Journal of Advanced Marine Engineering and Technology, Vol. 46, No. 5, pp. 204~211, 2022
J. Advanced Marine Engineering and Technology (JAMET)
https://doi.org/10.5916/jamet.2022.46.5.204

ISSN 2234-7925 (Print)
ISSN 2765-4796 (Online)
Original Paper

A study on performance characteristics and fabrication of a plate heat exchanger condenser for marine Organic Rankine Cycle

Dae Jung Hwang[1] • Tae Woo Lim[2] • Sang Kyun Park[3] • Jae Hoon Jee[4] • Eun Shin Bang[5] • Cheol Oh†

(Received August 24, 2022 ; Revised September 27, 2022 ; Accepted October 14, 2022)

Abstract: Research on improving the energy efficiency of ships has been conducted in various fields. Accordingly, the application of an Organic Rankine Cycle (ORC) power generation system for ships using exhaust gas-waste heat generated from the main engine is considered. A heat exchanger constituting an ORC system should be designed and manufactured such that it is suitable for stable operation and it is compatible with the system. In this study, to fabricate a plate-type condenser that can be used in a marine ORC system, Computational Fluid Dynamics (CFD) and Aspen Exchanger Design and Rating (EDR) modeling of a condenser designed based on a given single heat-transfer plate shape were compared and reviewed. Consequently, a similar heat-transfer performance between CFD and EDR models was confirmed. EDR condenser modeling was applied, and a basic marine ORC system was constructed and simulated using an R245FA refrigerant as the working fluid. The simulation results confirmed that the ORC system efficiency improved as the cooling-water temperature increased, but the heat-transfer rate of the condenser decreased. The turbine output and ORC system efficiency improved as the turbine inlet pressure increased. In particular, it was confirmed that the ORC system efficiency improved significantly in the range of 10–20 bar, which is a relatively low-pressure region compared with the high-pressure region. Moreover, the ORC system efficiency and heat-transfer rate of the condenser increased as the flow rate of the working fluid of the ORC system increased. In particular, it was confirmed that the liquid fraction in the condenser decreased from a specific flow rate of the working fluid owing to the change in the heat-transfer rate distribution occupied by the sensible and latent heat loads inside the condenser.

Keywords: Organic Rankine Cycle, Condenser, Plate heat exchanger, Efficiency, Heat transfer rate

1. Introduction

The International Maritime Organization (IMO) is enacting and revising various international conventions, such as the MARPOL Convention, to reduce greenhouse gas emissions from international shipping. As the main components of such regulations, energy efficiency management and an increase in the number of ships, such as the energy efficiency design index (EEDI) for new ships and the ship energy efficiency management plan (SEEMP), occupy a large part **[1]**.

To improve the energy efficiency of a ship, research related to the development of an Organic Rankine Cycle (ORC) system for ships using waste heat from the exhaust gas generated from the ship's main engine is being conducted.

As a representative study, Moon *et al.* (2021) examined the appropriate operating pressure representing the maximum net power by simulating ORC performance using waste heat generated from ships and seawater as cooling water **[2]**. Song *et al.* (2012) confirmed a power generation output of 2,400 kW through a simulation, using various eco-friendly organic refrigerants to optimize a marine ORC power generation system with the seawater temperature difference **[3]**. Ibrahim *et al.* (2017) simulated a combined ORC system using waste heat generated

† Corresponding Author (ORCID: http://orcid.org/0000-0002-2188-7291): Professor, Division of Marine System Engineering, Korea Maritime & Ocean University, 727, Taejong-ro, Yeongdo-gu, Busan 49112, Korea, E-mail: ohcheol@kmou.ac.kr, Tel: +82-51-410-4268
1 Researcher, Department of Maritime Safety, Korea Center for International Maritime Safety Cooperation, E-mail: hdaejung@imkmc.or.kr, Tel: +82-70-4923-4533
2 Professor, Division of Marine System Engineering, Korea Maritime & Ocean University, E-mail: kyunlim@kmou.ac.kr, Tel: +82-51-410-4256
3 Professor, Division of Marine AI & Cyber Security, Korea Maritime & Ocean University, E-mail: skpark@kmou.ac.kr, Tel: +82-51-410-4579
4 Professor, Division of Maritime Police, Mokpo National Maritime University, E-mail: jhjee@mmu.ac.kr, Tel: +82-61-240-7208
5 Ph. D. Candidate, Division of Marine System Engineering, Korea Maritime & Ocean University, E-mail: esbang627@kmou.ac.kr, Tel: +82-51-410-4274

from a 1,000-kiloWatt power-generation engine of a naval ship and confirmed the effect of saving 67.2 tons of carbon dioxide and the economic effect of fuel saving **[4]**.

The above study focuses on identifying the optimal operating conditions applicable to ships in terms of the overall efficiency of the system rather than the characteristics of the individual elements constituting the ORC system. Therefore, to ensure the stable operation of a power generation system, it is necessary to study the performance analysis of individual components, such as the heat exchanger, to reflect the characteristics of the marine ORC system. Sotirios *et al.* (2011) reviewed the changes in the heat-transfer coefficient and heat exchanger efficiency by analyzing the effect of major variables of ORC systems operating in the supercritical region on the heat exchanger performance **[5]**.

This study aims is to manufacture a 110-kiloWatt brazed-plate heat exchanger=type condenser that can be applied to an ORC power system for ships. For this purpose, research and development were conducted in the following order:

First, Computational Fluid Dynamics (CFD) modeling was performed using ANSYS CFX (v.18.1), a commercial numerical analysis program based on a single heat-transfer plate for use in manufacturing plate heat exchangers. Through simulation using CFD modeling, the condenser-design requirement conditions for stable operation were confirmed. Based on the design requirements, the ORC power generation system, including the condenser to which this requirement was applied, was simulated using Aspen Exchanger Design and Rating (EDR, v.12.1), a program for designing heat exchangers, and Aspen Plus (v.12.1), a process=simulation program. Based on the simulation results, analyses of the performance characteristics of the condenser according to the major factors of the ORC power-generation system were performed.

In this study, the heat-transfer performance of CFD and EDR models of a condenser were compared and reviewed, and the heat-transfer performance characteristics of the condenser were reviewed according to the turbine inlet pressure, cooling water temperature, and flow rate of the working fluid in the OCR system.

2. Modeling

2.1 Condenser modeling

Figure 1 shows the manufacturing process of the CFD modeling of a condenser. **Figure 1(a)** shows a single heat-transfer plate used to manufacture a condenser; the fin and hole are processed to improve the heat-transfer efficiency.

Figure 1 (b) and (c) shows the process of simplifying the detailed shape. This prevents excessive mesh generation during numerical analysis calculations and increases the efficiency of calculations. Three-dimensional (3D) modeling was performed using the ANSYS Design Modeler (v.18.1) program. **Figure 1 (d) and (e)** shows the process of determining the heat-transfer performance of fins through a numerical analysis. Through this process, a CFD condenser model with a single heat-transfer plate stacked in 25 layers, as shown in **Figure 1 (f)**, was created.

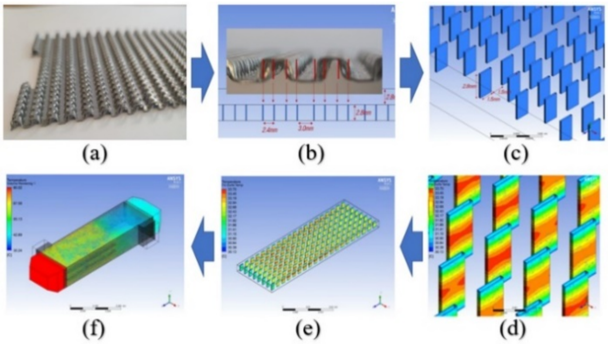


Figure 1: CFD Modeling of a condenser

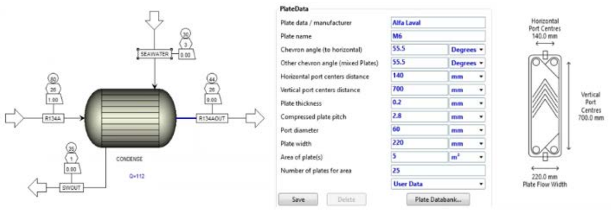


Figure 2: EDR Modeling of a condenser

Table 1: Comparative review of CFD and EDR modeling techniques

Modeling	Material	Content	Inlet	Outlet
CFD	R134a	Flow rate	0.642 kg/s	
		Pressure Drop	0.02 bar	
		Temperature	80.1℃	43.6℃
		Steam Quality	1	0
		Heat Transfer	106.1 kW	
	Water	Flow rate	4.8 kg/s	
		Temperature	30.0℃	33.1℃
EDR	R134a	Flow rate	0.642 kg/s	
		Pressure Drop	0.084 bar	
		Temperature	80.0℃	44.2℃
		Steam Quality	1	0
		Heat Transfer	112.3 kW	
	Water	Flow rate	4.8 kg/s	
		Temperature	30.0℃	34.8℃

Approximately 12,000,000 elements were used for condenser CFD modeling, as shown in **Figure 1 (f)**, and standard k-epsilon turbulence modeling was applied. To simulate the phase change occurring inside the condenser, from a gaseous state to a liquid state, a homogeneous model was applied to the material properties of the refrigerant, and the heat transfer rate between the refrigerant and water was calculated by applying the “Thin Material Heat Transfer” function found in the ANSYS CFX program. This condenser modeling uses a heat-transfer plate without a fin. Since the fins of a single heat-transfer plate are smaller than the heat transfer plate, if a plate heat exchanger-type condenser is laminated with a fin, it is difficult to perform a numerical analysis owing to excessive mesh generation. Therefore, the heat-transfer efficiency, according to the size of fins calculated through the process shown in **Figure 1 (d) and (e)** is reflected in the prototyping process, and in the modeling comparison and review process of this study, the shape of the heat-transfer plate without fins was used.

Figure 2 shows condenser EDR modeling using the Aspen Exchanger Design and Rating (EDR, v.12.1) program. The basic design method of the plate-heat exchanger provided in the Aspen EDR program is the chevron type. The shape of the plate-type heat exchanger condenser applied to the EDR model was made by stacking 25 single heat-transfer plates with a width of 220 mm and length of 700 mm. The height between the layers of a single heat-transfer plate is 2.8 mm, and the basic values of plate-heat exchangers with other specifications are applied to the heat exchanger model M6 of the Alfa Laval company provided by Aspen EDR. This somewhat differs from the curved shape of the single heat-transfer plate used in the EDR condenser modeling to be manufactured in this study and the bonding method in the manufacturing process. However, considering that similar calculation results are derived as a plate-heat exchanger through comparison and verification with the heat transfer performance of CFD modeling, reliability can be ensured in the calculation results through EDR condenser modeling. **Table 1** shows the results of the condensation heat transfer performance obtained through CFD and EDR modeling. By applying the same conditions to the thickness of a single heat plate, the number of stacked heat plates, and the horizontal and vertical sizes of the condenser, EDR modeling was produced to have an external condition similar to that of CFD modeling.

However, this is not a structure that perfectly matches the condenser to be manufactured. The working fluid was an R134a refrigerant, and water was used for cooling. The same flow rate, pressure, and temperature conditions were applied. Consequently, heat transfer performances of 112.3 kW and 106.1 kW were measured for EDR and CFD models, respectively. Additionally, similar supercooling degrees were confirmed through condenser outlet temperatures of 43.6°C and 44.2°C, for the EDR and CFD models, respectively.

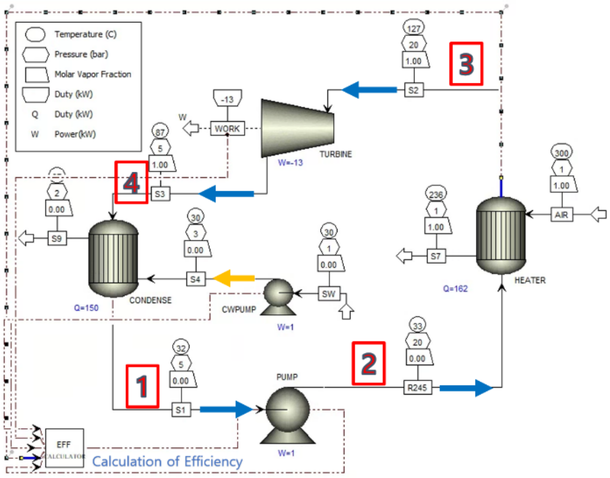


Figure 3: Modeling of the ORC system for ships.

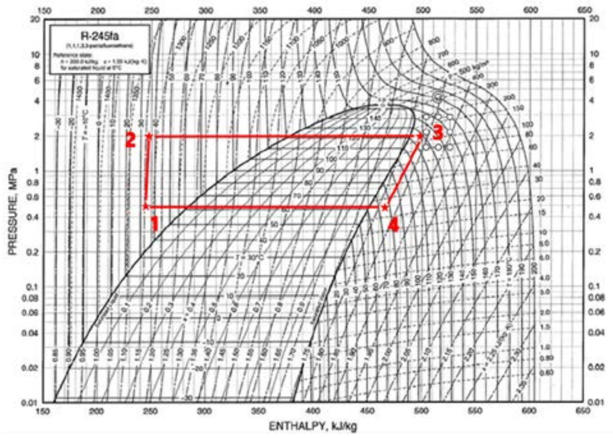


Figure 4: P-h Diagram on an ORC.

Table 2: Comparative review of CFD and EDR modeling

	1	2	3	4
Quality	0	0	1	1
T(°C)	32	33	127	87
P(kPa)	476	2,000	2,000	500
F(kg/s)	0.642	0.642	0.642	0.642
h(kJ/kg)	243.3	244.3	491.7	475.9
s(kJ/kg·K)	1.148	1.146	1.815	1.837
Heat(kW)	150 Condenser	-	162 Evaporator	-
Work(kW)	-	1 Pump	-	13 Turbine
Efficiency	6.18%			

2.2 ORC modeling

Figure 3 shows the ORC power-generation system configured using Aspen Plus (v.12.1). The EDR modeling described in Section 2.1 was applied to the condenser constituting an ORC system. The R245FA refrigerant with high chemical stability and eco-friendly performance was selected as the working fluid in several studies **[6][7]**.

The efficiencies of the working fluid and cooling water pump were 70% and 75 %, respectively. The superheating degree of the evaporator was maintained at 5 ℃, and an exhaust gas flow rate of 234.12㎥/min, from the main engine of the Korea Maritime and Ocean University training ship HANBADA, was applied for the exhaust gas used as a heat source **[8]**. The temperature of the cooling water in the condenser was set to 30 °C, considering the operating conditions of the vessel. The efficiency of the ORC system, calculated according to the above conditions, was 6.18%, and the power generation output of the turbine was 13 kW.

$$Q_{Heater} = \ h_3 - h_2 \tag{1}$$

$$Q_{Condenser} = \ h_4 - h_1 \tag{2}$$

$$Power_{Pump} = \ h_2 - h_1 \tag{3}$$

$$Power_{Turbine} = \ h_3 - h_4 \tag{4}$$

$$Work_{net} = Power_{Turbine} - Power_{Pump} \tag{5}$$

$$\eta_{th} = \frac{W_{net}}{Q_{Heater}} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)} \tag{6}$$

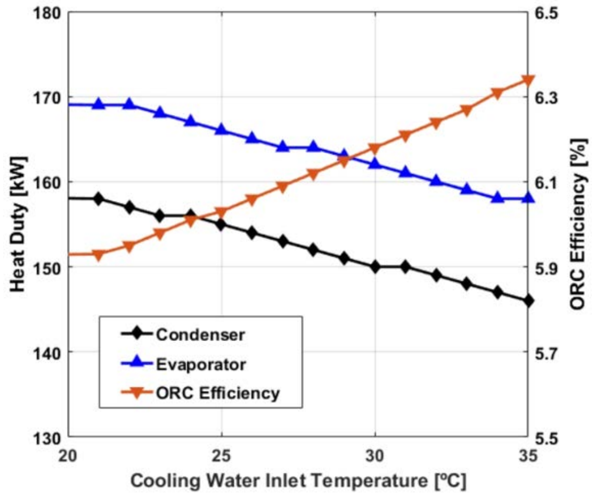
The efficiency of the ORC system was defined according to **Equation (1)**, and the power of the pump used was calculated by adding the power of the pump for the working fluid and that of the cooling water.

Figure 4 illustrates the Moliere diagram showing the operation process of the ORC system. The amount of heat transfer rate between the evaporator and condenser calculated by the difference in enthalpy is 162 kW and 150 kW, respectively, and the specific temperature and pressure are listed in **Table 2**.

3. Performance characteristics of the ORC

3.1 Effect of the temperature of cooling water

Considering that the ORC system was developed for ships, the cooling-water temperature is an important variable. As climatic conditions, such as seawater temperature, change according to the environment of the sea area in which the ship operates, the temperature of the cooling water of the ship using seawater also changes.



(a) Effect of cooling water temperature

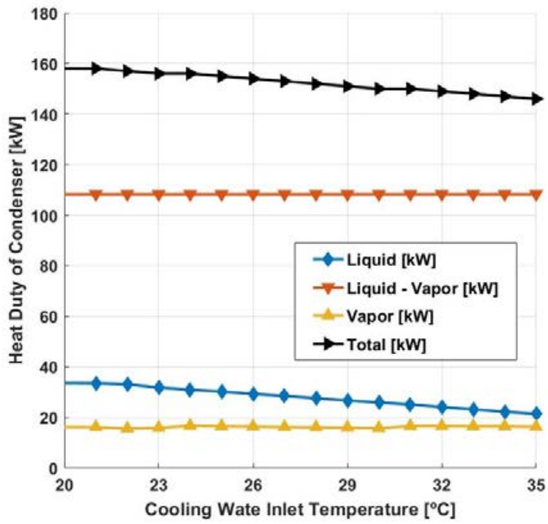


Figure 5: (b) Effect of cooling water temperature.

Figure 5 (a) shows the heat transfer rates of the evaporator and condenser, and the efficiency of the ORC system according to the increase/decrease in the cooling water temperature range of 20-35 °C. The heat transfer performance of the condenser was reduced as the cooling water temperature increased, showing a heat transfer rate of approximately 159 kW at a cooling water temperature of 20 °C and approximately 146 kW at a temperature of 35 °C. This is because, as the cooling water temperature increases, the average temperature difference with R245FA, which is the working fluid, decreases. As the cooling water temperature increases, the heat-transfer performance of the evaporator decreases accordingly. This phenomenon occurs when the liquid refrigerant, which is not sufficiently cooled in the condenser, flows into the evaporator, through the pump, as the temperature of the cooling water increases and then exchanges heat with the exhaust gas.

The efficiency of the ORC system showed a value of approximately 5.91 % at a cooling water temperature of 20 °C and approximately 6.34% at a temperature of 35 °C, and the system efficiency increased linearly as the cooling water temperature increased. The reason for this result is that the calculation condition was fixed at 5 °C, which was the superheat applied to the heater design.

Unlike the condenser, the heater does not reflect design factors such as the internal shape and size of the heat exchanger but only provides the 5 °C of the superheat condition of the heater outlet. Therefore, regardless of the properties of the refrigerant discharged from the condenser outlet, the outlet temperature of the heater was always constant at 127 °C, which was 5 °C overheated than the saturated R245fa temperature at a 20-bar operating pressure. Accordingly, as the temperature of the cooling water increases, the heat transfer performance of the heater decreases and the turbine power does not fluctuate; therefore, the efficiency of the ORC decreases according **to Equation (1)**.

Figure 5 (b) shows the heat transfer rate occupied by the latent and sensible heat in the condenser according to the increase/decrease in the cooling water temperature range of 20-35°C. The total heat transfer rate of the condenser decreased, there was no change in the transfer rate of latent heat, and the transfer rate of sensible heat by the liquid refrigerant decreased as the cooling water temperature increased. This is a phenomenon in which temperature change does not occur in the latent heat section, where gas and liquid coexist according to the vapor–liquid equilibrium, but in the sensible heat section.

From the results shown in **Figure 5 (b)**, it is possible to infer the stable operating state of the condenser. The condenser is accompanied by a phase change as the refrigerant in the gaseous state is discharged to a liquid state through the cooling process. If the gas refrigerant is discharged to the outlet owing to an insufficient cooling process, the pump installed at the end of the condenser has adverse effects such as reduced efficiency **[9]**. Therefore, it can be concluded that the condenser operates stably only when an appropriate amount of liquid is constantly present during operation. The condenser exhibited a heat transfer rate of 150 kW at a cooling water temperature of 30 °C, and the heat transfer rate by the liquid refrigerant was approximately 17% at 26 kW. According to the magnitude of the heat transfer rate of the liquid refrigerant, it can be inferred that approximately 17% of the liquid was present in the condenser.

3.2 Effect of the turbine inlet pressure

Figure 6 shows the power of the turbine, heat transfer rate of the condenser, and efficiency of the ORC system obtained by increasing and decreasing the turbine inlet pressure condition in the range of 10-32 bar. The power of the turbine was approximately 6 kW at an inlet pressure of 10 bar and approximately 16 kW at an inlet pressure of 32 bar. Accordingly, the power of the turbine increases as the inlet pressure increases. This is because, as the turbine inlet pressure increases, the work done by the turbine increases, showing that the total efficiency also increases.

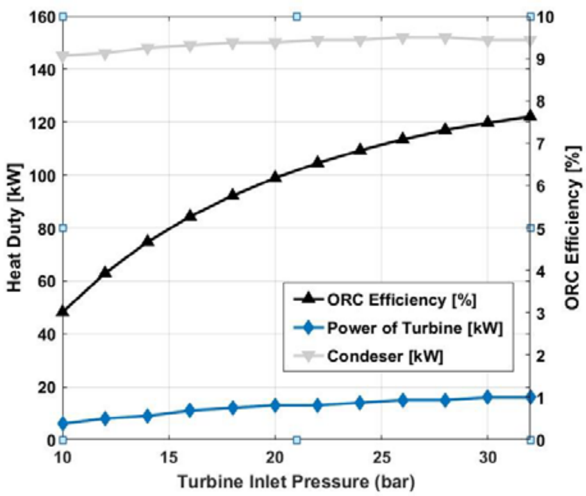
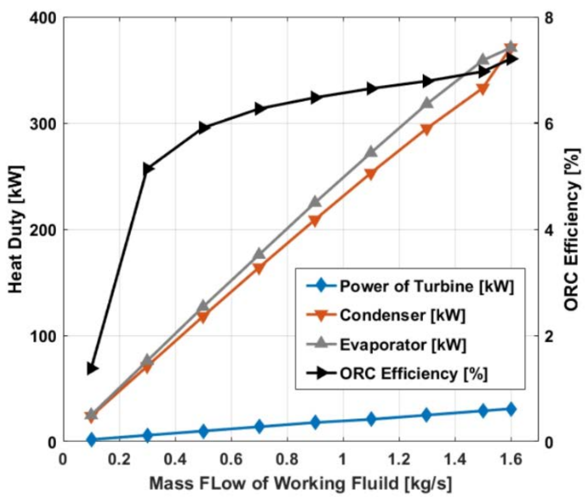


Figure 6: Effect of the turbine inlet pressure



(a) Effect of mass flow

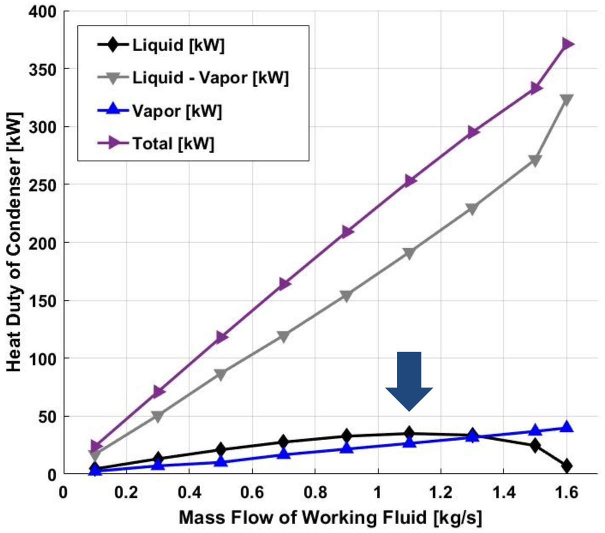


Figure 7: (b) Effect of mass flow

An increase or decrease in the turbine inlet pressure depends on the performance of the pump, and an increase in the electric power required by the pump to increase the pressure is inevitable. When the amount of electric power required for the pump increases, the overall efficiency of the ORC system is predicted to decrease according to **Equations (1)** and **(2)**; however, the overall efficiency is improved as the turbine produces more power than the amount required for the pump.

However, it does not increase linearly, and the rate of increase in the ORC efficiency gradually decreases with the increasing inlet pressure of the turbine. When the inlet pressure of the turbine was increased from 10 to 16 bar in the relatively low-pressure region, the ORC system efficiency increased by approximately 75% from 3.01% to 5.27%. However, when the pressure increased from 26 to 32 bar, the system efficiency increased from 7.09% to 7.63%, a slight increase of approximately 8%.

In addition to the turbine output and efficiency, an increase in the pressure of the working fluid has various effects on the heat-transfer performance and turbulent flow in the heat exchanger. Therefore, it is necessary to select an appropriate operating pressure considering the type of pump and physical properties of the working fluid **[10]**.

3.3 Effect of working fluid flow rate

Figure 7 (a) shows the power of the turbine, the heat transfer rate of the condenser, the heat transfer rate of the evaporator, and the efficiency of the ORC system against the change in mass flow rate in the R245FA flow condition, which is the working fluid, in the range of 0.1-1.6 kg/s.

A notable feature is the ratio of the increase in ORC system efficiency with increasing mass flow. In the case of the 0.1 kg/s flow rate of the working fluid, the efficiency of the ORC system is 1.38%; however, a flow rate change of only 0.2 kg/s, to 0.3 kg/s, increases the efficiency to 5.14%, which is an increase of approximately 3.7 times. The efficiency of the system is calculated using **Equation (1)**, and the power of the turbine and amount of heat absorbed by the evaporator are the main factors affecting the efficiency of the system. As shown in **Figure 7 (a)**, at a low flow rate of 0.1 kg/s, the efficiency is low because the 2-kiloWatt power of the turbine is small, compared to the amount of heat absorbed by the evaporator, and the efficiency increases as the flow rate increases, but the rate of increase is gradually reduced.

An increase in the operating flow rate affects the heat transfer rate and size of the plant, including the heat exchanger. Therefore, it is necessary to select the operating flow rate by considering not only the system efficiency, but also the location and economic feasibility of the unit installed on the ship.

Figure 7 (b) shows the heat transfer rate occupied by latent and sensible heat in the condenser by increasing and decreasing the flow condition of the R245FA refrigerant, which is the working fluid, in the range of 0.1-1.6 kg/s. As the flow rate increased, the heat-transfer rate of the condenser increased linearly. However, the heat transfer rate occupied by the refrigerant in the liquid state inside the condenser increases in the refrigerant flow condition in the range of 0.1-1.1 kg/s, but it decreases in the range of 1.1-1.6 kg/s. Although not shown in Figure 7 (b), when a working fluid flow rate of 1.7 kg/s was applied, the steam quality of the condenser outlet was 0.1, and the phenomenon of the gaseous refrigerant exiting the condenser outlet was confirmed by calculation.

As the flow rate of the working fluid increased, the superheated refrigerant flowed into the condenser through the evaporator and turbine. In the OCR system, because the evaporator is given a fixed superheat operating condition, it is discharged in a superheated state of 5°C, regardless of the flow rate increase. However, because the condenser is modeled by applying the shape and size of the heat exchanger to be manufactured in the future, according to the fundamental purpose of this study, the degree of supercooling at the condenser outlet varies according to the flow rate. The excessively superheated refrigerant may not be completely cooled in the condenser as the flow increases in the working fluid, and it may be discharged in a gaseous state from the condenser outlet.

As mentioned in Section 3.1, this phenomenon may adversely affect pump performance; therefore, an appropriate operating flow rate of R24fFA should be selected. From **Figure 7 (b)**, it is determined that the appropriate flow rate of the working fluid under the conditions of this calculation is 1.1 kg/s, which is the starting point at which the heat transfer rate of the liquid refrigerant decreases.

4. Conclusion

In this study, CFD and EDR modeling of a condenser constituting the marine ORC system were performed, and the heat transfer performances were compared. A marine ORC system using R245FA as the working fluid was designed, and the condenser performance characteristics were reviewed through the process simulation program, Aspen Plus (v.12.1). The following conclusions are drawn within the scope of the analysis and calculation conditions.

(1) When the superheat condition of the heater was applied at 5 °C, the ORC thermal efficiency increased as the cooling water temperature of the condenser increased.

(2) The turbine output and system efficiency improved as the inlet pressure of the turbine increased. In particular, the system efficiency improved significantly in the pressure range of 10–20 bar, which is a relatively low-pressure region, compared to the high-pressure range.

(3) As the flow rate of the working fluid increased, the ORC system efficiency and the heat transfer rate of the heat exchanger increased. Particularly, a decrease in the liquid fraction in the condenser was confirmed under a specific flow rate of 1.1 kg/s through the analysis of the heat transfer-rate distribution occupied by sensible and latent heat inside the condenser. Therefore, an appropriate working-fluid flow rate should be selected for stable condenser operation.

Acknowledgement

This research was conducted with the support of Korean Innovation Foundation funded by the R&D(2021-BS-RD-0115-201) in 2021.

Author Contributions

Conceptualization, D. J. Hwang; Methodology, D. J. Hwang and O. Cheol; Software, D. J. Hwang; Formal Analysis, T. W. Lim; Investigation, S. K. Park; Resources, J. H. Jee; Data Curation D. J. Hwang; Writing-Original Draft Preparation, D. J. Hwang; Writing-Review & Editing, E. S Bang; Visualization, E. S Bang; Supervision, D. J. Hwang and C. Oh; Project Administration, C. Oh; Funding Acquisition, C. Oh.

References

[1] E. -J. Kim, “A study on the improvement plans of the IMO's regulation for the greenhouse gas emissions from international shipping,” Journal of Regional Industry Review, vol. 41, no. 1, pp. 237-260, 2018 (in Korean).

[2] J. -H. Moon, H. -S. Lee, and J. -B Seo, “Performance analysis of ORC cycle using waste heat,” Journal of Power System Engineering, vol. 25, no. 5, pp. 43-50, 2021 (in Korean).

[3] C. Oh and Y. -U. Lee, “A optimization of the ORC for ship's power generation system,” Journal of the Korean Society of Marine Engineering, vol. 36, no. 5, pp. 595-602, 2012 (in Korean).

[4] I. Girgin and C. Ezgi, “Design and thermodynamic and thermoeconomic analysis of an Organic Rankine Cycle for naval surface ship applications,” Energy Conversion and Management, vol. 148, pp. 623-634, 2017.

[5] S. Karellas, A. Schuster, and A. -D. Leontaritis, “Influence of supercritical ORC parameters on plate heat exchanger design,” Applied Thermal Engineering, vol. 33-34, pp. 70-76, 2012.

[6] J. -S. Kim, D. -Y. Kim, Y. -T. Kim, and H. -K. Kang, “Performance analysis of an Organic Rankine Cycle for ocean thermal energy conversion system according to the working fluid and the cycle,” Journal of the Korean Society of Marine Engineering, vol. 39, no. 9, pp. 881-889, 2015.

[7] S. -Y. Cho and C. -H. Cho, “Effect on the cycle by the properties of working fluids using Organic Rankine Cycle,” The KSFM Journal of Fluid Machinery, vol. 18, no. 4, pp. 5-12, 2015.

[8] J. -S. Choi, S. -D. Lee, S. -Y. Kim, K. -W. Lee, K. -W. Chun, Y. -W. Nam, K. -S. Jung, S. -K. Park, and J. -H. Choi, “A study on the characteristics measurement of main engine exhaust emission in training ship HANBADA,” Journal of the Korean Society of Marine Environment & Safety, vol. 19, no. 6, pp. 658-665, 2013 (in Korean).

[9] J. -O. Mo, Y. -T. Kim, and Y. -H. Lee, “Numerical study on cavitation performance evaluation in a centrifugal pump impeller,” Journal of the Korean Society of Marine Engineering, vol. 36, no. 2, pp. 286-293, 2012 (in Korean).

[10] M. -B. Kim and C. -Y. Park, “Heat transfer and pressure drop characteristics of brazed plate heat exchangers with single-phase flow,” Journal of Air-Conditioning and Refrigeration Engineering, vol. 26, no. 11, pp. 522-528, 2014 (in Korean).