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Thermodynamic and Thermo-economic Analysis of Integrated Organic Rankine Cycle for Waste Heat Recovery from Vapor Compression Refrigeration Cycle

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Abstract

In the present study, an integrated air-conditioning-organic Rankine cycle (i-AC-ORC) system which combines a vapour compression cycle and an organic Rankine cycle is proposed. An organic Rankine cycle system is applied to recover the waste heat rejected by the condenser of air-conditioning system. The selection of optimal fluid pairfor the air-conditioning subsystem and organic Rankine cycle subsystem is investigated. Based on thermodynamic (energy and exergy) and thermo-economic analysis, R600a-R123 is chosen as the fluid pair for this integrated air-conditioning-organic Rankine cycle system. The thermodynamic model has been programmed using Engineering Equation Solver (EES). The combined coefficient of performance (COP) of integrated system can be improved from 3.10 to 3.54 compared with that of the standalone air-conditioning subsystem. The organic Rankine cycle subsystem can yield1.41 kW of net electricity with a thermal efficiency of 3.05%. The organic Rankine cycle subsystem operates with an exergy efficiency of 39.30%. In addition, energetic and exergetic performances of the integrated system are studied with variable external conditions.

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Keywords:Air conditioning;Vapor compression cycle; Organic Rankine cycle;Waste heat recovery;Working fluid;Thermodynamic and thermo-economic analysis

1. Introduction

Global warming associated with burning fossil fuels is continuously driving the growth of electricity consumption in air-conditioning (AC) domains. According to Li et al.[1], AC applications surges 30-50% of the total electricity consumption in urban areas in summer. Consequently, AC applications have accelerated the deterioration of urban micro-climate due to the rejected waste heat. Therefore, itis of great importance to promote waste heat recovery (WHR) through AC applications to enhance micro-environmental protection and energy saving.

Many research works have been done on waste heat recovery using heat sources like exhausted gas from power plants, solar power and geothermal energy[2]–[7]. However, for recovery of low-grade waste heat through AC systems, little work has been done.Heating watermay be an applied method which realizes the production of hot water by utilizing rejected waste heat in the AC system[8]. In recent research, organic Rankine cycle (ORC) system has been proved to be effective to utilize and recover the waste heat for electricity generation. An idea of combined AC-ORC system has been proposed and presented in [9]. As stated by many studies, the selection of working fluids iscrucial to optimize the performance of the ORC system[10]–[13]. Wang et al. [14] investigated and found that

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isentropic and dry fluids like R245fa, R141b and butane show attractive performance for low-grade waste heat recovery.In these studies, the energetic and exergetic analyses were considered as effective approaches and key parametric indicators [15]–[17].

2. System configuration and fluids selection

2.1. System configuration

As shown in Fig.1, the proposed i-AC-ORC system is a combination of a vapor compression cycle (VCC) on the left side and an ORC on the right side. It can be seen that the sharing heat exchanger (SHX) between the VCC and ORC works as the condenser inthe VCC as well as the evaporator in the ORC. Through applying i-AC-ORC system, waste heat rejected by the VCC can be converted into electricity by the ORC subsystem.

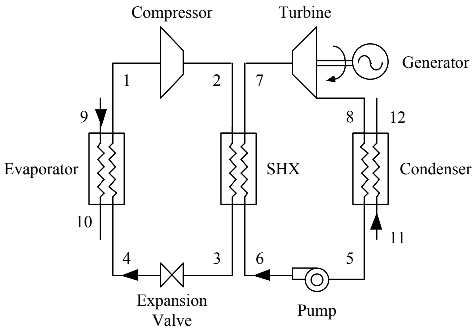

Fig.1. Schematic of i-AC-ORC waste heat recovery system.

2.2. Working fluids selection

Six refrigerants are selected as the potential working fuildsfor the ACsubsystem, which are R134a, R290, R404A, R407C, R600a andR410A. As mentionedabove, with growing interest in waste heat recovery, Bao et al.[18]presented the categoriesof the working fluids based on different heat source temperatures. Hence, six isentropic and dry fluids (butane, R123, R141b, R227ea, R245fa and R1233zd(e)) are chosen as the candidates for the ORC subsystem.

3. Modelling

Modelling equations on energetic, exergetic and thermo-economic analyses are developed and programmed in Engineering Equation Solver (EES). The computational modelling is based on the following general assumptions [19]. The system is considered under steady state. Friction, heat loss, changes inkinetic and potential energy are neglected.The pressure drops in heat exchangers and tubes are negligible. The working fluid in the VCC enters the compressor at saturated vapor state and exits the condenser at saturated liquid state. Expansion process in the VCC is adiabatic process. The designed condensation temperature of the AC subsystem is not affected by the ORC subsystem. The working fluid in the ORC exits the condenser at saturated liquid state and enters the turbine at saturated vapor state. Detailed design criteria are presented in Table 1.

3.1.Energy Analysis

Based on the aforementioned assumptions, the cooling capacity (Q_{evap}), the compressor work (W_{comp}), the rejected waste heat rate (Q_{waste}) and the initial COP of the AC subsystem (COP_{ini}) can be expressed as follows:

$$Q_{\mathrm{evap}} = \dot{m}_{\mathrm{AC}}(h_1 - h_4) \tag{1}$$

$$W_{\mathrm{comp}} = \dot{m}_{\mathrm{AC}}(h_2 - h_1) \tag{2}$$

$$Q_{\mathrm{waste}} = \dot{m}_{\mathrm{AC}}(h_2 - h_3) \tag{3}$$

$$COP_{\mathrm{ini}} = \frac{Q_{\mathrm{evap}}}{W_{\mathrm{comp}}} \tag{4}$$

where $\dot{m}_{\mathrm{AC}}$ is the mass flow rate of the refrigeration in the AC subsystem.

Table 1.Main proposed parameters for the i-AC-ORC system

Item	Value	Item	Value
Return chilled water temperatuer	12.0°C	Return chilled water pressure	0.101 MPa
Cooling water temperature	30.0°C	Inlet cooling water pressure	0.101 MPa
Designed AC condensation temperature	50.0°C	Compressor efficiency	70%
ORC condensation temperature	35.0°C	Pump efficiency	80%
Pinch point temperature difference in AC evaporator	5.0 °C	Turbine efficiency	80%
Pinch point temperature difference in the SHX	1.5 °C	Generator efficiency	95%
Pinch point temperature difference in the condenser	2.4 °C	Mass flow rate ofchilled water	1.7 kg/s
Dead state temperature	25.0°C	Cooling capacity of the AC subsystem	35.0kW

After applying the full condensing method, theentire waste heat in the SHX can be recovered by the ORC subsystem. Hence, in the ORC subsystem, the recovered waste heat rate (Q_{reco}) is in equilibrium withrejected waste heat rate (Q_{waste}). Meanwhile, the turbine power output (W_{turb}), the pump work (W_{pump}) are expressed as follows:

$$Q_{reco} = \dot{m}_{ORC}(h_7 - h_6) \tag{5}$$

$$W_{turb} = \dot{m}_{ORC}(h_7 - h_8) \tag{6}$$

$$W_{pump} = \dot{m}_{ORC}(h_6 - h_5) \tag{7}$$

Therefore, the net electricity output (E_{net}) and thermal efficiency of the ORC subsystem ($\eta_{thermal}$) can be written as:

$$E_{net} = W_{turb}\eta_g - W_{pump} \tag{8}$$

$$\eta_{thermal} = E_{net} / Q_{reco} \cdot 100\% \tag{9}$$

where η_g is the efficiency of the generator.

By applying waste heat recovery, the combined COP of the i-AC-ORC system (COP_{comb}) can be defined as:

$$COP_{comb} = \frac{Q_{evap}}{W_{comp} - E_{net}} \tag{10}$$

3.2.Exergy analysis

The exergy efficiency ($\eta_{ex,ORC}$) of the ORC subsystem can be given by:

$$\eta_{ex,ORC} = \frac{E_{net}}{W_{waste}} = \frac{E_{net}}{\dot{m}_{AC}[h_2 - h_3 - T_0(s_2 - s_3)]} \cdot 100\% \tag{11}$$

where W_{waste} is the exergy rate of the recovered waste heat.

3.3. Heat exchanger modeling

Counter-flow heat exchangers are selected and modelled by the Logarithmic Mean Temperature Difference (LMTD) method. For each heat exchanger, it is divided into certain zones based on different fluid phases, which are single phase flow, boiling and condensation [19]. The heat transfer coefficient U_i and heat exchange area A_i of each zone are calculated. The heat transfer coefficient U_i is obtained by considering two convective heat transfer resistances of the fluid onboth sides of the heat exchanger.

$$\frac{1}{U_i} = \frac{1}{a_{i,hf}} + \frac{1}{a_{i,cf}} \tag{12}$$

The heat transfer coefficient of single phase flow is estimated by non-dimensional relationship, which is developed by Wanniarachchi and Chisholm[20].

Hsieh and Lin correlation is employed for the boiling process [21].Han, Lee, and Kim correlation is used to study the heat transfer coefficient of the condensation process [22]. For each zone in single heat exchanger, the U_i value can be obtained. Further, $A_i$can be calculated. Total heat exchange area $A$is the sum of $A_i$of each zone.

3.4. Economic analysis

Correlations are given in Table 2 to estimate the cost of the components in the i-AC-ORC system. And, the economic performance of the whole system is evaluated by the specific investment cost (SIC) [23], which is given as:

$$\mathrm{SIC} = \frac{Cost_{\mathrm{total}}}{E_{\mathrm{net}}} \tag{13}$$

Table 2. Cost of the components [23]

Component	Factor	Cost (€)
Pump	Required pump work, kW	$900 \cdot (\dot{W}_{\mathrm{pump}} / 0.3)^{0.25}$
Heat exchangers	Heat exchanger area, m^2	$190 + 310 \cdot A$
Turbine	Volumetricflow rate, m^3/s	$1.5 \cdot (225 + 170 \cdot \dot{V}_{\mathrm{turb}})$
Compressor	Volumetric flow rate, m^3/s	$225 + 170 \cdot \dot{V}_{compressor}$
Hardware	/	500
Refrigerant	Mass, kg	$20 \cdot M$

4. Discussion and results

The energetic, exergetic and thermo-economic performances with different fluid pairs of the i-AC-ORC system are presented in Table 3. The comparison and selection of the working fluid pairs are conducted when the condensation temperature of the AC sub-system is maintained at 50°C. R600a-R123 is selected to be the optimal fluid pair when taking system performance in the first place. Without installing the ORC subsystem, the COP of the standalone AC subsystem is 3.10.

By recovering waste heat from AC system and utilizing it in ORC subsystem, electricity (1.41 kW) can be generated, which partially compensates the electricity required for the operation of the AC subsystem with a thermal efficiency of 3.05%. Consequently, the new COP can be increased to 3.54. Overall, the exergy efficiency of the ORC subsystem is 39.30%. However, in terms of the economic factors, R407c-Butane is more attractive for its lowest investment cost in the whole system.

Table 3. Performances of different fluid pairs

VCC Fluids	ORC Fluids	COP_{ini}	COP_{comb}	$\dot{E}_{\mathrm{net}}$ (kW)	Thermal Efficiency(%)	SIC[1](€/W)[1]	Payback Period[2](y=365d*24h)	ORC exergy efficiency (%)
R134a	Butane	3.01	3.42	1.40	3.00	12.68	9.96	37.58
	R123	3.01	3.43	1.42	3.04	17.19	13.49	38.13
	R141b	3.01	3.43	1.42	3.04	18.34	14.40	38.12
	R227ea	3.01	3.40	1.34	2.87	11.35	8.91	36.00
	R245fa	3.01	3.43	1.40	3.01	15.85	12.44	37.71
	R1233zd(e)	3.01	3.43	1.41	3.02	16.32	12.81	37.80
R290	Butane	2.93	3.32	1.40	3.00	12.03	9.44	37.49
	R123	2.93	3.33	1.43	3.03	16.42	12.88	38.04
	R141b	2.93	3.33	1.43	3.03	17.54	13.77	38.02
	R227ea	2.93	3.31	1.35	2.86	10.75	8.43	35.92
	R245fa	2.93	3.33	1.41	3.00	15.11	11.86	37.61
	R1233zd(e)	2.93	3.33	1.41	3.01	15.57	12.22	37.71
R404A	Butane	2.56	2.86	1.47	3.01	10.85	8.51	37.12
	R123	2.56	2.87	1.49	3.06	14.25	11.19	37.67

[1] 1€=1.074USD
[2] Pay back period is calculated in terms of Hong Kong's electricity charge.

	R141b	2.56	2.87	1.49	3.06	15.11	11.86	37.66
	R227ea	2.56	2.85	1.40	2.89	9.92	7.78	35.56
	R245fa	2.56	2.87	1.47	3.02	13.24	10.39	37.25
	R1233zd(e)	2.56	2.87	1.48	3.03	13.60	10.68	37.34
R407C	Butane	3.07	3.43	1.18	2.54	8.09	6.35	30.45
	R123	3.07	3.43	1.20	2.58	9.22	7.24	30.85
	R141b	3.07	3.43	1.19	2.58	9.40	7.38	30.82
	R227ea	3.07	3.41	1.14	2.45	8.23	6.46	29.33
	R245fa	3.07	3.43	1.18	2.55	8.97	7.04	30.54
	R1233zd(e)	3.07	3.43	1.19	2.56	9.05	7.10	30.61
R600a	Butane	3.10	3.53	1.39	3.01	13.33	10.46	38.73
	R123	3.10	3.54	1.41	3.05	18.53	14.54	39.30
	R141b	3.10	3.54	1.41	3.05	19.86	15.59	39.29
	R227ea	3.10	3.51	1.33	2.88	11.78	9.25	37.10
	R245fa	3.10	3.53	1.40	3.02	16.99	13.34	38.86
	R1233zd(e)	3.10	3.53	1.40	3.03	17.53	13.76	38.96
R410A	Butane	2.72	3.07	1.44	3.01	9.95	7.81	34.27
	R123	2.72	3.07	1.46	3.06	12.92	10.14	34.77
	R141b	2.72	3.07	1.46	3.06	13.69	10.74	34.76
	R227ea	2.72	3.05	1.38	2.89	9.15	7.18	32.83
	R245fa	2.72	3.07	1.45	3.02	12.02	9.43	34.38
	R1233zd(e)	2.72	3.07	1.45	3.03	12.35	9.69	34.47

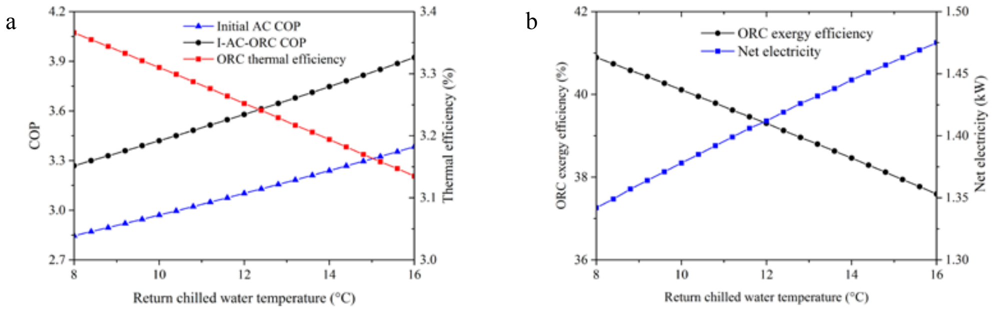

Fig. 2. (a)Variation of initial AC COP,i-AC-ORC COP and thermal efficiency with return chilled water temperature; (b) Variation of ORC exergy efficiency and net electricity of the i-AC-ORCsystem with return chilled water temperature.

In general, the proposed R600a-R123 i-AC-ORC system operates with steady volumetric flow rate at 0.033m^3/s. However, the indoor temperature varies with changing thermal load. Hence, evaluation on the system performance is further conducted with changing return chilled water temperature. In order to take a brief look into the system performance, overall UA values of the heat exchangers are considered to be temperature-independent. The effect on the energetic performance of the i-AC-ORC system with changing return chilled water temperature is shown in Fig.2. (a). It is notable that both initial COP of the AC subsystem and combined COP of the i-AC-ORC system increase with higher return chilled water temperature. However, considering the thermal efficiency, the performance degrades. Due to larger recovered waste heat input, the net electricity produced by the ORC subsystem rises as shown in Fig.2. (b) while the exergy efficiency of the ORC subsystem decreases.

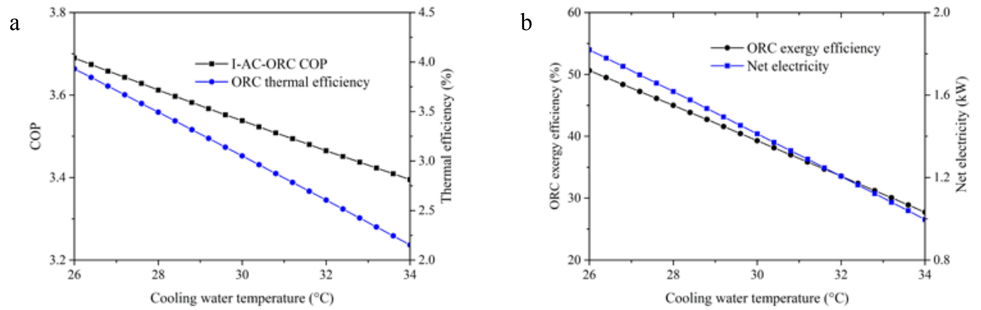

Fig. 3. (a) Variation of COP and thermal efficiency of the i-AC-ORC system with cooling water temperature; (b) Variation of theORC exergy efficiency and net electricity of the i-AC-ORC system with cooling water temperature.

Due to fluctuant outdoor conditions, the cooling water temperature varies. Therefore, energetic and exergetic analysis is conducted to evaluate the performance of the R600a-R123 i-AC-ORC system. In general, the changing operation conditions of the ORC subsystem may somehow affect the performance of the AC subsystem. Nevertheless, the condensation temperature of the AC subsystem is designed at 50°C. By using high efficiency heat exchanger and control strategy of the outlet pump pressure, the impact on the condensation temperature of the ACsubsystem can be neglected. Hence, the performanceofthe AC subsystem and the evaporation temperature of the ORC subsystem are regarded as unchanged with variable cooling water temperature. As can be seen in Fig.3. (b), the net electricity production drops sharply.Based on the predefined Eq. (9) and Eq. (10),significant drops in thermal efficiency and combined COP of the i-AC-ORC system can be observed in Fig.3. (a) with increasing cooling water temperaturedue to the decreasing electricity production.And, the falling net electricity leads to the degradation of the overall ORC exergy efficiency as shown in Fig.3. (b).

5. Conclusions

In this study, an i-AC-ORC system has been analyzed and proposed for waste heat recovery. In total, 36 working fluid combinations are formed and the system performances with these combinations are analyzed and compared. Based on the thermodynamic (energetic and exergetic) and economic analysis, the following conclusions can be drawn:

1 For low-grade waste heat recovery, the working fluids in the ORC subsystem make little difference when applying the same AC refrigerant. In terms of system performance, the optimal working fluid for the ORC subsystem is R123.
2 The performance of the i-AC-ORC system greatly depends on the initial performance of the AC subsystem. An AC system with better performance leads to better overall performance when combined with the ORC subsystem. R600a is selected as the working fluid for the AC subsystem when the operation is under restrict safety control due to its flammability.
3 Considering investment cost, R407c-Butane can be taken as the secondary fluid pair choice for its competitive performance.
4 When operated with variable return chilled water temperature, the COPs and the net electricity output rise with increasing return chilled water temperature. While, the thermal efficiency and exergy efficiency of the ORC subsystem drop contrarily.
5 Based on the assumptions made on the i-AC-ORC system, the combined COP, the thermal efficiency, the net electricity output and exergy efficiency decrease with higher inlet cooling water temperature.

References

[1] C. Li, J. Zhou, Y. Cao, J. Zhong, Y. Liu, C. Kang, and Y. Tan, "Interaction between urban microclimate and electric air-conditioning energy consumption during high temperature season," *Appl. Energy*, vol. 117, pp. 149–156, 2014.

[2] A. M. Delgado-Torres and L. García-Rodríguez, "Analysis and optimization of the low-temperature solar organic Rankine cycle (ORC)," *Energy Convers. Manag.*, vol. 51, no. 12, pp. 2846–2856, 2010.

[3] U. Drescher and D. Brüggemann, "Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants," *Appl. Therm. Eng.*, vol. 27, no. 1, pp. 223–228, 2007.

[4] Z. Shengjun, W. Huaixin, G. Tao, "Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation," *Appl. Energy*, vol. 88, no. 8, pp. 2740–2754, 2011.

[5] R. Rayegan and Y. X. Tao, "A procedure to select working fluids for Solar Organic Rankine Cycles (ORCs)," *Renew. Energy*, vol. 36, no. 2, pp. 659–670, 2011.

[6] E. Cayer, N. Galanis, M. Desilets, H. Nesreddine, and P. Roy, "Analysis of a carbon dioxide transcritical power cycle using a low temperature source," *Appl. Energy*, vol. 86, no. 7–8, pp. 1055–1063, 2009.

[7] H. D. Madhawa Hettiarachchi, M. Golubovic, W. M. Worek, and Y. Ikegami, "Optimum design criteria for an Organic Rankine cycle using low-temperature geothermal heat sources," *Energy*, vol. 32, no. 9, pp. 1698–1706, 2007.

[8] G. Gong, W. Zeng, L. Wang, and C. Wu, "A new heat recovery technique for air-conditioning/heat-pump system," *Appl. Therm. Eng.*, vol. 28, no. 17–18, pp. 2360–2370, 2008.

[9] S. Aphornratana and T. Sriveerakul, “Analysis of a combined Rankine-vapour-compression refrigeration cycle,” *Energy Convers. Manag.*, vol. 51, no. 12, pp. 2557–2564, 2010.

[10] C. He, C. Liu, H. Gao, H. Xie, Y. Li, S. Wu, and J. Xu, “The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle,” *Energy*, vol. 38, no. 1, pp. 136–143, 2012.

[11] B. F. Tchanche, G. Papadakis, G. Lambrinos, and A. Frangoudakis, “Fluid selection for a low-temperature solar organic Rankine cycle,” *Appl. Therm. Eng.*, vol. 29, no. 11–12, pp. 2468–2476, 2009.

[12] A. A. Lakew and O. Bolland, “Working fluids for low-temperature heat source,” *Appl. Therm. Eng.*, vol. 30, no. 10, pp. 1262–1268, 2010.

[13] Y. Dai, J. Wang, and L. Gao, “Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery,” *Energy Convers. Manag.*, vol. 50, no. 3, pp. 576–582, 2009.

[14] D. Wang, X. Ling, and H. Peng, “Performance analysis of double organic Rankine cycle for discontinuous low temperature waste heat recovery,” *Appl. Therm. Eng.*, vol. 48, pp. 63–71, 2012.

[15] H. Chen, D. Yogi Goswami, M. M. Rahman, and E. K. Stefanakos, “Energetic and exergetic analysis of CO2- and R32-based transcritical Rankine cycles for low-grade heat conversion,” *Appl. Energy*, vol. 88, no. 8, pp. 2802–2808, 2011.

[16] Ö. Kaşka, “Energy and exergy analysis of an organic Rankine for power generation from waste heat recovery in steel industry,” *Energy Convers. Manag.*, vol. 77, pp. 108–117, 2014.

[17] F. Heberle and D. Brüggemann, “Exergy based fluid selection for a geothermal Organic Rankine Cycle for combined heat and power generation,” *Appl. Therm. Eng.*, vol. 30, no. 11–12, pp. 1326–1332, 2010.

[18] J. Bao and L. Zhao, “A review of working fluid and expander selections for organic Rankine cycle,” *Renew. Sustain. Energy Rev.*, vol. 24, pp. 325–342, 2013.

[19] Y. Feng, T. Hung, K. Greg, Y. Zhang, B. Li, and J. Yang, “Thermoeconomic comparison between pure and mixture working fluids of organic Rankine cycles (ORCs) for low temperature waste heat recovery,” *Energy Convers. Manag.*, vol. 106, pp. 859–872, 2015.

[20] A. S. Wanniarachchi, D.Chisholm, “Plate heat exchangers: plate selection and arrangement,” *AIChE Meeting, Orlando, Florida, March 18-22*, 1990.

[21] Y. Y. Hsieh and T. F. Lin, “Saturated flow boiling heat transfer and pressure drop of refrigerant R-410A in a vertical plate heat exchanger,” *Int. J. Heat Mass Transf.*, vol. 45, no. 5, pp. 1033–1044, 2002.

[22] Y. S. Kim, “An Experimental Study on Evaporation Heat Transfer Characteristics and Pressure Drop in Plate Heat Exchanger,” *Diss. MS thesis, Yonsei University*, 1999. .

[23] S. Quoilin, S. Declaye, B. F. Tchanche, and V. Lemort, “Thermo-economic optimization of waste heat recovery Organic Rankine Cycles,” *Appl. Therm. Eng.*, vol. 31, no. 14–15, pp. 2885–2893, 2011.