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Research Article

OPTIMUM DESIGN OF ORGANIC RANKINE CYCLE FOR LOW TEMPERATURE HEAT SOURCES

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ABSTRACT:
The paper presents a study of new technology, called "Organic Rankine Cycle" (ORC), which can be operated likely Steam Rankine Cycle (SRC) but it can take more advantage than SRC. ORC can generate electricity with low temperature heat sources such as geothermal energy or waste heat released from the industry. Nevertheless, the thermo-economic optimization of this technology is still required. Therefore, in this study, the optimum design of ORC system by matching equipment and selection of appropriated working fluid are need to investigate in order to obtain the cost effective from low temperature heat resource. The operating condition of each working fluid was firstly investigated to match with the temperature of the heat sources. From the result, the optimum working temperatures of each working fluid can be suggested. Then the system design was calculated with the parametric survey on two types of heat sources; flue gas from industrial boiler and geothermal based heat source, which their temperatures were lower than 300℃. The variation of working fluids to obtain the optimum system with high thermal efficiency, high power output and short payback period were performed. The criterion of environmental friendly was one influence parameter taking into consideration to choose the appropriate working fluid in this study. The results can be concluded that R236ea, R245ca, R245fa were appropriate working fluid for ORC when it is applied with low temperature heat sources.

Keywords: *Organic Rankine Cycle, Working fluid, Low temperature heat source, Waste heat*

1. INTRODUCTION

The fluctuation of fossil fuel price and a limit of the supply have significant effect on the electricity generation which leads alternative energy resources, especially renewable energy and the heat recovery to be attractive. However, to employ the useful energy from low temperature heat sources (with temperature less than 300°C), new technology called "Organic Rankine Cycle" (ORC) is more appreciate than a conventional Rankine cycle [1]. The concept of ORC basically looks like steam Rankine cycle (SRC) in term of thermodynamic principle, but working fluid is different. High molecular mass fluid with a lower boiling temperature is used in the ORC system instead of water to convert the low temperature heat source into mechanical work and electricity, respectively. Therefore, the mass flow rate can be increased for the same size of turbine.

Even the advantage of ORC system makes it more interest but high investment cost is one of fence for the technology

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deployment of small scale ORC system [2]. The selection of working fluid and equipment matching are the important and primary concern to maximize the benefit from the ORC system. In ORC system, several working fluids can be employed, however, the working fluid selection could be relevant to its thermo-physical characteristics and the nature of heat source temperatures because not all working fluid appropriately operate in ORC power plant [3-4]. To use solar thermal energy as heat source, the optimization result show that R245fa can give better benefit than the other. This was because R245fa has low boiling temperature, which has high degree of thermal stability and compatible with common construction materials of ORC [2]. Therefore, it can produce higher power output and higher efficiency. On the other hand, to apply the ORC system with the geothermal heat source, R134a was the most appropriate one, while as R245fa can be better to convert renewable energy to electricity [4].

To decrease the risk of ORC system's investment, the feasibility study could be initially performed before the decision, especially when a heat source comes from low temperature heat source such as geothermal energy, or waste heat released from industry. Therefore, the objective of this study was to evaluate the appropriate design of ORC system to obtain higher useful power output and higher efficiency with shorter payback period by using a simple program "Microsoft Excel" for the analysis. In this study, the design of ORC system was based on a simple Rankine cycle. Geothermal energy as a free energy and waste heat from boiler stack for heat recovery and energy efficiency were intended as the heat sources to vaporize the working fluid. Finally, the proper equipment matching and appropriate working fluid, which is friendly to human and environment, were concluded at the end of this paper.

2. System Description

Rankine cycle is an ideal thermodynamic process of the heat engine which transforms heat into mechanical work by the operation of steam and using conventional thermal power plant to generate the heat. To operate steam Rankine cycle, it is restricted by higher temperature heat source to vaporize the water as compared with organic working fluid. To overcome the drawback of steam Rankine cycle, organic Rankine cycle, which uses a high molecular mass organic fluid as a working fluid, could take an advantage than SRC. Organic fluid can be vaporized at a lower temperature than the water and transfers energy from low temperature heat source in useful work.

In this study, a simple system design of ORC was intended. It basically consisted of four components: Pump, Evaporator, Turbine and Condenser, as shown in Figure 1. The appropriate selections were taken into consideration on their operating conditions such as the temperature, the mass flow rate, and the pressure. Figure 2 presents a Temperature-Entropy diagram of the system.

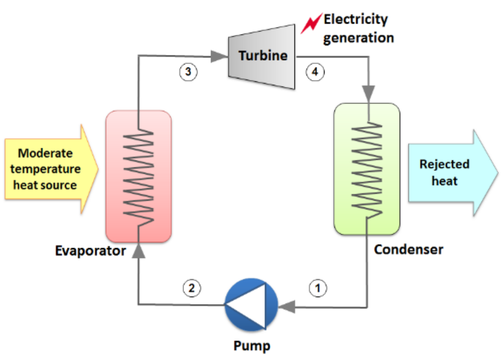

Fig. 1. Simple organic Rankine cycle.

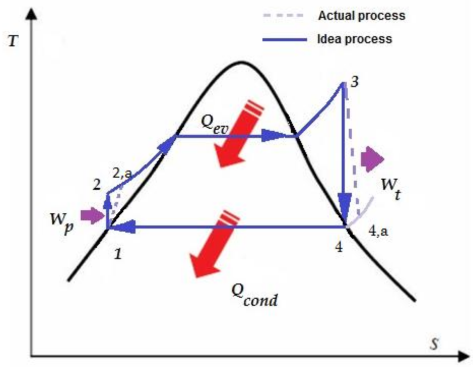

Fig. 2. Temperature-Entropy diagram of ORC.

To comparatively investigate the feasibility of ORC system for low temperature heat source, therefore, this study focused on 2 types of heat sources to drive the ORC system. The first heat source was the waste heat from the stack of boiler. The second one was the heat extract from geothermal energy which was carried out by secondary working fluid such as water.

3. MODELING

This section describes the model of each machine of the ORC system.

3.1 Pump model

Pump was used to compress and transfer working fluid to the evaporator. The process of pump was assumed with its isentropic efficiency of 0.8 ($\eta_p = 0.8$). To calculate the electrical consumption of pump, Eq.(1) can be applied when η_p was known.

$$\eta_P = \frac{W_P}{W_{P,a}} = \frac{v_{f1}(P_2 - P_1)}{h_{2a} - h_1} \tag{1}$$

where W_p and $W_{p,a}$ are the pump work consumption by theoretical and the actual one. P_1 and P_2 are suction and discharge pressures of the pump. v_{f1} is specific volume of saturated liquid at the pump suction. h_1 and $h_{2,a}$ are the enthalpy at the pump inlet and the real pump outlet.

3.2 Evaporator and condenser models

Working fluid was vaporized at the evaporator under the process of isobaric reversible heating. According to a small pressure drop in the evaporator when it was appropriately design. Therefore, pressure drop can be neglected in this study. This assumption was also applied to the condenser. The design of shell and tube heat exchanger was proposed when it was liquid-to-liquid heat transfer, while the case of gas-to-liquid heat transfer, tube bank cross flow heat exchanger was more appropriate in this study. To investigate the heating surface area, the different of Nusselt number (Nu) equations were applied for different types of heat exchangers as shown in Eq.(2) and Eq.(3).

For tube bank cross flow heat exchanger, the heat convection coefficient can be evaluated from [5]

$$Nu = \frac{D \cdot h}{k} = 0.35\left(\frac{s_T}{s_L}\right)^{0.2} (\mathrm{Re})^{0.8} (\mathrm{Pr})^{0.36} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25} \tag{2}$$

For shell and tube heat exchanger, it can find by using [5]

$$Nu = \frac{D \cdot h}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{\frac{1}{2}}\,\mathrm{Pr}^{\frac{1}{3}}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\mathrm{Re}}{282000}\right)^{\frac{5}{8}}\right]^{\frac{4}{5}} \tag{3}$$

The heat transfer area *(A_s)* can be calculated by using

$$A_s = \frac{\dot{Q}}{U \cdot \Delta T_{LMTD}} \tag{4}$$

where

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} \tag{5}$$

$$\Delta T_{LMTD} = \frac{T_1 - T_2}{\ln(T_1 / T_2)} \tag{6}$$

in which

$$T_1 = T_{h,in} - T_{c,out} \tag{7}$$

$$T_2 = T_{h,out} - T_{c,in} \tag{8}$$

3.3 Turbine model

High pressure and high temperature working fluid in vapor-phase passed through a turbine under adiabatic expansion process. Isentropic efficiency of turbine (η_t) at 0.8 was taken into account to make the investigation more reasonable. To find turbine work, it can be calculated from Eq.(9) where η_t was 0.8.

$$\eta_t = \frac{W_{t,a}}{W_t} = \frac{h_3 - h_{4,a}}{h_3 - h_4} \tag{9}$$

where W_t is the turbine work output by theoretical and $W_{t,a}$ is the actual turbine work output. h_3 and h_4 are the enthalpy at the turbine inlet and turbine outlet, while h_{4a} is the enthalpy of the actual turbine outlet.

Turbine selection was depended on the operating condition, including its capacity and pressure ratio, as summarized in Table 1. This research was focused on two types of turbines: radial flow turbine and screw expander because of a comprehensive range of working condition.

Table 1: The selection of expansion machine [6-11]

Type of expansion machine	Capacity (kW)	Pressure ratio
Radial inflow turbine	50 - 500	< 8
Screw expander	< 50	2.4 – 6.1
Scroll expander	1 - 10	2 – 12

3.4 System performance

The performance of the ORC system can be evaluated using

$$\eta_t = \frac{Net\ power\ output}{Heat\ input\ rate} = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_{ev}} \tag{10}$$

Where η_t is thermal efficiency of ORC system $\dot{Q}_{ev}$ is the heat input rate at the evaporator which can be calculated by using either Eq.(11) or Eq.(12).

$$\dot{Q}_{ev} = \dot{m}_r \left(h_3 - h_{2,a}\right) \tag{11}$$

$$\dot{Q}_{ev} = \varepsilon \cdot \dot{Q}_{source} \tag{12}$$

Where $\dot{m}_r$ is the mass flow rate of the organic working fluid, ε is the effectiveness of heat exchanger which is defined as the ratio of actual to maximum possible of heat transfer. $\dot{Q}_{source}$ is the heat transfer rate from the source. Thermal efficiency of ORC system was evaluated from the power output from turbine ($\dot{W}_t$) subtracted with the power supplied for the pump ($\dot{W}_t$) which can be determined from T-S diagram while the power supplied for cooling water system was excluded in this analysis.

3.5 Economic of ORC system

The economic evaluation of the ORC system was performed by using simple payback period method. The investment cost of the system was mainly from the cost of four main machines: evaporator, condenser, turbine, and pump which were varied depended on the sizing and the capacity and added with the fixed cost of piping and wiring system. The operating hour of the system was assumed at 24 hours per day with 300 days for a year. The net power output was connected with the generator to produce electricity and sell to the grid. The income benefit of the system can be calculated from the total electricity production which the unit price of electricity was 3 baht/kWh. Then the payback period can be investigated from the ratio of investment cost and the income.

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \tag{13}$$

$$Electricity\ production = \dot{W}_{net}\ x\ Operation\ hours\ per\ year \tag{14}$$

$$Income = Electricity\ production\ x\ Electricity\ price\ per\ unit \tag{15}$$

$$Payback\ period = \frac{Investment\ cost}{Annual\ Income} \tag{16}$$

4. Design Conditions for Working Fluid Operation

Design of operating conditions for various working fluids was initially performed before the feasibility study of ORC system. This was subject to clearly understand the appropriated selection of working fluid to the operating conditions such as the temperature, the pressure and the mass flow rate. The conditions were assumed by based on the typical waste heat from the boiler stack. Following assumptions were made:

- The exhaust gas from boiler was used as a heat source. The temperature was determined at 200^{o}C for the inlet temperature and not lower than 170^{o}C for the outlet temperature. A mass flow rate of exhaust gas was 4 kg/s which was equivalent to the boiler capacity of 700 Bhp.
- At the turbine inlet and the pump inlet, the states of working fluid were a saturated vapor and a saturated liquid, respectively.
- The condenser was cooled by cooling water, therefore, the outlet temperature of working fluid at the condenser can maintain at 40^{o}C under the capability of the cooling tower. While the outlet temperature at the evaporator was fixed at 140^{o}C.
- Working fluids were R11, R245ca, R245fa, R236ea, R134a, and R22.
- The effectiveness of heat exchangers were assumed at 0.65 for both evaporator and condenser.
- The isentropic efficiency of both turbine (η_t) and pump (η_p) were determined at 0.8.

The calculation result of R245fa was only explained in this section and the results of the others were summarized in Table 2. The calculation procedure, as descripted in Figure 3, was started by investigating power output from turbine, thermal efficiency, and the mass flow rate of R245fa in various turbine inlet temperatures. The result, as plotted in Figure 4, can be obviously noticed that power output at turbine and the thermal efficiency increased as the turbine inlet temperature increased, on the contrary the required mass flow rate of working fluid trended to decrease with the bottom point at the turbine inlet temperature of 140^{o}C. Design operating condition with lower mass flow rate of R245fa was preferable because the heat exchanger can become the compact design with lower maintenance cost. By comparing the optimum temperature and Pressure-Enthalpy diagram of R245fa, it can be noted that the optimum temperature of 140^{o}C was corresponded with the point on vapor saturation line of dry fluid at which its slope turned into the infinity, as shown in Figure 5.

This study investigated the optimum operating temperature of various working fluids. From the result, it can be noticed that the optimum temperature of each working fluid was always lower than their critical temperature. Table 2 summarizes the result of the optimum temperature of six working fluids of this study.

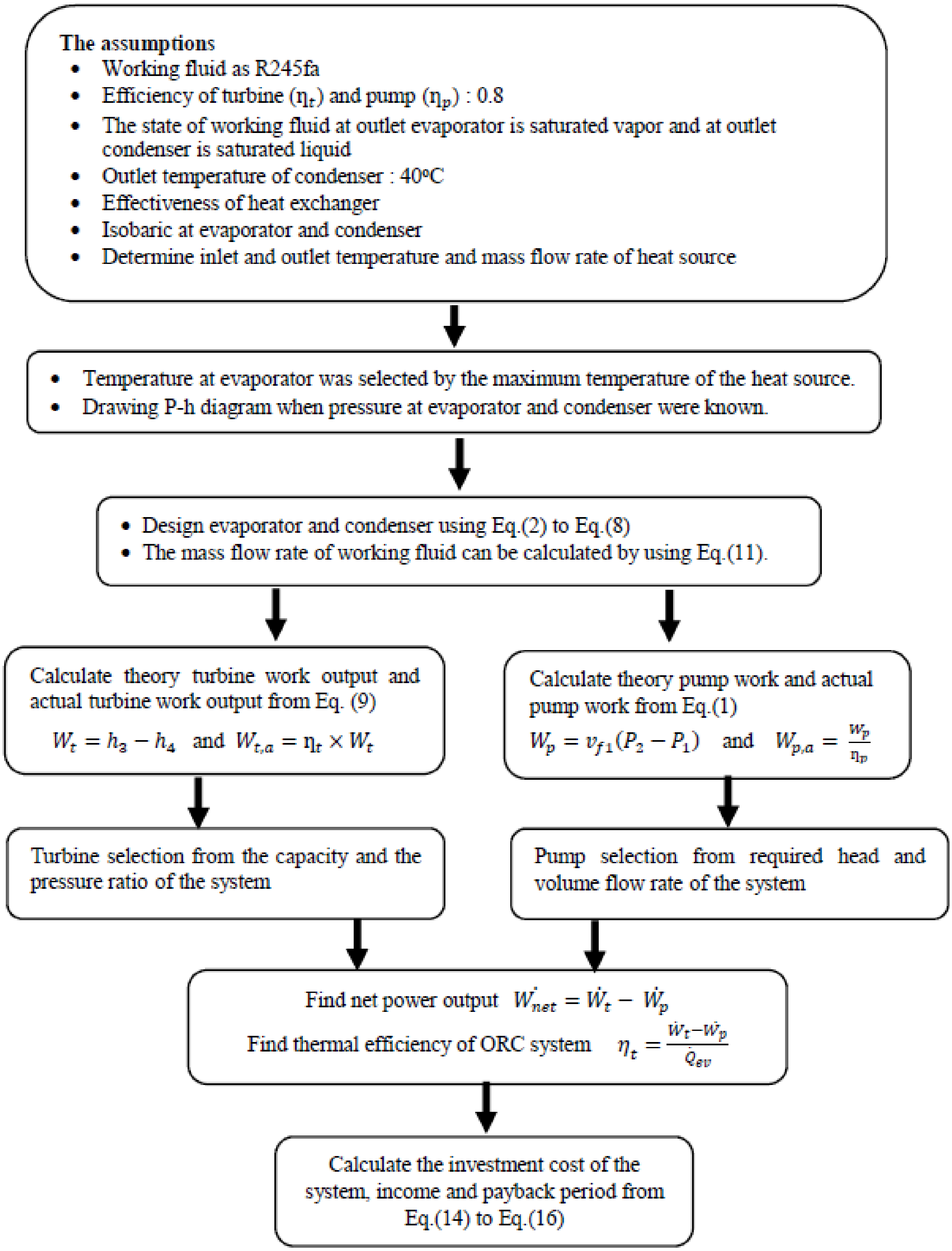

Fig. 3. Flow calculation of ORC system.

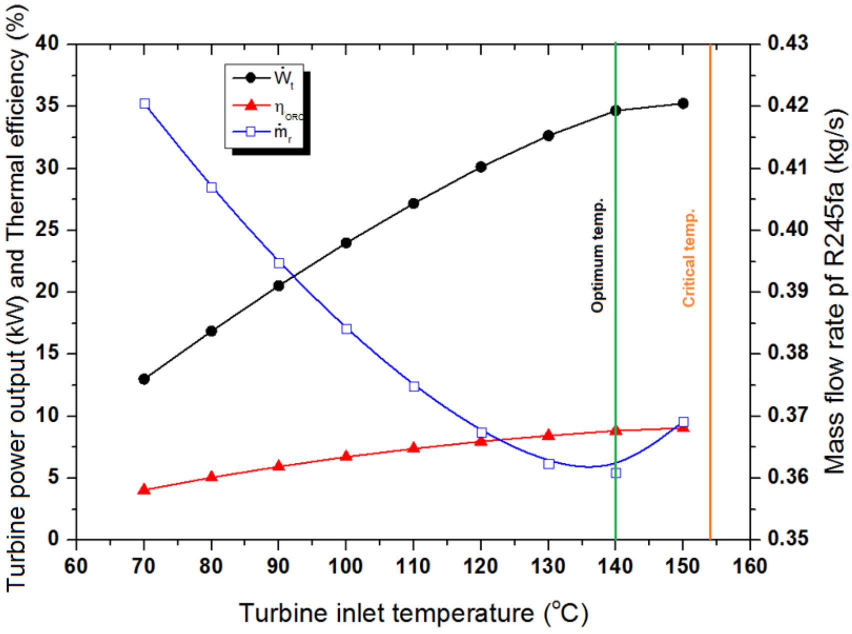

Fig. 4. Optimum temperature of R245fa for operation in the ORC system.

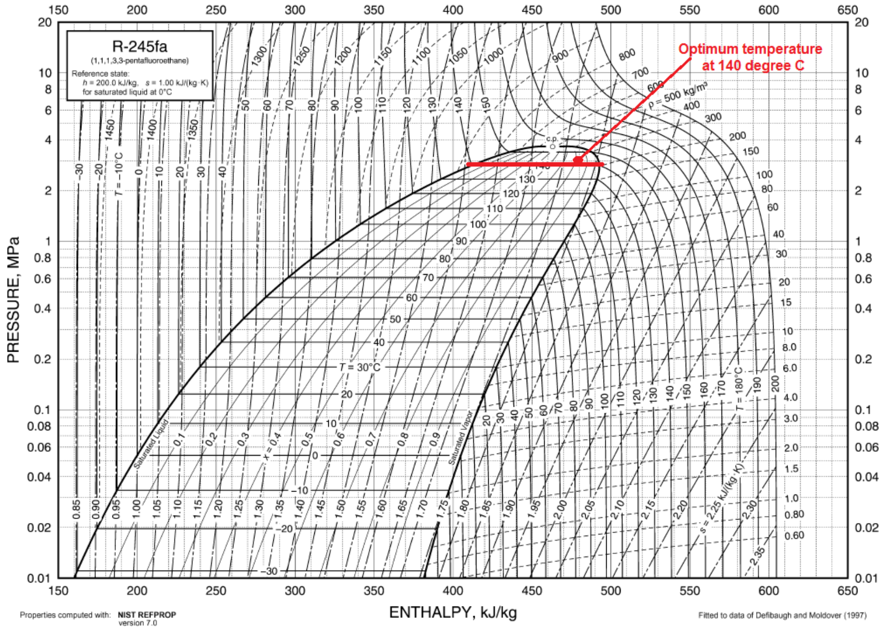

Fig. 5. Pressure – Enthalpy diagram of R245fa [18].

Table 2: The optimum temperatures for the operation of each working fluid from the calculation

Working fluid	Name	Critical temp. (°C) [10]	Optimum temp. (°C)
R11	Trichloromonofluoromethane	197.96	170
R245ca	1,1,2,2,3-Pentafluoropropane	174.42	156
R245fa	1,1,1,3,3-Pentafluoropropane	154.10	140
R236ea	1,1,1,2,3,3-hexafluoropropane	139.29	132
R134a	1,1,1,2-Tetrafluoroethane	101.06	76
R22	Chlorodifluoromethane	96.14	56

5. The Variation of Low Temperature Heat Sources

The second part of this study was to investigate the feasibility of ORC system when it was cooperated with low temperature heat source like exhaust gas from boiler or geothermal energy. Parametric surveys in various range of inlet heat source temperatures of both exhaust gas and geothermal energy accompanied with six different working fluids were performed to obtain the turbine power output, thermal efficiency of ORC system, and the investment return of a simple ORC system.

5.1 Heat source from exhaust gas of boiler

The exhaust gas from boiler was used as a heat source for the ORC system. The inlet temperature of the gas was varied in between 200 – 300^{o}C in order to investigate the effect of the heat source temperature to the net power output and the thermal efficiency. Figure 6 shows the result of net power output and the thermal efficiency with respect to the exhaust gas temperature, when the system was implemented with the different working fluids. Figure 7 represents the payback period to confirm the feasibility of the ORC system.

The results of exhaust gas used as the heat source show that the increase of inlet exhaust gas temperature led to the net power output increased, in the contrary, the thermal efficiency decreased as the result of more heat input than the useful power output. The payback period, which was calculated by using simple payback period method, was also inversely changed as the heat source temperature increased. This was due to the magnitude change of net power output was greater than the change of capital cost. However, it can be noticed that payback period at some point trended to increase which was caused by the change of turbine model when the capacity changed. This resulted into the increase of capital cost per unit power, and consequent to elongate the payback period.

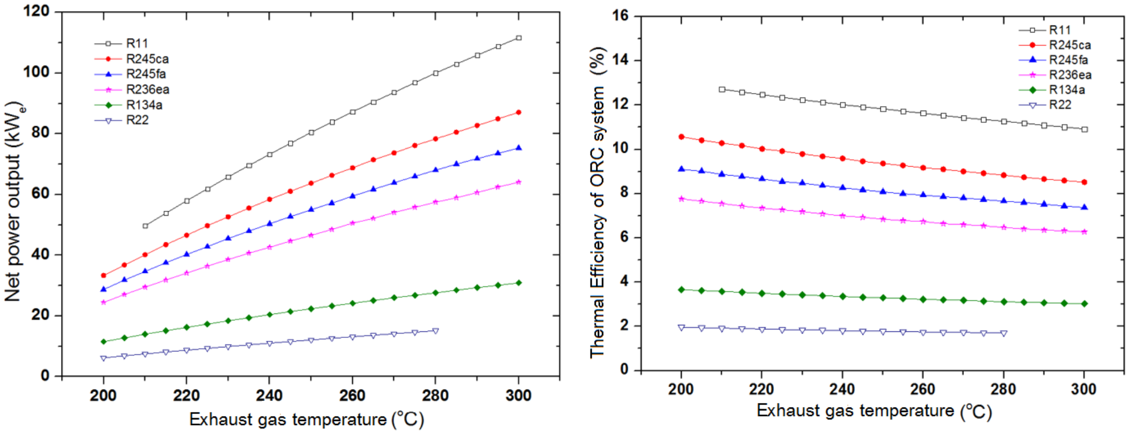

Fig. 6. Net power output and overall efficiency from ORC system with respect to exhaust gas temperature.

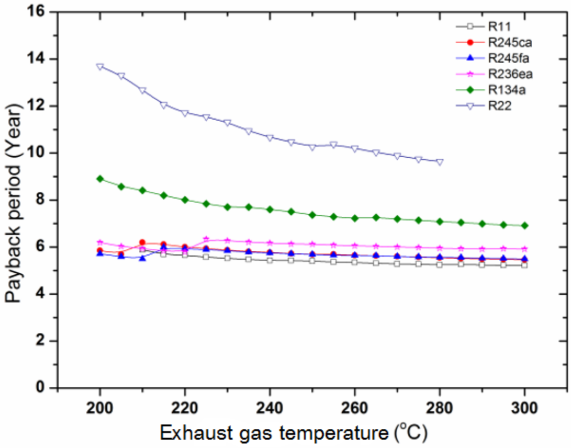

Fig. 7. Payback period of ORC system with respect to exhaust gas temperature.

The different working fluid was proposed to operate in the ORC system. The results show that the working fluid which has higher critical temperature can operate with greater useful turbine output and higher thermal efficiency. Those results also gave an impact on a shorter payback period. By using exhaust gas from boiler as the heat source, the appropriate working fluid can be arranged in order of the benefit as followed: R11, R245ca, R245fa, R236ea, R134a, and R22, respectively. From the order of appropriate working fluid, it can be obviously noticed that the sequence correlated with the critical temperature of working fluid.

5.2 Heat source from geothermal energy

The heat source was changed from waste heat from boiler to geothermal energy. Water, which is heavier and larger heat capacity, was used as intermedia working fluid to carry the heat from the underground. The temperature of intermedia working fluid was focused in the range of 140-160^{o}C and the water outlet temperature was fixed at 120 ^{o}C with the water flow rate of 18 m^3/h.

The results, as presented in Figure 8 and Figure 9, show that the tendency of power output, thermal efficiency, and the investment return of each working fluid was corresponding to the application of exhaust gas heat source. Power output increased, as the temperature increased, while the payback period and its efficiency decreased when the heat source temperature increased. The sequence of appropriate working fluid was also similar as the case of exhaust gas.

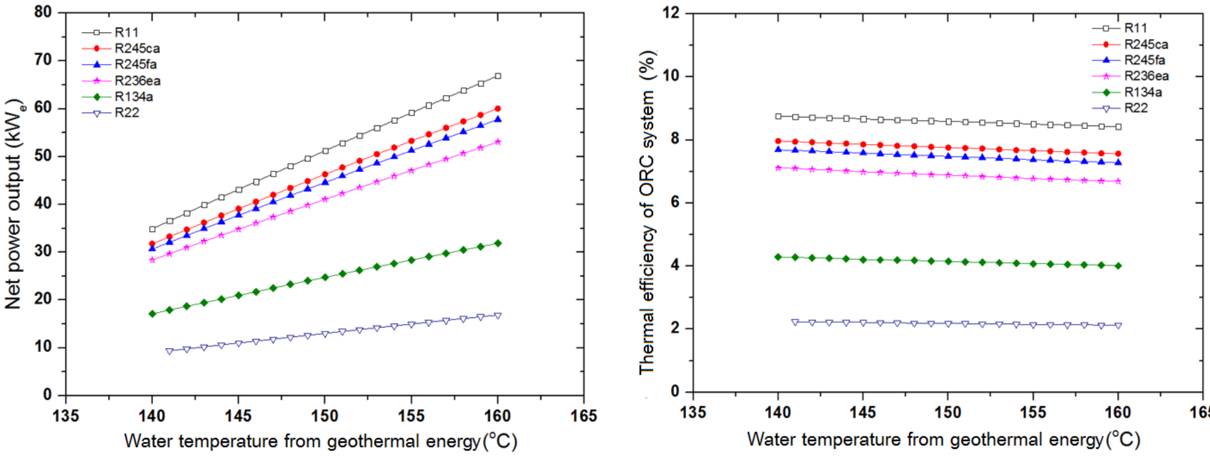

Fig. 8. Net power output and overall efficiency from ORC system using geothermal energy as heat source.

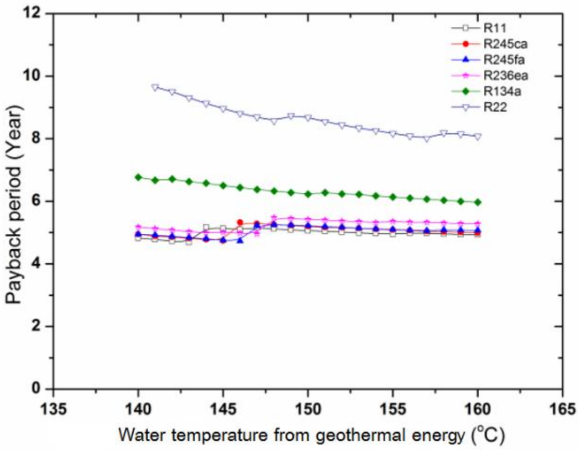

Fig. 9. Payback period of ORC system using geothermal energy as heat source.

6. WORKING FLUID PROPERTIES

The results in Section 5 show that R11 was the most suitable working fluid for the ORC system when it was applied for low temperature heat source. This was because the operating temperature of exhaust gas and geothermal energy in this study were specified in the range between 140 – 300^{o}C which was closed to the optimum working temperature of R11 and the reason of higher operating temperature, greater the benefit output. However, if an environmental friendly properties of working fluid was concerned in this study, a proper working fluid should be R245ca because it did not have the potential for Ozone Depletion Potential (ODP) and lower Global Warming Potential (GWP), as presented in Table 3. The most suitable working fluid for Organic Rankine Cycle at low heat source's temperature should be R245ca, R245fa and R236ea.

Table 2: The properties of working fluids [11-14]

Working fluid	ODP	GWP	Flammable	Toxic
R11	1	4000	None	Low
R245ca	0	640	None	Low
R245fa	0	950	None	Low
R236ea	0	1200	None	Low
R134a	0	1300	None	Low
R22	0.05	1700	None	Low

7. CONCLUSION

From the calculation by varying the working fluids, it was discovered that the increase of heat source temperature can increase the net power output, while as decrease the thermal efficiency of ORC system and shorter the payback period. The results show that the working fluids can be classified into 2 groups by considering the net power output, thermal efficiency, and the payback period. The first group was high potential working fluid and the second one was low potential working fluid. For the first group, high potential working fluids; including with R11, R236ea, R245ca, and R245fa, can contribute to higher thermal efficiency, higher net power output, and shorter payback period. However, if the selection of working fluid was considered on the effect to environment i.e. ODP, GWP, hazardous and flammability, the most suitable working fluid were only R245ca, R245fa and R236ea.

To utilize the waste heat from exhaust gas boiler in the ORC system and cooperate with the appropriate working fluids like R245ca, R245fa and R236ea which were concerned their effect on the environment, a simple ORC system design can generate the electricity around 20 – 75 kW_e with the thermal efficiency of 4.8 – 9.1% and able to get the return of investment within 4.7 – 6.7 years depended on the temperature of heat source and the type of working fluid.

For the free energy as geothermal energy which the temperature was exist around 140 - 160^{o}C, the electricity of 26.4 – 57.8 kW_e can be produced from a simple ORC system design with the thermal efficiency range of 6.2 – 7.7%. The payback period was around 4.7 – 5.7 years.

8. ACKNOWLEDGEMENT

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NOMENCLATURE

A	Area, m^2
D	Diameter, m
h	Enthalpy, kJ/kg
k	Thermal conductivity, W/m-K
$\dot{m}$	Mass flow rate, kg/s
Nu	Nusselt number, -
P	Pressure, Pa
Pr	Prandtl number, -

Q	Heat transfer, J
$\dot{Q}$	Heat transfer rate, W
Re	Reynolds number, -
S_L	Longitudinal distance between two consecutive tubes, m
S_T	Transverse distance between two consecutive tubes, m
T	Temperature, K
U	Overall heat transfer coefficient, W/m^2-K
W	Work, J
$\dot{W}$	Power, W
ε	Effectiveness, -
η	Efficiency, -
ν	Specific volume, m^3/kg

Subscripts

1	Pump inlet
2	Evaporator inlet
2a	Actual evaporator inlet
3	Turbine inlet
4	Condenser inlet
4a	Actual condenser inlet
c,in	Inlet cold working fluid
c,out	Outlet cold working fluid
e	Electric
ev	Evaporator
f1	Saturated liquid at the pump inlet
h,in	Inlet hot working fluid
h,out	Outlet hot working fluid
i	Inside
LMTD	Log mean temperature different
Net	Net
o	Outside
source	Source
r	Refrigerant
t	Turbine
p	Pump
t,a	Actual turbine
p,a	Actual pump

Abbreviations

GWP	Global Warming Potential
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
SRC	Steam Rankine Cycle

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