

Development of a 200-kW Organic Rankine Cycle Power System for Low-Grade Waste Heat Recovery

Taehong Sung and Kyung Chun Kim

Abstract—In this study, we designed a new plate-type heat exchanger system for the previous shell-and-tube evaporator with the same heat rate and additional turbine safety measure. The new evaporator system consists of two plate heat exchangers; one for preheating and the other for subsequent evaporation and superheating. The separate design reduces the liquid level of the evaporator and lowers the risk of liquid inflow into the turbine. For an additional driving stability, a co-current flow configuration was used for the preheater so that a subcooled liquid is introduced to the evaporator instead of a liquid-vapor mixture. A detailed design of the plate heat exchangers was carried out using a commercial software. The new heat exchangers are 75% smaller in volume and 83% smaller in weight. The designed heat exchangers were manufactured and installed in the previous ORC system. The performance was evaluated using an electric heater and a cooling tower as a heat source and heat sink. For the test, the working fluid pump was operated at the design point turbine inlet pressure of 2.13 MPa. Meanwhile, the condensation conditions were not matched to the design point due to the weather effect. Under the analysis condition, the new evaporator system showed an evaporator heat rate of 1662 kW, which is 91% of the design value with a pressure loss of 46 kPa. We calculated a pressure loss of 62.8 kPa for the design point using an equation for simple friction loss. The results showed that our new evaporator system successfully replaced the previous one without big performance losses.

Index Terms—Organic rankine cycle, preheater, plate heat exchanger, shell-and-tube heat exchanger.

I. INTRODUCTION

Many thermal energy systems discard the remaining energy to the environment without further utilization [1]. Utilizing these abandoned energies has a great impact on the economics of the system and also it can reduce environmental pollutions by improving the system efficiency. Efforts to improve the efficiency have been around for decades, and the remaining improvement option for now is to utilize low-grade heat sources [2]. Organic Rankine cycle (ORC) system is one of the potential technology for low-grade waste heat recovery [3]. The ORC system utilizes low-temperature heat source using organic working fluids with low critical temperature. Except for the working fluids, the working principle of the ORC system is similar to the conventional steam Rankine

cycle so that the ORC system can use the mature technology of the steam Rankine cycle. When dry or isentropic refrigerants are used for the working fluids, a low degree of superheating is available for the turbine safety margin. Therefore, the ORC system has much simple structure and simple control strategy compared to the steam Rankine cycle.

The design of the ORC system depends on the waste heat itself, the cooling method, the scale including technological availability, the safety and environmental issues and others. For the detailed information about ORC cycle design see the reviews written by Bao and Zhao [4], Tchanche *et al.* [5], and Quoilin *et al.* [6]. Bao and Zhao [4] summarized available working fluids reported in the literature for potential ORC applications with low-, medium- and high-temperature waste heat sources. They also summarized appropriate expanders for different system scale. Tchanche *et al.* [5] summarized layouts for the system integration between parent and ORC systems. They covered geothermal, biomass CHP, solar thermal [6], OTEC and WHR applications [7]. Quoilin *et al.* [8] provided a techno-economic survey of current ORC systems. In their report, major ORC manufacturer and their component technologies were summarized. Components and the system are developed for target heat source. For example, Yang *et al.* [9] developed an open-drive scroll expander for heat source temperature of 100 °C using R245fa as the working fluid. They tested the system performance under different operating conditions.

For the heat exchangers, both shell-and-tube and plate type heat exchangers are used. Although the former type has a relatively large configuration, it also has a large amount of liquid holdup which can reduce the effect of a sudden fluctuation of the heat source, so it is widely used in large systems where space is not limited. In the latter case, the size is relatively small and the liquid holdup is also small so that the reaction to heat source fluctuation is fast and an additional equipment such as a gas-liquid separator is required for preventing sudden liquid entering to the turbine used in ORC with a small degree superheating. Lee *et al.* [10] reported an experimental comparison of dynamic responses of shell-and-tube and plate evaporators. They found that plate evaporator shows less stability in ORC with a small superheating degree of less than 10 °C. The shell-and-tube heat exchanger didn't show those instabilities with a same degree of superheating. Both heat exchangers can achieve similar heat exchange performance but differ in size and weight. For the mobile application where a system as small as possible is required, a plate heat exchanger should be used but the stability must be ensured.

In this study, we developed a new evaporator system using plate heat exchangers for the shell-and-tube evaporator used

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in the previous ORC systems [11], [12]. We considered two different evaporator configurations: (a) a single evaporator system and (b) an evaporator with a preheater system. The new evaporator was designed for the same heat exchange specifications and selected considering the strategy of liquid entrainment prevention. The designed evaporator system was manufactured, installed in the previous ORC system and the performance was evaluated using an electric heater.

II. 200-KW ORGANIC RANKINE CYCLE

In this study, we developed a new plate evaporator for the previous 200-kW ORC system [11], [12]. The detailed development process for the cycle and turbine were reported in the previous studies. In this section, we summarized the important points for the evaporator replacement.

The ORC system was designed for a low-temperature waste heat of 140 °C. We used a pressurized hot water as a heat source material. For this heat source, an R245fa refrigerant was selected as the working fluid due to its good thermal performance with dry expansion characteristics, safety properties, and moderate environmental effect. A simple ORC configuration was used and the system consists of a two-stage back-to-back turbine, a generator, a shell-and-tube evaporator, a plate condenser, a refrigerant tank and a multistage centrifugal pump. System operating conditions of its evaporation temperature and degree superheating were first optimized for the maximum net power output. Then those were modified considering available component technologies. We manufactured an ORC system for the designed cycle (initial design), then the thermodynamic performance was measured. The manufactured ORC system showed less net power output due to the turbine inefficiency. Reflecting the turbine performance and the additional pressure losses in the heat exchangers, we modified the ORC design condition (modified design). Table I shows the operating conditions for the initial and modified designs. Fig. 1 shows the temperature-entropy diagram for the initial and modified designs. For the modified cycle, the condenser heat rate was increased 3.8% due to the turbine inefficiency, but the evaporator heat rate was sustained near constant as 1836 kW. The averaged evaporation pressure was increased as 25 kPa due to the additional pressure losses. We designed a new evaporator system for the modified design.

III. DESIGN OF PLATE EVAPORATOR

Fig. 2 shows three different evaporator systems considered in this study. Fig. 2(a) shows a single plate evaporator configuration which is similar to the previous shell-and-tube evaporator. We did not choose this configuration because this requires an additional liquid-vapor separator for preventing liquid overflow during transient conditions. Fig. 2(b) shows an evaporator with a counter-current preheater configuration. Heat loads of the evaporator and preheater were divided as 798 kW and 940 kW which correspond to the liquid saturation point. Compared to the single evaporator configuration, this second design shows low-pressure losses and small weight for the same heat rate. This is because each plate heat exchanger

can select the optimum type of plate. Fig. 2(c) shows an evaporator with a co-current preheater configuration. For this configuration, the preheater heat rate was maximized as 880 kW. Therefore, for the preheater, the working fluid outlet temperature approaches the hot water outlet temperature very closely.

TABLE I: DESIGN CONDITIONS OF 200-KW ORC SYSTEM.

Parameters	Design point (initial)	Design point (modified)	Test results
Turbine inlet pressure (kPa)	2,090	2,090	2130
Turbine outlet pressure (kPa)	220	220	192
Working fluid mass flow rate (kg/s)	7.2	7.2	6.16
Turbine speed (RPM)	15000	15000	14950
Turbine shaft power (kW)	261.0	214.9	213.3
Generator power output (kW)	247.9	204.1	210.1
Pump power (kW)	12.1	16.3	15.0
Net power output (kW)	235.8	187.9	195.0
Evaporator heat rate (kW)	1826	1836	1662
Condenser heat rate (kW)	1577	1637	1464
Turbine isentropic efficiency	0.85	0.7	0.717
Generator efficiency	0.95	0.95	0.9842
Pump total efficiency	0.85	0.65	0.609
Thermal efficiency	0.129	0.102	0.117
Pressure loss in evaporator working fluid side (kPa)	10	60	46
Pressure loss in condenser working fluid side (kPa)	10	20	10

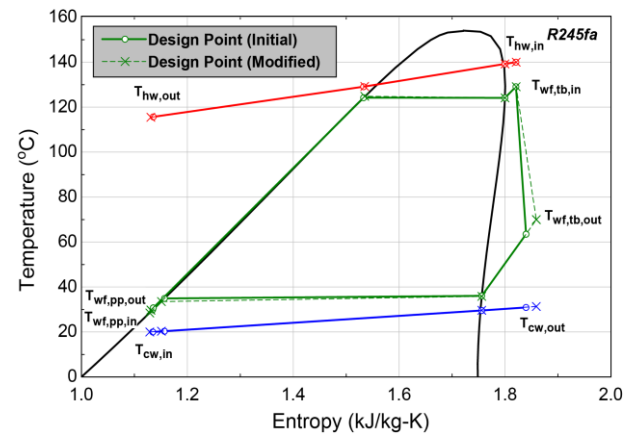


Fig. 1. Temperature-entropy diagram of the initial and modified design for 200-kW ORC system.

Fig. 3 shows all configurations considered above. As mentioned above, our ORC system has low superheating degree of 5 °C which was not enough for stable system operation with a plate type evaporator. So, we considered a separate evaporator design with the low liquid level in the evaporator which prevents sudden liquid entrainment to the turbine. Meanwhile, we considered a co-current flow configuration for the preheater. This configuration limits the evaporator inlet temperature of the working fluid below its saturation temperature about 10 °C so that no vapor is generated before it enters the evaporator. Therefore, to secure stable operation of the sequential evaporator, we adopted the

third design in Fig. 2(c) for our new evaporator.

Detail of the new evaporator system was designed using a commercial plate heat exchanger design software and the parameters are shown in Table II. For the compact design, we double the pressure losses in the hot water side as 34.4 kPa instead of the previous value of 17.5 kPa. The volume and weight were reduced by 75% and 83%. Fig. 4 shows a 3D CAD model of the ORC system adopting the new plate evaporator system.

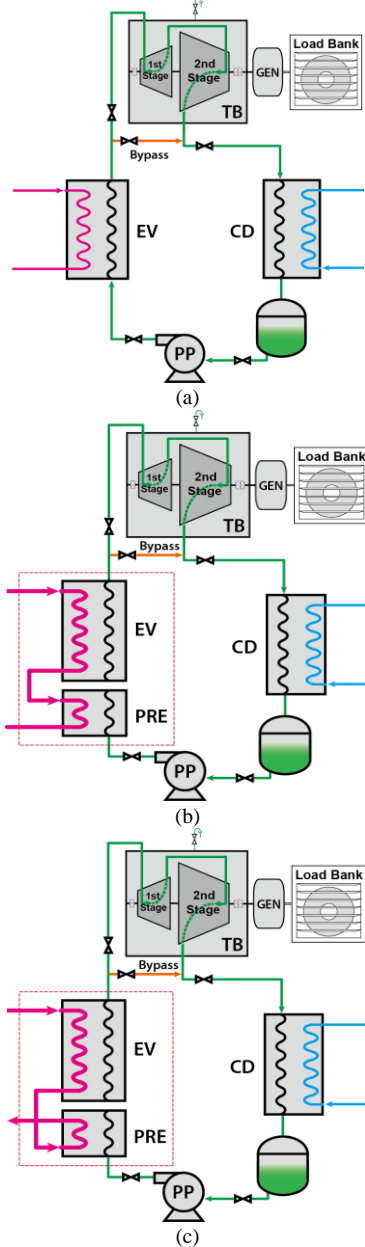


Fig. 2. ORC system with (a) a single evaporator; (b) an evaporator with counter-current preheater; (c) an evaporator with a co-current preheater.

TABLE II: PARAMETERS OF THE PLATE EVAPORATORS USED IN THE ORC-3

Parameters	Preheater	Evaporator
Plate type	B427M2	B633M
Dimensions (mm)	434 × 304 × 694	639 × 537 × 830
Heat load (kW)	880	809
Pressure losses in working fluid side (kPa)		12.1
Pressure losses in hot water side (kPa)		34.4

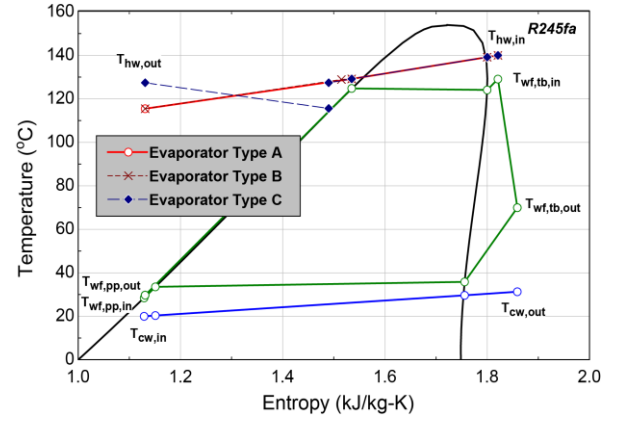


Fig. 3. Temperature-entropy diagram for the ORC systems shown in Fig. 2.

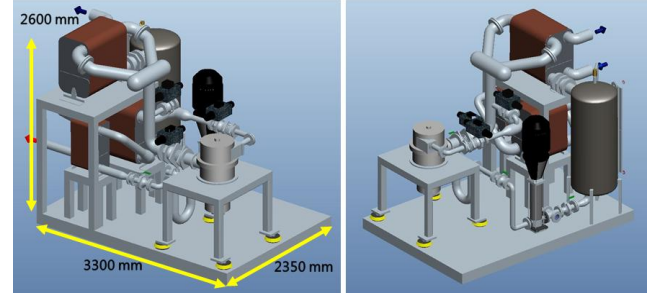


Fig. 4. 3D CAD model of the developed ORC system.

IV. RESULTS AND DISCUSSION

The performance of the ORC system with the new evaporator system was analyzed using an electric heater and cooling tower as a heat source and sink. The hot water temperature was set to 140 °C and the turbine inlet pressure was approached to 2130 kPa increasing the pump speed. The temperature-entropy diagram for this operating condition is shown in Fig. 5. This is a partial-load operating condition but we analyzed the performance at this point. This is because the turbine outlet pressure is hard to control with a cooling tower and the experiment was conducted during the winter season. The results are shown in Table I. The evaporation pressure was 40 kPa higher than the design point and the turbine outlet pressure was 28 kPa lower than the design point. Due to the lowered turbine outlet pressure (higher pressure ratio), the mass flow rate decreased from the design point value of 7.2 kg/s to 6.16 kg/s. The turbine shaft power was approached the design point value due to the higher pressure ratio and slightly higher turbine efficiency. Furthermore, the high value of generator efficiency provides the net power output of 195 kW which is higher than the value for the modified design. Therefore, we can conclude the evaporator system was successfully replaced the previous one.

Next, we analyzed the evaporator system itself. Due to its reduced mass flow rate, the evaporator showed only 91% heat rate of the design value. The pressure loss was also lower than the design value of 60 kPa as 46 kPa. Considering the general pressure loss in the pipe system, the pressure loss for the design point was calculated as 62.8 kPa which is slightly higher than the design value. However, the actual pressure loss may vary depending on the phase change interval in the evaporator. Therefore, additional experiments are needed to verify the design point performance of the new evaporator system.

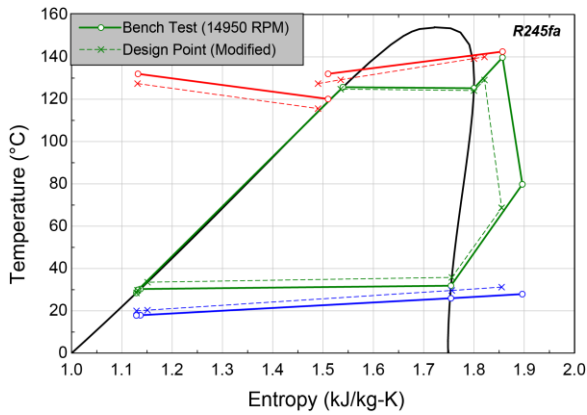


Fig. 5. Temperature-entropy diagram for the analysis point.

V. CONCLUSION

In this study, we developed a plate heat exchanger system to replace the shell-and-tube evaporator. To prevent unstable operation during a transient condition, we developed a counter-current evaporator with a co-current preheater system. The designed plate heat exchanger system was manufactured and integrated to the ORC system. The performance of the new system was evaluated using an electric heater and cooling tower. The results show our new compact evaporator system successfully replaced the previous shell-and-tube evaporator with a similar pressure loss and a great reduction in size (75% reduction in volume and 83% reduction in weight). Additional experiments are required for detailed verification of the full-load performance of the new evaporator system.

We have developed three ORC prototypes with different heat exchanger types. The first ORC system used shell-and-tube heat exchangers for both [11]. In the second ORC system [12] we replaced the condenser with plate one. The same type of evaporator was used to maintain a large holdup. And in this study, a plate evaporator was applied with safety measures. In past studies, we have successfully reduced the size of the ORC system, but in all cases, we used hot water as a heat source. Even when a hot gas heat source can be used, a heat transfer loop was introduced [12]. The heat transfer loop suppresses the fluctuation of the compressible heat source and also prevents the working fluid from being exposed to high temperatures and degradation. However, the heat transfer loop requires additional temperature differences, resulting in exergy loss. Removing the heat transfer loop and using a gas heat source can increase exergy efficiency. In addition, there are many gaseous waste heat sources in industrial sites [13]. Therefore, we will develop an ORC system that uses steam heat sources.

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