The International Journal of Energy & Engineering Sciences, 2023, 8 (2) 1-13
ISSN: 2602-294X - Gaziantep University

ANALYSIS OF AN ORC USING R245FA UNDER THE OPTIMUM DESIGN WORKING CONDITION OF CONDENSER

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ABSTRACT

This scientific paper presents a comprehensive performance analysis of an Organic Rankine Cycle (ORC) used for heat waste recovery. The study investigates the influence of condenser design on the overall performance of the (ORC) system. To achieve this, the R245fa refrigerant is chosen as the working fluid due to its advantageous properties, rendering it a suitable choice for this specific application. Through detailed analysis, the paper provides insights into the efficiency and effectiveness of the (ORC) system, shedding light on the significance of condenser design in optimizing its performance. The conclusion of the study provides an in-depth analysis of the ORC system using R245fa as the working fluid. The performance analysis considers ideal conditions and real conditions with pressure drops across heat transfer components. The design procedure for the shell and tube horizontal condenser is thoroughly explained, and eleven condenser designs are evaluated against design limitations. The optimal condenser design is chosen based on these limitations and minimum surface area. The study concludes that the overall performance of the ORC system in real conditions is lower due to pressure drop effects. The findings emphasize the crucial role of careful condenser design in improving the performance, efficiency, and cost-effectiveness of ORC systems, thus influencing investment choices. When the ideal and real cases are compared, it is seen that the thermal efficiency of ORC is 14.03% in the ideal situation, 13.94% in the real case, and the exergy efficiency is 31.50% in the ideal situation and 31.31% in the real case, due to the presence of pressure drop affecting the efficiency of the system.

Key Words: Condenser, design, analysing, optimising, ORC, R245fa

1. INTRODUCTION

The conventional sources of energy and their rising costs have prompted the exploration of energy systems that utilize lost or wasted energy and convert it into useful forms [1]. The Organic Rankine Cycle (ORC) has emerged as a promising option for heat recovery applications, offering high energy generation efficiency. Consequently, investments in (ORC) energy systems have seen a significant increase in recent years. However, while (ORC) operation costs are relatively low, the expenses associated with its components can be substantial, leading to investor hesitation and delayed investment decisions. Among these components, the heat exchanger unit has a high manufacturing cost comparatively and the design of it plays a crucial role in the overall performance of the (ORC) system. To tackle this challenge, scientists and engineers have intensified their efforts to enhance both the performance and cost-effectiveness of (ORC) systems [2]. Starting from this critical point, this paper focuses on the influence of heat exchanger unit design on the performance of (ORC), shedding light on the ongoing endeavours to improve the efficiency and cost-effectiveness of this energy conversion technology. The findings contribute to advancing (ORC) energy systems and inform decision-makers in the investment and design of efficient heat recovery systems [3,6].

This research was undertaken to conduct an extensive performance analysis of an Organic Rankine Cycle (ORC) employed for heat waste recovery, specifically to examine the impact of condenser design on the overall performance of the ORC system. The study aimed to elucidate significant insights pertaining to the efficiency and effectiveness of the ORC system, particularly in the context of real operating conditions wherein pressure drops occur across heat transfer components.

2. SYSTEM DESCRIPTION

This paper describes a power plant based on the Organic Rankine Cycle (ORC) principle, in which the temperature of exhaust gas that is generated from industrial operations is utilized to generate electricity for lighting a city in Turkiye. The (ORC) system includes six main components: condenser, main pump, preheater, evaporator, superheater, and turbine [4,5]. Fig. 1 illustrates the flow of the system. The high-temperature of exhaust gas enters the (ORC) system through the superheater and then passes through the evaporator before exiting at a lower temperature from the preheater.

Simultaneously, the refrigerant is pumped by the main pumping station to the preheater, where the first stage of heat transfer with the exhaust gas is initiated. This causes the refrigerant to become a saturated liquid. Subsequently, in the evaporator, the saturated liquid of the refrigerant undergoes the second stage of heat transfer with the exhaust gas, transforming into a saturated vapour. The vapour then proceeds to the superheater, where it undergoes the third stage of the heating transfer process, becoming superheated. The superheated vapour is then directed to the turbine, where the expansion process takes place, resulting in the generation of electricity. Following this, the superheated vapour exits the turbine and enters the condenser, initiating the refrigerant condensation process. The refrigerant condenses into a saturated liquid state, which is subsequently pumped back into the preheater, and the cycle repeats.

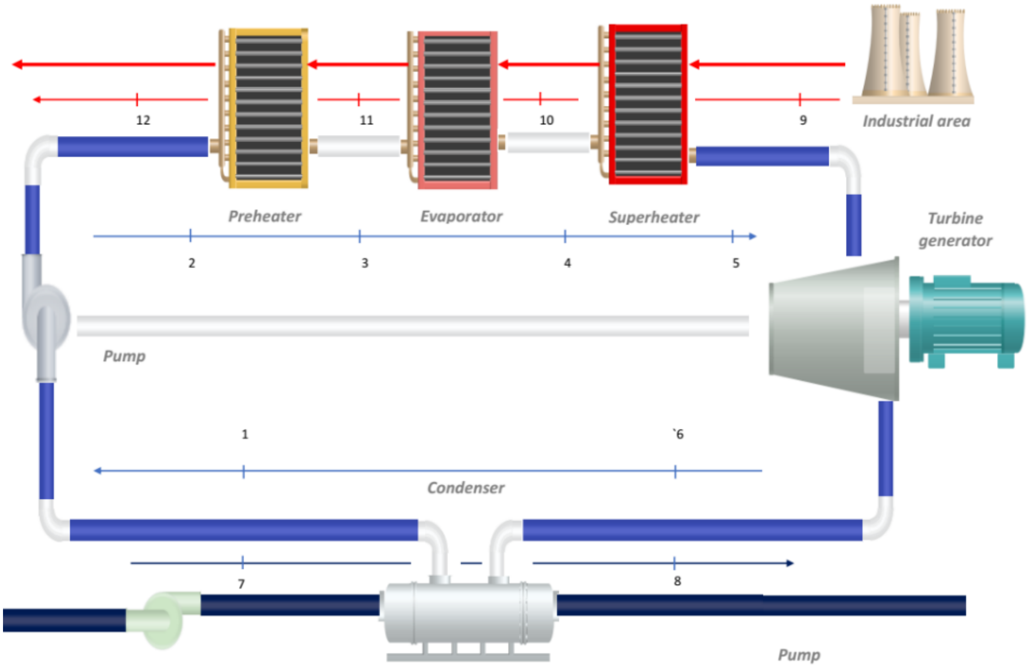

Fig. 1. The schematic layout of the Organic Rankine cycle

2.1. Refrigerant selection criteria

R245fa, also known as 1,1,1,3,3-pentafluoropropane, is a refrigerant that is widely employed in a variety of applications due to its advantageous qualities. It is a hydrofluorocarbon (HFC) and is extensively used as a working fluid in heat transfer and energy conversion systems. 245fa has good thermal features like as a low boiling point and high heat transfer coefficients, making it well-suited for application in organic Rankine cycle (ORC)heat recovery systems. Because of its modest critical temperature and pressure, it enables effective heat exchange and energy generation. Furthermore, R245fa has great environmental properties because it contains no chlorine and is non-ozone depleting. It has a low global warming potential (GWP) when compared to other refrigerants, which helps to reduce greenhouse gas emissions. In summary, R245fa is a favoured choice for heat recovery applications, notably in (ORC) systems, due to its unique mix of thermodynamic efficiency, environmental compatibility, and safety. Its use helps to maximize energy efficiency while limiting environmental effects, which aligns with the increased emphasis on sustainable energy practices.

Table 1. R245fa Physical and Environmental Properties [6,7]

Organic Fluid	MM kg/kmol	NBP (@1 bar) °C	T_{crit} °C	P_{crit} bar	ODP	GWP	ASHRAE 34* Safety Group
R245fa	134.03	14.81	154.01	36.5	0	1030	A1

MM: Molecular Mass, NBP: Normal Boiling Point, ODP: Ozone Depletion Potential (relative to R11), GWP: Global Warming Potential (relative to CO_2)
*A low toxicity, B high toxicity; 1 no flame; 2 lower flammability, 3 higher flammability

Table 2. Exhaust's (dry air) accepted nominal parameters:

Fluid	**$\dot{m}a$ kg/s**	**T_9 °C**	**T_{12} °C**	**h_9 kJ/kg**	**h_{12} kJ/kg**
Exhaust (dry air)	21.8	375	150	784.3018	550.7663

$\dot{m}a$: Mass flow rate; T9: Exhaust temperature coming from industrial operations; T12: Exhaust temperature exits from the system; h: Enthalpy

Table 3. Refrigerant's accepted nominal parameters:

Organic fluid	**P_1 bar**	**P_2 bar**	**x_1**	**x_3**	**x_4**	**h_1 kJ/kg**	**h_3 kJ/kg**	**h_4 kJ/kg**
R245fa	3*	20*	0	0	1	260.6501442	376.3823071	486.1891629

P1: pump inlet pressure; P2: pump outlet pressure; x: gas quality
*The pressure is assumed in values in which the cycle operates in a good performance.
3* minimum applicape pressure inlet; 20* moderate value of pumping pressure.

3. MATHEMATICAL MODELLING

Presented below (Eqs. 1-14) are the equations that govern the mass, energy, and exergy balance of the (ORC) system, ensuring equilibrium under steady-state conditions.

The key equations are as follows [8,9]:

Condenser (6-1):

$$\dot{Q}_{con} = \dot{m}_R(h_1 - h_6) = \dot{m}_W\, cp\, \Delta T_{cooling} \quad (1)$$

Preheater (2-3):

$$\dot{Q}_{pre} = \dot{m}_R(h_3 - h_2) = \dot{m}_a(h_{11} - h_{12}) \quad (2)$$

Evaporator (3-4)

$$\dot{Q}_{evap} = \dot{m}_R(h_4 - h_3) = \dot{m}_a(h_{10} - h_{11}) \quad (3)$$

Superheater (4-5):

$$\dot{Q}_{sup} = \dot{m}_R(h_5 - h_4) = \dot{m}_a(h_9 - h_{10}) \quad (4)$$

Total energy input to the system:

$$\dot{Q}_{total} = \dot{Q}_{pre} + \dot{Q}_{evap} + \dot{Q}_{sup} = \dot{m}_a(h_9 - h_{12}) = \dot{m}_R(h_5 - h_2) \quad (5)$$

Turbine (5-6):

$$\dot{W}_{tur} = \dot{m}_R(h_5 - h_6) \quad (6)$$

Pumping station (1-2):

$\dot{W}_{pump} = \dot{m}_R(h_2 - h_1)$	(7)
Net power generated:	
$\dot{W}_{net} = \dot{W}_{tur} - \dot{W}_{pump}$	(8)
Thermal efficiencies for turbine and pump:	
$\eta_{th,tur} = \frac{h_5 - h_{s6}}{h_5 - h_6}$	(9)
$\eta_{th,pump} = \frac{hs2 - h1}{h2 - h1}$	(10)
(ORC)'s thermal and exergy efficiency:	
$\eta_{th,ORC} = \frac{\dot{W}_{net}}{\dot{Q}_{total}}$	(11)
$\eta_{ex,ORC} = \frac{\dot{W}_{net}}{Ex_{in}}$	(12)
$Ex_{in} = \dot{m}_a(\psi_{in} - \psi_{out})$	(13)
$\psi = (h - h_0) - T_0(s - s_0)$	(14)

The pressure, temperature, enthalpy, and mass flow of the state points in (ORC) with R245fa and analysis results are given in Table 4 and Table 5, respectively.

Table 4. Thermodynamic flow parameters for (ORC)

State	Fluid	Pressure	Temperature	Enthalpy	Mass flow
		bar	°C	kJ/kg	kg/s
1	R245fa	3	45.58	260.65	21.30805
2	R245fa	20	46.52	262.31	21.30805
3	R245fa	20	121.77	376.38	21.30805
4	R245fa	20	121.77	486.19	21.30805
5	R245fa	20	131.77	501.23	21.30805
6	R245fa	3	72.66	466.07	21.30805
7	Water	1	20.00	84.01	69.81058
8	Water	1	35.00	146.70	69.81058
9	Exhaust gas	1	375.00	784.30	69.81058
10	Exhaust gas	1	361.14	769.60	69.81058
11	Exhaust gas	1	258.67	662.27	69.81058
12	Exhaust gas	1	150.00	550.77	69.81058

Table 5. (ORC) thermodynamic analysis results with R245fa

$\dot{Q}_{pre}$ kW	$\dot{Q}_{evap}$ kW	$\dot{Q}_{sup}$ kW	$\dot{Q}_{con}$ kW	$\dot{W}_{pump}$ kW	$\dot{W}_{tur}$ kW	$\dot{W}_{net}$ kW	$\eta_{th,ORC}$ %	$\eta_{ex,ORC}$ %
2430.74	2339.77	320.57	4376.99	35.29	749.37	714.08	14.03%	31.50%

3.1. Shell and tube horizontal condenser

A horizontal shell and tube condenser is a type of heat exchanger that is often used in industrial applications to condensate vapours or gases into liquids. It is made up of a shell (an exterior cylindrical vessel) and a bundle of tubes (inner tubes) that are positioned horizontally within the shell. The condensed fluid circulates around the tubes in the shell, while the cooling medium (water) travels through the tubes.

Table 6. Condenser designing steps: [10-13]

Heat transfer rate for condensation process	Description
$\dot{Q}_{con} = \dot{m}_R (h_6 - h_1) = \dot{m}_w \, c_{pw} \, \Delta T_{cooling}$ (15)	
$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\left[\frac{T_1 - t_2}{T_2 - t_1}\right]}$ (16)	
$A = \frac{\dot{Q}_{con}}{U_{0,assumed} \, LMTD}$ (17)	T_1 and T_2 are hot fluid inlet and outlet temperature respectively t_1 and t_2 are cold fluid inlet and outlet temperature respectively
LMTD_corrected = F * LMTD (If two fluid procced no isothermal process) see ***Appendix-A, Figure A1.*** The typical U-value = 200-300 W/m^{2}K. for a horizontal shell and tube condenser using R245fa refrigerant	
Assign the tube properties and geometries	
$A = \pi \, d_{out} \, L \, N_t$ (18)	
$D_b = d_{out} \left(\frac{N_t}{K1}\right)^{(1/n1)}$ (19)	
$D_s = D_b + Clearance$ (20) K1 and n1 values; ***Appendix-A, Table A1*** Clearance; ***Appendix-A, Figure A2***	
Heat transfer coefficient inside the tube hi for single phase flow, we use Seider-Tate and Hausen equations	
$Re = \left(\frac{G x d_{in}}{\mu}\right)$ (21)	
$h_i = 0.023 \, Re^{0.8} Pr^{\frac{1}{3}} \left(\frac{k}{d_{in}}\right)$ (22) 4000<Re	
$h_i = 0.116 \left(R_e^{\frac{2}{3}} - 125\right) Pr^{\frac{1}{3}} \left(1 + \left(\frac{d_{in}}{L}\right)^{\frac{2}{3}}\right) \left(\frac{k}{d_{in}}\right)$ (23) 2300<Re<4000	G = mass velocity; kg/m^{2}s
$h_i = 1.86 \left(Re \, Pr \, \frac{D}{L}\right)^{\frac{1}{3}} \left(\frac{k}{d_{in}}\right)$ (24) Re>4000	

The estimating of shell side (refrigerant side) could be by using the following formula:

$$(h_c)b = 0{,}95k_L\left[\frac{\rho l(\rho l-\rho v)g}{\mu_L\Gamma_n}\right]^{\frac{1}{3}}(N_\gamma')^{(-\frac{1}{6})}$$ (25)

$$\Gamma_n = \frac{\dot{m}}{L\times N_t}$$ (26)

$$N_\gamma' = (\frac{2}{3}N_r)$$ (27)

$$N_r = \frac{D_b}{p_t}$$ (28)

$(h_c)b$ = the mean coefficient for a tube bundle; (W/m²K)
Γ_n = the condensate flow per unit length of tube; (kg/ms)
N_r= number of tubes in the centre raw
N_γ' = average number of tubes in a vertical tube row
p_t = tube pitch; m
(Recommended, p_t=1.25d_{out})

$U_{o,est}$ overall heat transfer coefficient which can be written as

$$\frac{1}{U_{o,est}} = \left[\frac{1}{h_o} + R_{do} + \frac{A_O}{A_i}\left(\frac{d_{in}-d_{out}}{2k_W}\right) + \frac{A_O}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_O}{A_i}R_{di}\right]$$ (29)

R_{do} and R_{di} = fouling factors; m²°C/W;

Appendix-A, Table A2

The error design percentage could be calculated by using the following formula:

$$E\% = \frac{Uo.est-Uo.assumed}{Uo.\ assumed}\times 100<10\%$$ (30)

The pressure drop calculations for both the tubes and shell side for the shell and tube horizontal condenser are performed by using the following equations:

$$\Delta P_{tube} = \Delta P_{tube,f} + \Delta P_{turing\ pass}$$ (31)

$$\Delta P_{tube,f} = f_{darcy}\left(\frac{L}{d_{in}}\right)\frac{\rho {v_t}^2}{2}N_{tube,passes}$$ (32)

$$\Delta P_{turing\ pass} = K\frac{\rho v^2}{2}N_{tube,passes}$$ (33)

$$\Delta P_s = 8\,(jh)\,(\frac{Ds}{De})\,(\frac{L}{BS})\,\frac{\rho vs^2}{2}$$ (34)

jh: friction coefficient chart
(Appendix-A, Figure A3)

$$D_e = \frac{1.27}{d_{out}}(p_t^2 - 0.785d_{out}^2)$$ (35)

for square pitch type

$$D_e = \frac{1{,}1}{d_{out}}(p_t^2 - 0.917d_{out}^2)$$ (36)

for triangular pitch type

$$B_s = 0.4D_s$$ (37)

$$v_s = (\frac{G_s}{\rho})$$ (38)

$$G_s = \frac{\dot{m}_s}{A_s}$$ (39)

$$A_s = \frac{p_t-d_{out}}{p_t}x\ D_s\ x\ B_s$$ (40)

K=Minor friction coefficient
B_s = baffle space; m
G_s= shell side mass velocity; kg/m² s

Condenser accepted fixed dimensions:

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Table 7. Condenser fixed properties and tube geometries

d_{out} (mm)	d_{in} (mm)	Material	Pitch (mm)	Tube layout	Shell Passes No	Tube Passes No
25.4	21.3	Carbon steel	1.25 d_{out}	Triangular-square	1	4

Condenser design results

Fig. 2 and Fig. 3 represent 11 different condenser designs' refrigerant side pressure drop and required cooling surface area respectively:

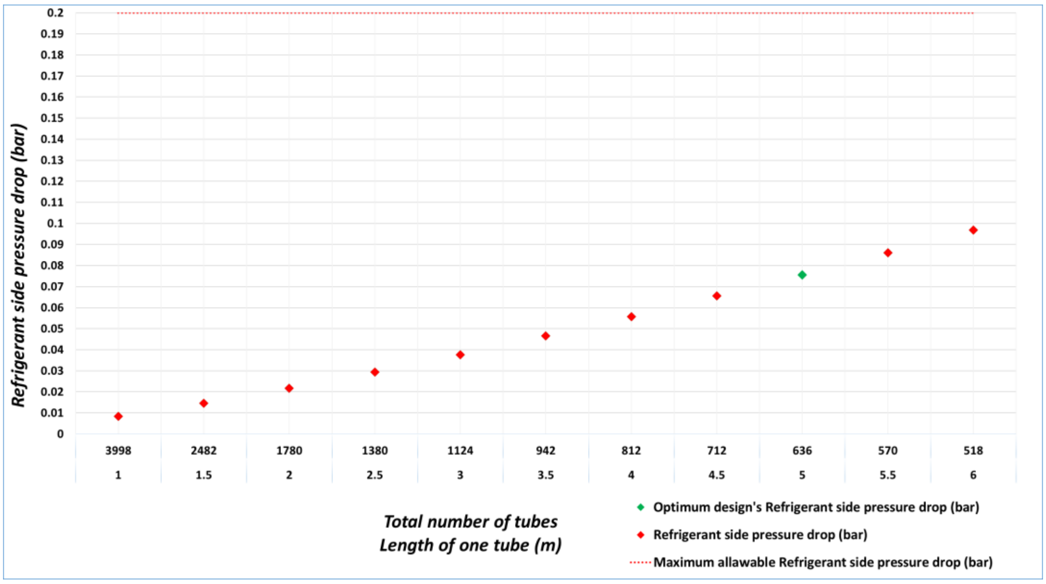


Fig. 2. Refrigerant side pressure drops (bar) for different condenser designs

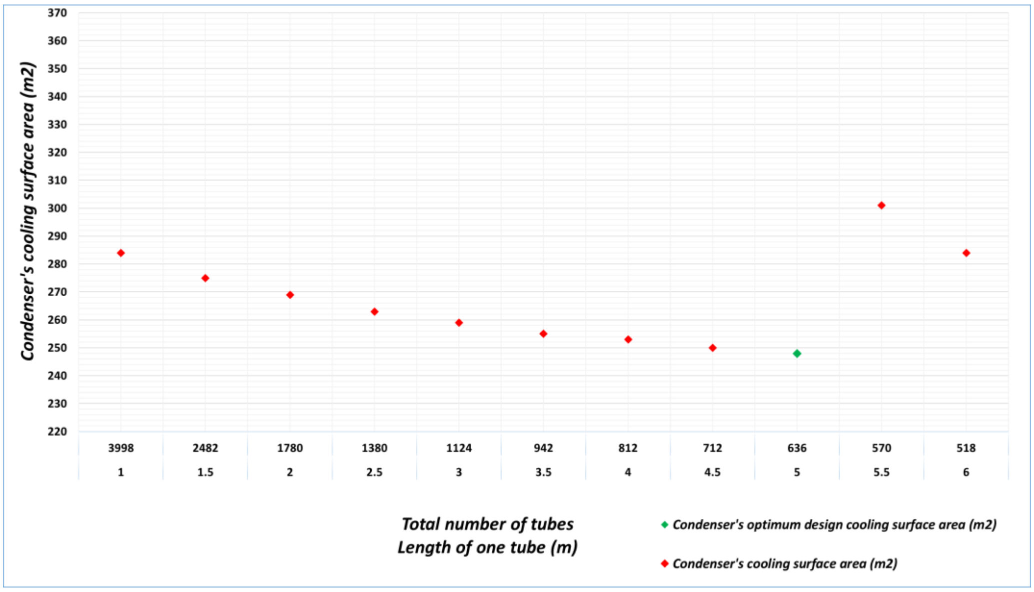


Fig. 3. Required cooling area (m^2) for different condenser designs

The green symbol indicates the best design among all designs that are represented as red symbols, as all these designs are subjected to several design enhancing tests and the optimum design is able to pass all these limitations.

Table 8. Condenser design limitations [12]

Error (E%)*	v_t (m/s)**	L_t/D_s***	ΔP_t (bar)	ΔP_s (bar)
<10%	1.5<v_t < 2.5	5 < L_t/D_s<10	<2	<0.2

*To avoid overdesign for condenser

**low-velocity causes low heat transfer; high velocity leads to tube erosion and more pressure drop

***to have exemplary geometries of the design

3.2. Performance analysis of (ORC) with the optimum design of condenser

Here in this section, the pressure drop of the optimum design is taken in all the calculations. However, the pressure drop of all other heat transfer units remains zero.

Table 9. Thermodynamic flow parameters for (ORC) (Real case)

State	Fluid	Pressure	Temperature	Enthalpy	Mass flow
		bar	°C	kJ/kg	kg/s
1	R245fa	2.90	44.54	259.23	21.18
2	R245fa	20	45.49	260.89	21.18
3	R245fa	20	121.77	376.38	21.18
4	R245fa	20	121.77	486.19	21.18
5	R245fa	20	131.77	501.23	21.18
6	R245fa	3	72.66	466.07	21.18
7	Water	1	20.00	84.01	69.88
8	Water	1	35.00	146.70	69.88
9	Exhaust gas	1	375.00	784.30	21.8
10	Exhaust gas	1	361.22	769.68	21.8
11	Exhaust gas	1	259.36	662.99	21.8
12	Exhaust gas	1	150.00	550.77	21.8

Table 10. (ORC) thermodynamic analysis results with R245fa (Real case)

$\dot{Q}_{pre}$ kW	$\dot{Q}_{evap}$ kW	$\dot{Q}_{sup}$ kW	$\dot{Q}_{con}$ kW	$\dot{W}_{pump}$ kW	$\dot{W}_{tur}$ kW	$\dot{W}_{net}$ kW	$\eta_{th,ORC}$ %	$\eta_{ex,ORC}$ %
2446.38	2326.01	318.68	4381.31	35.20	744.96	709.76	13.94%	31.31%

4. CONCLUSIONS

The study which is provided in this paper has included an in-depth performance analysis of an Organic Rankine Cycle (ORC) with R245fa as the working fluid. Firstly, a performance study has been carried out under ideal conditions, assuming zero

The International Journal of Energy & Engineering Sciences, 2023, 8 (2) 1-13
ISSN: 2602-294X - Gaziantep University

pressure drop across all heat transfer components. Following that, the design procedure for the shell and tube horizontal condenser has been thoroughly explained.

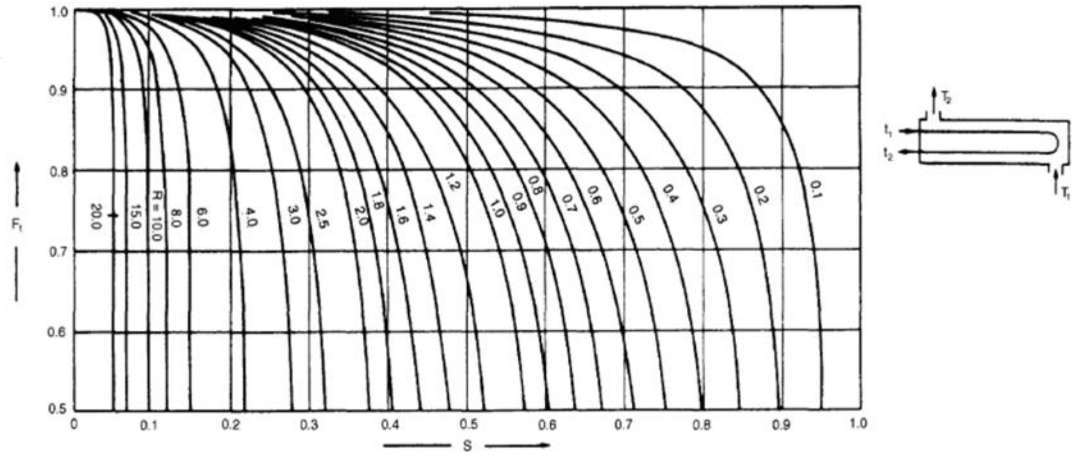
In the design process, eleven distinct condenser designs have been developed and evaluated against several design limitations. Following this, the most optimal condenser design is chosen based on the performed design limitations and minimum surface area. Finally, an (ORC) performance study has been performed based on real conditions, which includes the optimum condenser design, and this analysis aims to determine the influence of condenser design on the overall (ORC)'s performance. The results show that the overall performance of the (ORC) is lower in real conditions with the effect of condenser design than in the ideal case as shown in Table 11, due to the presence of pressure drop, which reduces the overall performance of the (ORC). As a result, the findings highlight the significant impact of heat exchanger unit design in improving both the performance and cost-effectiveness of the (ORC), therefore directly influencing investment choices.

In conclusion, this study emphasizes the crucial relevance of careful condenser design and its major effect on the performance, efficiency, and cost-effectiveness of (ORC) systems.

Table 11. (ORC) performance analysis recorded values for both ideal and real cases with R245fa

Parameter	$\dot{Q}_{pre}$	$\dot{Q}_{evap}$	$\dot{Q}_{sup}$	$\dot{Q}_{con}$	$\dot{W}_{pump}$	$\dot{W}_{tur}$	$\dot{W}_{net}$	$\eta_{th,ORC}$	$\eta_{ex,ORC}$
Unit	kW	kW	kW	kW	kW	kW	kW	%	%
State	↖	↘	↘	↖	↘	↘	↘	↘	↘
Ideal case	2430.74	2339.77	320.57	4376.99	35.29	749.37	714.08	14.03%	31.50%
Real case	2446.38	2326.01	318.68	4381.31	35.20	744.96	709.76	13.94%	31.31%

Appendix-A



Temperature correction factor: one shell pass; two or more even tube 'passes

Figure A1. Temperature correction factor [10]

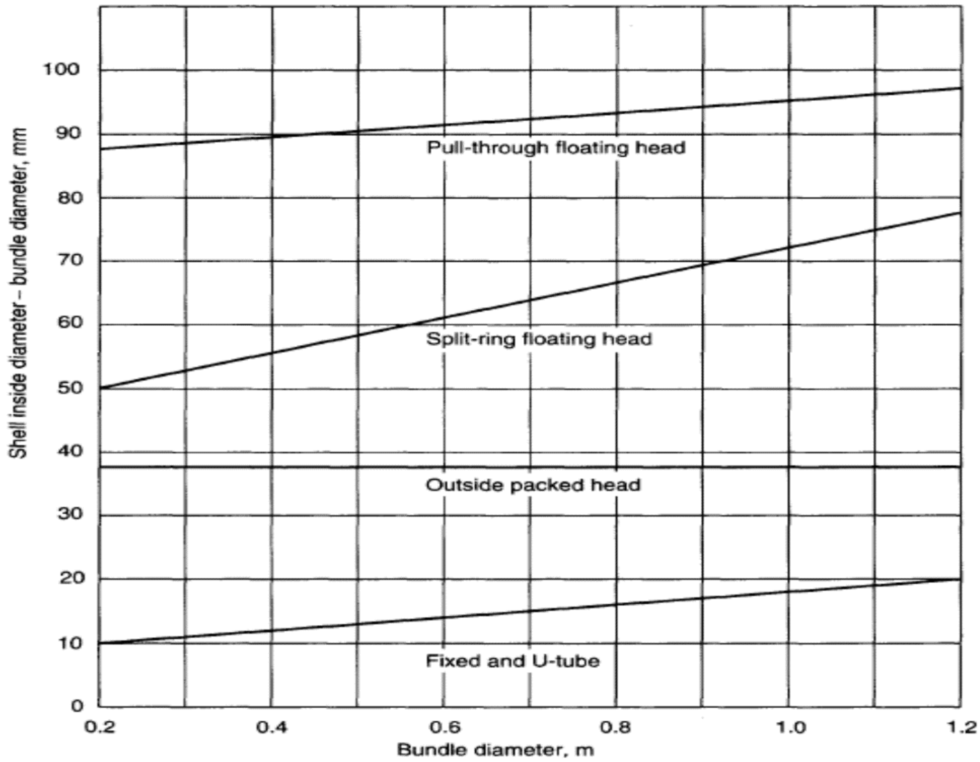


Figure A2. Bundle diameter and shell and clearance diagram [10]

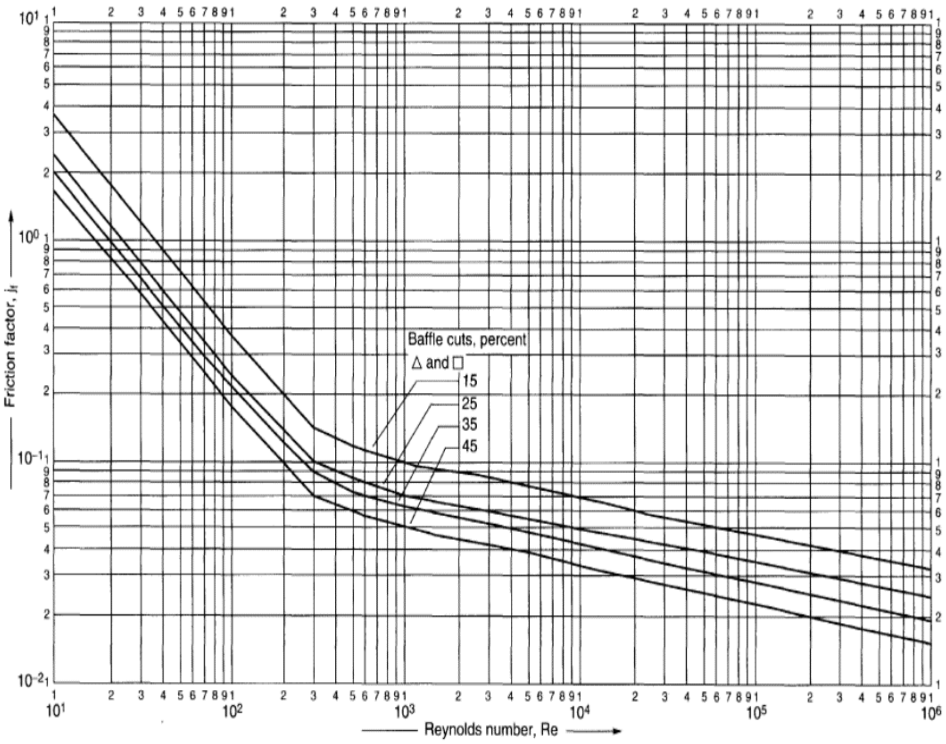


Figure A3. Shell side friction factor jh chart [10]

Table A1. K1 and n1 values [7]

Triangular pitch, *pt=1.25do*					
No. passes	1	2	4	6	8
K1	0.319	0.249	0.175	0.0743	0.0365
n1	2.142	2.207	2.285	2.499	2.675
Square pitch, *pt=1.25do*					
No. passes	1	2	4	6	8
K1	0.215	0.156	0.158	0.0402	0.0331
n1	2.207	2.291	2.263	2.617	2.643

Table A2. fouling factors and coefficient [12]

Fluid -	Coefficient W/m² °C	R_{do} m² °C/W
River water	3000-12000	0.0003-0.0001
Sea water	1000-3000	0.001-0.0003
Cooling water	3000-6000	0.0003-0.00017
Refrigerant	3000-5000	0.0003-0.0002
Air and industrial gases	5000-10000	0.0002-0.0001
Organic vapors	5000	0.0002
Organic liquids	5000	0.0002

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