\dot{m}$[kg/s]		5.0	16.0	52.3	5.0
Inlet	T[°C]	12.2	75.0	5.0	24.2
	P[kPa]	479.4	-	-	91.5
Outlet	T[°C]	60.0	-	-	12.0
	P[kPa]	462.3	-	-	89.2
Heat duty	Q[kW]	1161.0		1044.0	

Table 2: Design requirements of turbine and pump

		Turbine	**Pump**
Working Fluid		R245fa	
$\dot{m}$[kg/s]		5.0	
Inlet	T[°C]	59.8	12.0
	P[kPa]	456.6	102.6
Outlet	T[°C]	24.3	-
	P[kPa]	95.2	492.8
PR/ΔP[kPa]		4.8/361.4	4.8/390.2

Hybrid type heat exchangers, developed by Innowill, are adopted for both the evaporator and condenser. A Grundfos pump (CRN15-4) was chosen for the present system. A twin radial type turbo-generator was designed and fabricated by Jinsol Turbomachinery. The turbo-generator has twin radial turbines directly coupled with a high speed synchronous generator (Fig. 4). The yellow arrows in Fig. 4 indicate the flow direction of the vapor refrigerant. Details of the turbo-expander are discussed by Yang (2015).

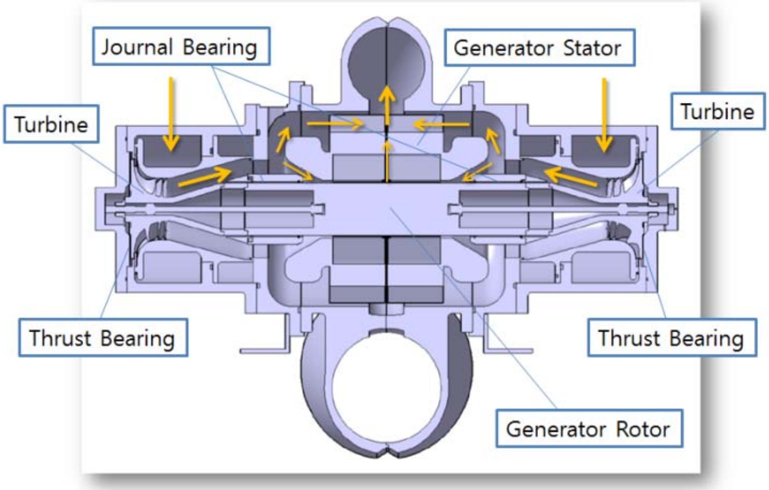

Figure 4: Mechanical Layout of the 100kW turbo-generator for ORC plant (Yang et al., 2015)

A 3ton/hr steam boiler and a plate heat exchanger and hot water tank are used to simulate heat source in Seokmo Island (Fig. 4). The hot water temperature to evaporator was adjusted by controlling steam valve, and the hot water flow rate is fixed. A 500RT cross-flow cooling tower supplies cooling water.

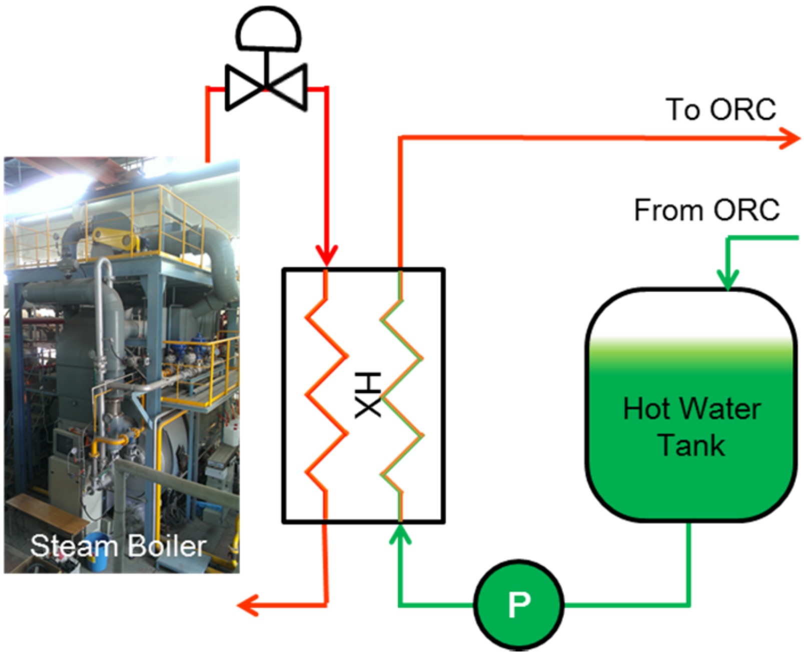

Figure 5: Schematics of hot water supply system

Figure 6 presents the photo of a 100kW ORC turbo-generator installed at KIER. A separator is located just downstream of the evaporator to remove liquid droplet and prevent the possibility of flooding. A small tank is positioned beneath the separator. If the tank is filled with liquid R245fa, an automated control valve opens, so liquid refrigerant can be discharged to receiver tank. Then, hot refrigerant vapor enters the turbines and expands. Since the expanded vapor has very low temperature, it designed to cool the generator instead of external air. Therefore, the vapor refrigerant flows through the generator as shown in Fig. 4. After that it reaches to the condenser and returns to the receiver tank. The receiver tank secures acceptable pump operation pressure for avoiding cavitation and keeps the system stable during unexpected transient behavior, such as over heat input and/or failures in cooling system. A bypass pipe is installed parallel to the turbine.

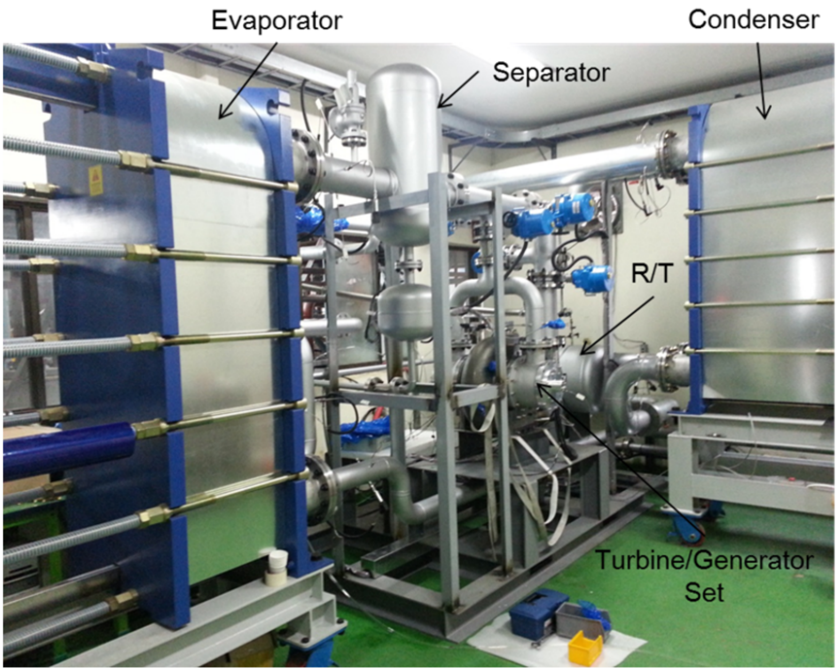

Figure 6: Photo of a 100kW Geothermal ORC Turbo-generator

3.2 Performance test

For the test and control, pressure transducers with the full scale range of 10bar (GE Druck, PT5072), and RTDs (PT100, 3-wire) are installed as shown in Table 3. The full scale range of the selected pressure transducers is more than double of the operating range. The reason is that the system is pressurized up to 10bar with N_2 gas during the leakage test. The refrigerant flow rate was measured by using Coriolis mass flow meter (OVAL, CA010L) located between the pump and the evaporator. The flow rates of the heat source and cooling water are measured by using magnetic flow meters. The generated electric power was measured at the generator outlet. Precision power analyzer (Yokogawa, WT3000) is used to measure the high frequency electric power.

Figure 7 shows the generator power output (P), refrigerant mass flow rate ($\dot{m}$) and cycle efficiency (η) during the startup test. When the turbine starts, the bypass valve was partially opened, and part of the refrigerant flows through the bypass line. Therefore, the measured flowrate is higher than that enters the turbines. With increasing of the refrigerant mass flow rate, both the power output and the cycle efficiency increase. When the power output increases up to 70kW, the cycle efficiency reaches maximum value and then stays constant. The effects of mass flow rate are shown in Fig. 8. The condensing temperature rose slightly during the test run. The maximum power output (gross) was recorded 91.22kW and the cycle efficiency was 7.14% at 4.17kg/s of mass flow rate. Both the power output and the cycle efficiency are lower than expected. One of the possible reasons is the absence of insulation. The disagreement between heat release of the source and heat gain of the working fluid is about 10~15% during the test. Based on heat gain of the working fluid, the cycle efficiency was rises up to 9.17%.

Figure 9 compares the designed cycle and test results. Dashed lines indicated designed cycle and solid lines refers the test results. During the test, heat sink temperature was higher than that of design condition. To keep the temperature difference between the heat source and heat sink, the heat source temperature. As a result, the tested condition shifted upward compared to the designed cycle, due to higher heat sink temperature. As shown in Fig. 9, unexpected superheating and subcooling are observed. The degrees of superheat and subcooling were 10.2°C and 5.6°C, respectively. The estimated uncertainty of the measured electric power and calculated cycle efficiency were less than ±1.0% and ±1.46%, respectively (Kline and McClintock, 1995).

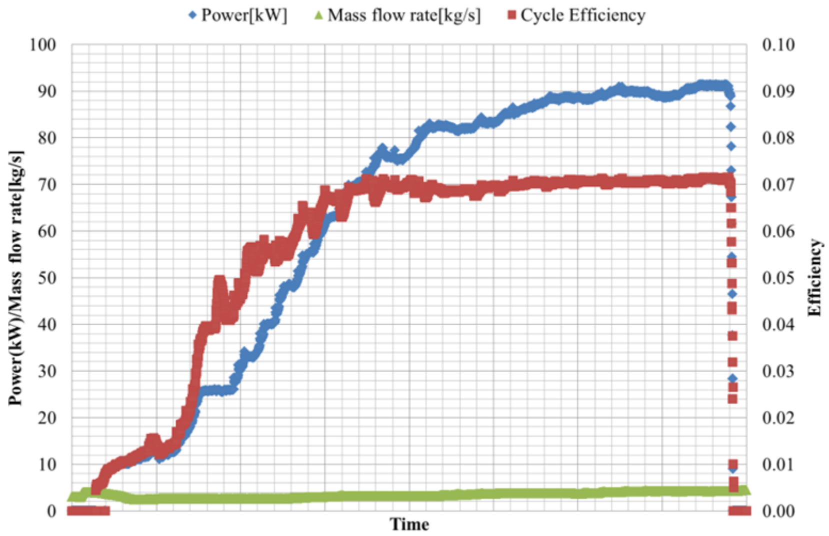

Figure 7: Power and efficiency change during startup

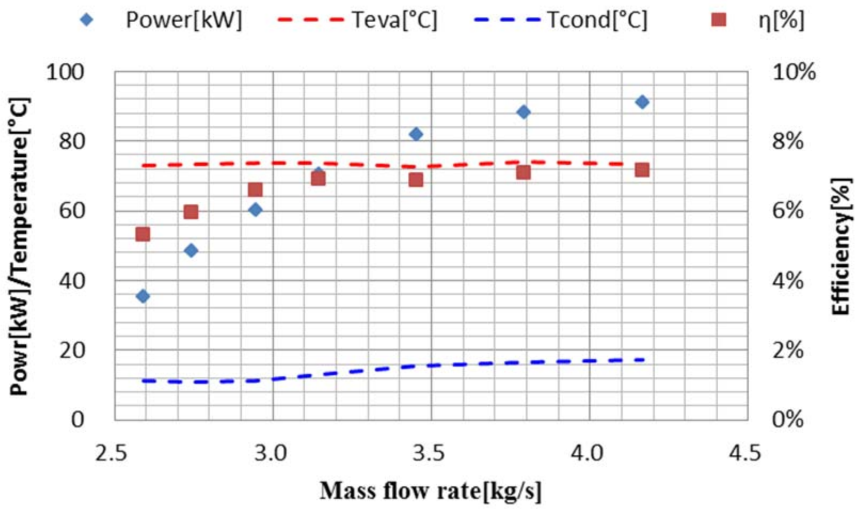

Figure 8: Performance results of 100kW ORC generator

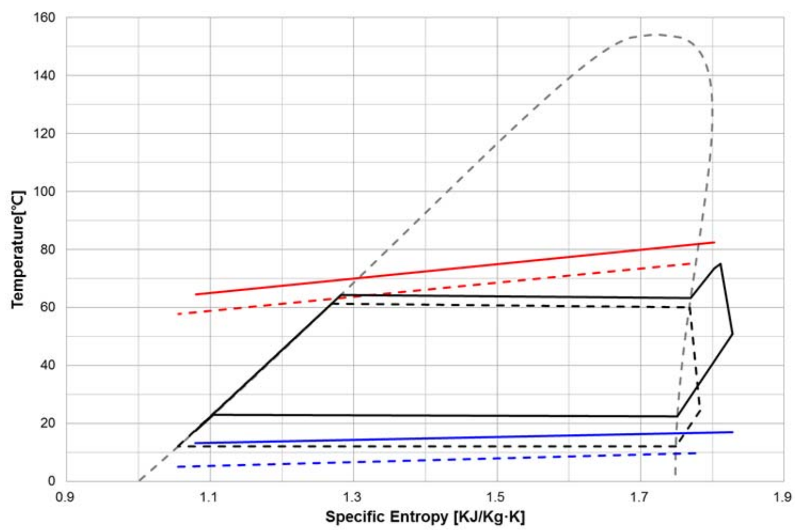

Figure 9: Comparison of designed and tested cycles

6. CONCLUSIONS

A 100kW ORC system adopting a high speed turbo expander was developed for low temperature geothermal heat applications. Performance test has been made and the results are follows:

- The ORC turbo-generator is designed for 113.1kW of gross electric power and 9.74% of cycle efficiency with the refrigerant mass flow rate of 5kg/s.
- The performance test show maximum electric power output was 91.22kW with the cycle efficiency of 7.14%, and the measured refrigerant flow rate was 4.17kg/s.
- High heat sink temperature and heat losses of the evaporator are the possible reason of low cycle efficiency. Based on heat gain of R245fa, the cycle efficiency was 9.17%, which is almost the same with expected value.

NOMENCLATURE

T	temperature	(°C)
P	pressure	(kPa)
$\dot{m}$	mass flow rate	(kg/sec)
η	cycle efficiency	(%)

Subscript

eva	evaporation
cond	condensation
h	heat source
c	heat sink

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