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THERMODYNAMIC ANALYSIS OF DIFFERENT WORKING FLUIDS USED IN ORGANIC RANKINE CYCLE FOR RECOVERING WASTE HEAT FROM GT-MHR

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Abstract

In this paper, the performance of 13 working fluids in two Organic Rankine Cycles, which operate as the bottoming cycles for recovering waste heat from gas turbine modular helium reactor (GT-MHR), is investigated. Working fluids are classified in three dry, isentropic and wet fluids. The effect of varying pump temperature and evaporator pressure on the thermal efficiency, total exergy loss of the combined cycle is studied for each category, and the results are compared. The results are calculated for an optimum pressure ratio in which thermal efficiency is maximum. According to the results, dry fluids show a higher thermal efficiency while wet fluids have the lowest values. However, the highest value for thermal efficiency is for R141b, which is an isentropic fluid. Furthermore, the results indicate that pump temperature increase, reduces the total thermal efficiency and increases the total exergy loss of the combined cycle. Increasing evaporator pressure leads to an optimum pressure that maximizes total thermal efficiency. According to the optimized pressure ratio and evaporator pressure, R141b in isentropic fluids, R123 in dry fluids and R717 in wet fluids have the highest thermal efficiency values.

Keywords: GT-MHR, ORC, working fluids, comparison, thermal efficiency

1. Introduction

The gas turbine modular helium reactor (GT-MHR) which is the combination of a gas-cooled modular helium reactor (MHR) and a high efficiency modular Brayton cycle gas turbine (GT) energy conversion system, is a nuclear-power system that

Nomenclatures

$\dot{E}$	exergy rate, MW
h	specific enthalpy, kJ/kg K
$\dot{m}$	mass flow rate, kg/s
P	pressure , kPa
PR	pressure ratio
$\dot{Q}$	heat transfer rate, MW
T	temperature, ◦C
v	specific volume, m^3/kg
$\dot{W}$	power generation rate, MW

Greek Symbols

η_I	First law efficiency
η_{II}	second law efficiency
$\eta_{p,t}$	turbine polytropic efficiency
$\eta_{p,c}$	compressor polytropic efficiency
ε	effectiveness, %

Subscripts

0	reference environment
1,2,…	cycle locations
c	compressor
con	condenser
core	reactore core
L	loss
eva,1	evaporator 1
eva,2	evaporator 2
GT	gas turbine
HP	high pressure
IC	intercooler
in	inlet
LP	low pressure
MHR	Modular helium cooled reactor
net	net
ORC	Organic Rankine Cycle
p	pump
pc	precooler
rec	recuperator
rel	relative
s	Isentropic process
t	turbine

is designed to offer high thermal efficiency, safety, proliferation/terrorist resistance, economic competitiveness and environmental advantages. The environmental predominance as well as lower thermal discharge (waste heat) of GT-MHR is because of its high thermal efficiency, and high fuel burned up capability. It also eliminates the need to large heat exchangers to produce steam for electricity generation, because of the helium's high thermal capacity. Because helium is naturally inert and single-phase, the helium-cooled reactor can operate at much higher temperatures than today's conventional nuclear plants. The performance of this cycle has been studied by several researchers [1-4].

On the other hand, the Organic Rankine cycle (ORC) has been given a lot of attention as it has the potential to be used for power generation from low-temperature heat sources. It is a Clausius Rankine cycle that exploits organic working fluid instead of steam. Some researchers studied the application of the ORCs in the combined systems [5, 6]. Wang et al. [7] investigated a combined Organic Rankine Cycle and a conventional vapor compression cycle. The results have shown that by recovering waste heat from diesel engines and other power cycles, the system can generate cooling as well as power to improve the overall efficiency and the utilization of fuel. Chacartegui et al. [8] studied the low-temperature Organic Rankine Cycles as bottoming cycle in medium and large scale combined cycle power plants. The results showed that ORCs are an interesting and competitive option when combined with high efficiency gas turbines with low exhaust temperature.

Moreover, performance analysis of different working fluids in ORCs have been analyzed [9-16]. Saleh et al. [17] studied 31 pure component working fluids for Organic Rankine Cycles assuming the BACKONE equation of state. Dai et al. [18] showed that the cycles with organic working fluids are much better than the cycle with water in converting low grade waste heat to useful work. Mago et al. [19] investigated a second-law analysis for the use of different organic working fluids in ORC to convert waste energy to power from low-grade heat sources. According to the results, the higher the boiling point temperature of the organic fluid, the better the thermal efficiency that will be achieved by the ORC. Rayegan and Tao [20] compared capabilities of 117 working fluids used in solar Oeganuc Rankine cycles with similar working conditions. Calculation results showed that the effect of regeneration on the exergy efficiency of the cycle is fluid dependent while the effect of collector efficiency improvement on the exergy efficiency of the cycle is nearly independent of fluid type.

Although a lot of efforts have been done, there is not enough research work to study the GT-MHR cycle combined with other cycles. Yari and Mahmoudi [21] studied the combination of GT-MHR with two Organic Rankine Cycles. The effect of the different parameters on the first and second-law efficiencies and on the exergy destruction rate of the combined cycle were investigated. The results showed that by adding ORC to GT-MHR, the exergy destruction rate of the cycle is about 5% lower than that of the GT-MHR. Moreover, the first-law efficiency for GT-MHR/ORC cycle is 2.64–2.93%-points higher than that of the GT-MHR cycle. In another study, Yari and Mahmoudi [22] investigated the different arrangements of three Organic Rankine Cycle with a gas turbine-modular helium reactor from the point of view of thermodynamics and economics. It was found that the GT-MHR/SORC cycle had the highest performance among the studied arrangements. They also expressed that GT-MHR/RORC cycle had the potential of being used for cogeneration purposes.

The aim of this paper is to study the performance of different working fluids in Organic Rankine Cycles to recover waste heat from the GT-MHR. Two ORCs are used as the bottoming cycles of the GT-MHR cycle. Parametric studies and optimization of the cycle working conditions are necessary to compare the operation of different working fluids in a cycle. The working fluids are classified in dry, isentropic and wet fluids. The effect of each category on the thermal efficiency and total exegy loss of the combined cycle is studied separately and compared. The results of the optimization according to pressure ratio and evaporator pressure are also presented.

2. Combined Cycle Performance and Assumptions

The schematic and T-S diagrams of the combined GT-MHR and ORC cycle are shown in Figs. 1 and 2.

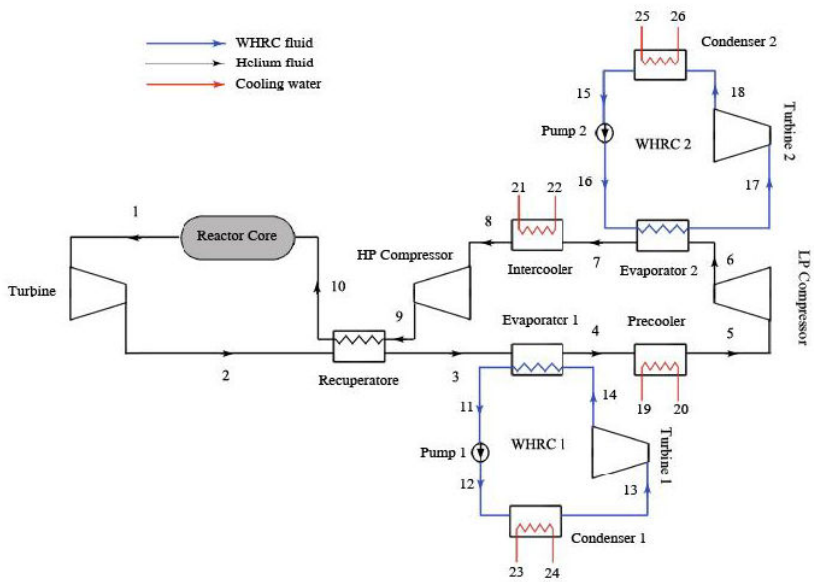


Fig. 1. Schematic diagram of GT-MHR/WHRCs system.

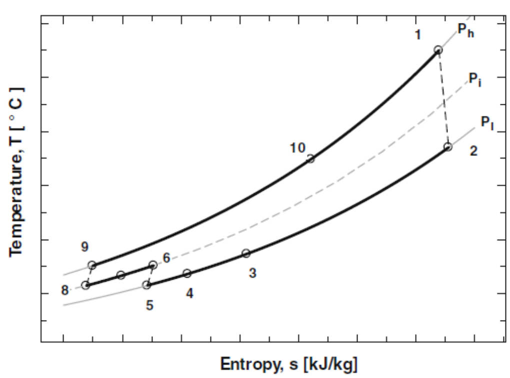


Fig. 2. T-S diagram for the combined GT-MHR/WHRCs system.

The working principle of the cycle is described as follows: Pressurized heated helium coolant leaving the core, flows through cooling channels and exits into the hot duct at the center of the cross vessel and is expanded through the turbine in the Power Conversion System (PCS). A gas turbine, an electric generator and gas compressors on a common, vertically oriented shaft supported by magnetic bearings are included in PCS. Furthermore, it is included recuperator, precooler and internal heat exchangers. The mechanical energy produced in the turbine is used to drive the electrical generator and the high and low pressure compressors. The helium exiting the turbine, flows into the hot side of the recuperator. After cooling, the helium flows through the precooler, after passing through the evaporator 1, to reject heat to the main heat sink of the cycle for further cooling. The cool low-pressure helium is compressed in a low-pressure compressor and after passing through evaporator 2 and intercooler, flows towards a high pressure compressor. It is then directed to the cold, high pressure side of the recuperator where it is preheated. Although, the cooling processes in the cycle leads to a reduction in the required work for the compressor, a majority of the removed energy is not lost from the power cycle and is recovered in the recuperator. The main assumptions for the simulation of the combined cycle are presented in Table 1.

The following assumptions are considered in this study:

- The combined cycle runs in a steady state condition.
- No pipe pressure drop is considered.
- The kinetic and potential energies are neglected.
- Isentropic efficiencies are considered for the ORC turbine and pump.
- Polytropic efficiencies are considered for the GT-MHR turbine and compressor.
- The effectiveness was considered for intercooler, recuperator and precooler.

Table 1. Main assumptions for simulating the combined cycle.

Assumption	Value
P_0 (kPa)	100
PR_c	1.668
$\dot{Q}_{core}$ (MW)	600
T_0 (∘C)	20
T_1 (∘C)	750
T_c (∘C)	30
$P_{eva,1}$ (kPa)	2000
$P_{eva,2}$ (kPa)	2000
η_p (%)	85
η_t (%)	80
ε (%)	90
ΔP_{core} (kPa)	100
ΔP_{eva}, ΔP_{IC}, ΔP_{pc} (kPa)	40
$\Delta P_{rec,HP}$ (kPa)	80
$\Delta P_{rec,LP}$ (kPa)	50

The basic equations obtained from the conservation law of energy in the components are presented in Tables 2 and 3.

Table 2. Energetic and exergetic equations for different components.

Cycles	Components	Energy equations	Exergy balances
GT-MHR	Compressor	$\eta_{p,c} = 0.916 - 0.01175\ln(PR_c)$[23] $PR_c = \frac{P_6}{P_5}$ $\dot{W}_c = \dot{m}_5(h_6 - h_5)$	$\dot{E}_{L,c} = \dot{E}_5 - \dot{E}_6 + \dot{W}_c$
	Intercooler	$\dot{m}_7(h_7 - h_8) = \dot{m}_{21}(h_{22} - h_{21})$ $\varepsilon_{IC} = \frac{T_7 - T_8}{T_{22} - T_{21}}$	$\dot{E}_{L,IC} = \dot{E}_7 - \dot{E}_8 + \dot{E}_{21} - \dot{E}_{22}$
	Precooler	$\dot{m}_4(h_4 - h_5) = \dot{m}_{19}(h_{20} - h_{19})$ $\varepsilon_{pc} = \frac{T_4 - T_5}{T_4 - T_{19}}$	$\dot{E}_{L,pc} = \dot{E}_4 - \dot{E}_5 + \dot{E}_{19} - \dot{E}_{20}$
	Reactor core	$\dot{Q}_{core} = \dot{m}_1(h_1 - h_{10})$	$\dot{E}_{L,core} = \dot{E}_{10} - \dot{E}_1 - \dot{Q}_{core}$
	Recuperator	$\dot{m}_2(h_2 - h_3) = \dot{m}_9(h_{10} - h_9)$ $\varepsilon_{rec} = \frac{T_{10} - T_9}{T_2 - T_9}$	$\dot{E}_{L,rec} = \dot{E}_2 - \dot{E}_3 + \dot{E}_9 - \dot{E}_{10}$
	Turbine	$\eta_{p,t} = 0.932 - 0.0117\ln(PR_t)$[23] $\dot{W}_t = \dot{m}_1(h_1 - h_2)$	$\dot{E}_{L,t} = \dot{E}_1 - \dot{E}_2 - \dot{W}_t$
ORC	Condenser	$\dot{m}_{12}(h_{12} - h_{13}) = \dot{m}_{23}(h_{24} - h_{23})$	$\dot{E}_{L,con} = \dot{E}_{12} - \dot{E}_{13} + \dot{E}_{23} - \dot{E}_{24}$
	Evaporator	$\dot{m}_3(h_3 - h_4) = \dot{m}_{11}(h_{11} - h_{14})$	$\dot{E}_{L,eva} = \dot{E}_3 - \dot{E}_4 + \dot{E}_{14} - \dot{E}_{11}$
	Pump	$\eta_p = \frac{v_{11}(P_{12} - P_{11})}{h_{12} - h_{11}}$	$\dot{E}_{L,p} = \dot{E}_{11} - \dot{E}_{12} + \dot{W}_p$
	Turbine	$\eta_{t,ORC} = \frac{h_{13} - h_{14}}{h_{13} - h_{14s}}$ $\dot{W}_{t,ORC} = \dot{m}_{13}(h_{13} - h_{14})$	$\dot{E}_{L,t,ORC} = \dot{E}_{13} - \dot{E}_{14} - \dot{W}_{t,ORC}$

Table 3. Performance evaluation of the combined cycle.

First law efficiency	$\eta_{\mathrm{I}} = \frac{\dot{W}_{net}}{\dot{Q}_{core}}$
Net power generation	$\dot{W}_{net} = \dot{W}_{net,GT-MHR} + \dot{W}_{net,ORC} =$ $(\dot{W}_t + \dot{W}_c)_{GT-MHR} + (\dot{W}_t + \dot{W}_c)_{ORC,1} + (\dot{W}_t + \dot{W}_c)_{ORC,2}$
Relative power generation	$\dot{W}_{rel} = \frac{\dot{W}_{ORC,1} + \dot{W}_{ORC,2}}{\dot{W}_{net}}$
Relative heat transfer rate	$\dot{Q}_{rel} = \frac{\dot{Q}_{ORC,1} + \dot{Q}_{ORC,2}}{\dot{Q}_{net}}$
second law efficiency	$\eta_{\mathrm{II}} = \frac{\dot{W}_{net}}{\dot{E}_{in}} = \frac{\dot{W}_{net}}{\dot{Q}_{core}}$ $\dot{E}_{in} = \dot{W}_{net} + \dot{E}_D$, $\dot{E}_{in} = \dot{Q}_{core}$
Exergy destruction rate	$\dot{E}_D = \dot{E}_{D,GT-MHR} + \dot{E}_{D,ORC,1} + \dot{E}_{D,ORC,2}$

3. Choosing Working Fluid

Choosing the appropriate working fluid is the first step to design a cycle. Although, in a cycle design, it is impossible to satisfy all desired general requirements. Non-fouling, non-corrosiveness, non-toxicity, and non-flammability are a few preferable physical and chemical characteristics that the working fluids should satisfy. Moreover, the working fluid should be stable and not dissociate at the pressures and temperatures in the cycle. Ozone depletion potential (ODP), global warming potential (GWP) and the atmospheric lifetime (ALT) are three important factors that should be regarded as the environmental effects of the working fluids. ODP of a chemical compound indicates the potential of a substance to destroy the ozone layer compared with CFC-11 having an ODP of 1.0. ODP is often used in conjunction with a compound's GWP which is defined as the potential of a substance to contribute to global warming (usually measured over a 100-year period). It is compared with the same mass of carbon dioxide whose GWP is by convention equal to 1. ALT is the approximate amount of time it would take greenhouse gases to leave the atmosphere.

The working fluids for ORC's are generally categorized in three groups based on their slope of saturation vapour curves in T-S diagram. The fluids having the positive slope are dry fluids. The fluids having the negative slope are wet fluids, while isentropic fluids have infinitely large slopes. Many investigations concerning different working fluids for ORC systems have been carried out [23-27]. Hung [28] showed that the dry and the isentropic fluids are the most preferred working fluid for the ORC cycle. Furthermore, Somayaji et al. [28] concluded that dry and isentropic fluids show better thermal efficiencies. This is because dry or isentropic fluids are superheated after isentropic expansion. Therefore, there is no concern for existing liquid droplets at the turbine outlet which causes erosion in the turbine blades, whereas the wet fluid leads to droplets at the later stages of turbine and requires superheating at the turbine inlet.

Table 4 shows physical properties and environmental data of fluids for wet, isentropic and dry fluids. Only one criterion was considered here: having ODP equal to zero or very close to zero according to the Montreal Protocol, an international treaty for the protection of the stratospheric ozone layer, and the EC regulation 2037/2000 which restrict the use of ozone depleting substances.

Table 4. Physical properties and environmental data of the studied working fluids.

Refrigerant Number	Type of fluid	Physical data				Safety data	Environmental data		
		P_{cr}	T_{cr}	T_{bp}	M	ASHRAE 34 safety group	GWP (100 year)	ODP	ALT (year)
		(MPa)	(◦C)	(◦C)	(g/mol)				
R143a	**wet**	3.76	84.04	-47.2	72.7	4470	0	52	A2
R152a	**wet**	4.52	66.05	-24	113.3	124	0	1.4	A2
R290	**wet**	4.25	44.1	-42.1	96.7	~20	0	0.041	A3
R717	**wet**	11.33	17.03	-33.3	132.3	<1	0	0.01	B2
R124	**isentropic**	3.62	136.48	-12	122.3	77	0.02	5.8	A1
R134a	**isentropic**	4.06	102.03	-26.1	101.1	1430	0	14	A1
R141b	**isentropic**	4.21	116.95	32	204.4	725	0.12	9.3	N.A
R142b	**isentropic**	4.06	100.5	-9.1	137.1	2310	0.07	17.9	A2
R123	**dry**	3.66	152.93	27.8	183.7	77	0.02	1.3	B1
R245fa	**dry**	3.650	134.05	15.1	154.0	1030	0	7.6	B1
R600	**dry**	3.8	58.12	-0.5	152	~20	0	0.018	A3
R600a	**dry**	3.63	58.12	-11.7	134.7	~20	0	0.019	A3
R601a	**dry**	3.38	72.15	27.8	187.2	~20	0	0.01	A3

4. Validation

A program for the simulation of the combined GT-MHR and ORC using EES software [29] was developed. The results are validated and compared with the results of the GT-MHR/ORC introduced by Yari and Mahmoudi [21]. According to the results of the comparison which are shown in Table 5, a good agreement is obtained.

Table 5. Validation of the numerical model with the previously published data [21].

Parameters	Present work	Yari and Mahmoudi
PR_C	3.244	3.244
$T_{eva,1}$ (◦C)	98.7	98.7
$T_{eva,2}$ (◦C)	78.96	78.96
η_I (%)	49.21	49.82
η_{II} (%)	49.21	49.82
$\dot{W}_{net,ORC1}$ (MW)	13.38	13.255
$\dot{W}_{net,ORC2}$ (MW)	6.007	5.789
$\dot{W}_{net}$ (MW)	295.3	298.93
$\dot{m}_{He}$ (kg/s)	275.2	282.2
$\dot{E}_D$ (MW)	304.7	301

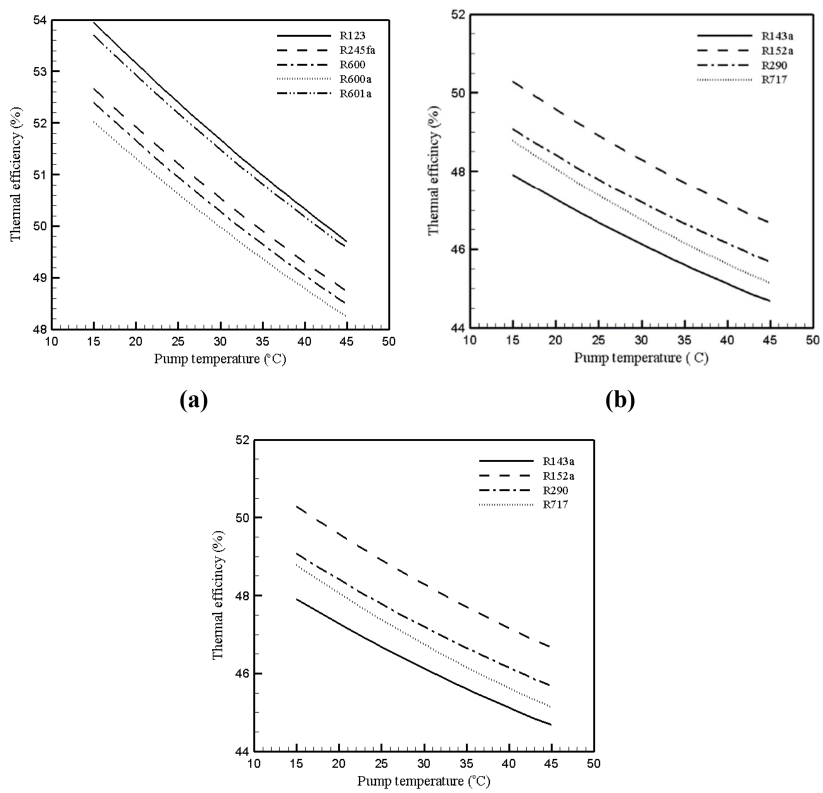
5. Results and discussion

The detailed data of the analysed cycle for 13 different working fluids, based on the assumptions in Table 1, are presented in Table 6. It should be noted that the simulation is conducted for the optimum pressure ratio in which the thermal efficiency is maximum. The results show that R123, R141b and R152a have the maximum amounts of thermal efficiencies in dry, isentropic and wet working fluids.

Table 6. Results of the comparison of studied working fluids.

Element	Unit	R143a	R152a	R290	R717	R124	R134a	R141b	R142b	R123	R245fa	R600	R600a
$\dot{m}_{ORC,1}$	(kg/s)	998.1	533.4	445.5	134.6	907.4	828.9	514.6	661.2	662.6	628.7	332.8	384.7
$\dot{m}_{ORC,2}$	(kg/s)	595.8	318.1	265.7	80.48	540.7	494.3	308.1	394.1	395.1	374.9	198.3	230.4
$\dot{W}_{net,ORC1}$	(MW)	4.723	12.87	8.768	7.052	16.14	11.31	28.44	18.03	25.6	21.34	20.37	19.18
$\dot{W}_{net,ORC2}$	(MW)	2.82	7.677	5.229	4.217	9.616	6.742	17.03	10.75	15.26	12.73	12.14	11.49
$\dot{Q}_{net}$	(MW)	1740	1726	1732	1737	1720	1728	1703	1717	1705	1712	1714	1718
$\dot{W}_{net}$	(MW)	276.7	289.7	283.2	280.5	294.9	287.2	314.7	298	310	303.2	301.7	299.9
$\dot{Q}_{relative}$	-	0.2783	0.2722	0.275	0.2778	0.2696	0.2733	0.2641	0.2684	0.2634	0.2665	0.2667	0.2704
$\dot{W}_{relative}$	-	0.02783	0.07093	0.04943	0.04018	0.08732	0.06284	0.1445	0.09659	0.1318	0.1123	0.1077	0.1023
$\dot{E}_D$	(MW)	323.3	310.3	316.8	319.5	305.1	312.8	285.3	302	290	296.8	298.3	300.1
η_{th}	(%)	46.12	48.25	47.2	46.75	49.15	47.87	52.45	49.66	51.67	50.54	50.28	49.98
η_{exe}	(%)	46.12	48.29	47.2	46.75	49.15	47.87	52.45	49.66	51.67	50.54	50.28	49.98

The effect of pump temperature increment on the thermal efficiency of the cycle is shown in Fig. 3. It is apparent that for all working fluids, the thermal efficiency of the combined cycle decreases with increasing pump temperature. This is since, the power generation in ORCs decreases as the pump temperature rises. The graphs show that, in general, dry fluids have the highest values of thermal efficiency compared with other types, and the wet fluids have the lowest amounts. The reason is the lower temperature at the turbine inlet temperature. However, the maximum amount is for R141b, which is an isentropic one. For the studied dry fluids, R123 with the values of 53.94 % and 49.7% and R600a with the values of 52.02% and 48.24% for the lowest and highest pump inlet temperatures, have the maximum and minimum amounts. For isentropic fluids, it is 54.85% and 50.35% for R141b and 49.78% and 46.31% for R134a. For wet fluids, R152a with the values of 50.28% and 46.66% has the highest value, and R143a has the lowest values with the values of 47.91% and 44.68%.

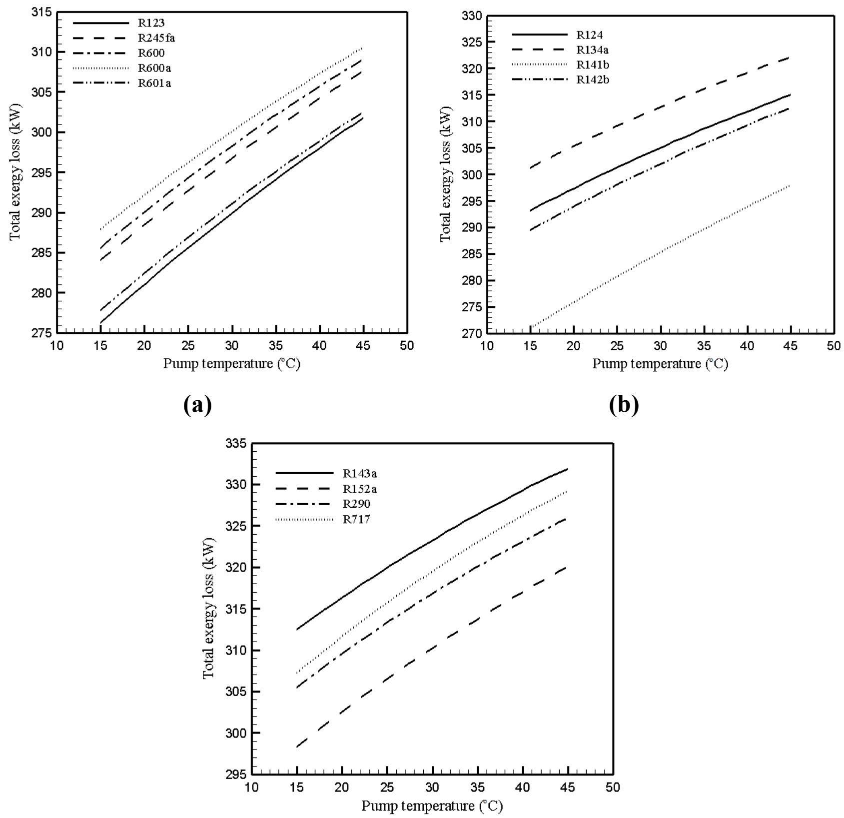


(a) **(b)** **(c)**

Fig. 3. Effect of pump temperature increment on the efficiency (a) dry, (b) isentropic, (c) wet fluids.

Figure 4 shows the variation of the total exergy loss with the pump inlet temperature. As a matter of fact, the lower the total exergy loss, the higher the thermal efficiency. So R123, R141b and R152a that have the highest values of dry, isentropic and wet fluids, respectively, have the lowest values of total exergy loss. As the pump temperature goes up, the total net power of the cycle increases which leads to the reduction of the total exergy loss of the cycle.

Figure 5 shows the effect of the evaporator pressure on first law efficiency of the combined cycle for dry, isentropic and wet working fluids, respectively. From figures, it can be seen that, the thermal efficiency of the combined cycle increases as the evaporator pressure rises and for all working fluids, there is an optimum value of evaporator pressure that maximizes the first law efficiency. It is due to the fact that, increasing evaporator pressure leads to the decrease in the mass flow rate of the ORC and the evaporator heat as well as the increase in inlet enthalpy of the ORC turbine. Furthermore, the varying evaporator pressure has no effect on the power generation of the GT-MHR cycle but the power generation in ORCs increases and therefore, leads to the increase of the combined cycle's total power. Among the studied dry and isentropic working fluids, R123 and R141b have the maximum values. For wet working fluids, the evaporator pressure range can be divided into two parts, less than 5000 kPa and higher than 5000 kPa. In the first part, R152a has the highest value but in the second part, R717 surpasses the other working fluids. The highest value of thermal efficiency id for R141b that is an isentropic fluid.

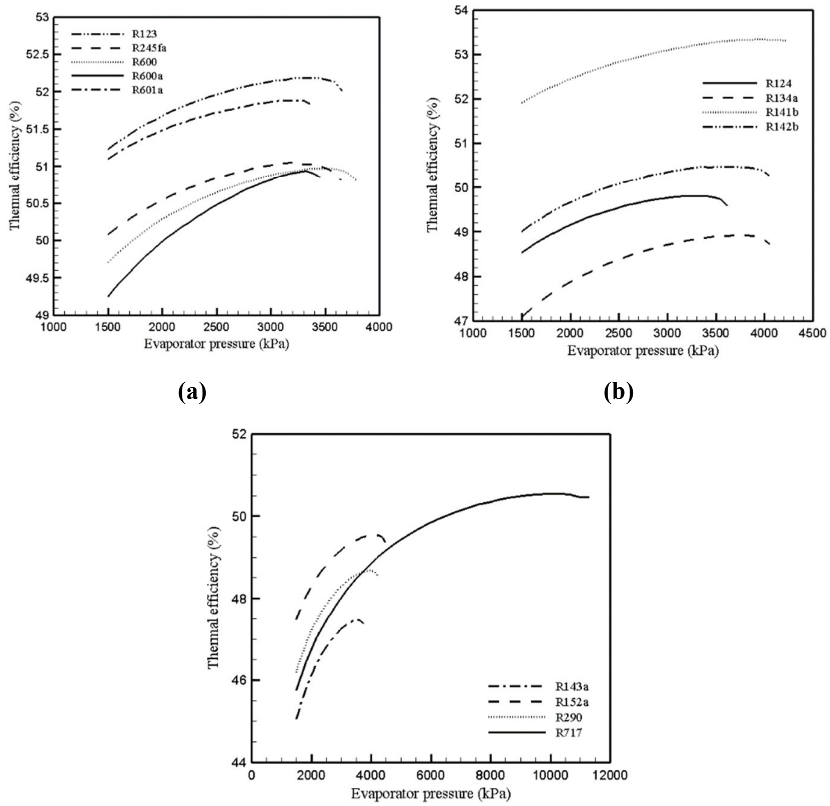


(a) (b)

(c)

Fig. 4. Effect of pump temperature increment on the total exergy loss (a) dry, (b) isentropic, (c) wet fluids.

Figure 6 illustrates total exergy loss of the combined cycle with varying evaporator pressure. It is shown that the total exergy loss decreases with the increment in the evaporator pressure for all working fluids. As the evaporator pressure is increased, since the temperature of the stream entering evaporator is constant, the difference between the evaporator temperature and the stream entering the evaporator is reduced. Although, evaporator pressure increasing leads to the increment in exergy loss of the precooler, intercooler, compressors and ORC turbines, the percentage of the reduction in the exergy loss of the evaporators is higher than that increase and therefore, decreases the total exergy loss of the combined cycle. The figures reveal that the total exergy loss decreases first and then decreases for each working fluid. There is an optimum value that minimizes the total exergy loss.



(a) **(b)**

(c)

Fig. 5. Effect of evaporator pressure increment on the thermal efficiency (a) dry, (b) isentropic, (c) wet fluids.

Optimization of the thermal efficiency according to the optimum evaporator pressure and optimum pressure ratio is shown in Table 7. The results indicate that at optimum conditions, R141b as an isentropic fluid, has the highest value of the thermal efficiency among the studied fluids. The reason of this predominance, is the maximum amount of the power that produces in ORCs. Moreover, it has the lowest amount of the produced heat rate. It also can be seen that, all of the dry fluids have the efficiency over 50%. One of the reasons why dry and isentropic fluids show better thermal efficiencies than wet fluids is because wet fluids may be at a two-phase state after the fluid goes through the turbine which leads to the droplets that may damage the turbine blades and therefore, for wet fluids, superheating is necessary in Organic Rankine cycle.

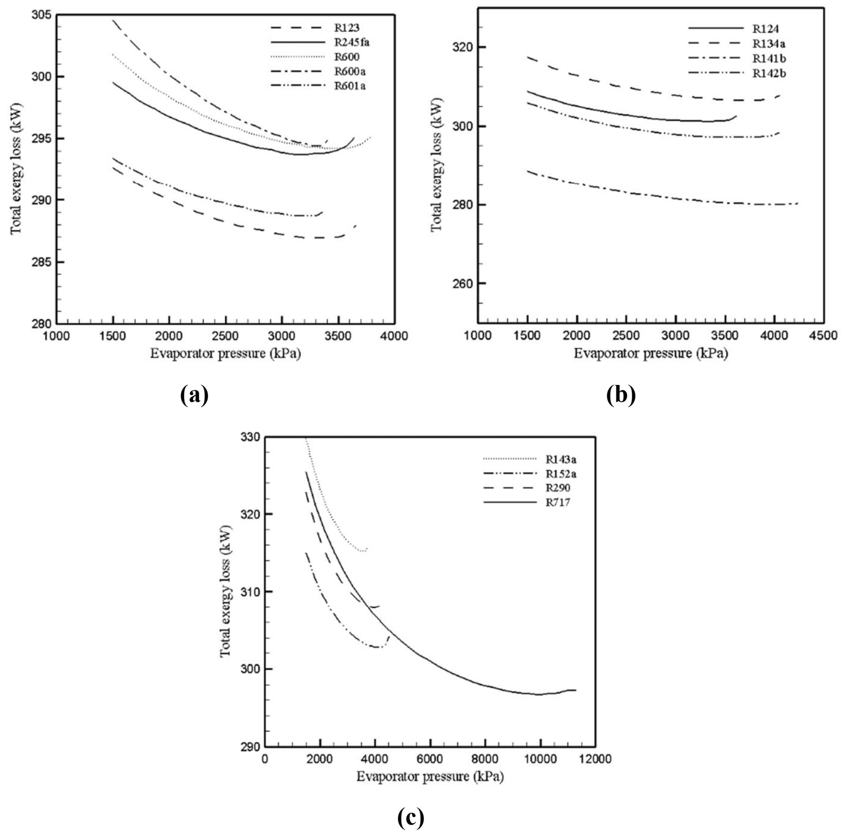


Fig. 6. Effect of evaporator pressure increment on the total exergy loss (a) dry, (b) isentropic, (c) wet fluids.

Table 7. Comparison of the studied working fluids under the optimum evaporator pressure and optimum pressure ratio.

Element	Unit	R143a	R152a	R290	R717	R124	R134a	R141b	R142b	R123	R245fa	R600	R600a	R601a
P_e	(kPa)	3517	4095	3948	9948	3254	3703	3957	3610	3362	3205	3474	3325	3224
$\dot{m}_{ORC,1}$	(kg/s)	1110	570.9	465.7	158.1	921.9	875.5	519.9	675.1	661.5	622.5	328.7	390.6	289.2
$\dot{m}_{ORC,2}$	(kg/s)	656.9	338.1	275.3	93.31	547.3	519.1	311.4	400.4	393.1	370.3	195	233.9	171.9
$\dot{W}_{net,ORC1}$	(MW)	9.827	17.66	14.42	21.48	18.64	15.32	31.77	21.15	27.57	23.3	23.03	22.76	26.43
$\dot{W}_{net,ORC2}$	(MW)	5.817	10.46	8.523	12.67	11.06	9.081	19.03	12.54	16.38	13.86	13.67	13.63	15.71
$\dot{Q}_{net}$	(MW)	1727	1714	1719	1706	1714	1719	1698	1710	1701	1708	1707	1712	1702
$\dot{W}_{net}$	(MW)	284.8	297.2	292.1	303.3	298.8	293.5	320	302.8	313.1	306.3	305.8	305.6	311.3
$\dot{Q}_{relative}$	-	0.2701	0.265	0.2662	0.2604	0.2657	0.2672	0.2618	0.2634	0.2602	0.2636	0.2625	0.268	0.2611
$\dot{W}_{relative}$	-	0.05494	0.09461	0.07854	0.1126	0.0994	0.08312	0.1587	0.1113	0.1404	0.1213	0.12	0.1191	0.1354
$\dot{E}_D$	(MW)	315.2	302.8	307.9	296.7	301.2	306.5	280	297.2	286.9	293.7	294.2	294.4	288.7
η_{th}	(%)	47.46	49.54	48.68	50.54	49.81	48.92	53.34	50.47	52.18	51.05	50.97	50.94	51.88
η_{exe}	(%)	47.46	49.54	48.68	50.54	49.81	48.92	53.34	50.47	52.18	51.05	50.97	50.94	51.88

6. Conclusion

In this study, the effect of the different working fluids including dry, isentropic and wet fluids on the performance of the combined turbine modular helium reactor (GT-MHR) and Organic Rankine Cycle (ORC) are studied. A computer code based on the EES software has been developed to analyze the combined cycle. Different parameters such as thermal efficiency, total exergy loss, net power generation is investigated and compared for all fluids. The main results obtained from this study are as follows. It should be noted that all calculations have been performed for an optimum pressure ratio in which thermal efficiency is maximum.

- Increasing pump temperature leads to the reduction of the total thermal efficiency of the combined cycle.
- Increasing pump temperature leads to the increment of the total exergy loss.
- Increase in evaporator pressure, first increases the thermal efficiency and then reduces it and causes an optimum pressure in which thermal efficiency is maximum.
- Increase in evaporator pressure leads to an optimum amount that minimizes the total exergy loss.
- In general, dry fluids show a higher thermal efficiency than isentropic and dry fluids, but the highest value is for R141b, which is an isentropic fluid.
- Wet fluids have the lowest values of thermal efficiencies.
- At the optimum point, R141b in isentropic fluids, R123 in dry fluids and R717 in wet fluids have the highest thermal efficiency values.

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