Performance Analysis of an organic Rankine Cycle applied on Marine Engine for Fourteen Working Fluid

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Abstract– Under the world-wide concern about the depletion of fossil fuels and different environmental issues, it is necessary to develop more techniques to enhance the efficiency of energy conversion of fuel and reduce the environmental global warming. Organic Rankine Cycle is an effective method for waste heat recovery from engines. In this study, the performance of an ORC operated by the exhaust of marine engine is numerically evaluated. Commercial software EES32 was used to calculate system performances under specified operating conditions with 14 different working fluids. Various system components were modeled. Different comparisons are carried out between different working fluids, according to their performance, system cost, volume and mass. Toluene was found to be the most promising working fluid with the highest performance indices regarding first low and second low efficiencies. It can produce 117.5 kW power at marine cruising speed with engine power output of 500 KW. The ORC saves 31,169 L of diesel fuel (US$24490) at the end of 1000 operating hours.

Keywords-- Organic Rankine cycle-Waste heat recovery-Marine Engine-Sizing-Cost

I. Introduction

Energy is one of the most significant concern worldwide. Cost , recourses, and environmental issues(air pollution, acid rain, and global warming) associated with energy receive great focus academically and politically . Fossil fuel is the foremost source of energy [1] . It is still used in Rankine cycles, gas turbines , piston engines, and many other industries.

Recently, Many researchers is paying attention to engines waste heat recovery systems (WHRS) motivated by potential emissions legislation and raise in fuel cost. Different technologies of waste heat recovery can be used onboard ships, which can be turbo charging of air into the engine, absorption refrigeration, thermoelectric generation, or combined power cycles [2]. Zhou et al. [3] reviewed waste heat recovery from the marine engine with highly efficient bottoming power cycles by studying Trade-offs between working fluid characteristics, cycle configuration, size, cost and WHR potential.

Most of the related work has focused on WHRS based on the Rankine cycle [4]–[6] because they result in an increase in the engine efficiency and a in the engine emissions [7]. The Organic Rankine Cycle (ORC) system was found to be the most suitable WHR method for two key advantages which are its simplicity and components availability [8]. Organic Rankine cycle is similar to the ordinary Rankine cycle except that it uses an organic fluid that has a low boiling point compared with water to recover waste heat from low temperature heat sources. Confined and small scale power production is becoming possible by ORC technology.

For the regaining and conversion of low-grade heat energy, ORC plays a major role for the simple, compact and low cost system components with small sizing and the properties of organic fluids that can exploit low and variable temperature heat sources [9]–[14].Therefore, this technology is excellent because to its low cost and outstanding thermal efficiency [15].

ORCs have been investigated in many previous works: Badr et al. [16], Gu et al. [17], Dai et al. [18], compared the performance of an ORC working with a number of contestant working fluids using straightforward thermodynamic models considering constant efficiencies of both pump and expander. They concluded that the cycle efficiency is very responsive to the evaporator pressure. Larjola [19] studied the use of an integrated high speed, oil-free turbogenerator-feed pump for a 100 kWeWHR ORC. Sophisticated cycle configurations have also been investigated: Gnutek et al. [20] proposed multiple pressure levels ORC cycle with sliding vane expansion machines using R123 in order to get the most out of the use of the heat source.

Wang et al [21] proposed a dual loop ORC system which essentially has two cascaded ORCs to recover energy from the engine's exhaust gases and coolant separately. Mahmoudi et al [22] concluded that the most effective thermo-physical properties of organic fluids affecting the cycle performance are critical state, sensible heat and ratio of vaporization latent heat. Thus, this different fluid mixing is important to improve the cycle performance.

Chen et al. [23] investigated 35 working fluids that are more suitable for the LTRC. They indicated the criteria in their work for choosing a working fluid for a LTRC. By examining the properties of these fluids, according to the possible operating conditions and the criteria mentioned in [23] many working fluids are excluded such as R170, R744, R41, R23, R116, R32, R125, R143 and R218 for low critical temperature because low critical temperature (especially, below 300 K) require a condensation temperature lower than the ambient temperature, and hence they have a difficult condensation and it will increase the pinch point temperature difference. HC-270, R1270, and Propyne are excluded because they have relatively low molecular weight implies a larger system size compared to those fluids with higher molecular weight

From previous literature, Toluene is a good choice for recovering high-temperature heat, but the ORC cycle efficiency decreases with low temperature exhaust gas [24][25]. Furthermore, Sanaye et al [26] compared toluene and water in standings of economical and thermal terms, where water serves to be better with respect to power output and payback period, while toluene shows better exergy destruction ratio and lower cost. Mashadi et al [27] applied the thermodynamic analysis to 19 different working fluids, appropriate matching conditions between the low-temperature Rankine cycle with the engine cooling system and the properties of the working fluid were examined. It was concluded that, Ammonia had the highest compatibility with the proposed system. According to Farhat et al [28]. Fig. 1 shows the distribution of working fluid study among 14 papers. Therefore, no single working fluid is best for all ORC's, as the selection of the proper working fluid requires consideration of operating conditions, environmental concerns and economic factors [29].

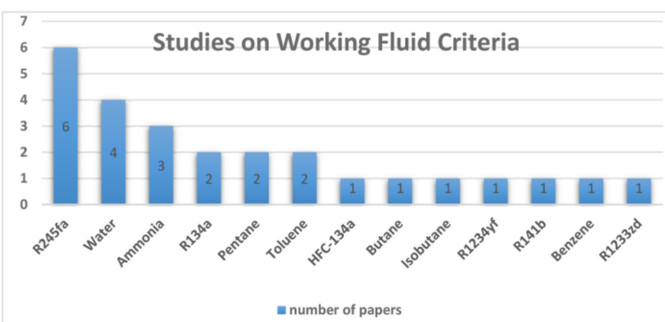

Fig. 1. Distribution of working fluid study among some papers [28]

From previous literature, it is many researches were devoted to compare the ORC performance with different working fluid. The main objective of this work is to analyze the first and second law performance of ORC for waste heat recovery of a marine diesel engine .

II. RECOVERY SYSTEM DESCRIPTION AND MODELLING

In this section, the System structure, modelling, boundary conditions, and working fluid of the studied Organic Rankine Cycle (ORC) are presented and explained

A. ORC Structure

A typical simple ORC used for waste heat recovery from exhaust gases, and a typical T-s diagram is shown in Fig. 2.

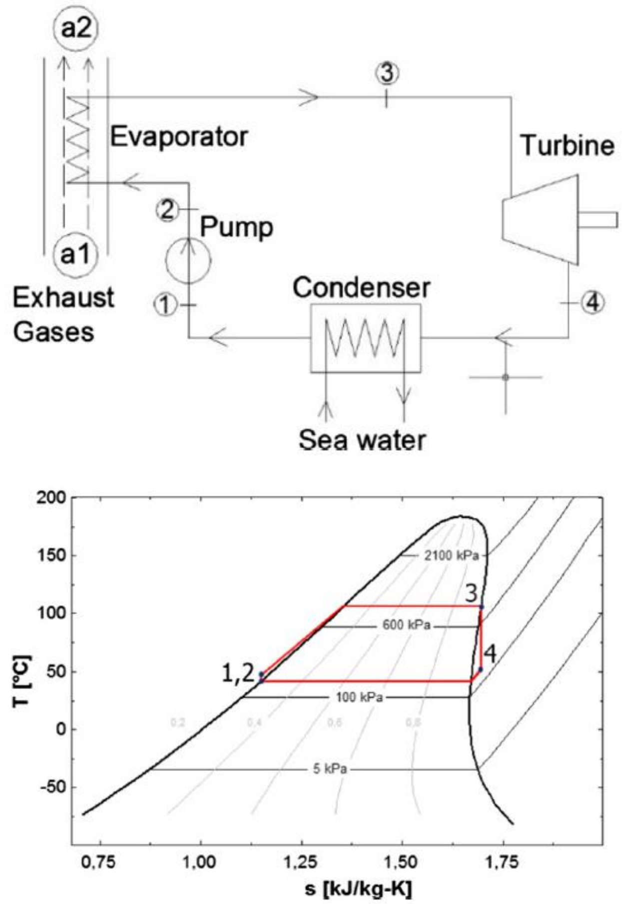

Fig. 2. The simple ideal organic Rankine cycle [1].

ORC consists of four main components. A heat exchanger (evaporator) for recuperating waste heat from exhaust gases, to heat the working fluid and convert it from compressed liquid to saturated vapour. A turbine for expansion of the working fluid from the evaporator pressure to the condenser pressure and producing mechanical work which is converted to electrical work at the generator connected to the turbine. A condenser for rejecting heat to the environment to satisfy the 2nd law of Thermodynamics such that some amount of heat must be rejected to surrounding environment to change phase of the working fluid to saturated liquid at an ideal Rankine cycle. A pump for increasing the pressure of the working fluid from condensation pressure to evaporator pressure.

TABLE I
Thermodynamic model of first and second laws [27].

Process&Equipment	Modelling overview	
1–4: Pump	$W_{P,rev} = \dot{m}_{RC}(h_{1,s} - h_4)$	(1)
	$W_{P,ac} = \dot{m}_{RC}(h_1 - h_4) = \eta_P W_{P,ac}$	(2)
	$\dot{I}_P = \dot{m}_{RC} T_{amb}(s_1 - s_4)$	(3)
1–2: evaporator	$Q_{Evap} = \dot{m}_{RC}(h_2 - h_1)$	(4)
	$\dot{m}_{RC} = \frac{\dot{m}_{cf} c_{cf}(T_{cf,in} - T_{cf,out})}{q_{Evap}}$	(5)
	$\dot{I}_{Evap} = T_{amb}\dot{m}_{RC}\left((s_2 - s_1) - \frac{(h_2 - h_1)}{T_w}\right)$	(6)
2–3: expander	$x = \frac{s_3 - s_{f3}}{s_{fg3}}$	(7)
	$W_{Exp,rev} = \dot{m}_{RC}(h_2 - h_{3,s})$	(8)
	$W_{Exp,ac} = \dot{m}_{RC}(h_2 - h_3) = \eta_{Exp} W_{Exp,rev}$	(9)
	$\dot{I}_{Exp} = T_{amb}\dot{m}_{RC}(s_3 - s_2)$	(10)
		(11)
3–4: condenser	$Q_{Cond} = \dot{m}_{RC}(h_4 - h_3)$	(12)
	$\dot{I}_{Cond} = T_{amb}\dot{m}_{RC}\left((s_4 - s_3) - \frac{(h_4 - h_3)}{T_{amb}}\right)$	(13)
$W_{net} = W_{Exp,ac} - W_{P,ac}$		(14)
$\eta_{1stl} = \frac{W_{net}}{Q_{Evap}}$		(15)
$\dot{I}_t = \dot{I}_P + \dot{I}_{Eva} + \dot{I}_{Exp} + \dot{I}_{Cond}$		
$\eta_{2ndl} = \frac{W_{Exp,ac}}{W_{Exp,ac} + \dot{I}_t}$		(16)

B. Thermodynamic analysis

The four components of the considered ORC are assumed to be steady flow steady flow system. By investigating the energy balance in each component of the cycle, the pump and the expander works, and the transferred heat through condenser and evaporator can be obtained as shown in table 1. From the perspective of the 2nd law of thermodynamics, by investigating the exergy destruction in each component of recovery system, it can be determined how much of the available energy has been recovered. The 1st and 2nd law of thermodynamics applied on each component are presented in Table 1. These equations were adopted from [27]).

A simple ORC with a regenerator and T-s diagram of the cycle is shown in Fig. 3. The Regenerator exchange heat between the compressed water prior to the evaporator (state 2 to 6) and the turbine exhaust steam prior to the condenser (state 4 to 5) For an ideal regenerator assuming no heat loss to the environment:

$$h_6 - h_2 = h_4 - h_5 \qquad (17)$$

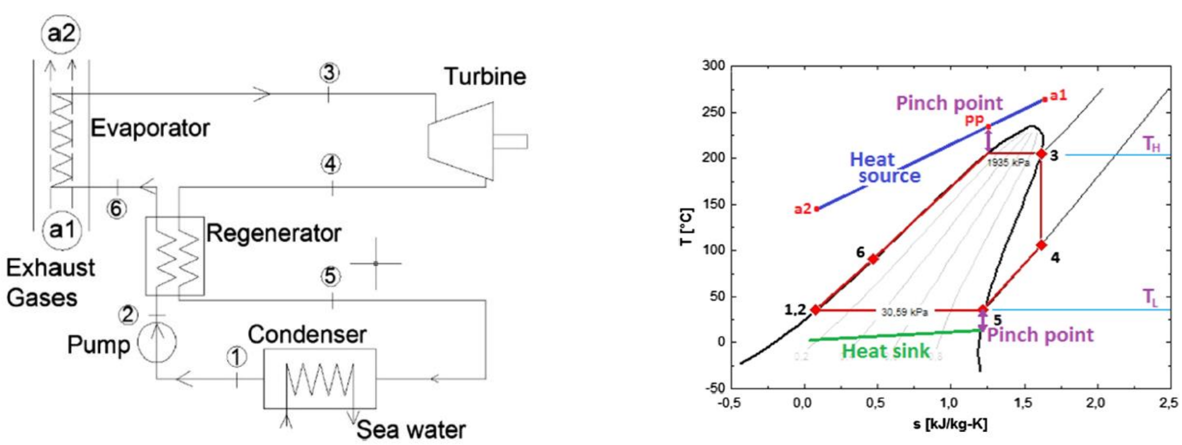

Fig. 3. Organic Rankine cycle with regenerator

Regenerator effectiveness is defined as $\varepsilon = (h_4 - h_5)/(h_4 - h_{p5,T2}) \qquad (18)$

The pinch point temperature (TPP) of exhaust gases shown in Fig. 3 can be calculated as follows:

$$m_{ex}C_{Pex}(TPP\text{-}T_{a2})_{air}=m_{OR}(h_{f,TH}\text{-}h_6) \quad (19)$$

where $h_{f,TH}$ is the enthalpy of the saturated ORC liquid at the temperature T_H, and T_{air},Tpp is the air temperature at the enthalpy of air, hpp.

In the heat transfer analysis of heat exchangers, it is convenient to express the heat transfer rate between the hot and cold streams as:

$$Q=UA_sLMTD \quad (20)$$

$$LMTD=(\Delta T_1\text{-}\ \Delta T_2)/\ln(\Delta T_1/\ \Delta T_2) \quad (21)$$

where As is the heat transfer area, U is the overall heat transfer coefficient, LMTDis log mean temperature difference between hot and cold fluid, and ΔT_1 and ΔT_2 are the temperature differences of the fluids at the inlet and the outlet, respectively.

C. Operating conditions

Quoilin et al [30] reported that the slope of the saturated vapor curve is much closer to vertical for organic fluids. As a consequence, the limitation of the vapor quality at the end of the expansion process disappears in an ORC cycle, and there is no need to superheat the vapor before the turbine inlet. However the working fluid quality is checked during calculations.

The evaporator temperature of the working fluid is determined during calculations such that the pinch point TPP should be within a certain limit. Ozdil et. al [31] found the exergy efficiency increased with the decrease of temperature at pinch point of heat source due to the drop of exergy destruction rate. In this work the pinch point temperature is determined to be minimum as possible while preserving the cycle efficiency.

The pump is used to control the working fluid mass flow rate. The electrical motor is connected to an inverter which allows varying the rotating speed. In positive- displacement pumps, the flow rate is roughly proportional to the rotating speed, while in centrifugal pumps this flow rate also depends on the pressure head (i.e. the difference between evaporating and condensing pressure). The pump efficiency is a crucial parameter in low temperature cycles. Few pump efficiencies are reported in the literature, and they are usually quite low for low capacity unit [31]. In this work the performance of the cycle was calculated for isentropic pumping and expansion.

The average electric power consumption of the case ship is about 500 kW while the average exhaust temperature for this power production has been measured approximately as 294 ^oC. Mass flow rate of the exhaust gases is calculated about 160 kg/min. The exit temperature of the exhaust gases from the evaporator has been selected as 158 ^oC as a design criterion [1].

The condenser temperature of the working fluid depends on the temperature of the heat sink, which is seawater for this case. According to Loydu's [32] Rules for the Classification of Naval Ships, the selection, layout and arrangement of all shipboard machinery, equipment and appliances should ensure faultless continuous operation under the seawater temperature of -2 ^oC to +32 ^oC [33]. Therefore, the condensation temperature of the working fluid has been assumed to be 35 ^oC for the analysis [1].

D. Working Fluid

The working fluids listed in Table 2 were selected for further examination. The calcification of the fluid as wet, isentropic or dry is carried out according to chen et al [23]

Table 2
Properties of the used working fluids

SHRAE No.	Name	Type	Molecular weight	TC [K]	PC[bar]	Vapor Cp [kJ/kgK]	Latent Heat [kj/kg]	$\xi=\frac{ds}{dT}$ [kj/kgK2]
R152a	1,1-Difluoroethane	wet	66.05	386.41	4.52	1036.52	249.67	-1.14
	Benzene	isentropic	78.11	562.05	4.89	1146.27	418.22	-0.7
	Toluene	isentropic	92.14	591.75	4.13	1223.9	399.52	-0.21
R245 fa	1,1,1,3,3-Pentafluoropropane	isentropic	134.05	427.2	3.64	980.9	177.08	0.19
R 227ea	1,1,1,2,3,3,3-Heptafluoropropane	isentropic	170.03	375.95	3.00	1013.00	97.14	0.76
R600a	Isobutane	dry	58.12	407.81	3.63	1981.42	303.44	1.03
R-c 318	Octafluorocyclobutane	dry	200.03	388.38	2.78	896.82	93.95	1.08
R123	2,2-Dichloro-1,1,1-trifluoroethane	isentropic	152.93	183.68	36.6	738.51	161.82	0.26
R141b	1,1-Dichloro-1-fluoroethane	isentropic	116.95	477.50	4.21	848.37	215.13	0.00
R601a	Isopentane	dry	72	460.35	3.37		343.14	
	n-Hexane	dry	86.17	507.82	3.04		335	
R 236fa	1,1,1,3,3,3-Hexafluoropropane	dry	152	398	3.2		160	
R717	Ammonia	wet	17.03	405.4	11.33	3730.71	1064.38	-10.48
R-601	n-pentane	dry	72.14	469.6	3.36	1824.12	349.00	1.51

III. RESULTS AND DISCUSSIONS

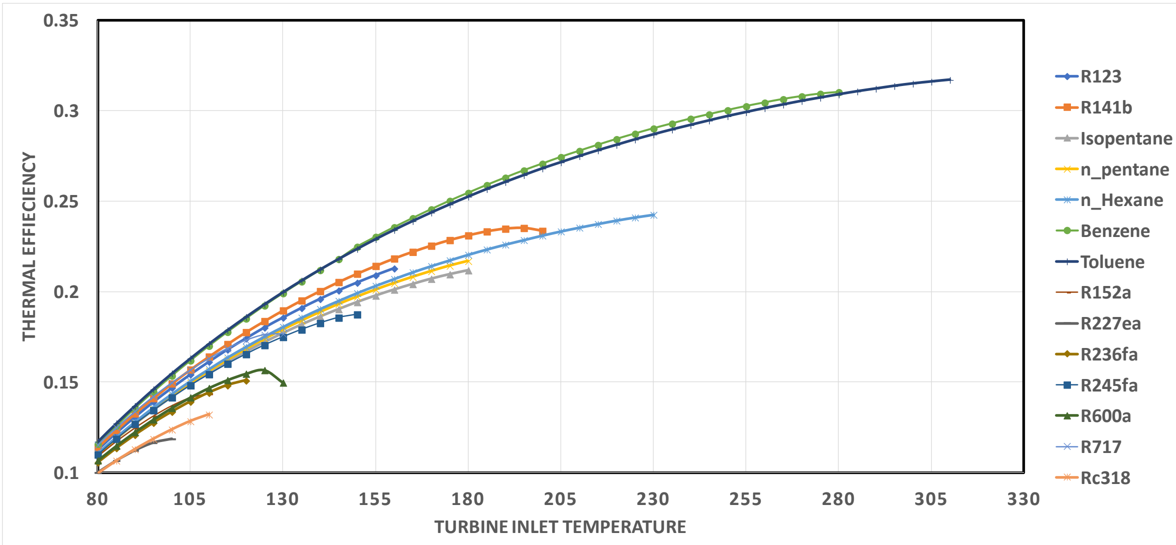

Fig. 4 The efficiency of the ORC without regeneration against turbine inlet temperature

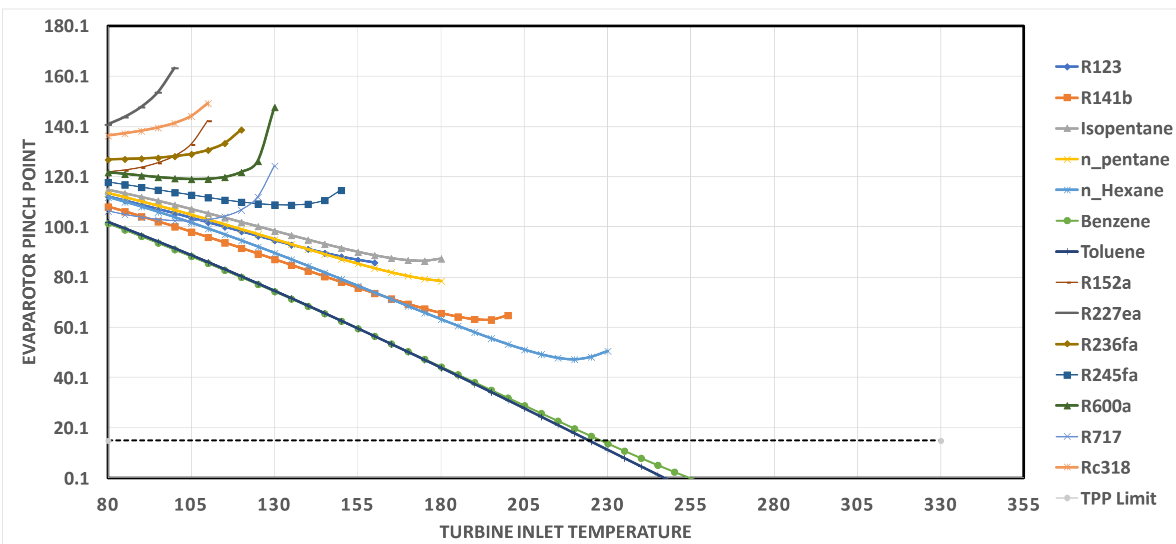

Fig. 5 The evaporator pinch point of the ORC without regeneration against turbine inlet temperature

The condensation temperature for the ORC fluid in the condenser is fixed to be 35 ^{o}C as a design parameter. So the efficiency of the cycle can be increased by increasing the evaporation temperature by increasing the outlet pressure of the pump. However evaporative temperature temperature of the ORC fluid in the evaporator is limited because it should be less the fluid critical temperature and it should attain the minimum evaporator pinch point.

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The efficiencies of the ORC's with and without regenerator are shown in Fig 6. It is observed from the Fig that the efficiencies of dry fluids increase dramatically as a regenerator is used. At relatively low evaporator temperatures, the efficiency of n-hexane with regenerator is highest. But as the evaporator temperature increases, since thebcritical temperature of toluene is highest among all, toluene with regenerator reaches maximum efficiency. So both the regenerator and the turbine inlet temperature affect the efficiency.

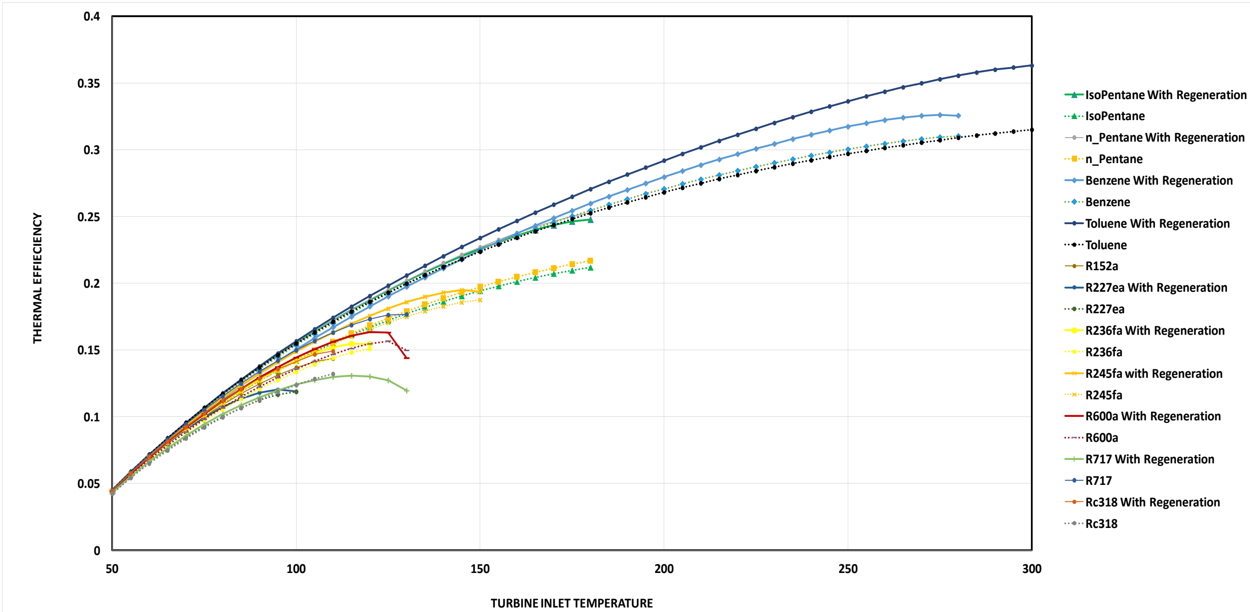

Fig. 6. The efficiency of the ORC with and without regeneration

The evaporator temperature and pressure can be determined according results shown in Figs 4, 5 and 6 such that it attains maximum efficiency with minimum pinch point. Table 3 shows the evaporative temperature $T_3$of the fourteen fluid. As T_3 is calculated the performance parameters are calculated as shown in table 3.

The average electric power consumption of the case ship is about 500 kW while the average exhaust temperature for this power production has been measured approximately as 294 ^{o}C. The thermal power absorbed by ORC working fluid from exhaust gases, which has a mass flow rate 160 kg/min, has been calculated as 372 kW for cruising conditions of the case naval ship. The power that the ORC system can produce with the working fluids is seen in Table 3 and Fig. 7. . Regeneration has not been calculated for R123 and R141b since both they are isentropic fluids. Maximum power that can be recovered from waste heat of exhaust gases and maximum cycle efficiency was calculated as 118 kW and 0.32, respectively with the fluid toluene using an ideal regenerator. Overall exergy efficiency and irreversibility are presented for the case ORC cycles in Table 3, Fig. 8, and Fig. 9. It is seen on the Figs that the cycle with toluene and regenerator has the maximum power, maximum exergy efficiency and minimum irreversibility among the selected case fluids.

The electric power output of the cycle with each fluid and regenerator is caculated. In this case the power of the combined system can be calculated by

$$Power_{\ combined\ System}=Power_{\ DE}+Power_{\ ORC}\ \ [kW] \qquad (22)$$

Where the diesel engine power ($Power_{\ DE)}$=500 kW

The efficiency of the combined system is calculated from

$$\eta_{\ combined\ System}= Power_{combined\ System}.3600/(g_e.\ Power_{\ DE}.Hl) \quad (23)$$

where g_e is the specific fuel consumption of the main diesel engine which is 0.287 kg/kW.hr. Hl is the lower heating value of the fuel which is 42,700 kJ/kg.

The fuel saving is calculated from the difference between the combined system efficiency and the diesel engine efficiency. This fuel saving can be represented in tons or in volume units Also the cost of fuel saving is calculated based on a price of 0.786 $/L of saved fuel.

For toluene with regeneration the ORC produces 117.5 kW. which increase the efficiency of the combined system to 0.363 compared to 0.294 of the diesel engine alone. The ORC cycle is used to generate power from waste heat, 31,200 L of diesel fuel (US$24,870) can be saved.

Table 3

The efficiency, power, overall exergy efficiency, and irreversibility of the ORC on cruising

Fluid	R123	R141b	Isopentane		n-Pentane		n-Hexane		Benzene		Toluene	
	T3=160 C	T3=195 C	T3=175 C		T3=180 C		T3=220 C		T3=240 C		T3=235 C	
	Without Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration
Efficiency	0.2129	0.2353	0.2436	0.2097	0.2531	0.217	0.302	0.2394	0.3081	0.2957	0.3158	0.2897
Power Kw	79.19	87.55	90.62	78.02	94.14	80.74	112.4	89.06	114.6	110	117.5	107.8
Exergy Efficiency	0.5374	0.5942	0.6151	0.5295	0.6389	0.5479	0.7625	0.6044	0.7778	0.7464	0.7974	0.7313
Volume of fuel saved per year L/1000 hr	22397	24410	25135	22112	25955	22774	30057	24767	30546	29536	31169	29044
Cost of fuel saved $/1000 hr	17598	19179	19749	17374	20393	17894	23616	19460	24000	23207	24490	22820
Irreversability	68.16	59.79	56.72	69.32	53.21	66.61	34.99	58.28	32.74	37.36	29.85	39.59

Fluid	R152a	R227ea		R236Fa		R245fa		R600a		R717		Rc318	
	T3=70 C	T3=75 C		T3=50 C		T3=135 C		T3=105 C		T3=105 C		T3=50 C	
	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration	With Regeneration	Without Regeneration
Efficiency	0.0905	0.00936	0.09265	0.1019	0.04343	0.1859	0.1791	0.1508	0.1416	0.1199	0.1568	0.04445	0.04246
Power Kw	33.67	36.96	34.47	37.89	16.16	69.14	66.61	56.09	52.67	44.59	58.33	16.53	15.79
Exergy Efficiency	0.2285	0.2509	0.2339	0.2571	0.1097	0.4692	0.4521	0.3807	0.3575	0.3026	0.3959	0.1122	0.1072
Volume of fuel saved per year L/year	10334	11276	10564	11539	5128	19900	19258	16524	15612	13412	17115	5244	5016
Cost of fuel saved $/year	8120	8860	8300	9066	4029	15636	15132	12983	12266	10538	13447	4120	3941
Irreversability	113.7	110.4	112.9	109.5	131.2	78.21	80.73	91.25	94.67	102.8	89.01	130.8	131.5

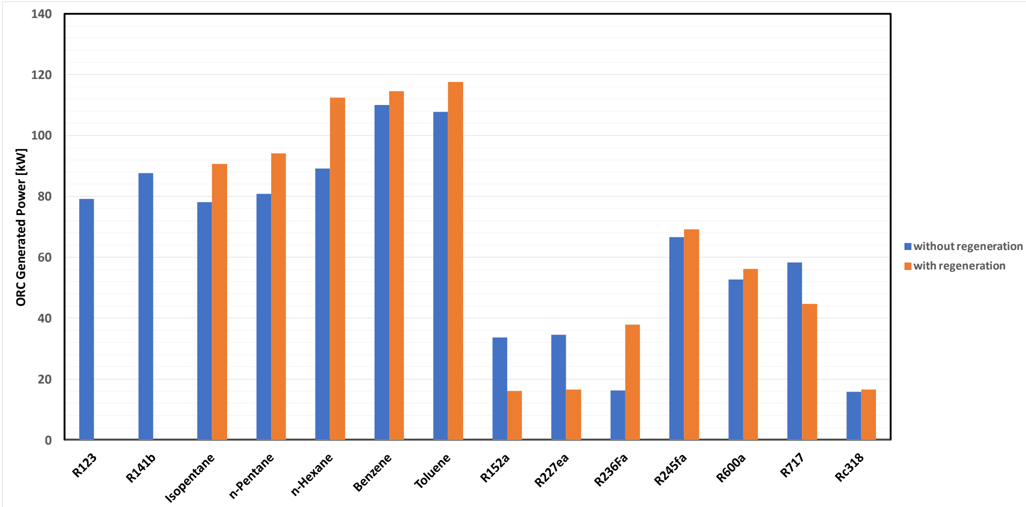

Fig. 7. The power that can be generated from ORC on cruising

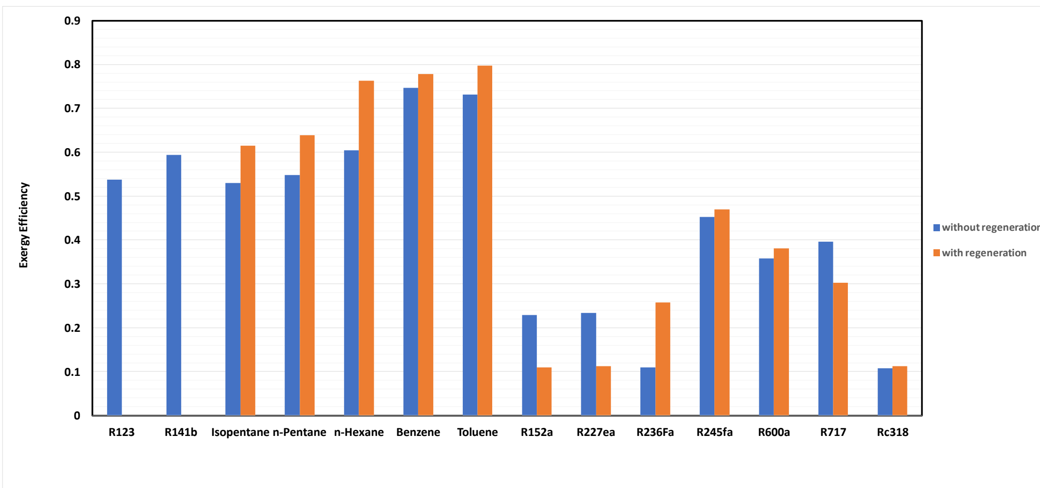

Fig. 8 Exergy efficiency of the cycles

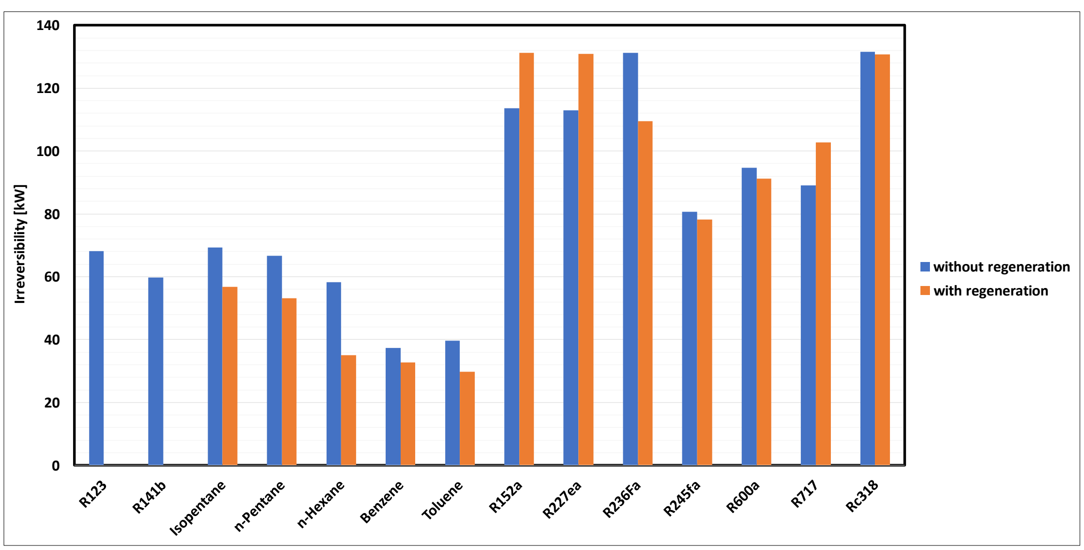

Fig. 9. Irreversibility of the cycles.

IV. Conclusions

This work investigated the first and second low efficiencies of fourteen working fluids as a waste heat recovery of a marine Diesel engine operating at 500kW. Specific conclusions can be Summarized as follows:

1. The study estimated the efficiency between 0.04246 for Rc318 without regeneration to 0.3158 for toluene with regeneration depending on the type of the fluid, the inlet temperature of the turbine, and regenerator.
2. The powers of the ideal ORC's were calculated and compared. The results show that a net power can be obtained from the cycle between 15.8 for Rc318 and 117.6 kW for toluene with regeneration depending on the selected fluids and the operating conditions.
3. Among the selected working fluids, the ORC cycle working with toluene with regenerator was found to have the maximum generated power and exergy efficiency.
4. Assuming an isentropic efficiency for both the turbine and the pump of the case ORC equal to 1 and neglecting the losses at the ORC electric generator, the electric power output of the cycle with toluene and regenerator becomes 117.5 kW. In this case the power of the combined system is calculated as 617.5 kW while the efficiency of the combined system is equal to about 0.363
5. For toluene as the working fluid of the ORC cycle is used to generate power from waste heat,31169 L of diesel fuel US$24,490) can be saved per 1000 operating hours.

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