Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46152

DESIGN AND EXPERIMENTAL STUDY OF ORGANIC RANKINE CYCLE (ORC) AND RADIAL TURBINE

Seok Hun Kang*
Korea Institute of Energy Research
Daejeon, Korea

Dae Hun Chung
Korea Institute of Energy Research
Daejeon, Korea

ABSTRACT

The Organic Rankine Cycle (ORC), which generates electric power by using a low temperature heat source, is developed and an experimental study is conducted. A radial turbine and high speed DC generator are also designed and developed in this study. R245fa is adopted as a working fluid, considering the operation conditions of the cycle and its environmental friendly characteristics. Experiments are conducted to analyze the operation characteristics and performance of the developed ORC. The efficiencies of the cycle and turbine, electric power and temperature versus entropy diagram of the developed ORC with respect to the operation condition are investigated through experiments. The factors that influence the efficiency of the developed ORC are analyzed and discussed.

INTRODUCTION

Recently, oil prices have been fluctuating dramatically and environmental regulations concerning the greenhouse gas emissions are become severer. Under these circumstances, research on high efficiency energy technology including the organic Rankine cycle (ORC) is garnering more attention. The ORC generates electricity by utilizing low temperature heat sources without emitting any environmental pollutants or exhaust gases such as CO, CO_2, NO_X, etc. The ORC is structurally similar to a typical Rankine cycle but uses organic fluids as a working fluid instead of water. Organic fluids are suitable for the ORC because their specific vaporization heat is much lower than that of water. This enables the ORC to produce electricity by using low temperature heat sources. The ORC is mainly used as a power generation system utilizing low-grade heat sources such as geothermal energy and solar and waste heat, the available temperature range of which is from 60℃ to 200℃ [1, 2, 3, 4].

Yamamoto et al. designed an ORC by using an electric evaporator instead of an external heat source [3]. R123 and water were used as a working fluid and experiments were conducted to compare each fluid. The maximum cycle efficiency and electric power were shown to be 1.25% and 150W, respectively. Hung et al. parametrically analyzed the effects of various working fluids such as benzene, ammonia, R11, R12, and R134a on the efficiency of the ORC. They showed that the slopes and shapes of the saturation vapor curves of the fluids primarily affected the system efficiency [5, 6]. Hettiarachchi et al. presented a cost-effective optimum design criterion for ORCs utilizing low-temperature geothermal heat sources. The optimum cycle performance was compared for working fluids of ammonia, HCFC123, n-Pentane, and PF5050 [7]. Larjola developed an ORC using a high-speed oil free turbogenerator-feed pump in which turbomachines (turbine and pump) and electric generator are directly coupled [8].

In the present study, an ORC that generates 30kW electric power is designed and developed. R245fa is selected as a working fluid on the basis of the following characteristics. It is not subject to greenhouse gas emission regulations and it is developed to replace R123 which has limitation on the regulations. And it is suitable for cycle operation conditions, that is, the evaporation and condensation pressures and temperatures, and is non-flammable and non-toxic. Table I shows the thermodynamic properties of R245fa [9]. A radial turbine is used as an expander and is designed by considering the thermodynamic properties of the working fluid and the operation conditions of the cycle. A high speed synchronous generator is also developed and is directly coupled with a turbine rotor blade.

Experiments to analyze the performance and operation characteristics of the developed ORC are conducted. The cycle, turbine efficiencies, and electric power of the developed ORC with respect to the operation conditions are analyzed through

* Corresponding author: Tel.: +82-42-860-3545.
E-mail address: shkang@kier.re.kr (S. H. Kang)

experiments. The factors that influence the efficiency of the developed ORC are analyzed and discussed.

Table I: Properties of R245fa (HFC-245fa).

Molecular name	Molecular weight	Critical pressure	Critical temperature
CF3CH2CHF2	134 g/mol	3640 kPa	427.2 K

The designed ORC is developed for the experimental study, so it is tested in design and off-design points before considering its application to the real site. Most of the commercial ORCs whose working fluid temperature range in evaporator is from 100℃ to 150℃ use R245fa instead of R123 in these days. But there is no journal paper that deals with experimental study on the ORC and expander that uses R245fa as a working fluid. This paper is the first one that presents the results of an experimental study carried out on ORC and radial turbine that uses R245fa as a working fluid [11].

NOMENCLATURE

η cycle efficiency ignoring the pump work
η_t turbine efficiency
η_{th} cycle efficiency
h specific enthalpy [kJ/kg]
$h_{p,i}$ specific enthalpy of working fluid at pump inlet [kJ/kg]
$h_{p,o}$ specific enthalpy of working fluid at pump outlet [kJ/kg]
$h_{\sup,o}$ specific enthalpy of working fluid at superheater outlet [kJ/kg]
$h_{t,i}$ specific enthalpy of working fluid at turbine inlet [kJ/kg]
$h_{t,o}$ specific enthalpy of working fluid at turbine outlet [kJ/kg]
$h_{t,o,is}$ specific enthalpy of the working fluid at the turbine outlet when the entropy is the same as that at the turbine inlet [kJ/kg]
$h_{\tan k}$ specific enthalpy of liquid state working fluid in storage tank [kJ/kg]
s specific entropy [kJ/kg-K]
T temperature [℃]
P pressure [bar]

DESING OF EXPERIMENTAL APPARATUS

(1) Design of cycle condition

As a basis of the cycle design, condensation pressure is considered first and it is set to the saturation pressure of the working fluid at 30℃, that is, 1.78bar, and the condensation temperature is 5 ~ 10℃ higher than atmospheric temperature in autumn.

Generally, cycle efficiency increases as the pressure ratio at the turbine inlet and outlet rises. However, the turbine is limited in terms of increasing the pressure ratio. A radial type of turbine with a pressure ratio at its inlet and outlet set to 4.11 is designed in this study. Therefore, the evaporation pressure, which is considered to be the same as the turbine inlet pressure, is set to 7.32bar under consideration of the condensation pressure and the pressure ratio at the turbine inlet and outlet, which is 4.11 (= 7.32 / 1.78). The turbine efficiency is assumed to be 75% in the design procedure.

The evaporation pressure is the saturation pressure of the R245fa in the evaporator when the evaporation temperature is 77℃. The ORC is designed to superheat the working fluid to 80℃ in order to increase its quality. The typical Rankine cycle that uses water as a working fluid requires a superheater to avoid a decrease of quality during the expansion process in the turbine. On the other hand, the present ORC that uses R245fa does not need a superheater, because saturation vapor curve in the temperature versus entropy (T-S) diagram for R245fa has a positive slope, as can be seen in Fig. 1, and liquid droplets are not formed during the expansion process, in contrast to water. Nevertheless, a superheater is used in this study to avoid decreased quality owing to heat loss by insufficient pipe insulation and pressure drop in the pipe wall between the evaporator and turbine inlet.

Figure 2 shows the T-S diagram of the designed cycle, and Table II lists the thermodynamic properties of the working fluid at designed points. The designed ORC is assumed to generate 30kW theoretically when it operates with the above conditions and its working fluid flow rate is 1.53kg/s. The cycle efficiency is assumed to be 8.76% when pump work, pressure drop, and heat loss in the entire system are ignored. The cycle efficiency is calculated via the following equation [10].

$$\eta = \frac{h_{t,i} - h_{t,o}}{h_{\sup,o} - h_{\tan k}} \tag{1}$$

where η is the cycle efficiency ignoring the pump work, $h_{t,i}$ is the specific enthalpy of the working fluid at the turbine inlet, $h_{t,o}$ is the specific enthalpy of the working fluid at the turbine outlet, $h_{\sup,o}$ is the specific enthalpy of the working fluid at the superheater outlet, and $h_{\tan k}$ is the specific enthalpy of the liquid state working fluid in the storage tank. In this calculation, enthalpies at the turbine inlet and superheater are assumed to be the same since heat and pressure losses are ignored.

Table II Thermodynamic properties of working fluid at design points shown in Fig. 2.

Design points	T (℃)	P (bar)	h (kJ/kg)	s (kJ/kg-K)
(a)	30	1.78	239	1.14
(b)	77	7.32	305	1.34
(c)	77	7.32	460	1.78
(d)	80	7.32	463	1.79
(e)	48	1.78	444	1.75
(f)	30	1.78	427	1.14

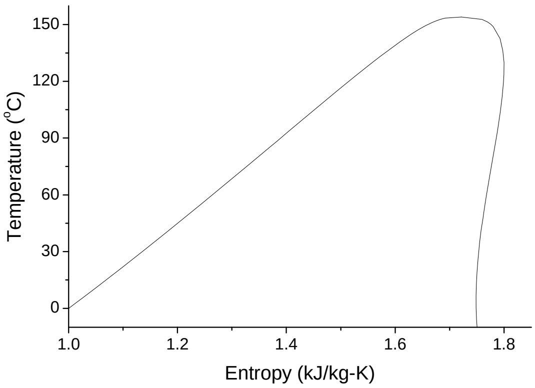


Figure 1: Positive slope of saturated vapor curve of R245fa in T-S diagram

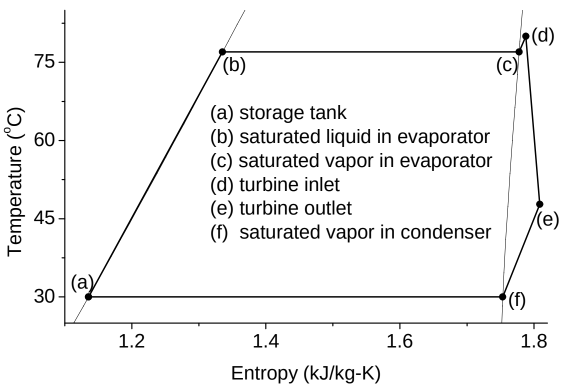


Figure 2. T-S diagram of designed cycle

(2) Design of cycle layout

The developed ORC consists of heat exchangers—a preheater, evaporator, superheater, and condenser—that heat or cool the working fluid, a turbine that generates electricity through the expansion process of the working fluid, and a pump that supplies low pressure working fluid from the storage tank to the evaporator. A schematic diagram of the ORC is shown in Fig. 3. Figures 4 and 5 show an illustration and photograph of the experimental apparatus, respectively.

The liquid working fluid is heated to a superheated vapor through the three heat exchangers, i.e., the preheater, evaporator, and superheater. The saturated steam of 150℃, 4.76bar, which is produced from the boiler located outside, is used as a heat source for the three heat exchangers. The two-phase state working fluid at the turbine outlet is cooled to a liquid state through the condenser. The water is used as a coolant for the condenser and is cooled in the cooling tower.

The working fluid temperature during the heating process is controlled by valves installed in the three heat exchangers, and the steam flow rate is adjusted with these valves. The valves are controlled by using PID logic on the basis of the working fluid temperature at the three heat exchanger outlets.

The high and low pressures of the cycle are controlled by the saturation temperatures of the working fluid in the evaporator and condenser, respectively. The flow rate of the working fluid is set by the pressure ratio between the turbine inlet, outlet, and the area of nozzle throat.

Compact brazed type heat exchangers are used for the preheater and the condenser, considering compactness of the system. A shell-and-tube type heat exchanger is used for the evaporator, since it must contain working fluid in liquid and vapor states simultaneously to produce saturated vapor. The level of the liquid state working fluid in the evaporator is controlled by using a float type level sensor. A shell-and-tube type heat exchanger is also used for the superheater, considering minimization of pressure drop and ease of connectivity with the evaporator, as it is also the same type of heat exchanger.

The turbine is made of aluminium and the number of turbine blades is 12. Figures 6 and 7 show illustrations of the rotor blade and nozzle of the designed turbine, respectively. Figure 8 shows the developed rotor blade and turbine coupled with a high speed generator. The generator is a synchronous type and employs a permanent magnet rotor. Its shaft speed is 20,000rpm at normal operation condition. The storage tank is designed with a cylindrical shape and its volume is 180ℓ. An impeller type of vertical pump is used.

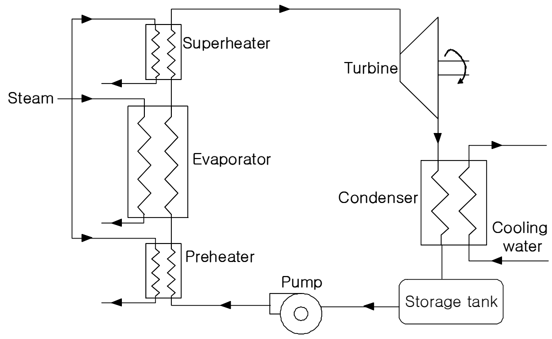


Figure 3. Schematic diagram of a developed organic Rankine cycle

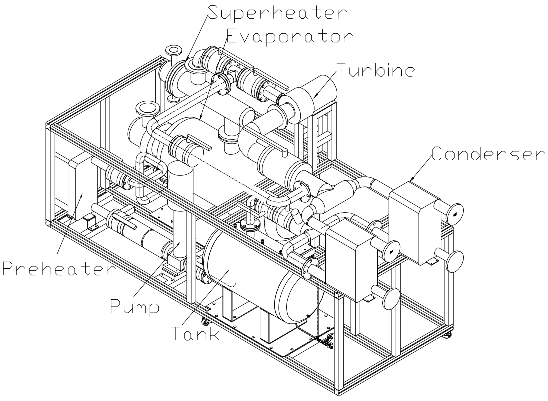


Figure 4. Drawing of the experimental apparatus

Figure 5. Experimental apparatus

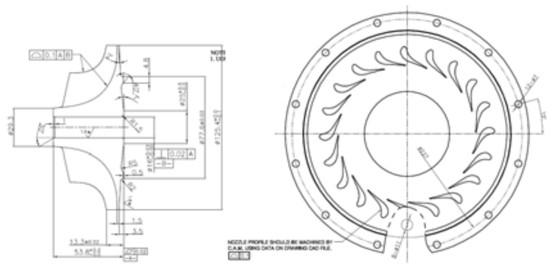


Figure 6. Drawing of turbine rotor (meridional view) and nozzle

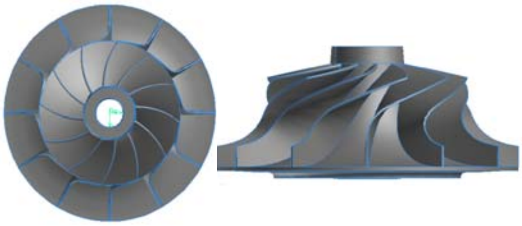
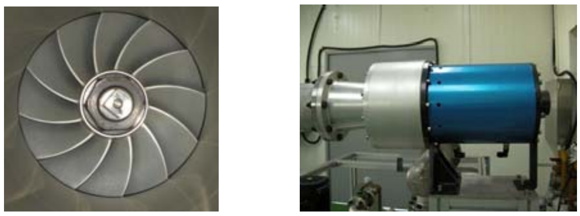


Figure 7. Three dimensional drawing of turbine rotor



(a) Rotor blade (b) Turbogenerator

Figure 8. Developed rotor blade and turbogenerator

EXPERIMENTAL METHOD

The experimental data are measured for 11 evaporation temperature steps. A steady state is maintained during 15 minutes to gather data at each evaporation temperature step. The temperature of the working fluid at the superheater outlet is set to be 3℃ higher than that in the evaporator.

The heat transfer quantity obtained by the working fluid from the heat source is measured by temperature and pressure sensors installed in the inlet and outlet of each heat exchanger. Temperature and pressure sensors are installed at the inlet and outlet of the turbine to calculate the working fluid enthalpy at each position. The flow rate of the working fluid is measured by using a turbine type flowmeter installed in the pipe located between the condenser outlet and the storage tank inlet. The generated electric power is measured by using a power analyzer (YOKOGAWA WT3000).

RESULTS AND DISCUSSION

Figure 9 presents the experimental results of the evaporation pressure versus the evaporation temperature of R245fa in the evaporator, and they are similar to the reference data obtained from REFROP 8.0. Figure 10 shows the electric power output of the cycle versus the turbine inlet temperature (TIT). The electric power output increases proportionally to the TIT, because the enthalpy difference of the working fluid between the turbine inlet and outlet also increases proportionally to the TIT. Figure 11 shows the flow rate of R245fa versus TIT. The flow rate also increases proportionally to the TIT for the same reason as described above. Figure 12 shows the turbine efficiency versus TIT, and it also increases as the TIT increases. The turbine efficiency is calculated with the following equation [10].

$$\eta_t = \frac{h_{t,i} - h_{t,o}}{h_{t,i} - h_{t,o,is}} \tag{2}$$

Where η_t is the turbine efficiency and $h_{t,o,is}$ is the specific enthalpy of the working fluid at the turbine outlet when the entropy is the same as that at the turbine inlet. Figure 13 shows the cycle efficiency versus TIT. The cycle efficiency is calculated using the following equation [10].

$$\eta_{th} = \frac{(h_{t,i} - h_{t,o}) - (h_{p,o} - h_{p,i})}{h_{\sup,o} - h_{p,o}} \quad (3)$$

Where η_{th} is the cycle efficiency, $h_{p,i}$ is the specific enthalpy of the working fluid at the pump inlet, and $h_{p,o}$ is the specific enthalpy of the working fluid at the pump outlet. The cycle efficiency also increases with the TIT.

It is found that the pressure at the turbine outlet is larger than that in the storage tank. Fig. 14 shows the temperature versus entropy diagrams and pressures at various measurement position when the cycle efficiency is 5.22%. The pressure at the turbine outlet is shown to be 3.5bar and it is larger than that in the storage tank that is 2.1bar as shown in Fig. 14 when cycle efficiency is 5.22%. The pressure at the flowmeter inlet which is located between the condenser outlet and the storage tank inlet is shown to be 2.7bar and it is also larger than that in the storage tank as shown in Fig. 15. The increases of pressure at turbine outlet and flowmeter are also shown when cycle efficiency is 3.81% in Fig. 15. These pressure increases are owing to the pressure drop in condenser. And the cycle and the turbine efficiencies are decreased because the pressure drop reduces enthalpy and pressure differences between turbine inlet and outlet. The system is modified by changing the condenser to resolve the pressure drop phenomenon. Fig. 16 shows the plate type heat exchangers. The pressure drop is occurred in left one and it is solved in the right one which is redesigned to replace the left one. The cycle efficiency of modified system is shown to be increased to about 20% more than that of the previous system, see Fig. 17, because pressure drop problem is resolved. Fig. 18 shows the temperature versus entropy diagrams obtained from modified system when the cycle efficiency is 6.1%. This shows the pressure difference between turbine outlet and condenser outlet is nearly zero.

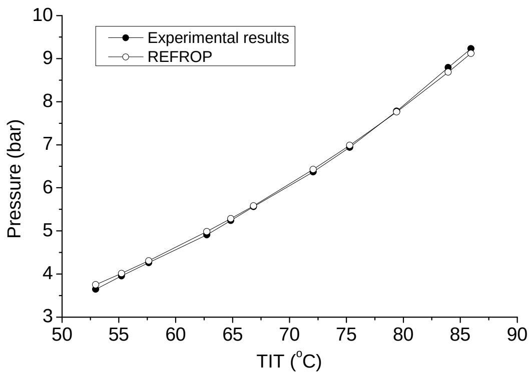


Figure 9. Evaporation temperature versus TIT

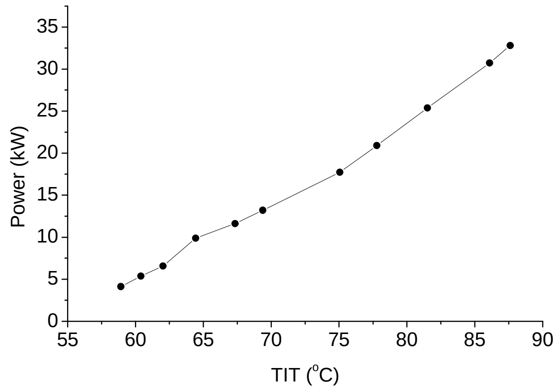


Figure 10. Electric power output versus TIT

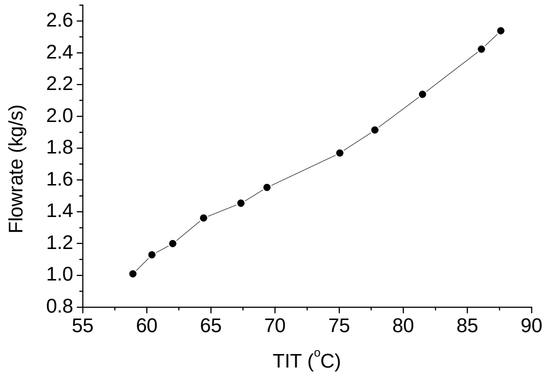


Figure 11. Mass flowrate of working fluid versus TIT

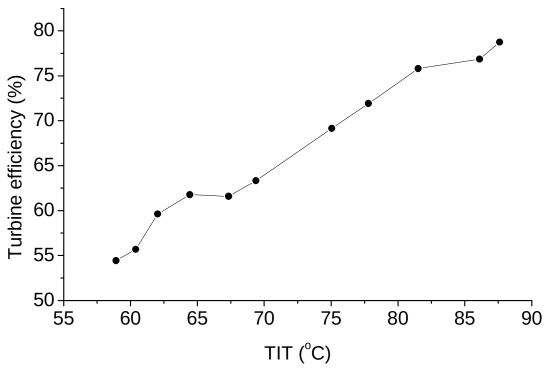


Figure 12. Turbine efficiency versus TIT

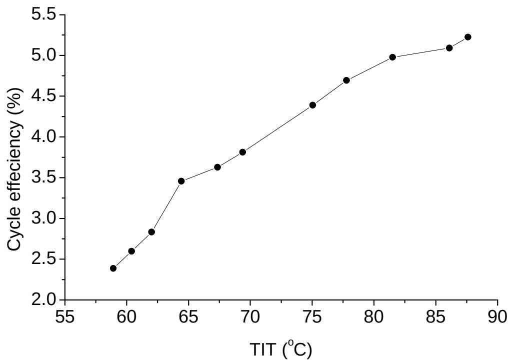


Figure 13. Cycle efficiency versus TIT

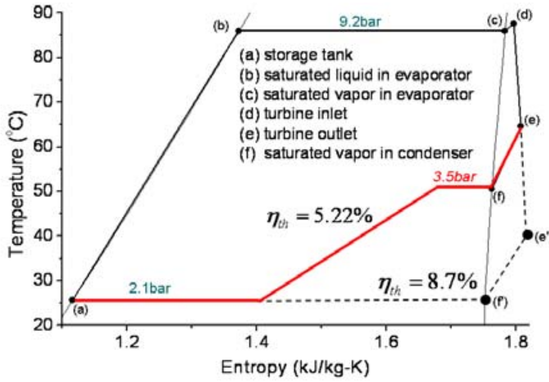


Figure 14. T-S diagrams when cycle efficiency is 5.22%

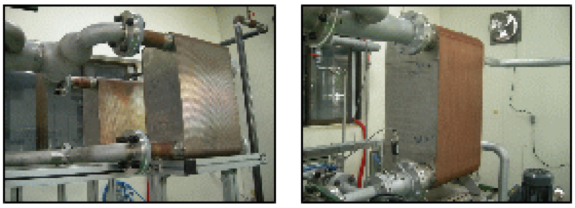


Figure 16. Condenser modified to solve the pressure drop phenomenon.

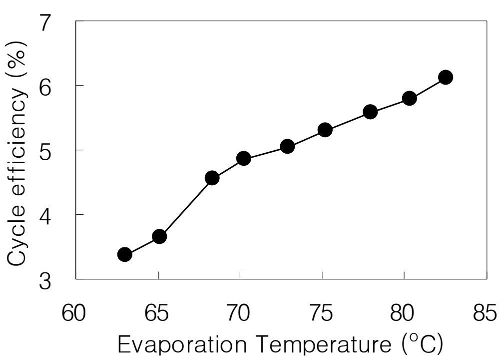


Figure 17. Cycle efficiency versus TIT of modified system

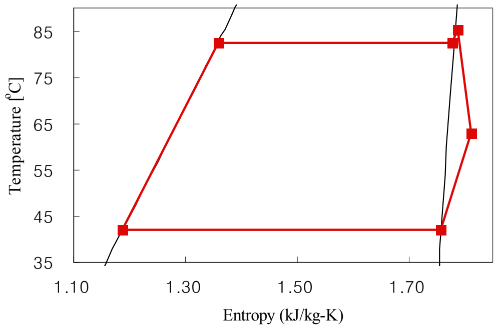


Figure 18. T-S diagrams obtained from modified system when cycle efficiency is 6.1%

CONCLUSION

In this study, an ORC that generates 30kW electric power and uses R245fa as a working fluid is designed and developed. A radial turbine that is directly coupled with a high speed generator is also designed by considering the thermodynamic properties of the working fluid and the cycle conditions. The performance and characteristics of the developed cycle are investigated experimentally. The maximum cycle, turbine efficiencies, and electric power are shown to be 6.1%, 78.7% and 32.7kW, respectively.

REFERENCES

[1]Yiping Dai, Jiangfeng Wang, Lin Gao, 2009, Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery, Energy Conversion and Management, Vol. 50, pp. 576-582.

[2]Donghong Wei, Xuesheng Lu, Z. Lu, J. Gu, 2007, Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery, Energy Conversion and Management, Vol. 48, pp1113-1119.

[3]T. Yamamoto, T. Furuhata, N. Arai, K. Mori, 2001, Design and testing of the Organic Rankine Cycle, Energy, Vol. 26, pp239-251.

[4]H. Kang, 2009, Organic Rankine cycle technology, Journal of the KSME, Vol. 49, No 9, pp47-52.

[5]T. C. Hung, T. Y. Shai, S. K. Wang, 1997, A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat, Energy, Vol. 22, No 7. pp. 661~667.

[6]T. C. Hung, S. K. Wang, C. H. Kuo, B. S. Pei, K. F. Tsai, 2010, A study of organic working fluids on system efficiency of an ORC using low-grade energy sources, Energy, Vol. 35, pp. 1403-1411.

[7]H. D. Madhawa Hettiarachchi, M. Golubovic, W. M. Worek, Y. Ikegami, 2007, Optimum design criteria for an organic Rankine cycle using low-temperature geothermal heat sources, Energy, Vol. 32, pp. 1698-1706.

[10] J. Larjola, 1995, Electricity from industrial waste heat using high-speed organic Rankine cycle (ORC), Int. J. Production Economics, Vol. 41, pp. 227-235.
[9]E.W. Lemmon, M.L. Huber and M.O. McLinden, NIST REFPROP Standard reference database 23, Version 8.0, User's Guide, 2007.
[10]G. V. Wylen, R. Sonntag, C. Borgnakke, 1993, Fundamentals of classical thermodynamics, John Wiley & Sons, Inc., 4th edition.
[11]V. Lemort, S. Quoilin, C. Cuevas, J. Lebrun, 2009, Testing and modeling a scroll expander integrated into an Organic Rankine Cycle Applied Thermal Engineering, Vol 29, pp. 3094-3102.

