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A case study of working fluid selection for a small-scale waste heat recovery ORC system

PIOTR KLIMASZEWSKI*
DAWID ZANIEWSKI
ŁUKASZ WITANOWSKI
TOMASZ SUCHOCKI
PIOTR KLONOWICZ
PIOTR LAMPART

Institute of Fluid Flow Machinery Polish Academy of Sciences,
Centre of Heat and Power Engineering, Turbine Department, Fiszera 14,
80-231 Gdańsk, Poland

Abstract The paper illustrates a case study of fluid selection for an internal combustion engine heat recovery organic Rankine cycle (ORC) system having the net power of about 30 kW. Various criteria of fluid selection are discussed. Particular attention is paid to thermodynamic performance of the system and human safety. The selection of working fluid for the ORC system has a large impact on the next steps of the design process, i.e., the working substance affects the turbine design and the size and type of heat exchangers. The final choice is usually a compromise between thermodynamic performance, safety and impact on natural environment. The most important parameters in thermodynamic analysis include calculations of net generated power and ORC cycle efficiency. Some level of toxicity and flammability can be accepted only if the leakages are very low. The fluid thermal stability level has to be taken into account too. The economy is a key aspect from the commercial point of view and that includes not only the fluid cost but also other costs which are the consequence of particular fluid selection. The paper discusses various configurations of the ORC system – with and without a regenerator and with direct or indirect evaporation. The selected working fluids for the considered particular power plant include toluene, DMC (dimethyl carbonate) and MM (hexamethyldisiloxane). Their advantages and disadvantages are outlined.

*Corresponding Author. Email: pklimaszewski@imp.gda.pl

Keywords: Waste heat recovery; Organic Rankine cycle, ORC fluids; Heat exchangers; Turboexpanders

Nomenclature

h	–	specific enthalpy, kJ/kgK
h_1	–	enthalpy of vapour of working fluid, kJ/kgK
h_2	–	enthalpy of vapour at turbine outlet, kJ/kgK
h_3	–	enthalpy of fluid at condenser outlet, kJ/kgK
h_4	–	enthalpy of pressure liquid, kJ/kgK
$\dot{m}$	–	mass flow rate, kg/s
P	–	power, W

Greek symbols

η_p	–	overall pump efficiency
η_T	–	overall turbine efficiency

Subscripts

s	–	exhaust gas
m	–	thermal oil
c	–	cooling water
p	–	pump
T	–	turbine
sr	–	heat source

1 Introduction

Recent years have been a period of dynamic growth in environmentally friendly power systems. The organic Rankine cycle (ORC) technology definitely fits in this trend very well thanks to its potential of improving energy efficiency through waste heat recovery. The application of an ORC module usually improves the industrial economy. This effect is often possible to achieve through subsidiary legislations which favour energy efficiency. The scope of application of ORC is relatively wide as it can be applied to a number of different heat sources with various parameters [1].

There are a number of available fluids that can be applied as working media in ORC systems. The literature shows analysis for different fluids, a few of them were selected as the best candidates for the mentioned application. Selection of working fluid for a 2 kW solar ORC power plant was considered by Tchanche *et al.* [2]. Authors chose 20 fluids as potential candidates. It is concluded that the most suitable fluids are R134a, R152a, R600, R600a, and R290. Mikielewicz and Mikielewicz [3] investigated 20 fluids for small scale micro combined heat and power units. They chose ethanol, R123 and R141b as the most suitable for these applications. The paper describes a postulated thermodynamic criterion helpful in preliminary fluid selection and presents main parameters of cycle working for different fluids. Publication by Bandean *et al.* [4] presents the problem of selection of working fluids with several aspects: thermophysical characteristics, environmental characteristics, toxicity classification, flammability classification. Authors create a special database with an implemented algorithm for selection of fluids for use in organic Rankine cycle.

A number of fluid selection procedures and methods for ORC systems have been proposed and described in the literature. One of them is optimization of boiling temperature, Mikielewicz *et al.* [5]. In this paper authors optimized three working fluids in terms of evaporating temperature using analytic equation. Other authors presented a method of selection of working fluid with a criterion of obtaining maximum power, Nowak *et al.* [6]. They presented two specific indicators of power. These indicators characterize ORC plant and have primary importance for selection of working fluid. Paper by Qiu [7] presents the comparison of eight most popular fluids used in micro-ORC systems. The author recommended the following fluids: HFE7100, HFE7000, PF5050, R123, n-pentane, R-245fa, R134a and isobutene as suitable for such applications. The proposed criteria of the fluid selection are: low environmental impact, high enthalpy drop, positive slope of saturation line, moderate dimensions of system components, thermophysical properties, availability and low cost. Each criterion is evaluated in the form of an assigned rating point. The sum of points indicates the most suitable fluid. In the mentioned paper it is the solvent HFE7100.

Vescovo and Spagnoli [8] consider the problem of fluid selection for high temperature sources. They point out the technical and economic potential of high temperature ORC units. Thermal stability of working fluids is regarded as the main limitation. Paper states that the most suitable fluid is a diphentyl – diphentyl oxide mixture which may operate at the temperature of 400 °C. The selection method based on source temperature was presented in paper of Thurairaja *et al.* [9]. The authors design a simple model of ORC system and show some potential heat sources and possible examples of working fluids. The publication shows a dependence of the recommended working fluid on temperature related to the heat source type. Authors conclude that the next important aspects of fluid selection are:

safety, economic feasibility and environmental friendliness.

The working fluid selection process for waste heat recovery from regasification of liquefied natural gas (LNG) was presented by Yu *et al.* [10]. The paper underlined desirable fluid properties: no ozone depletion potential (ODP), low global warming potential (GWP), positive slope of saturation line, no condensation at the outlet of turbine, high chemical stability. Net power output was chosen as the main criterion to evaluate working fluids and 5 independent decision variables were proposed.

The selection process is not a generalized approach and is rather tailored for a specific application. The paper presents a process of working fluid selection for an exemplary waste heat ORC system with nominal electric power of about 30 kW. As a thermodynamic criterion the system net power is assumed. Four variants of the ORC system are compared, including cases with or without regenerator as well as with direct or indirect evaporation. The list of potential working fluids is discussed, especially from the point of view of thermodynamic performance and safety issues.

2 Considered working fluids

There are many potential criteria of fluid selection but the most important include: safety, impact on the environment, good thermodynamic performance and low price. The choice of a substance is very important because it determines all of the consequent design aspects such as heat exchangers and turbine design, materials, sealing types, etc. In any commercial application, the most important fluid selection criterion is safety. It is the reason why a lot of fluids, despite their good thermodynamic performance, have to be rejected.

The chemical substances can be classified by the system introduced by the United Nations. The globally harmonized system of classification and labelling of chemicals (GHS) is being currently used in over 60 countries and the whole European Union. The GHS contains unified criteria of classification of the substances and mixtures for physical hazards. They are always described in a fluid specification sheet, the appropriate GHS symbol must also be shown on the fluid container. Table 1 shows the list of GHS symbols and their meaning. Table 2 presents the fluids that were considered as potential working fluids for the 30 kW ORC system with critical temperatures between 200 °C and 400 °C. There exist a lot of fluids that can be used in ORC systems; the fluids presented in the table were chosen from the NIST Thermodynamic Database – Refprop [11], which is currently one of the leading libraries of this type. Out of the considered fluids, substances such as octane, methylcyclohexane, cyclohexane, heptane, hexane, 2-methylpentane, 2,2,4-trimethylpentane, nonane and octane seem to be considerably dangerous (Tab. 2). However, they have also been included in the calculations for the purpose of comparison.

Table 1: The GHS codes and their meaning [12].

GHS code	Physical hazard
GHS01	Explosive
GHS02	Flammable
GHS03	Oxidizing
GHS04	Compressed Gas
GHS05	Corrosive
GHS06	Toxic
GHS07	Harmful
GHS08	Health hazard
GHS09	Environmental hazard

Table 2: Substances considered as potential working fluids for 30 kW ORC power plant and their properties.

No.	Short name	Full name	Critical temperature [°C]	Critical pressure [kPa]	Kind of danger	Flash point [°C]	Freezing temperature [°C]
1	2	3	4	5	6	7	8
1	Acetone	Propanone	234.95	4700.0	GHS02, GHS07	20	-95.4
2	Benzene	Benzene	288.87	4907.3	GHS02, GHS07, GHS08	-11	5,5
3	Cyclohex	Cyclohexane	280.45	4080.5	GHS02, GHS07, GHS08, GHS09	152	33
4	Cyclopen	Cyclopentane	238.57	4571.2	GHS02	-42	-93.3
5	C1CC6	Methylcyclohexane	299.05	3470.0	GHS02, GHS07, GHS08, GHS09	-4	-126.6
6	C3CC6	Propylcyclohexane	357.65	2860.0	GHS02	90	-70
7	DMC	Dimethyl Carbonate	283.85	4908.8	GHS02	16.7	0.5-4.7

1	2	3	4	5	6	7	8
8	Ebenzene	Ethylbenzene	343.97	3622.4	GHS02, GHS07, GHS08	15	-95
9	Ethanol	Ethyl alcohol	241.56	6268.0	GHS02, GHS07	12	-114.5
10	Heptane	Heptane	266.98	2736.0	GHS02, GHS07, GHS08, GHS09	-4	-90.5
11	Hexane	Hexane	234.67	3034.0	GHS02, GHS07, GHS08, GHS09	-22	-94.3
12	Ihexane	2-Methylpentane	224.55	3040.0	GHS02, GHS07, GHS08, GHS09	-40	-154
13	Isooctane	2,2,4-Trimethylpentane	270.85	2572.0	GHS02, GHS07, GHS08, GHS09	-12	-107
14	MDM	Octamethyltri-siloxane	290.94	1415.0	GHS02, GHS09	51	18
15	Methanol	Methyl alcohol	239.45	8103.5	GHS02, GHS06, GHS08	10	-98
16	MM	Hexamethyldi-siloxane	245.60	1939.0	GHS02, GHS09	4	-68
17	Mxylene	1,3-dimethylbenzene, m-Xylene	343.74	3534.6	GHS02, GHS07, GHS08	27	-48
18	Nonane	Nonane	321.40	2281.0	GHS02, GHS07, GHS08, GHS09	31	-53
19	Octane	Octane	296.17	2497.0	GHS02, GHS07, GHS08, GHS09	13	-57
20	Oxylene	1,2-dimethylbenzene, o-Xylene	357.11	3737.5	GHS02, GHS07, GHS08	30	-25.2
21	Pxylene	1,4-dimethylbenzene, p-Xylene	343.02	3531.5	GHS02, GHS07	24	13.3
22	R113	Trichlorotrifluoro-ethane	214.06	3392.2	GHS07, GHS09	195	-35
23	R141b	1,1-Dichloro-1-fluoroethane	204.35	4212.0	GHS04, GHS07	–	-103.5
24	Toluene	Methylbenzene, Toluene	318.60	4126.3	GHS02, GHS07, GHS08	4	-9
25	Water	Water	373.95	22064.0	–	–	0

3 ORC variants and modeling

A gas engine powered compressor system stored in a container has been selected as the waste heat source. It is fuelled by mine gas containing between 30% and 80% of methane. The gas engine is connected through a coupling with an air compressor and the exhaust gas is directed to the ORC module (Fig. 1). Heat recovery from internal combustion engine by means of ORC systems has been extensively studied in recent years with respect to many aspects [13–16,18]. Rosset, Mounier *et al.* [13] consider the recovery of waste heat from coolant flow and exhaust gases from a passenger vehicle engine and present multiobjective optimization to select fluid with boiling point close to the heat sink temperature, high critical pressure and high molecular weight. The maximum temperature of exhaust gases was equal to 706 °C. Luo *et. al* [15] identified the temperature of exhaust gases from a combustion engine as ranging between 230.8 °C and 548.2 °C. The efficiency of the combined system (vehicle engine and ORC) in the most optimistic variant can increase up to 40% as compared to the efficiency of the engine itself in full load operation equal to 35%.

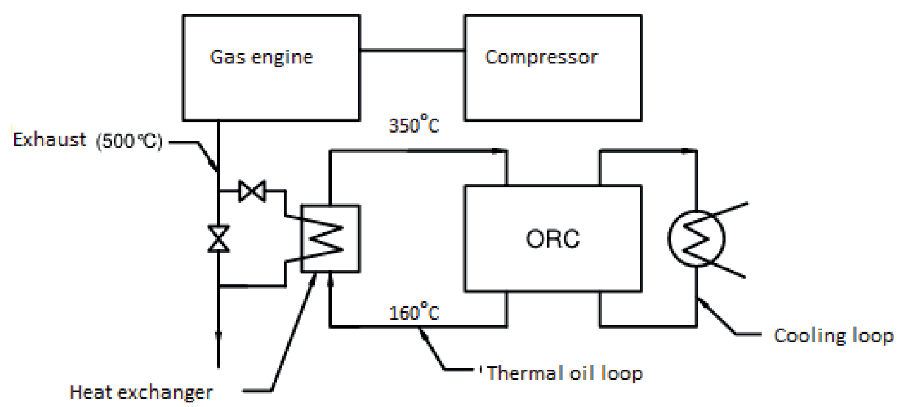

Figure 1: The schematic of the a gas engine powered compressor system with an ORC module.

In the case of our engine considered in the presented investigations, the exhaust gases have the temperature of 510 °C. This is a significant waste of heat if these exhaust gases are not utilized. Therefore, the waste heat can be directed to the ORC module through a heat exchanger and an intermediary thermal oil loop (in the case of indirect evaporation) with a lower temperature between 160 °C and 190 °C and upper temperature between 330 °C and 350 °C. The thermal oil becomes then the heat source for the ORC module.

The cycle calculations were conducted according to a number of assumptions, which are presented in Tabs. 3, 4, and 5. The exhaust gas temperature, which is equal to 510 °C is a measured value. It was also assumed that the exhaust gas cannot be cooled below 125 °C in order to avoid condensation and potential corrosion problems in the exhaust. The exhaust gas composition was also measured and is shown in Tab. 6.

Table 3: Assumptions for indirect heating.

Parameter	Value	Unit
Minimum condensing temperature	50	°C
Cooling water temperature	25	°C
Hot oil temperature	310	°C
Minimum condensing pressure	20	kPa (a)
Pinch temperature difference in the exhaust gas – oil heat exchanger	40	°C
Pinch temperature difference in the evaporator	40	°C
Pinch temperature difference in the condenser	20	°C

Table 4: Assumptions for direct heating.

Parameter	Value	Unit
Minimum condensing temperature	50	°C
Cooling water temperature	25	°C
Minimum condensing pressure	20	kPa (a)
Pinch temperature difference in the evaporator	80	°C
Pinch temperature difference in the condenser	20	°C

The model does not include analysis of heat exchangers, because such analysis requires many additional assumptions (e.g., heat exchanger type). The issues related to the design of microchannel heat exchangers were presented by Mikielewicz and Mikielewicz [17]. The use of regeneration in ORC installations was considered by Borsukiewicz-Gozdur [19]. The author analysed

Table 5: Assumptions for the ORC loop.

Parameter	Value	Unit
Maximum pressure in ORC loop	1500	kPa (a)
Overall turbogenerator efficiency	65	%
Main pump efficiencies	40	%
Overheating in evaporator	5	°C
Overcooling in condenser	5	°C
Head of cooling water pump	10	m
Cooling water pump efficiencies	60	%
Head of oil pump	20	m
Oil pump efficiency	60	%

Table 6: Assumptions for the exhaust gas parameters.

Parameter	Value	Unit
Exhaust gas temperature	510	°C
Exhaust gas cooling limit	125	°C
Exhaust gas mass flow	0.65	kg/s
Mass fraction CO_2	0.067	–
Mass fraction O_2	0.061	–
Mass fraction N_2	0.741	–
Mass fraction H_2O	0.131	–

three variants of power plant: without recuperation, with internal and external recuperation. The application of an internal regenerator increases the power by approximately 5%.

Four configurations of an ORC system are considered in the course of calculations:

- Indirect evaporation (oil thermal loop), no regenerator.
- Indirect evaporation (oil thermal loop), with regenerator.
- Direct evaporation, without regenerator.
- Direct evaporation, with regenerator.

First two variants presented in Figs. 2 and 3 have two main advantages: protection of a working fluid from overheating and a good thermal stability of the ORC unit (the thermal oil loop works as a thermal buffer). The disadvantages include a decreased system efficiency as compared to the case of direct evaporation, depending on the fluid. The costs are also higher as the oil loop makes up a significant fraction of the total costs of installation.

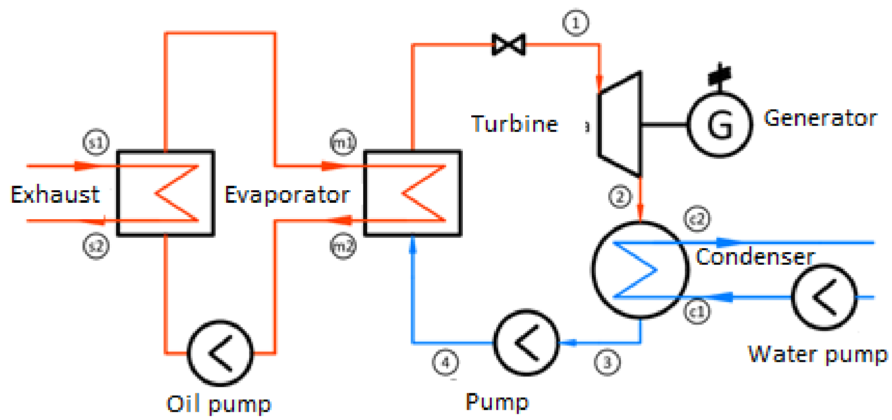

Figure 2: ORC case with indirect evaporation: 1 – vapour of working fluid, 2 – vapour at turbine outlet, 3 – fluid at condenser outlet, 4 – high pressure liquid, c1 – cooling liquid (inlet), c2 – cooling liquid (outlet), m1 – hot oil (inlet), m2 – cold oil (outlet), s1 – hot exhaust, s2 – cold exhaust.

The second variant is more complex as it contains a regenerator. That additional device increases the efficiency but requires a large heat exchanger and makes the control of the installation more difficult (the heat dynamics is worse). That variant is the most complex of all four and has the largest number of elements.

The third option (Fig. 4) does not feature an oil loop and evaporation of the working fluid takes place directly in the exhaust gas heat exchanger which is a significant system simplification. That variant is expected to bring a better efficiency of heat recovery than the first two variants. The main drawback of this solution is a risk of fluid overheating or even explosion, so the system design is more demanding.

The last variant (Fig. 5) is a modification of the third variant as it contains a regenerator. This is the most efficient option.

The computational assumptions were slightly changed from variant to variant as they contain different elements. For the first two variants of indirect evaporation the boundary was a minimum difference of temperature

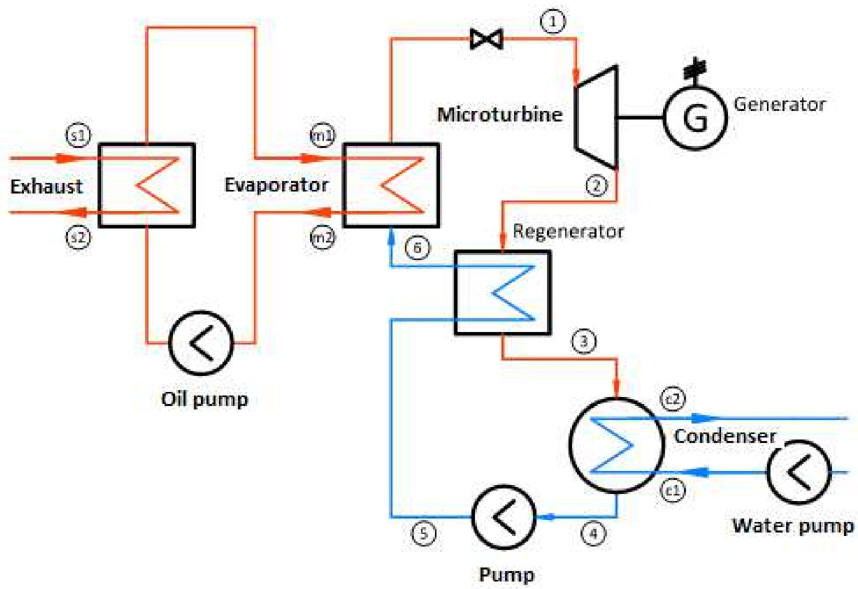

Figure 3: ORC case with indirect evaporation and regenerator: 1 – vapour of working fluid, 2 – vapour at turbine outlet, 3 – vapour after cooling in regenerator, 4 – fluid at condenser outlet, 5 – high pressure liquid, 6– warm fluid at regenerator outlet, c1 – cooling liquid (inlet), c2– cooling liquid (outlet), m1 – hot oil (inlet), m2 – cold oil (outlet), s1 – hot exhaust, s2 – cold exhaust.

at the heat exchanger (exhaust gas/thermal oil). The value of temperature difference was established at 60 °C or 40 °C. The temperature of exhaust gases could not become lower than 125 °C to avoid condensation of water in both cases. The optimized value was net generated power of the ORC power plant. That kind of optimization is justifiable if the aim is heat recovery and if the ORC does not plan to work in cogeneration. The criterion of maximum efficiency of the ORC cycle itself is not satisfied in that case, however the efficiency of heat recovery with respect to the available heating power in the heat source as defined below will also become an optimum value. The cost of installation is also very important. With this respect, indirect evaporation is certainly more expensive than direct evaporation, which results from a necessity of using an additional oil loop. Specific fluids also involve, e.g., specific dimensions, materials, sealings. Thus, precise costs connected with different fluids and variant configurations are impossible to estimate at this stage of the project and will not be discussed further.

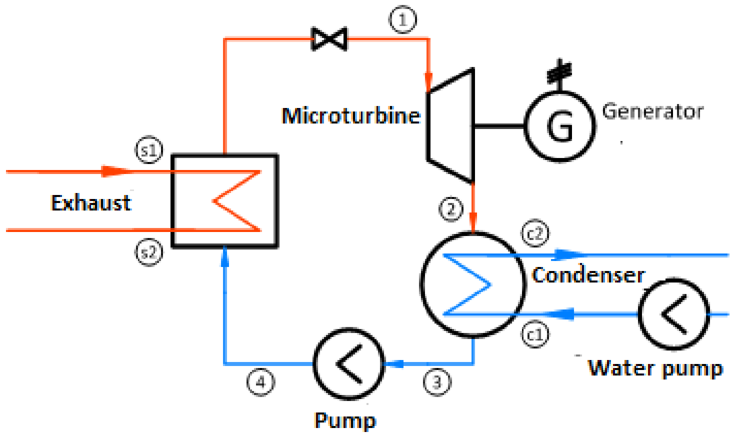

Figure 4: ORC case with direct evaporation: 1 – vapour of working fluid, 2 – vapour at turbine outlet, 3 – fluid at condenser outlet, 4 – high pressure liquid, c1 – cooling liquid (inlet), c2 – cooling liquid (outlet), s1 – hot exhaust, s2 – cold exhaust.

The net power was calculated based on a difference of turbine power and pump power.

The turbine power is

$$P_T = \dot{m}\left(h_1 - h_2\right) = \dot{m}\eta_T\left(h_1 - h_{2s}\right) \ . \tag{1}$$

The pump power can be calculated as

$$P_p = \dot{m}\left(h_4 - h_3\right) = \frac{\dot{m}\left(h_{4s} - h_3\right)}{\eta_p} \ . \tag{2}$$

The system net power is equal to

$$P_{\mathrm{NET}} = P_T - P_P \ . \tag{3}$$

The net efficiency of heat recovery is then

$$\eta_{\mathrm{NET}} = \frac{P_{\mathrm{NET}}}{P_{sr}} \ . \tag{4}$$

On the other hand, the net efficiency of the ORC cycle can be written as

$$\eta_{\mathrm{ORC}} = \frac{P_{\mathrm{NET}}}{\dot{m}\left(h_1 - h_4\right)} \ . \tag{5}$$

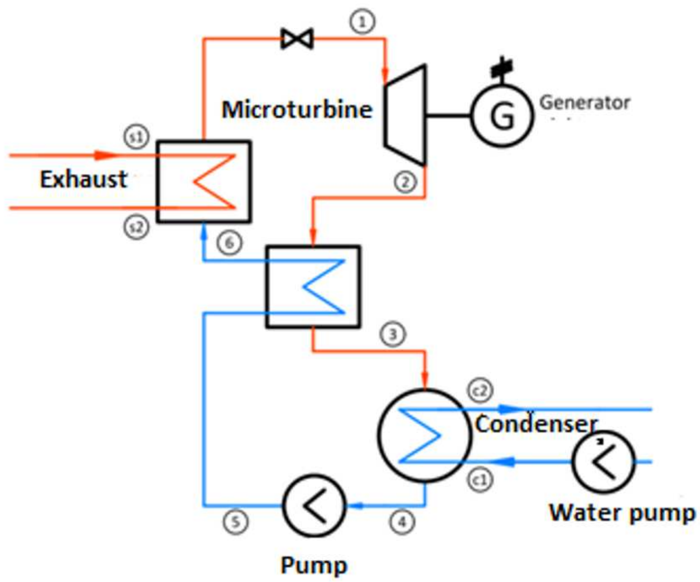

Figure 5: ORC case with direct evaporation and regenerator: 1 – vapour of working fluid, 2 – vapour at turbine outlet, 3 – vapour after cooling in regenerator, 4 – fluid at condenser outlet, 5 – high pressure liquid, 6 – warm fluid at regenerator outlet, c1 – cooling liquid (inlet), c2 – cooling liquid (outlet), s1 – hot exhaust, s2 – cold exhaust.

The calculations are performed in commercial Matlab environment (a high-level language and interactive environment for numerical computation, visualization, and programming) [24] and fluid properties were obtained via the NIST Refprop interface [11]. The study is meant to select a narrower group of candidate fluids for which a more detailed analysis could be performed.

4 Results and discussion

The calculation results showing the system net power output are illustrated in Figs. 6 to 9. The results correspond to the fluids presented in Tab. 2. The optimized objective function was the net generated power of the system and the decision variable was the evaporation temperature. A constraint in the form of the exhaust gas temperature was imposed.

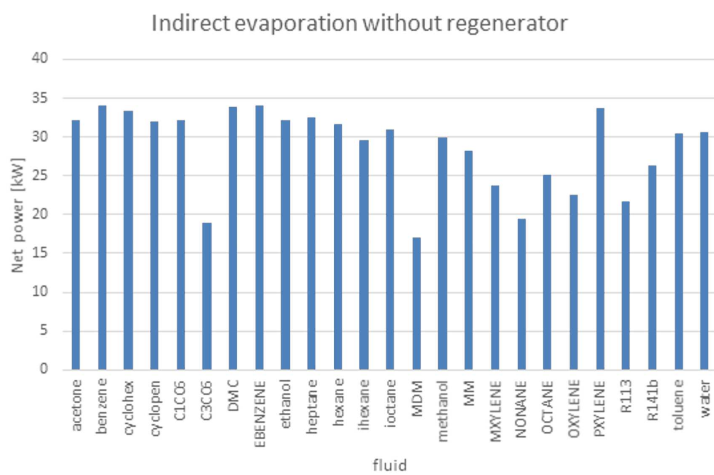

Figure 6: Net power comparison of fluids in case of indirect evaporation without regenerator.

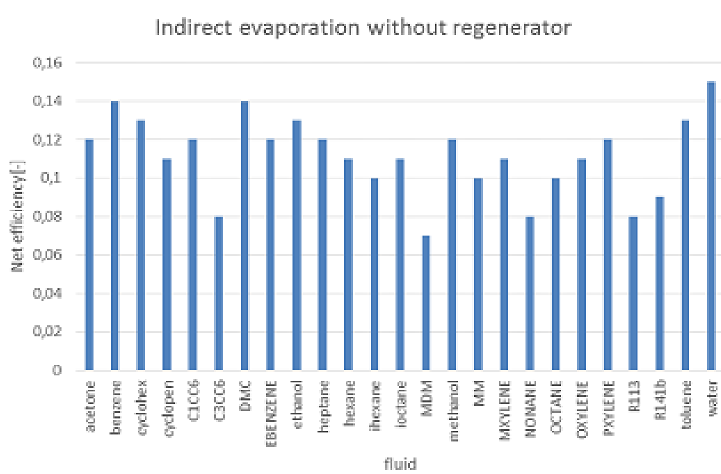

Figure 7: Net ORC efficiency comparison of fluids in case of indirect evaporation without regenerator.

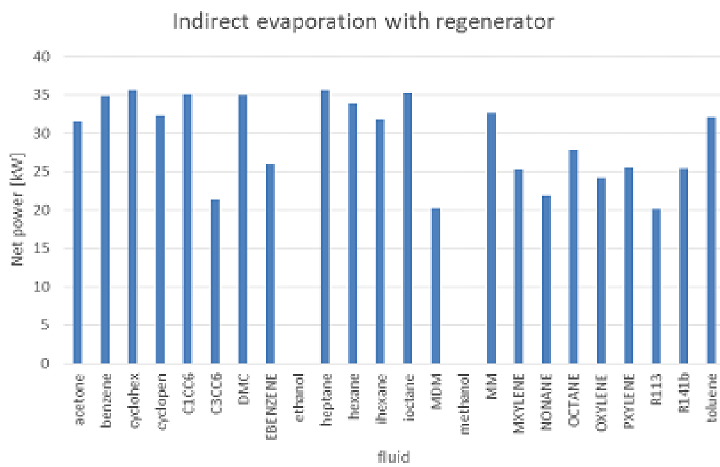

Figure 8: Net power comparison of fluids in case of indirect evaporation with regenerator.

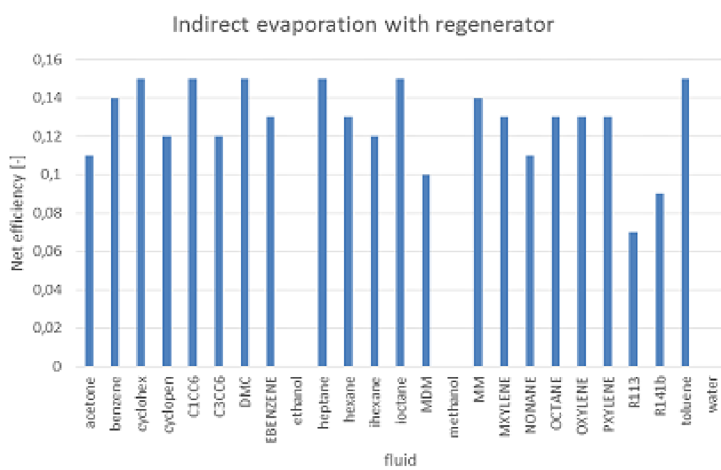

Figure 9: Net ORC efficiency comparison of fluids in case of indirect evaporation with regenerator.

The range of generated power is from about 15 kW to 50 kW. Benzene seems to be one of the most suitable fluids from the thermodynamic point of view as it yields the highest or nearly highest net power. A good performance can also be observed for toluene and dimethyl carbonate. Acceptable performance has been achieved for many fluids, e.g. ethanol, acetone, cy-

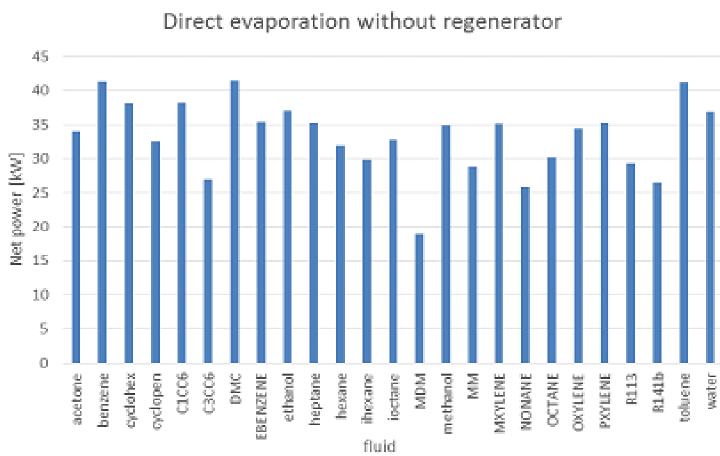

Figure 10: Net power comparison of fluids in case of direct evaporation without regenerator.

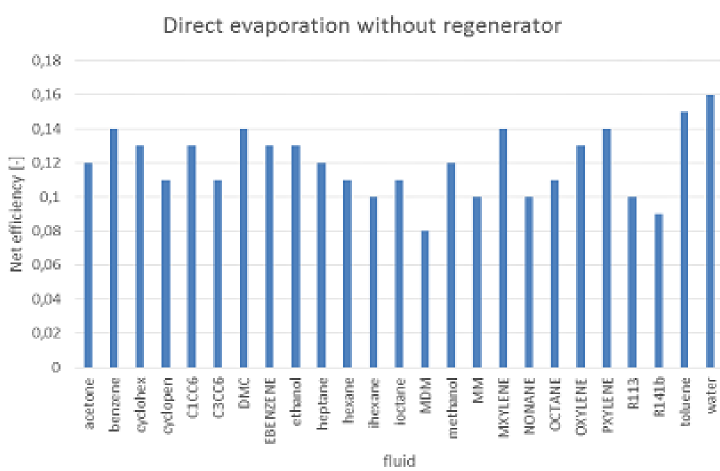

Figure 11: Net ORC efficiency comparison of fluids in case of direct evaporation without regenerator.

clopentane and MM. Specifically, for the first variant the best performance has been achieved by benzene, ethylbenzene and dimethyl carbonate, the worst by propylcyclohexane, MDM and nonane. For the second variant the best performing fluids included cyclohexane, heptane and isooctane, the

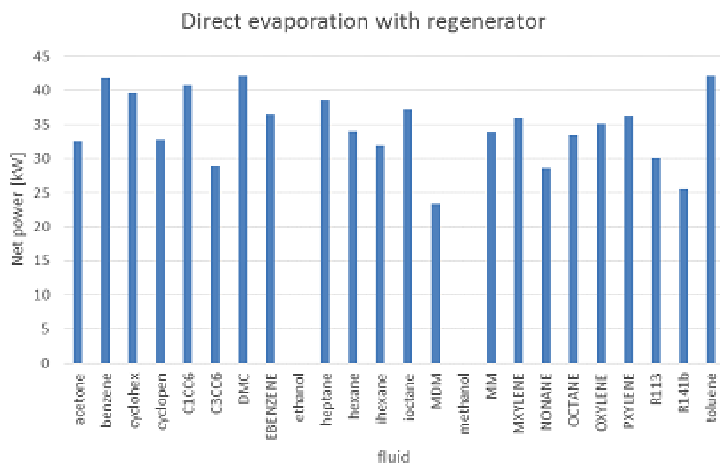

Figure 12: Net power comparison of fluids in case of direct evaporation with regenerator.

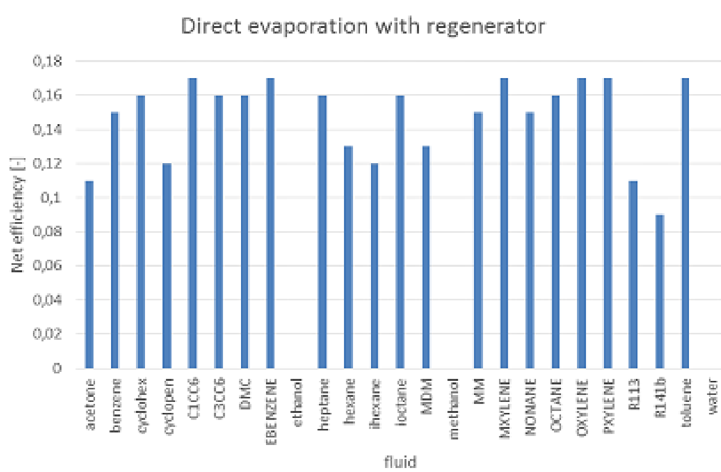

Figure 13: Net ORC efficiency comparison of fluids in case of direct evaporation with regenerator.

worst included propylcyclohexane, MDM and R113. For the third variant the highest net power has been observed for benzene, dimethyl carbonate and toluene, the poorest for: MDM, nonane and R141b. Finally, in the fourth case the most thermodynamically attractive fluids were: benzene, dimethyl carbonate and toluene, the least attractive were propylcyclohexane, MDM and R141b.

Out of twenty five considered fluids the 8 most promising have been selected and presented in Tab. 7. Besides the thermodynamic performance, other selection criteria also need to be considered, including the cost of fluids, health hazards, thermal stability, environmental impact, flammability and the already obtained experience from experimental rigs. Most of these fluids have previously been recommended in techno-economic analyses of ORC systems [20].

Table 7: Comparison of selected working fluids.

Working fluid	Advantages	Disadvantages
Acetone	low price, isentropic fluid - regenerator is not required, high condensation pressure	relatively poor efficiency, possible condensation in turbine, flammable
Cyclopentane	high condensation pressure	low efficiency, relatively low power, high price, flammable
Ethanol	low price	high enthalpy drop (more turbine stages), condensation in turbine, flammable
MM	silicone oil - low enthalpy drop, commercially used	high price, big regenerator, low condensation pressure, environmental hazard, flammable
Toluene	low price, very high efficiency, recommended in a lot of publications, excellent thermal stability, commercially used	health hazard: respiratory sensitization category 1, carcinogenicity categories 1A, 1B, 2
m-Xylene	high efficiency, low price	harmful: acute toxicity category 4, skin irritation categories 2, 3, eye irritation category 2A
DMC	high efficiency	poor thermal stability, melting temperature equal to 4°C
Benzene	high efficiency, recommended in a lot of publications [21,22] , low price	harmful: acute toxicity category 4, skin irritation categories 2, 3

The fluids selected after the performed analysis include toluene, DMC and MM (toluene is already commercially used). Benzene has been excluded here due to its acute toxicity (see Tab. 7). Toluene has a lot of advantages such as low price and good thermal stability. The analysis of the ORC model indicates that the toluene loop generates the highest net power. The main disadvantage is health hazard in the form of respiratory sensitization and carcinogenicity. MM is a silicone oil which is not toxic nor health hazardous. It also represents a satisfactory level of the net generated power. The optimum calculated rotational speed of the turbine for MM exhibits a value lower by 20% than that for DMC or m-xylene and by 22% lower than for toluene, and is the lowest of all the considered fluids. That implicates a lower structural load of the turbine impeller for MM as compared to other fluids for the same generated power.

The analysis confirms that applying an ORC system to a gas engine can significantly increases a system efficiency. The installation of ORC enables us to recover from 22 to 44 kW of power (Figs. 6, 8, 10, and 12). The generated power changes from variant to variant of the ORC system and depends on the selected working fluid. For example toluene lets us generate additional 10 kW as compared to MDM while both are commercially used.

DMC as working fluid provides a considerably high efficiency of the ORC cycle. The results of calculated power are similar to those for toluene. This fluid is more safe then toluene. A disadvantage is poor thermal stability for the assumed range of temperatures of the heat source. Hexamethyldisiloxane (MM) can also be regarded as one of the promising working fluids for ORC units. The paper of Weiss *et al.* [23] describes the results from a small high-speed turbine operating in an ORC system with MM as working fluid. The performance of the tested turbine seems promising so it is a strong argument towards recommending this fluid for potential applications.

5 Conclusions

The process of fluid selection for an ORC application is a complex and multidisciplinary task and it is often an iterative process. This is a key problem of ORC power systems as the selected fluid determines every element of the module such as heat exchangers, expansion unit and feed pump. As it was shown, there is usually no obvious choice and many aspects of a particular system have to be considered.

Consideration of different thermodynamic and physical properties allows for a rational selection of the working fluid. Several criteria were applied in the present paper for the comparison of fluids working in an ORC heat recovery system designed for a compressor unit driven by a gas engine: net generated power, ORC cycle efficiency, economic benefits and health hazards. 25 potential fluids and four variants of ORC installation (without or with regeneration, with direct or indirect heating) were considered in the paper. The objective was to pick up the best configurations of fluids and ORC variants featuring the highest net power generation from the available heat source. The obtained results were presented in the form of bar graphs. Based on the net generated power criterion, 8 fluids were appointed for further selection to consider other criteria including health hazards involved. Three fluids were finally chosen as the most suitable for application in the ORC heat recovery system – toluene, DMC and MM. Toluene is a fluid widely used in chemical industry; it has quite good thermodynamic properties and low price. DMC as working fluid has good thermodynamic properties and it provides a considerably high efficiency of the ORC cycle. MM is a fluid that is not harmful for humans and also lets us avoid high structural load of the turbine impeller, however, it is potentially hazardous for the environment.

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