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Analysis of a possible application of a small power plant in the Organic Rankine Cycle, which apply waste heat from a combustion engine

Abstract
Some considerations on the waste heat treatment, heat and power combined generation and principles of the Organic Rankine Cycle and its application are produced. A thermodynamic screening of eight working fluids for Organic Rankine Cycle (ORC) was carried out. All investigated agents were single component fluids. The considered ORC systems operated at the upper temperature in the range Tmax = 146–185°C and the lower one Tmin = 27–37°C. The highest pressure in the cycles equalled pmax = 3,500 kPa and pmin = 85 kPa. The cycles were run at subcritical pressures and the saturated vapour line was of the overhanging type in all cases. Heat regeneration was not considered in this paper. The heat source for the ORC system was heated air exhausted from the cooling system of an opposed-piston engine. The temperature of this heat at the inlet to ORC was equal to Th,in = 230°C. Air was also the heat sink for the ORC with the inlet temperature Tcold,in = 20°C. The power output of the ORC, neglecting electric generator efficiency, was kept in the range N = 10–11 kW. Calculated thermal efficiencies η held values between 0.121 and 0.164.

Keywords: Organic Rankine Cycle, working fluids, waste heat.

1. Introduction

The term ORC is an abbreviation for Organic Rankine Cycle, which means Rankine cycle. The working medium in such systems is an organic agent. The classical Rankine cycle take place in large thermal power plants, in which steam spins a steam turbine which drives an electrical generator. The source of heat energy in this system is either a boiler utilizing a fossil fuel or a nuclear reactor. In both cases the working fluid is water steam. In Fig. 1 a diagram of the thermal circuit is shown and the corresponding cycle scheme in Fig. 2. In the boiler, between state points 5–1 in the scheme, steam is produced which drives a steam turbine (points 1–2). Then, in the turbine it expands and flows to the condenser (state points 2–4c). This waste heat is usually disposed of with cooling water or in cooling towers in large power plants. After condensation the water is pumped (state point 4c–5) back to the boiler and the cycle of the working medium is closed. The set of devices: the boiler, the turbine, the condenser, and the pump constitutes a thermodynamic cycle, which consists of four transformations: isobaric evaporation, isentropic vapour expansion, isobaric liquefaction, isentropic water compression. In power plants, the steam is superheated to temperatures much above the boiling point. The pressure is also much higher than atmospheric. Increasing the steam parameters results in an increase in energy efficiency, which increases with the increase of the steam temperature. Operational reasons are also important. Leaving the steam at atmospheric pressure and the temperature slightly exceeding $T = 100°C$, causes the presence of small, unevaporated residual drops of water in the steam. These droplets flowing with the steam, hit the turbine blades causing their erosion. Hence, steam parameters in high-power plants are high, i.e. the pressure is several MPa and the temperature is about $T = 550^{o}C$. Polish newest power blocks (Jaworzno, Opole) incorporate supercritical steam parameters, i.e. the pressure $P = 22$ MPa, and the temperature $T = 550°C$.

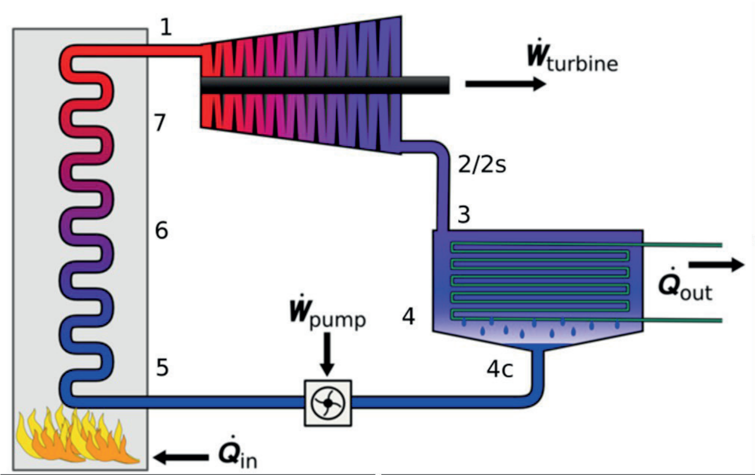

Fig. 1. Condensation power plant circuit

High power steam circuits are very difficult to miniaturize. For instance, utilization of waste heat from internal combustion engines with steam power units is not applied. Low power of turbines is associated with low temperatures, slightly exceeding the boiling point, which leads to the presence of micro-droplets of water in the steam. As a result erosion appears on the turbine blades. There may be also conditions that the waste heat flux is insufficient to evaporate the required amount of water. A significant improvement of this situation is obtained by replacing water with another agent with a lower boiling point. This makes possible to superheat the steam at temperatures much lower than for water. In addition, depending on the type of working medium, position of the wet-dry separation curve may be such that during expansion in the turbine, steam flowing out of the turbine remains dry. In Fig. 2, the right saturation coexistence curve separating the two-phase region from the superheated vapour region is shown. It is so called overhanging saturation line ("o") that distinguish many ORC fluids form the bell-shaped ("b") coexistence curve of water (Saleh, Koglbauer, Wendland, & Fischer, 2007; Lai, Wendland, & Fischer, 2011). It appears that after the expansion in the turbine (state points 1 to 2), point 2 still remains on the right side of the saturation curve. It is a safe expansion zone for the proper machine work. Therefore, ORC systems may not need vapour extensive superheating contrary to Rankine steam cycles. This way, the erosion of the machine is avoided and this should be considered reasonable from exploitation point of view.

Water has an opposite inclination of the right boundary curve. It causes that at the end of expansion low-pressure parts of steam turbine may be exposed to wet steam — detrimental to the turbine rotor blades. Point 2 shown in Fig. 2 of "o" type fluid would be probably on the left side of the saturation curve for water ("b") type system. Typically, steam turbine expansion ends at a vapour quality $x = 80–90\%$. Functionally, the ORC circuit diagram for the ORC system is the same as for the Rankine steam circuit, as shown in Fig. 1.

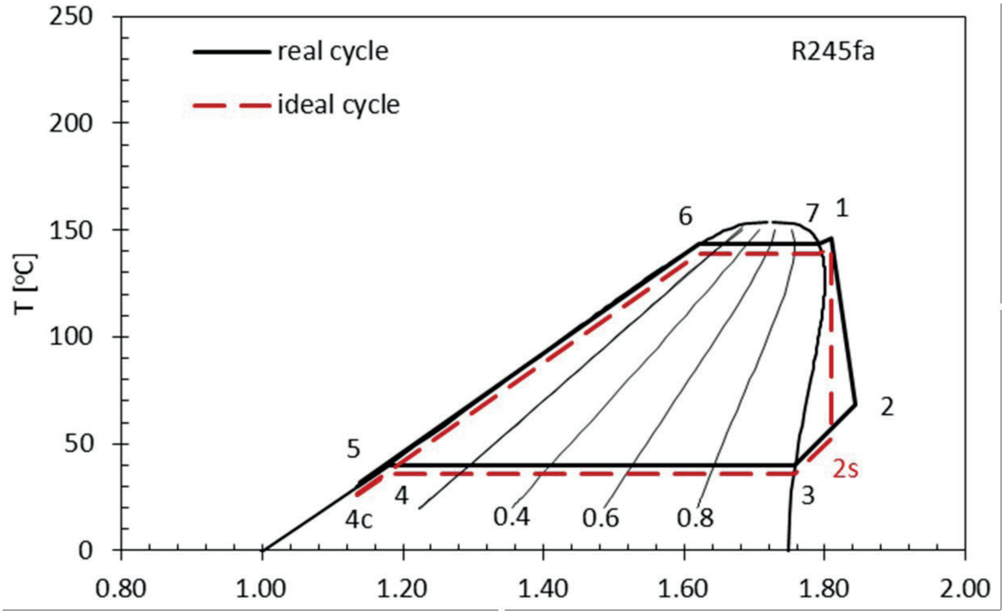

Fig. 2. T-s diagram of the ORC system

It is also important that at these same thermal parameters, the density of low-boiling agent is higher than the steam density. It results in a higher power density of low boiling agents than that of the water steam at equal rates of mass fluxes. These features make ORC suitable in applications of a low-temperature heat recovery, i.e. waste heat from internal combustion engines. This include also geothermal heat, solar collectors, etc. An alternative to ORC systems is entire heat delivery to heating purposes without electric energy production.

The generation of heat and electricity in small power systems is less energy-efficient if the systems are separated. Heat production for heating almost entirely goes to the final destination. Then heat losses to the environment are small. In the production of electricity in energy systems, a large part of the generated heat must be discharged into the environment. Separate production means doubled heat production — once for heating, twice for the needs of the electricity generation. Both processes can be combined into one technological line, in which the waste heat from the production of electricity is used for the heating purpose. Thus, the same quantity of the primary heat is used to generate electricity and the useful heat. The diagram of possible energy savings is shown in Fig. 3. The conclusion is that due to the combined heat and power generation, over 90% of the original heat is used. This level of efficiency is not available with separate heat and energy production.

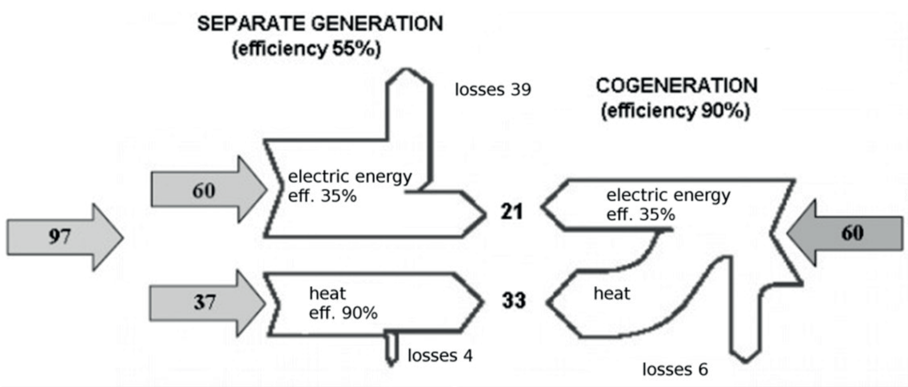

Fig. 3. Separate and combined heat and power generation

If the heat source of the ORC system is a waste heat from (for example) an internal combustion engine, then the heat is twice utilised. First as a primary energy source for heat and energy production in the ORC system, and second as the ORC output heat for heating purposes.

In the field of internal combustion engines, there is a noticeable research trend devoted to opposed-piston engines, i.e. engines in which counter stroking pistons work in one cylinder. This layout is characterized by a better scavenging of the cylinders and a longer stroke expansion. An example of conceptual considerations over an inline engine conducted at the Silesian University of Technology is shown in Fig. 5.

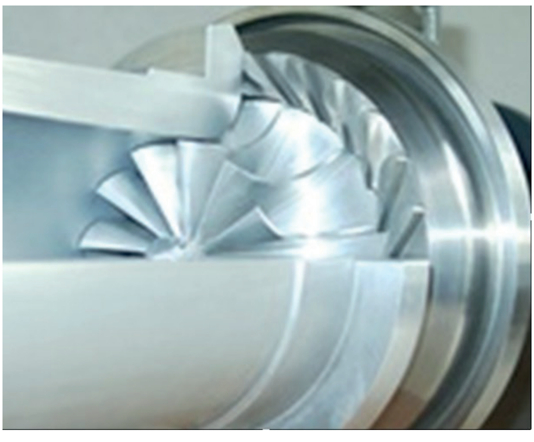

Fig. 4. Radial gas micro turbine for ORC applications

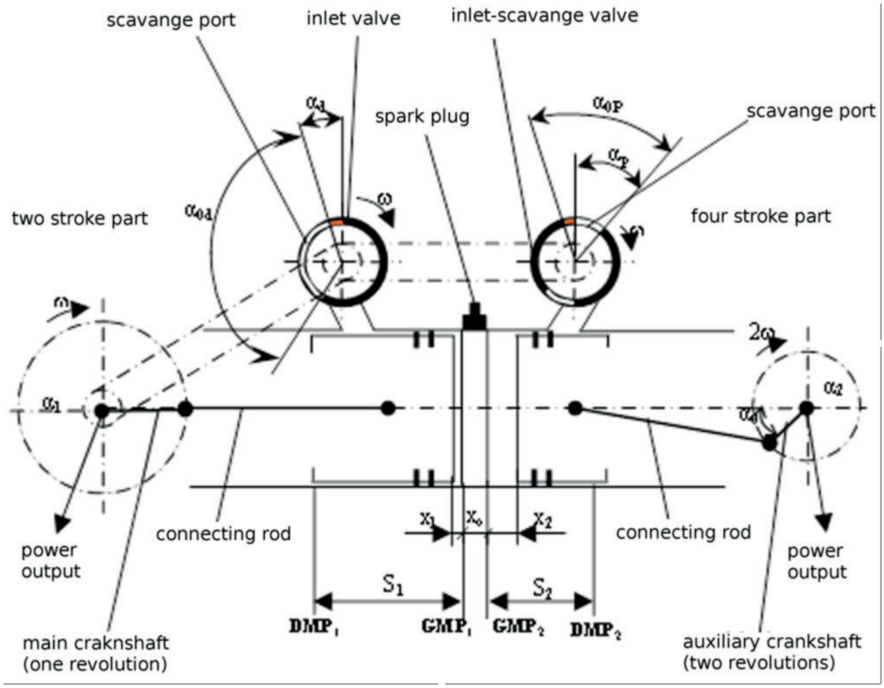

Fig. 5. Combined two-stroke four-stroke opposed-piston engine; Adam Ciesiołkiewicz, Silesian Universtiry of Technology

Recently several research institutes and commercial companies put some focus on the application of turbines or other expanders as power generators for the ORC circuits. An example of a micro-turbine designed for ORC systems is shown in Fig. 4. Size and weight of a small power turbine (N = 1 kW) may be roughly characterised by its diameter $D \approx 0.1$ m, length $L = 0.15$ m and weight $m = 5$ kg.

In the process of designing a small power system, the following stages can be distinguished:

1. Selection of a power generator depending on energy needs.
2. Selection or design of the ORC system.
3. Selection of the working medium taking into account: the waste heat temperature, shape of the saturation curve, freezing temperature, chemical stability temperature, evaporation heat, operational chemical properties such as chemical aggressiveness, toxicity, self-ignition, etc.
4. Selection of an expander — a selection should be done among piston engines, scroll expanders, turbines, etc., it is necessary to choose parameters of the expander.
5. Selection of a heat exchanger in which waste heat from an engine or another waste heat source is transferred to the working medium.

2. Analysis of a small power output engine waste heat recovery with ORC

A few variants of ORC systems with net power output $N \approx 10$–11 kW were examined. For each system a different fluid was taken into consideration. The calculation focus on the input heat Q_{in}, output heat Q_{out} and the cycle efficiency η. It was assumed that the heat is delivered to the system from the hot air with the temperature $T_{h,\ in}$. Another stream of air recovers the output heat Q_{out} from the system at the initial temperature $T_{c,\ in}$ = 20°C. The efficiencies of the turbine and the pump were presumed η_{turb} = 0.75, η_{pump} = 0.65 respectively. The efficiency of the generator was out of consideration in this paper.

All chosen for investigation fluids are characterised by the so called overhanging ("o") saturation line Fig. 6 (Saleh et al., 2007; Lai et al., 2011). Water on the contrary has a well known bell-shaped ("b") coexistence curve. Few other organic fluids also feature "b" saturation lines, among others: R128, R227ea, RC318, R236fa, etc. (Saleh et al., 2007). It is natural for "o" shaped ORC systems to run either on supercritical or slightly superheated subcritical mode. The latter case was investigated in this paper. The reason for strong superheating of the water stream is to avoid expansion in the turbine in the two phase region. It may be substantially important since organic fluids are usually aggressive chemicals. But in case of overhanging coexistence curves only slight superheating is required to hold the entire expansion in the turbine in the vapour zone. Moreover, an increase in superheating in "o" subcritical cycles decreases the thermal η efficiency of the cycle (Saleh et al., 2007).

Thus subcritical "o" ORC cycles were chosen for investigations. Heat regeneration of the expander exhaust stream (points 2–3) was not included in calculations. The scheme of characteristic point of the ORC is shown on a T-s diagram, Fig. 2. This enumeration pattern was taken from the paper of/by (Yang, Zhang, Bei, Song, & Wang, 2015; Yang, Zhang, Song, Bei, Wang, & Wang, 2015). They investigated a diesel engine waste heat recovery. The heat was fed into a low boiling ORC systems.

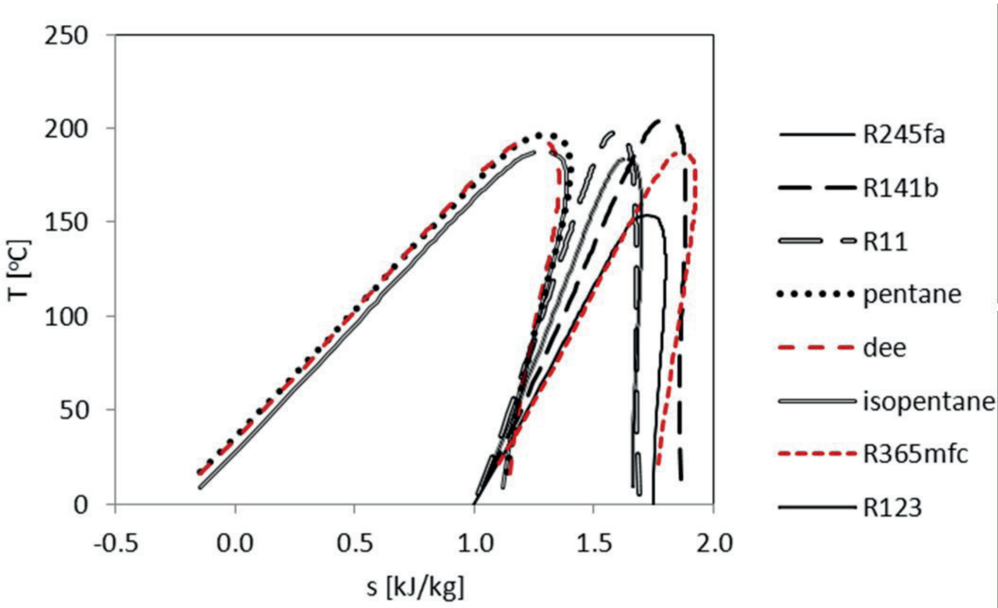

Fig. 6. T-s diagram of the selected fluids, dee — diethyl ether

3. Conclusions

The basic parameters of the investigated thermodynamic cycles are produce in Tab. 1. The condensation pressure p_{min} for seven agents chosen out of eight is larger than the ambient one. Only for R365mfc the condensation pressure is lower than that and amounts to p_{min} = 85 kPa. It is important due to possible air penetration. However, it is still more than generally reported lowest value $p_{min} \approx 30$ kPa for small ORC systems. The output powers of the investigated systems were kept between N = 10 to 11 kW by means of mass fluxes $\dot{m}_{ORC}$ settings. Cycles efficiencies varied between $\eta = 0.145$ and $\eta = 0.164$. Substantially lower value was achieved on the cycle with the R245fa agent. Though, it might be increased if the outlet temperature of the cooling air $T_{pow2} = 50^{o}$C was decreased. Maximum pressures of the investigated cycles did not exceed p_{max} = 3000 kPa. Thus, the systems parameters are rather typical to low pressure refrigeration systems. Agents maximal fluxes were in the range:

$$\dot{m}_{ORC} = 0.1 \div 0.4\,\text{kg/s}\ .$$

Pinch points issues may only refer to R141b and R11 agents and the problem is located on the heating curve at the point 6 on Fig. 2. However, careful parameters settings eradicated such possibilities so that, cycle heating curves were at every point below the temperature of the heating air. Taking into account thermodynamic parameters T_{max}, p_{max}, p_{min}, $\dot{m}_{ORC}$, none of the investigated agents should be favoured at the moment.

Nomenclature

- D – diameter [m]
- $\dot{m}_{ORC}$ – ORC mass flux [kg/s]
- $\dot{m}_{cold}$ – mass flux of the cooling air [kg/s]
- $\dot{m}_{hot}$ – mass flux of the heating air [kg/s]
- m – mass [kg]
- N – ORC power output, turbine power output [kW]
- p_{max} – maximal pressure of ORC circuit [kPa]
- p_{min} – minimal pressure of ORC circuit [kPa]
- $\dot{Q}_{in}$ – heat supplied to ORC [kW]
- $\dot{Q}_{out}$ – outlet heat [kW]
- T_{max} – maximal temperature of ORC circuit [^{o}C]
- T_{vap} – evaporation temperature of the ORC fluid [^{o}C]
- T_{cond} – condensation temperature of the ORC fluid [^{o}C]
- T_{min} – minimal temperature of the ORC circuit [^{o}C]
- $T_{h,\,in}$ – inlet temperature of the heating air [^{o}C]
- $T_{h,\,out}$ – outlet temperature of the heating air [^{o}C]
- $T_{c,\,in}$ – inlet temperature of the cooling air [^{o}C]
- $T_{c,\,out}$ – outlet temperature of the cooling air [^{o}C]
- x – vapour quality
- η – ORC thermal efficiency

Table 1. Basic parameters of ORC systems based on different working fluids

Fluid	p_{max}	p_{min}	T_{max}	T_{vap}	T_{cond}	T_{min}	$\dot{m}_{ORC}$	$\dot{Q}_{in}$	N	η	$T_{hot,in}$	$T_{hot,out}$	$\dot{m}_{hot}$	$T_{cold,in}$	$T_{cold,out}$	$\dot{m}_{cold}$
	p_5–p_1	p_2–p_{4c}	T_1	T_6–T_7	T_3–T_4	T_c					Heating air			Cooling air		
	[kPa]	[kPa]	[°C]	[°C]	[°C]	[°C]	[kg/s]	[kW]	[kW]		[°C]	[°C]	[kg/s]	[°C]	[°C]	[kg/s]
R245fa	3000	250	146	143.2	39.9	30.0	0.3616	91.54	11.03	0.121	230	50	0.500	20	40	3.9987
R141b	3000	140	184	181.6	41.6	30.0	0.2218	68.66	10.87	0.158	230	92	0.488	20	38	3.1890
R11	3500	170	185	182.2	39.2	35.0	0.31476	71.36	11.71	0.164	230	90	0.500	20	38	3.2920
pentane	2700	100	184	182.2	35.7	27.0	0.13091	76.89	11.72	0.152	230	50	0.420	20	38	3.5963
diethyl ether	2700	110	176	174.6	36.8	31.0	0.13954	76.45	11.57	0.151	230	42	0.400	20	38	3.5810
izo-pentane	2700	130	174	172.7	35.2	31.0	0.14732	79.27	11.73	0.148	230	35	0.400	20	38	3.7270
R365mfc	2400	85	171	169.2	35.4	30.0	0.24997	77.26	11.17	0.145	230	40	0.400	20	38	3.6470
R123	3000	160	173	171.3	41.1	37.0	0.34097	76.89	11.38	0.148	230	50	0.420	20	38	3.6150

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Acknowledgments

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