THERMODYNAMIC INVESTIGATION OF ORGANIC RANKINE CYCLE ENERGY RECOVERY SYSTEM AND RECENT STUDIES

by

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Recently, new environment-friendly energy conversion technologies are required for using energy resources valid to power generation. Accordingly, low-grade heat sources as solar heat, geothermal energy, and waste heat, which have available temperatures ranging between 60 and 200 °C, are supposed as applicants for recent new generation energy resources. As an alternative energy source, such low-grade heat sources usage generating electricity with the help of power turbine cycles was examined through this study. Such systems have existing technologies applicable at low temperatures and a compact structure at low cost, however, these systems have a low thermal efficiency of the Rankine cycles operated at low temperatures. An organic Rankine cycle is alike to a conventional steam power plant, except the working fluid, which is an organic, high molecular mass fluid with a liquid-vapor phase change, or boiling point, at a lower temperature than the water-steam phase change. The efficiency of an organic Rankine cycle is about between 10% and 20%, depending on temperature levels and availability of a valid fluid.

Key words: *organic Rankine cycle, energy recovery, energy efficiency, energy recovery system*

Introduction

Organic Rankine cycle (ORC) is the recent type of an environmentally friendly leading technological system, applying the principle of the steam Rankine cycle, but can utilize relatively low-grade heat sources below 150 °C, fig. 1. This system uses an organic substance as a working fluid, therefore, it is named as *Organic* Rankine. Thus, the system utilizes low-grade heat sources for economic energy production and consists of an evaporator (heating area), a turbine and a condenser (cooling area). Utilizing different kinds of working fluid instead of water/steam provides the opportunity of constructing miniature, compact and portable thermal power plants. The selection of the architecture will rely on heat source temperature stage and also sort of working fluid, dry or wet. The ORC based new type of power plant system is designed to use low and mid-temperature waste heat sources and to convert them into electric energy – efficiently, economically and CO_2 free. The ORC systems are typically used for four major applications: waste heat recovery, geothermal power plants, biomass combustion plants and solar thermal plants [1-3].

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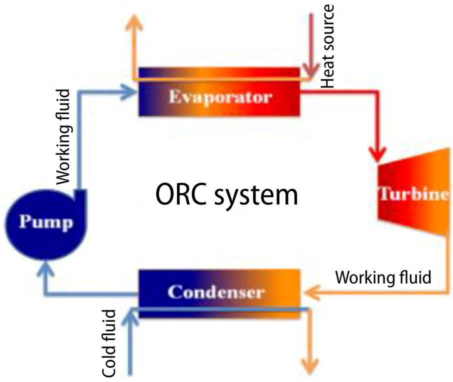

Figure 1. The ORC overview

The most developed application for an ORC system is the so-called *waste heat recovery*. The term *waste heat recovery* may be utilized to express the use of any heat overall ejected to the environment. Thus, the ORC system is a useful way of heat recovery in the temperature range between 150 to 200 °C; especially if no other utilization for the waste heat is present on the site [4-6].

The ORC is simpler and economically more feasible than the steam Rankine cycle. Thus, many commercial and test plants can be made by using organic Rankine cycle due to its advantages. The ORC system exhibits great flexibility, high safety and low maintenance requirements in recovering this grade of waste heat. Integrating the ORC to the energy system, such as power plants, could achieve using low-grade energy (waste heat) to generate high-grade energy (power), easing the power burden and enhancing system efficiency. Since the ORC consumes virtually no additional fuel, for the same added power, the emission of environmental pollutants such as CO_2, SO_2, and so on would be decreased. What is more, according to the local demand, the exhaust heat exiting from the ORC could be further utilized to drive chillers such as absorption chillers to supply cooling capacity. The ORC technology is a newly emerged technology since 1961. Thus, its usage is not so common in our country [7-9].

The ORC transforms thermal energy into mechanical shaft power. The advantage of ORC systems is the return of useful energy, often as electrical turnout, from low energy resources such as the low pressure steam affiliated with steam-driven turbines used for electricity generation [10-12].

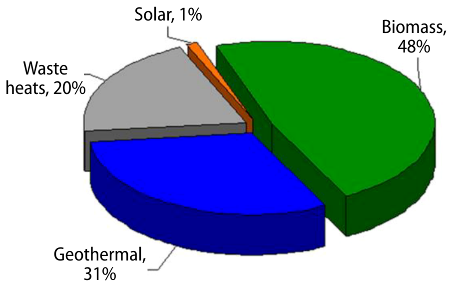

Figure 2. Pie chart of percentage of ORC energy sources [6]

Figure 2 is an approximate sketch of the proportion of energy sources for ORC systems. According to the pie chart seen in fig. 2, the percentage of biomass energy source is 48%, the percentage of geothermal energy source is 31%, the percentage of waste heats energy source is 20%, and the percentage of solar energy source is 1%.

The ORC energy recovery system overview

An ORC is energy recovery system shown in fig. 1, like a traditional Rankine steam system, except the organic working fluid, which is high molecular mass fluid with a liquid-vapor phase change, or boiling point at a lower temperature than the water-steam phase transition [13-15].

The low temperature heat is transformed into useful work to be transformed into electricity. Selection of the fluid of ORC system relies on the temperatures of the thermal sink and also the thermal source [16-18].

The ORC energy recovery system is a closed loop consisting four fundamental functions:

– 1–2 Pump compression: A feed pump make liquid working fluid is pressurized.
– 2–3 Boiler vaporization: The liquid working fluid takes thermal energy in and vaporizes to the steam phase. Evaporators provide the heat transfer from the heat carrier fluid to the working fluid.

– 3–4 Expansion by an expander: An expander transforms the heat energy of the working fluid into mechanical energy, which is the power generating operation. Then, an alternator makes this mechanical energy to be transformed into electricity.
– 4–1 Condensation by a condenser: The steam fluid condenses into the liquid phase, which is the heat-refusing operation [19-21]. In fig. 3 general system layout is given.

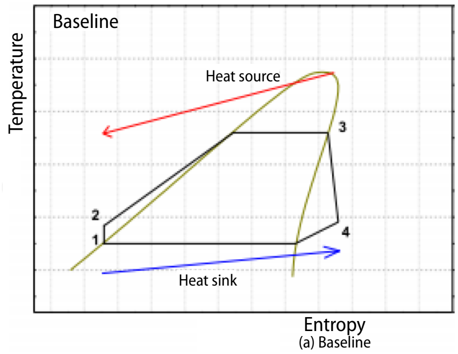

Figure 3. General system layout; a typical *T-s* process diagram for the ORC system utilizing organic fluid [16]

Selection of working fluids

Working fluid selection is the significant concept in order to further develop the design of the ORC system. Since the working fluid selection involves the thermodynamic design and effectiveness of all parts in an ORC, it is a restricting aspect of ORC system design. Additionally, the required pressure grade and material class of all components within the system is also affected by the refrigerant [22-24].

The common specifications which the working fluid should ideally suit are various and significant. Some of the preferred physical and chemical properties are non-flammability, stability, non-corrosiveness, non-fouling, and non-toxicity. Moreover, the low boiling point of the working fluid is another property to be used in a binary power cycle as involving with low-temperature geothermal waters [25-27].

Considering working fluids that are appropriate for a binary geothermal power cycle, it can be expressed that: Ammonia is toxic with exposures of more than 500 ppm presenting an immediate hazard to life and health. The N-Pentane is highly flammable. Chlorofluorocarbons has ozone depletion characteristics. The PF5050 is non-toxic, non-flammable and it has zero ozone depletion capability [28-30].

Features of various organic fluids can be inspected. Accordingly, R114 (C2Cl2F4) is non-flammable, non-toxic, generates an overpressure to the condenser, gives fine process efficiency, has adequate thermal stability and permits the utilization of a low-cost, single stage turbine. It should be noticed that the full lack of oil (due to high-speed technology) essentially enhances the thermal stability of R114. However, since R114 is a CFC-compound, it is essential to discover other options. Because the process is hermetic, there is no actual hazard for the ozone layer, yet by utilizing a CFC-compound it is hard to satisfy the international requirements. Thus, other fluids, fluorine-85 and toluene ($C_6H_5CH_3$), can be analyzed for the other prototype builts. The amount of thermal decomposition of toluene in oil-free circumstances is very low up to about 400 °C. In terms of low temperature applications, isobutane (C_4H_{10}) is found to be rather appropriate. In those circumstances, it follows better than toluene to be cooled. Common refrigerants utilized within ORC systems were outlined and summarized in tab. 1 [31-33].

The R-134a is readily accessible, low cost and widely used in the refrigeration industry as a working fluid. The thermodynamic efficiency of R-134a at temperatures higher than 50 °C was detected to be amazingly deficient. Higher process pressures also declined the safety factor of experimental components. Therefore, a more profitable refrigerant can be selected. A refrigerant mix called HFC-M1 can be chosen for utilization with the ORC test bed. The HFC-M1 is a mix of 50% R-245fa and 50% R-365mfc. A boiling temperature of 30

Table 1. Properties of some common ORC working fluids [17]

Shrae number	Name	Molecular weight	T_c [K]	P_c [MPa]	Vapor C_p [Jkg^{-1}K^{-1}]	Latent heat L [kJkg^{-1}]	ξ [Jkg^{-1}K^{-2}]
R-21	Dichlorofluoromethane	102.92	451.48	5.18	339.85	216.17	–0.78
R-22	Chlorodifluoromethane	86.47	369.30	4.99	1069.13	158.46	–1.33
R-23[a]	Trifluoromethane	70.01	299.29	4.83	3884.02	89.69	–6.49
R-32	Difluoromethane	52.02	351.26	5.78	2301.61	218.59	–4.33
R-41[a]	Fluoromethane	34.03	317.28	5.90	3384.66	270.04	–7.20
R-116[a]	Hexafluoroethane	138.01	293.03	3.05	4877.91	30.69	–5.54
R-123	2,2-Dichloro-1,1,1-trifluoroethane	152.93	456.83	3.66	738.51	161.82	0.26
R-124	2-Chloro-1,1,1,2-tetrafluoroethane	136.48	395.43	3.62	908.70	132.97	0.26
R-125	Pentafluoroethane	120.02	339.17	3.62	1643.89	81.49	–1.08
R-134a	1,1,1,2-Tetrafluoroethane	102.03	374.21	4.06	1211.51	155.42	–0.39
R-141b	1,1-Dichloro-1-fluoroethane	116.95	477.50	4.21	848.37	215.13	0.00
R-142b	1-Chloro-1,1-difluoroethane	100.50	410.26	4.06	1036.52	185.69	0.00
R-143a	1,1,1-Trifluoroethane	84.04	345.86	3.76	1913.97	124.81	–1.49
R-152a	1,1-Difluoroethane	66.05	386.41	4.52	1456.02	249.67	–1.14
R-170[a]	Ethane	30.07	305.33	4.87	5264.72	223.43	–8.28
R-218	Octafluoropropane	188.02	345.02	2.64	1244.87	58.29	0.45
R-227ea	1,1,1,2,3,3,3-Heptafluoropropane	170.03	375.95	3.00	1013.00	97.14	0.76
R-236ea	1,1,1,2,3,3-Hexafluoropropane	152.04	412.44	3.50	973.69	142.98	0.76
R-245ca	1,1,2,2,3-Pentafluoropropane	134.05	447.57	3.93	1011.26	188.64	0.60
R-245fa	1,1,1,3,3-Pentafluoropropane	134.05	427.20	3.64	980.90	177.08	0.19
HC-270	Cyclopropane	42.08	398.30	5.58	1911.81	366.18	–1.54
R-290	Propane	44.10	369.83	4.25	2395.46	292.13	–0.79
R-C318	Octafluorocyclobutane	200.03	388.38	2.78	896.82	93.95	1.05
R-3-1-10	Decafluorobutane	238.03	386.33	2.32	928.83	77.95	1.32
FC-4-1-12	Dodecafluoropentane	288.03	420.56	2.05	884.25	86.11	1.56
R-600	Butane	58.12	425.13	3.80	1965.59	336.82	1.03
R-600a	Isobutane	58.12	407.81	3.63	1981.42	303.44	1.03
R-601	Pentane	72.15	469.70	3.37	1824.12	349.00	1.51
R-717	Ammonia	17.03	405.40	11.33	3730.71	1064.38	–10.48
R-718	Water	18.00	647.10	22.06	1943.17	2391.79	–17.78
R-744[a]	Carbon dioxide	44.01	304.13	7.38	3643.72	167.53	–8.27
R-1270	Propene	42.08	365.57	4.66	2387.36	284.34	–1.77
	Propyne	40.06	402.38	5.63	2100.54	431.61	–1.87
	Benzene	78.11	562.05	4.89	1146.72	418.22	–0.70
	Toluene	92.14	591.75	4.13	1223.90	399.52	–0.21

[a] The critical temperature of the fluid is below 320 K, and the data is given based on 290 K.

°C at atmospheric conditions also reinforces the facility of management of this refrigerant as shown in tab. 1 [34-36].

Lower boiling point and dry saturation curve are some of the various advantages of the refrigerant offer over water which is advantageous for the system efficiency since the steam flowing through the turbine is not a wet mixture [36-38].

Thus, high density and low latent heat are the properties which the organic substance chosen for working fluid must have. In order to boost the turbine inlet mass-flow rate, such properties are favored. The thermal properties of these working fluids can be observed from tab. 1 [39-41].

Thermodynamic modeling

A thermodynamic model of the conceptual design can be developed in engineering equation solver and validated against experimental data [28]. Then, the model can be adapted to utilize both the convenient resource temperatures and prospected performance characteristics of the system components. Assessments could then be performed for the function of each of the four major components within the system [42-44]. The system efficiency for an ORC system is a key performance parameter and is defined:

$$\eta_t = \frac{\dot{W}_{\text{gen}} - \dot{W}_{\text{input}}}{Q_{\text{in}}} \tag{1}$$

The thermal efficiency can be evaluated as 5.7%. The efficiency of an ORC system is typically very low due to the low resource temperatures used. The theoretical maximum efficiency for a thermodynamic cycle is the Carnot efficiency of a cycle:

$$\eta_t = \frac{T_h - T_c}{T_h} \tag{2}$$

Thermodynamic efficiency is often utilized as a gauge for ORC system design, yet is not the only concern. Maintenance, facility of manufacture, essential costs and environmental effect all need to be estimated when optimizing the system design. These concerns are equalized with thermal efficiency during the design operation and concluded in the design efficiency of 5.7% [45-47].

Exergy efficiency

Exergy efficiency for the binary geothermal power plants is described as the rate of exergy output to the exergy input. In binary geothermal cycles, the geothermal fluid leaving the evaporator is reinjected into the ground and its exergy is lost [48, 49]. Thus, efficiency is formed on the primary heat source exergy as given by eq. (3):

$$\eta_{ex} = \frac{E_{\text{out}}}{E_{\text{heat source}}} \tag{3}$$

where

$$E_{\text{heat source}} = (h_{\text{HWI}} - h_0) - T_0(s_{\text{HWI}} - s_0) \tag{4}$$

and

$$E_{\text{out}} = (h_{\text{HWI}} - h_{\text{HWO}}) - \frac{\dot{m}_{\text{HW}}}{\dot{m}_{\text{CW}}}\left\{273.15 c_p \exp\left[\frac{\dot{m}_{\text{HW}}}{\dot{m}_{\text{CW}}} x\ (s_{\text{HWI}} - s_{\text{HWO}}) - s_{\text{CWI}}\right] - h_{\text{CWI}}\right\} \tag{5}$$

Equation (5) is evolved utilizing energy and entropy balances for the whole cycle.

The thermal efficiency of the ORC is the ratio of the net power output to the heat inclusion. However, this definition could be deceptive when considering various working fluids under particular operating conditions if the heat source of the inlet temperature and the pinch point are set. Nevertheless, the diversities of the thermal efficiency is straightly related to the variations of $\dot{W}_{\text{net}}$ since the inlet, outlet temperatures and the flow rate of the heat source are set. Furthermore, the exergy efficiency is analyzed herein, which can be utilized to gauge the efficiency for waste heat recovery since the thermal efficiency can not indicate the capability to transform energy from low-grade waste heat into functional work [50, 51]. Regarding P_0 and T_0 to be the ambient pressure and temperature, respectively, as the particular dead reference state, the exergy of the state point can be expressed:

$$\dot{E}_i = \dot{m}\left[(h_i - h_0) - T_0(s_i - s_0)\right] \tag{6}$$

The exergy efficiency of ORC system can be considered:

$$\eta_{\text{exg}} = \frac{\dot{W}_{\text{net}}}{\dot{E}_{\text{input}}} \tag{7}$$

The ORC power plant design and testing

The major modules can be designed as standard modules, which constitute the ORC energy plant, which means the power plant can simply be conformed to any waste heat origin by only transforming the central cycle to the particular industry plant. The switch module is designed for stable operating process circumstances. Furthermore, the power plant can be designed to only utilize a limited surface area, due to a rather compact constitution [52, 53].

Temperature, pressure, and flow rate data is obtained throughout the system to permit the utilizer to process the system and attain a steady-state at the requested process conditions. A CompacDAQ can be utilized for data acquisition and also permits the system performance to be calculated. Presently, the system will require to be commanded by the utilizer, however, as the system dynamics are comprehended better PID control will able to be enabled. Constitution of the system can be accomplished throughout nearly one year and then the system can be prompt for testing. Most of the components chosen are readily accessible which shortened the fabrication time plan. Extra caution should be taken throughout the constitution and assembly operations in order to assure no leakage in the system, which avoids a deficit of refrigerant [54-56].

The compact and modular design of the power plant is demonstrated in fig. 4: all components requiring monitoring or maintenance (electro-mechanical parts such as turbine, pumps, generator, valves, *etc.*) are fixed at the lowest stage. The median floor stage includes every static systematic component as process heat exchangers, piping. Additionally, the condensers are beared by the top of the steel body [57, 58]. The system can be merged in almost any industrial processes due to its modular design. The two-cycle system provides an adaptable integration in current plants. The flexibility in its function permits for economic utilization of waste heat with temperatures beginning from 150 °C to produce power from 500 kW to double-digit MW [59, 60].

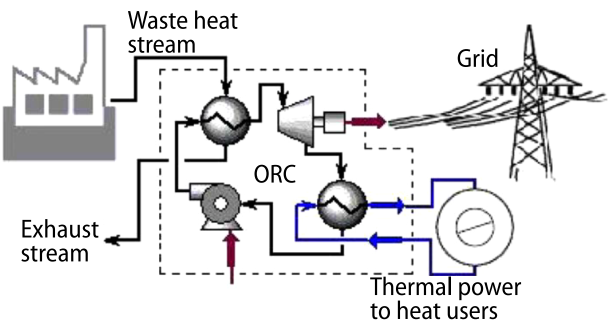

Figure 4. Illustration of ORC power plant [40]

The ORC power plant is – from the cement maker's perspective – an installation of secondary significance, thus it must not intervene in or even disrupt the main process of cement manufacturing. This requirement is performed by fixing the heat exchangers in a bypass mode. If the ORC power plant should somehow be out of service, while the kiln is under full operation, the waste gas bypasses the heat exchanger and takes the regular path through the cooling tower. In bypass mode, the clinker cooler air is cooled in the existing cooler before it arrives the dust precipitator [61-63].

Thermodynamic investigation of our ORC

In the numerical study, thermodynamic modeling of ORC using 1-D code has been implemented for investigating an efficiency, network and power output of the system as illustrated in fig. 5.

The performance of the ORC system is shown in fig. 6 at given operating conditions. In this case, the situation was examined in the simulation involved a cold start of the system. The heat source temperature was selected 973 K, and the mass-flow rate was 0.23 kg/s. Both the pressure and the temperature eventually converged in the model. After the model was validated, further optimization of the ORC system was based on the observations of the possible behavior of the selected real system. For these models, ORC analyses should include bypass valves for the expander and heat exchanger based on the experimental set-up which will be selected.

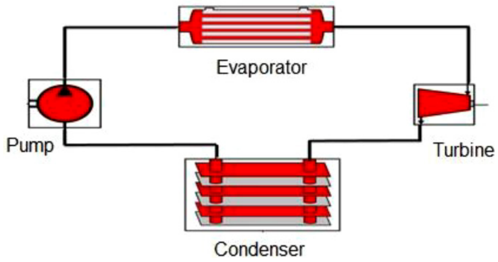

Figure 5. Sketch of our ORC system for thermodynamic analysis

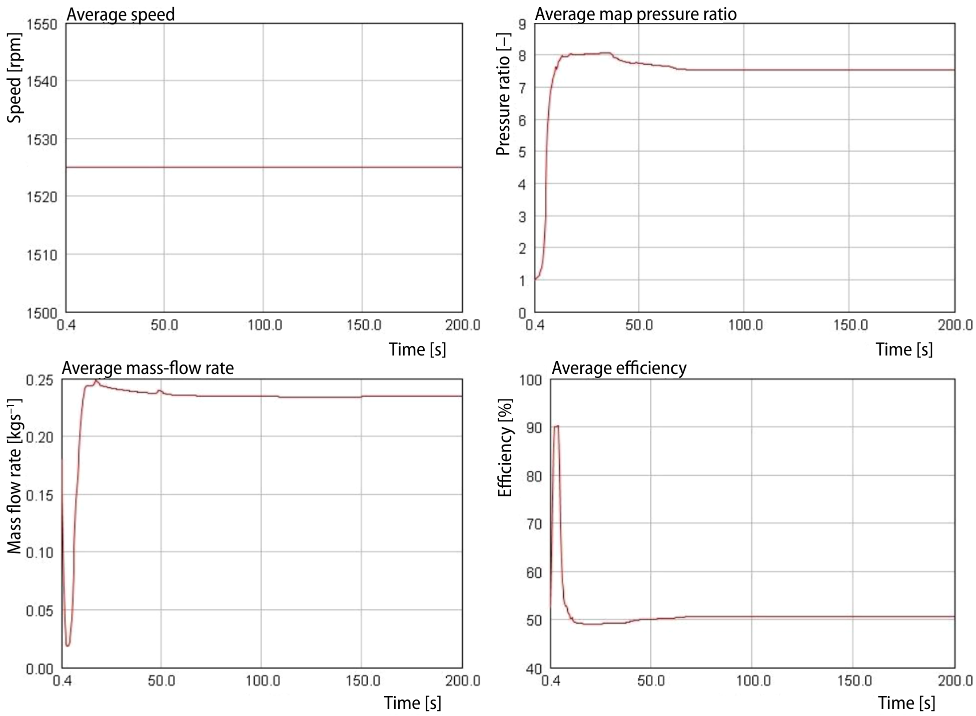

Figure 6. Performance of our ORC system

Findings according to our thermodynamic analysis are given in Appendix.

Pump conditions:	
Average speed bar	1750.0
Average pressure rise [bar]	20.3864
Average mass-flow rate [kgs^{-1}]	0.23438
Average inlet volume flow rate [Lmin^{-1}]	10.9015
Average total (isentropic) efficiency %	61.980045

Slave evaporator conditions:	
Average pressure drop [bar]	0.4436
Mass averaged temperature (inlet) [K]	16.96
Mass averaged temperature (outlet) [K]	478.09
Average wall temperature [K]	86.81
Combined energy rate out of fluid [kW]	–78.44
Average mass-flow rate (inlet) [gs^{-1}]	234.37
Average volume flow rate (inlet) [Ls^{-1}]	0.181

Turbine conditions:		*Master condenser conditions:*	
Average speed bar	1525.0	Average pressure drop [bar]	0.2797
Average map pressure ratio	7.533982	Mass averaged temperature (inlet) [K]	442.2
Average mass-flow rate [kgs^{-1}]	0.2343876	Mass averaged temperature (outlet) [K]	315.44
Average efficiency [%]	50.65889	Average wall temperature [K]	311.603
Average power (incl. shaft if modeled) [kW]	5.1654296	Combined energy rate out of fluid [kW]	72.9
Average efficiency (ideal power weighted) [%]	50.65889	Average mass-flow rate (inlet) [gs^{-1}]	234.4
		Average volume flow rate (inlet) [Ls^{-1}]	20.91

Conclusions

This paper investigates an ORC system for the industrial waste heat as the focus is on the waste heat recovery application. The ORC is generally recognized as an applicable technology to transform low temperature heat into electricity. Moreover, ORC are designed for the unmanned process with a small amount of maintenance. Industrial waste heat, solar heat, geothermal energy and biomass combustion heat are recoverable energy resources. The ORC energy recovery system was modeled and reviewed. In addition, a comprehensive literature survey is carried out. Thus, many recent papers about ORC have studied over accordingly in the literature section. Then, they were expressed in this paper in order, to sum up. A thermodynamic analysis was also carried out on an ORC system throughout this paper. Binary fluid cycles utilize a blend of water/ammonia, yet are rather complicated compared to Rankine cycles. Additionally, the organic substance chosen for the working fluid must have low latent heat and high density to boost the turbine inlet mass-flow rate. Typical features of an ORC module are indicated as heat source temperature, power output, thermal efficiency, *etc.* The maximum thermal efficiency of a module is about 25%. The choice of a module is basically founded upon operation, heat source temperature and aimed power output. The selection of the sort of ORC machine is commonly affected by the nature, condition, and temperature of the heat source. Thus, the temperature is a critical property during selection in case studies. According to results, the system varies according to technology, size and cost aspects. Moreover, machine, engineering, system integration, capital costs are contained in the investment cost of an ORC project and also the investment cost is closely related to the application.

Since the ORC consumes virtually no additional fuel, the emission of environmental pollutants such as CO_2, SO_2, and *etc.* would be decreased. Moreover, the exhaust heat exiting from the ORC could be further used to drive chillers such as absorption chillers to supply cooling capacity if required by the local demand. Moreover, further work is needed to enhance the ORC performance, such as decreasing the heat and pressure losses in the whole system and developing the performance of the common parts such as the turbine, generator, and heat exchangers.

Nomenclature

C_p – specific heat, [Jkg^{-1}K^{-1}]
E – exergy, [kJ]
h – specific enthalpy, [kJkg^{-1}]
h_{CWI} – specific enthalpy of cooling water inlet, [kJkg^{-1}]
h_{HWI} – specific enthalpy of geothermal water inlet, [kJkg^{-1}]
h_{HWO} – specific enthalpy of geothermal water exit, [kJkg^{-1}]
L – latent heat, [kJkg^{-1}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – pressure, [kPa]
P_c – critical pressure, [MPa]
$\dot{Q}$ – heat rate, [kW]
Re – Reynolds number, [–]
s – specific entropy, [kJkg^{-1}K^{-1}]
s_{CWI} – specific entropy of cooling water inlet, [kJkg^{-1}K^{-1}]
s_{HWI} – specific entropy of geothermal water inlet, [kJkg^{-1}K^{-1}]
s_{HWO} – specific entropy of geothermal water exit, [kJkg^{-1}K^{-1}]

T – temperature, [K]
T_c – critical temperature, [K]
$\dot{W}$ – work rate, [kW]
$\dot{W}$ – power, [kW]
x – vapour quality

Greek symbols

η – energy efficiency
ξ – molecular latent heat, [Jkg^{-1}K^{-2}]
ψ – exergy efficiency

Subscripts

c – cooling
gen – generator
h – heating
ORC – organic Rankine cycle
1-46 – state numbers
0 – ambient (or reference environment) condition

Appendix

Table 3. Findings according to our thermodynamic analysis

Compressor turbine pump fan			Heat exchanger		
			Nusselt number correlation		
	PumpRefrig	TurbineRefrig		PumpRefrig	TurbineRefrig
Type of device	Pump	TurbineRefrig	Type of device	Pump	TurbineRefrig
Speed [rpm]	1750	1525		Cond_Heat Transfer: Condenser _Master	Evap_HeatRate: Evaporator _Master
Pressure Ratio (static)	8.51	7.57	Prandtl Number	--	0.6946
Pressure ratio	8.51	7.53	Min hA_ratio	--	0.0786
Mass-flow rate [kgs^{-1}]	0.23	0.23	Max hA_ratio	--	0.3995
Power [kW]	0.6	5.2	Slave Nusselt Correlation	Slave	Slave
Efficiency [%]	62	50.7	Re lower limit	67	--
Inlet pressure [bar]	2.71	22.66	Laminar Re number limit	180.977	--
Outlet pressure [bar]	23.1	2.99	Transition Re number limit	182.193	--
Inlet temperature [K]	315	478	Re upper limit	190.059	--
Outlet temperature [K]	317	442	Slave Nusselt correlation	Slave	Slave
Pressure rise [bar]	20.39	-19.66	Re lower limit	67	--
Heat exchanger			Laminar Re number limit	180.977	--
Nusselt number correlation			Transition Re number limit	182.193	--
	Cond_Heat-Transfer: Condenser _Master	Evap_HeatRate: Evaporator _Master	Re upper limit	190.059	--
Master Nusselt correlation	Master	Master	Laminar exponent	0.905	--
Re lower limit	--	890	Transition exponent	0.9541	--
Laminar Re number limit	--	3144	Turbulent exponent	0.928	--

→

Table 3. (continuation)

Compressor turbine pump fan			Heat exchanger		
			Nusselt number correlation		
	PumpRefrig	TurbineRefrig		PumpRefrig	TurbineRefrig
Type of device	Pump	TurbineRefrig	Type of device	Pump	TurbineRefrig
Transition Re number limit	--	3328	Turbulent coefficient	0.0151	--
Re Upper limit	--	4074	Prandtl number	28	--
Laminar exponent	--	1	Mean relative error (%)	0.8963	--
Transition exponent	--	0.9866			
Turbulent exponent	--	0.6128			
Turbulent coefficient	--	0.1798			

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