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Thermodynamic Analysis of ORC Configurations Used For WHR from a Turbocharged diesel engine

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Abstract

This paper illustrates a comparison between different solutions for waste heat recovery from a specific internal combustion engine using an Organic Rankine Cycle (ORC). The purpose of the present work is to find the cycle configuration and the fluid that can provide the maximum mechanical power for given conditions of waste heat recovery (WHR) from flue gas and motor cooling water of a Turbocharged Diesel Engine Electric Generator. The thermodynamic analysis is conducted for ten working fluids from different chemical classes: hydrofluorocarbons (HFC) and the new subclass hydrofluoroolefins (HFO), hydrocarbons (HC), siloxanes and alcohols applied on six ORC configurations, three conventional ones (basic ORC, regenerative ORC and preheater ORC), a regenerative dual-loop ORC (DORC), a mixture of two traditional ORC (one used for WHR from the motor cooling water and the other used for flue gas WHR) and a dual ORC but with a common condenser configuration. One of the working fluids investigated is a new HFO, named R1336mzz, which has very low GWP, zero ODP and good safety properties. This novel fluid also shows good results in our investigation. Future work and development perspectives are discussed.

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Keywords: Organic Rankine Cycle; Waste heat recovery; Working fluid; Diesel Engine; Thermodynamic analysis

1. Introduction

The '70s oil crisis and increasing level of concern about environmental pollution led to the search for alternatives to fossil fuels or equipment that use fossil fuels in all areas [1]. Waste heat recovery and increasing energy efficiency for installations that are using electrical energy or fossil fuels were also studied.

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Nomenclature

WHR	waste heat recovery
ORC	Organic Rankine Cycle
ICE	internal combustion engine
ODP	ozone depletion potential
GWP	global worming potential
t_{cr}	critical temperature of a fluid
p_{cr}	critical pressure of a fluid
$\dot{Q}_o$	evaporation heat flow
$\dot{Q}_{PH}$	preheating heat flow
$\dot{m}_{fl}$	working fluid mass flow rate
h_i	enthalpy value of a state point
$\dot{W}_D$	power output
t_{gae}	evaporator outlet flue gas temperature
t_o	evaporation temperature
η_D	expander isentropic efficiency
c_{ga}	flue gas specific heat
c_w	ICE cooling water specific heat
Δt_{ga}	difference between the inlet and outlet flue gas temperatures
Δh_i	enthalpy difference between two state points
Δt_w	difference between the inlet and outlet ICE cooling water temperatures

One solution to relieve the energy deficit and environmental pollution problems is the efficient use of thermal energy at low and medium temperature, which is found in large quantities [3] in various fields: petrochemical industry, steel industry, food industry, automotive industry, pharmaceutical industry.

While there is no single technique that provides excellent results for all heat sources regardless of the temperature or heat available, literature [1,2,3,12] shows that ORC systems can provide an attractive combination of efficiency and accessibility for heat recovery. Other systems used to recover waste heat or heat from renewable sources are: Stirling, Kalina, Bryton cycles, electrical effects, absorption or adsorption refrigeration or heat pumps.

From about 20 years ORC systems are intensively studied and analysed. Numerous articles, studies and books are published that present different ORC: configurations [5,6,8], working fluids [2,4,7,8,11] expanders [2,11] or mathematical models for analysis and optimization [8,9,10]. The present conclusion is that for each heat source an individual analysis is required to determine the optimal working fluid and configuration.

The present paper seeks to establish the ORC configuration and working fluid for maximizing the production of mechanical energy using waste heat from the diesel engine model 4TNV98TGGEHR manufactured by Yanmar [27].

2. Waste heat source parameters

The work is based on a hybrid micro-cogeneration group which includes an electric generator based on a 40 kW Turbocharged Diesel Engine, flue gas and the engine jacket cooling water were considered as possible ORC heat source.

From past experimental research on this engine was acquired the data presented in table 1.

Table 1. Heat source parameters.

Engine type	Diesel[27]
Engine model	4TNV98TGGEHR[27]
Manufacturer	YANMAR[27]
Engine maximal mechanical power	40 kW[27]
Flue gas temperature	480 °C[27]
Flue gas mass flow	0,05 kg/s[28]
Engine cooling water temperature	75 °C[28]
Engine cooling water mass flow	0,3 kg/s[28]

3. ORC configurations

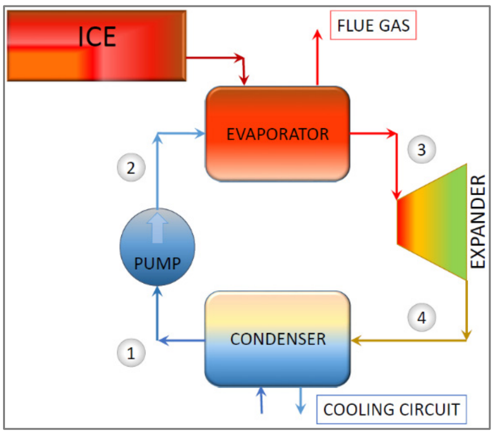
Six ORC configurations suitable for using waste heat from an internal combustion engine are presented in Fig.1 through their schemes and the temperature entropy diagrams.

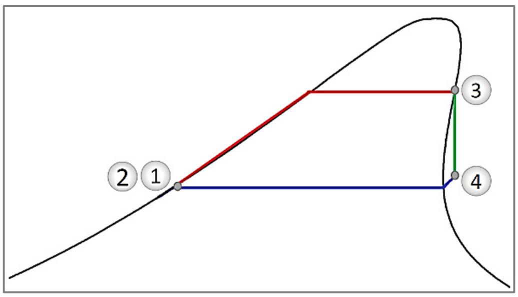
3.1. Basic ORC – BORC

A system built on the principle of an organic Rankine cycle has four main components: evaporator, expander, condenser and pump. BORC represents the elementary configuration for an Organic Rankine Cycle. The working fluid takes the heat flow from the heat source in the evaporator, evaporates and the high pressure vapour are then expanded in the expander or turbine to a lower pressure thereby generating mechanical work, low pressure vapour output enters the condenser gives up heat to the cooling fluid and the resulting liquid is pumped from the condensing pressure to the evaporation pressure thus rerunning the cycle, Fig.1 (a).

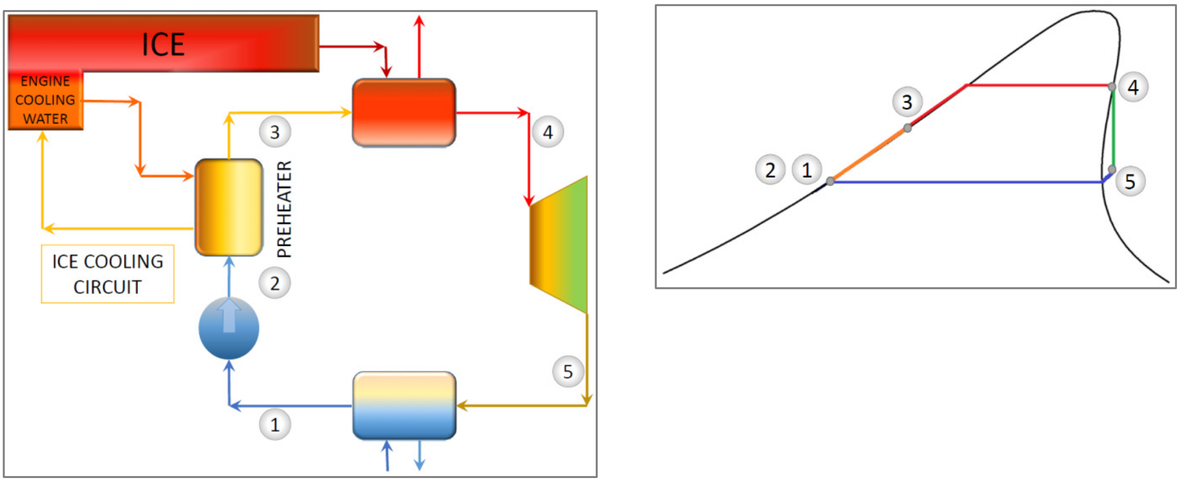
3.2. Preheater ORC – PHORC

For this configuration a heat exchanger between the pump and the evaporator it's used to preheat the liquid by means of heat from the engine cooling water circuit. This is an improved system because it's using also part of the waste heat from the cooling circuit together with the waste heat from the exhaust gas, Fig.1 (b).

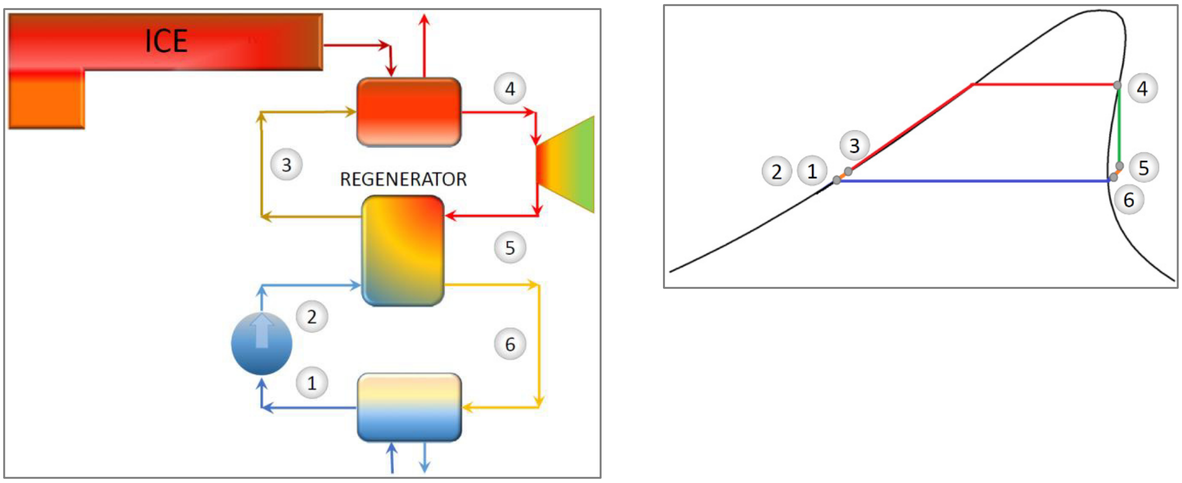




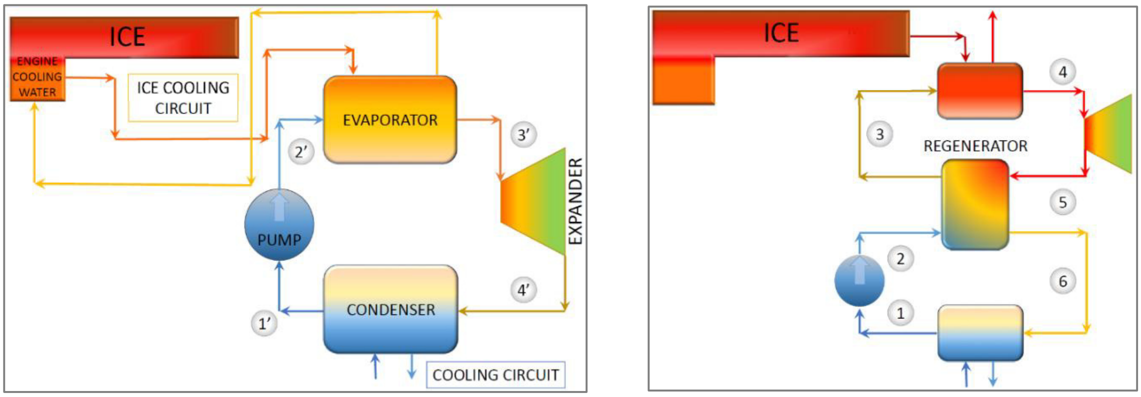
(a)


ICE
ENGINE COOLING WATER
ICE COOLING CIRCUIT
PREHEATER
1
2
3
4
5

(b)


ICE
REGENERATOR
1
2
3
4
5
6

(c)


ICE
ENGINE COOLING WATER
ICE COOLING CIRCUIT
EVAPORATOR
EXPANDER
PUMP
CONDENSER
COOLING CIRCUIT
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4'
ICE
REGENERATOR
1
2
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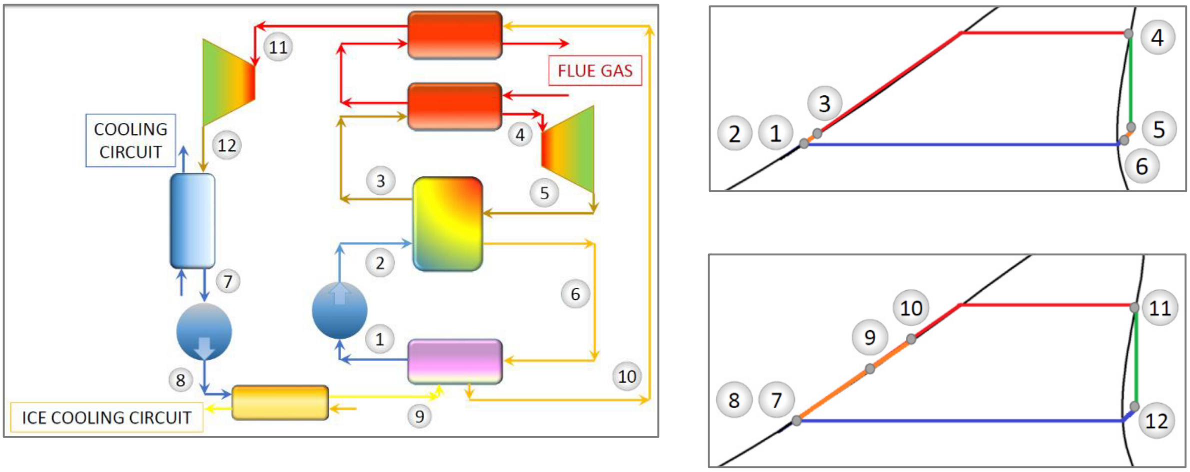
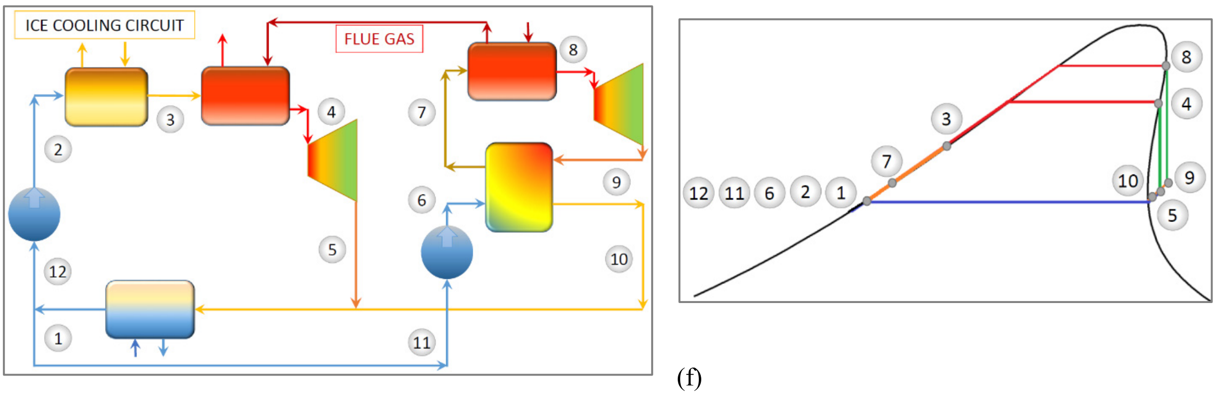
6

(d)



(e)



(f)

Fig. 1. ORC configuration schemes and the temperature-entropy diagrams: (a) BORC; (b) PHORC; (c) RORC; (d) 2ORC; (e) DORC; (f) DRPHORC.

3.3. Regenerative ORC – RORC

Similar to the preheater ORC it also adds a heat exchanger to preheat the liquid before the evaporator but it uses heat from the low pressure vapour at the expander outlet which will be cooled before the condenser inlet, reducing as well the condenser load, Fig.1 (c).

3.4. Two complementary ORC – 2ORC

Another solution but much more expansive is to use two ORC systems one RORC configuration for the flue gas and another BORC system for WHR from the ICE cooling circuit, Fig.1 (d). This way all heat available at the engine water cooling circuit can be used unlike in PHORC where because of the need to match the working fluid flow in the evaporator with the preheater flow only a part of the ICE cooling circuit heat can be used.

3.5. Double ORC – DORC

This cycle proposed by Gequn Shu et al. [14] uses also two ORC in a complex configuration where the flue gas enters first in the high temperature ORC and after in the low temperature ORC, Fig.1 (e). In the low temperature ORC part of the engine coolant heat and the condensing heat from the high temperature ORC are used to preheat the liquid. This system has a higher investment cost because of the two expanders, two pumps and the six heat exchangers but it's better matching the evaporation temperature with the flue gas temperature.

3.6. Double regenerative & preheater ORC – PHRDORC

This system it's using also two ORC to better match the flue gas temperature with the evaporation temperature but with a common condenser, Fig.1 (f).

4. Working fluid

Because of the many requirements for the working fluid: good thermo-physical properties, environment friendly [13], thermal stability, compatibility with materials, non-toxic and non-flammable, low cost, there isn't a general solution for all ORC applications regardless of heat source features. One of the most important classification criteria for the working fluids is the slope of the vapour saturation curve in the temperature – entropy diagram. If the slope is negative the fluid is wet, if the slope is positive than the working fluid is dry or if the slope it's vertical the fluid is isentropic. For wet fluids it is necessary to superheat the vapours at the evaporator outlet to avoid condensation in the expander.

For determining the ORC configuration and the working fluid that can reach the maxim production of mechanical energy of an ORC system coupled with the given diesel engine, 8 working fluids have been chosen (table 2), the selection criteria are:

- the working fluid should not affect the ozone layer, null ozone depletion potential (ODP);
- two of them can be used in the field of subcritical vapor at a temperature of 200 °C to permit analysis of the two configurations DORC and DRPHORC;
- at least one of them has the global worming potential (GWP) value below 150 [15] to comply with the EU directive on greenhouse gases [15];
- at least one working fluids considered to be a new agent that has not previously been the subject of a special study, to meet this criterion R1336mzz was chosen, a recent developed fluid for which the production on a commercial scale was announced for 2014;
- the selection to include substances belonging to various chemical classes: hydrocarbons (n-pentane, n-dodecane, toluene), hydrofluorocarbons HFC (R245fa, SES36), hydrofluoroolefins HFO (R1336mzz), alcohols (ethanol), siloxanes (MM - hexamethyldisiloxane);
- for this analysis dry and isentropic fluids are preferred because they require no superheating, exception made only by ethanol (because it is an alcohol and wanted the study to include this chemical class).

Table 2. Selected working fluids properties.

Working fluid	Slope	t_{cr}[°C]	p_{cr}[bar]	ODP	GWP_{100}	Safety class
R245fa	Isentropic[16]	154[16]	36,51[16]	0	1030[15]	A1[23]
SES36	Dry[19]	177,55[18]	28,49[18]	0[17]	3710[17]	A1[17]
R1336mzz	Dry [20]	171,3[20]	29[20]	0[20]	9[15]	A1[20]
MM	Dry [16]	245,5[16]]	19,39[16]	0	-	slight toxic, serious flammable[24]
Ethanol	Wet[10]	240,8[16]	61,48[16]	0	1[21]	severe flammable[26]
Toluene	Isentropic[10]	318,6[16]	41,26[16]	0	3[21]	moderate toxic, serious flammable[40]
n-Pentane	Dry [10]	196,5[16]	33,64[16]	0	5[15]	A3[39]
n-Dodecane	Dry [10]	385[16]	18,17[16]	0	2[21]	moderate toxic, moderate flammable[25]

5. Mathematic model

The equations used in the thermodynamic analysis for each configuration are shown in table 3, along with the assumptions and limitations that were taken into account.

Table 3. Equation used for the thermodynamic analysis.

Parameter	ORC type	Equation
Inlet heat flow $\dot{Q}_i$	BORC	$\dot{Q}_o = \dot{m}_{fl} \cdot (h_3 - h_2)$
	RORC	$\dot{Q}_o = \dot{m}_{fl} \cdot (h_4 - h_3)$
	PHORC	$\dot{Q}_o = \dot{m}_{fl} \cdot (h_4 - h_3)$
		$\dot{Q}_{PH} = \dot{m}_{fl} \cdot (h_3 - h_2)$
	2ORC	$\dot{Q}_{o1} = \dot{m}_{fl1} \cdot (h_4 - h_3)$
		$\dot{Q}_{o2} = \dot{m}_{fl2} \cdot (h_{3'} - h_{2'})$
	DORC	$\dot{Q}_{o1} = \dot{m}_{fl1} \cdot (h_4 - h_3)$
		$\dot{Q}_{o2} = \dot{m}_{fl2} \cdot (h_{11} - h_{10})$
		$\dot{Q}_{PH} = \dot{m}_{fl2} \cdot (h_9 - h_8)$
	DRPHORC	$\dot{Q}_{o1} = \dot{m}_{fl1} \cdot (h_8 - h_7)$
		$\dot{Q}_{o2} = \dot{m}_{fl2} \cdot (h_4 - h_3)$
		$\dot{Q}_{PH} = \dot{m}_{fl2} \cdot (h_3 - h_2)$
Expander work $\dot{W}_D$	BORC	$\dot{W}_D = \dot{m}_{fl} \cdot (h_3 - h_4)$
	RORC	$\dot{W}_D = \dot{m}_{fl} \cdot (h_4 - h_5)$
	PHORC	$\dot{W}_D = \dot{m}_{fl} \cdot (h_4 - h_5)$
	2ORC	$\dot{W}_{D1} = \dot{m}_{fl1} \cdot (h_4 - h_5)$
		$\dot{W}_{D2} = \dot{m}_{fl2} \cdot (h_{3'} - h_{4'})$
	DORC	$\dot{W}_{D1} = \dot{m}_{fl1} \cdot (h_4 - h_5)$
		$\dot{W}_{D2} = \dot{m}_{fl2} \cdot (h_{11} - h_{12})$
	PHRDORC	$\dot{W}_{D1} = \dot{m}_{fl1} \cdot (h_8 - h_9)$
		$\dot{W}_{D2} = \dot{m}_{fl2} \cdot (h_4 - h_5)$
Limitations:	$t_{cr} > t_o$; $t_{gae} = 130$°C	
Assumptions:	$\eta_D = 100\%$; $c_{ga} = 1{,}14$[kJ/kg · K]; $c_w = 4{,}2$[kJ/kg · K] [22]; the working fluid state at the evaporator outlet is on the vapors saturation curve and at the condenser outlet is on the saturated liquid curve; pressure loses in the system are neglected; pump work is neglected	

Thermodynamic properties of analysed working fluids were acquired from the database of Engineering Equation Solver software [16], except in the case of R1336mzz for which there were determined using log p-h diagram from reference [20]. For the purpose of comparing in similar conditions the 6 configurations and 8 working fluids the condensing temperature was set at 30°C except for the high temperature stage of DORC where the condensing temperature is 100°C to permit the cooling of the condenser with the preheated fluid from the low temperature stage. For the same reasons the evaporation temperature was set at 120°C except for the high temperature stage of DORC and PHRDORC where the evaporation temperature is 200°C and for the second cycle of 2ORC where the heat source is the ICE cooling water with a temperature of 75°C and the evaporation temperature was set 50°C.

Working fluid flow $\dot{m}_{fl}$ was calculated from the heat balance applied to the evaporator:

$$\dot{m}_{fl} = \frac{\dot{m}_{ga} \cdot c_{ga} \cdot \Delta t_{ga}}{\Delta h} \quad (1)$$

ICE cooling circuit water flow that can be used for preheating $\dot{m}_w$ was calculated from the heat balance applied to the preheater:

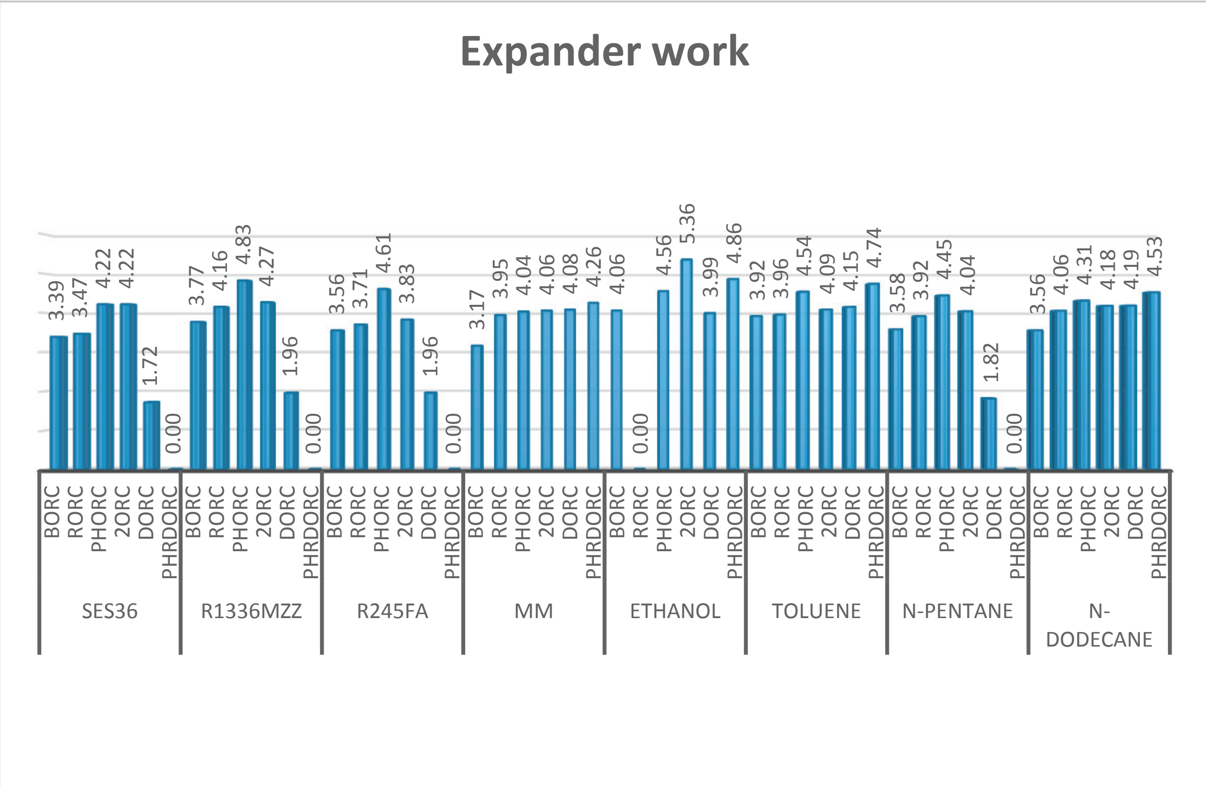
$$\dot{m}_w = \frac{\dot{m}_{fl} \cdot \Delta h}{c_w \cdot \Delta t_w} \tag{2}$$

6. Results

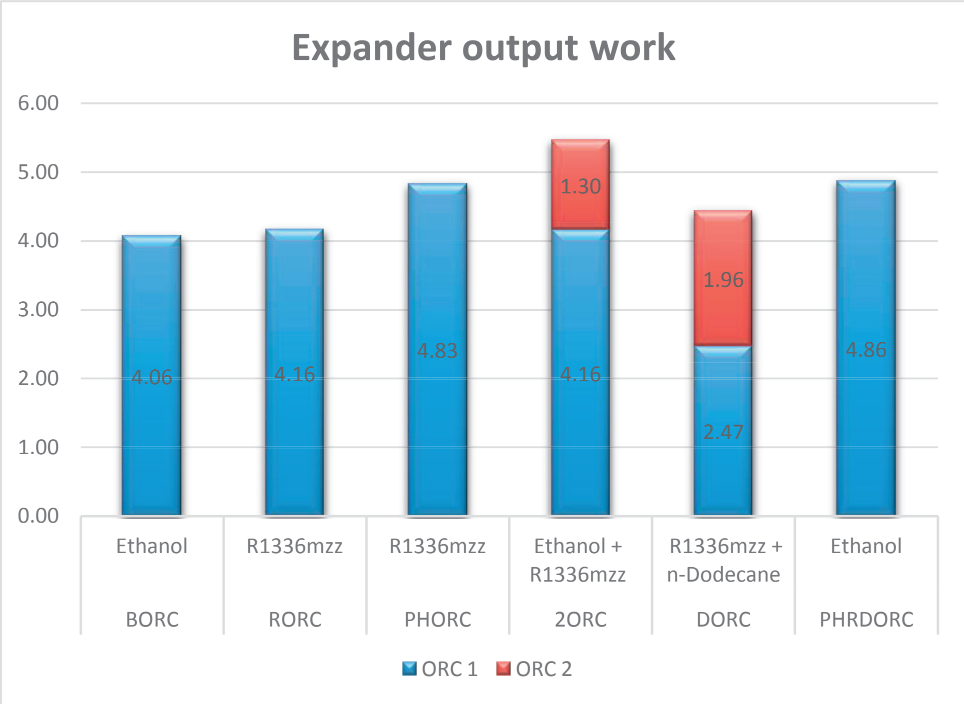
Mathematical model equations for each configuration and for each working were used to create a program in Engineering Equation Solver software [16], values are presented in Fig. 2(a).

For the PHRDORC the working fluids with the critical temperatures smaller than the second stage evaporation temperature weren't evaluated and show the value zero in Fig.2 (a), for the same fluids the work output values for DORC configuration were evaluated only for the low temperature stage because of this SES36, R1336mzz and R245fa have much smaller expander work for DORC system compared with the rest of the fluids. In RORC configuration ethanol which is a wet fluid couldn't be evaluated.

Fig.2 (b) shows the maximum work value for each configuration, in the case of 2ORC and DORC for each cycle was selected the fluid with the highest value of expander work output.



(a)



(b)

Fig. 2. (a) expander work output values; (b) highest value of expander work output for each configuration.

Conclusions

The highest value of work output resulted for 2ORC configuration in the case of using R1336mzz in the low temperature cycle and ethanol in the high temperature one. With approximately 11% less expander work output is the PHRDORC system using ethanol and very close with less than 12% difference is PHORC configuration for R1366mzz as working fluid. The preheater cycle shows very good performances with most of the analysed working fluids and also has considerable lower investment cost compared with the more complex cycles 2ORC, DORC and PHRDORC.

The good results achieved with the new HFO R1336mzz and also its positive safety and environmental characteristics are promising for the future of this working fluid in ORC development.

The next step in the research is to set up an experimental ORC system based on the optimal configuration regarding work output revealed by the present thermodynamic analysis and the investment cost which is the PHORC. Based on the experimental results the development of an optimization model for different ICE working conditions is intended.

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