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Analysis of the use of waste heat from a glass melting furnace for electricity production in the organic Rankine cycle system

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Abstract In most production plants, waste heat is usually discharged into the environment, contributing to a reduction in the energy efficiency of industrial processes. This is often due to the low thermal parameters of the carriers in which this energy is contained, such as oils, water, exhaust gases or other post-process gases, which means that their use for electricity production in a conventional Rankine cycle may prove to be economically unprofitable. One of the technologies enabling the use of low- and medium-temperature waste heat carriers is the organic Rankine cycle (ORC) technology. The paper present results of calculations performed to evaluate potential electricity production in ORC using waste heat from a natural gas-fired glass melting furnace. The analysis was carried out assuming the use of a single-stage axial turbine, whose efficiency was estimated using correlations available in the literature. The calculations were carried out for three working fluids, namely hexamethyldisiloxane, dimethyl carbonate, and toluene for two scenarios, *i.e.* ORC system dedicated only to electricity production and ORC system working in cogeneration mode, where heat is obtain from cooling the condenser. In each of the considered

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cases, the ORC system achieves the net power output exceeding 300 kW (309 kW for megawatts in the cogenerative mode to 367 kW for toluene in the non-cogenerative mode), with an estimated turbine efficiency above 80%, in range of 80,75 to 83,78%. The efficiency of the ORC system, depending on the used working fluid and the adopted scenario, is in the range from 14.85 to 16.68%, achieving higher efficiency for the non-cogenerative work scenario.

Keywords: Energy efficiency; Distributed generation; Organic Rankine cycle; ORC; Industrial waste heat

Nomenclature

A_i	–	coefficient for the efficiency correlations
F_i	–	terms of the efficiency correlations
h	–	mass specific enthalpy, kJ/kg
$\dot{m}$	–	mass flow, kg/s
N	–	electric power, kW
p	–	absolute pressure, kPa(a)
$\dot{Q}$	–	rate of heat, kW
SP	–	dimensionless size parameter
T	–	temperature, °C
$\dot{V}$	–	volume flow rate, m^3/s
V_r	–	volume ratio

Subscripts

c	–	coolant
cog	–	cogeneration
$COND$	–	condenser
CP	–	coolant pump
CT	–	cooling tower
ECO	–	economiser
$EVAP$	–	evaporator
HS	–	heat source
i	–	number of point in layout
j	–	number of element in correlation
in, 1	–	inlet
is	–	isentropic
max	–	maximal
min	–	minimal
MP	–	working fluid pump
net	–	netto
o	–	oil
OP	–	oil pump
ORC	–	organic Rankine cycle

out, 2	–	outlet
REG	–	regenerator
s	–	flue gas
sup	–	superheating
sub	–	subcooling
T	–	turbine
X	–	vapour quality

Greek symbols

η	–	efficiency

1 Introduction

Industrial processes are accompanied by emission of waste heat in a form of radiation and excess energy contained in fluid and solid streams [1, 2]. Excess energy in process output streams can occur in form of raised temperature (in most cases, but it can also occur in streams with temperature lower than environment, *e.g.*, at liquefied natural gas (LNG) terminals), pressure or chemical composition of the stream (mostly flammable constituents from unburned fuel or other raw materials fed into the process) [3]. According to [1] in 2015 there was 304.13 TWh/year of technically available waste heat to harvest, which represents 9.5% of total energy consumption in EU industry that year (3200 TWh/year). In most cases this energy has to be removed from process and can be disposed to environment, although some heat carriers can have parameters high enough (mostly temperature) that harvesting their energy can be seen as great opportunity for improving energy efficiency and hence economic profitability of industrial plants. One of technologies allowing for conversion of industrial waste heat into electricity is the organic Rankine cycle (ORC).

The main difference between ORC and classical Rankine cycle is the working medium. In ORC hydrocarbons, silicone oils or refrigerants (also referenced as organic fluids) are used instead of water. In most cases they have lower evaporation temperature and/or lower enthalpy of evaporation, what allows to utilise heat sources at low temperature levels or with relatively small electricity generation potential (less than 10 MW). Over last 50 years this technology have seen an increasing number of manufacturers and commercial applications with power ranging from less than 1 kW to 18 MW [4, 5]. The ORC technology is considered as one of interesting in case of waste heat recovery due to its scalability and suitability to wide range of heat sources ranging from geothermal, solar, biomass and different industrial heat sources like gas turbines, internal combustion engines or different types of furnaces [6].

The aim of this work is to estimate possible electricity production in ORC using waste heat form glass melting furnace in two scenarios: purely for power generation and for cogeneration. Presented approach can be considered as preliminary for working fluid selection.

2 Glass industry

In glass industry melting furnaces, annealing ovens and tempering furnaces [7] are used to process raw material into different types of glass (flat, container, fibre and others more specialised glass types), where the main source of waste heat are exhaust gases. All of those devices work at high temperatures which provides an opportunity for turning waste heat into electricity. Nowadays modern glass plants can use regenerative and recuperative furnaces to achieve high efficiency, which can reduce available rate of waste heat for power generation [7, 8]. In these types of furnaces waste heat is mostly used to preheat air for combustion and can be also used to preheat batch and cullet materials. Recuperation allows to cool down exhaust gasses to 982°C and regeneration even to the range from 316°C to 593°C. Without any waste heat recovery process the temperature of flue gases from furnace can exceed 1315°C [8].

According to [9] in EU27 glass works producing only flat glass (container glass and other glass products were not included due to lack of energy audits), there is possibility to install 78.5 MW of total gross ORC power with significant reduction greenhouse gases emissions by 140 333 metric tons, with focusing only on ORC technology. For the United States and China waste heat to power potentials were estimated respectively as 340 MW (float and container) [7] and 190 MW (float only) [8] including a wide range of waste heat recovery technologies.

According to data provided in [4] in 2017 summarised power of ORC units dedicated to converting waste heat from industrial plants was 376 MW with another 39 MW under construction. Eight of those projects were dedicated for recovery of waste heat at glass works with 4.7% share in total power capacity installed in ORC systems on industrial plants.

3 Data, assumptions and methodology

An ORC system equipped with a regenerator and an intermediate oil loop was adopted for the analysis (Fig. 1). The intermediate oil loop protects working fluid from overheating, acts as a thermal buffer and improves thermal stability of the ORC unit. In comparison to direct evaporation this solution results in a decreased cycle efficiency due to lower temperature at the inlet of evaporator and raises the cost of ORC unit. The regenerator improves cycle efficiency but makes it harder to control due to worse heat dynamics and is more expensive than without regeneration. Regeneration also increases the power output of ORC unit by approximately 5% [10]. This layout has been chosen to ensure safety of operation and high efficiency.

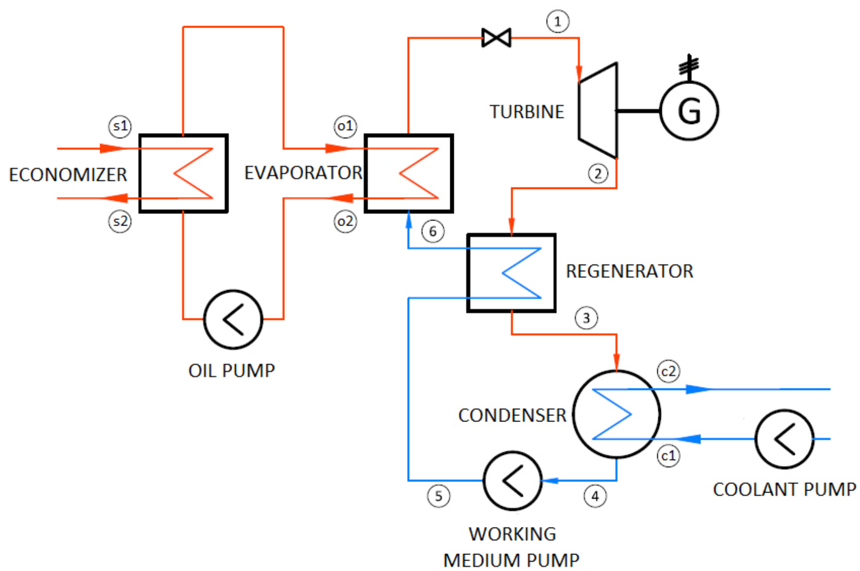

Figure 1: Adapted layout of ORC system: s1 – hot exhaust gas at economiser inlet, s2 – cold exhaust gas at economiser outlet, o1 – hot oil at economiser outlet and evaporator inlet, o2 – cold oil at evaporator outlet and oil pump inlet, o3 – cold oil at pump outlet and evaporator inlet, 1 – vapour of working medium at evaporator outlet and turbine inlet, 2 – vapor at turbine outlet and regenerator inlet, 3 – vapour at regenerator outlet and condenser inlet, 4 – liquid working medium at condenser outlet and medium pump inlet, 5 – high pressure liquid working medium at pump outlet and regenerator inlet, 6 – high pressure liquid at regenerator outlet and evaporator inlet, c1 – coolant at pump outlet and condenser inlet, c2 – heated up coolant at condenser outlet directed to cooling tower, G – electric generator.

Although many different fluids can be used in ORC, only three were preselected based on the literature review. Calculations were performed for following working fluids: toluene, hexamethyldisiloxane (MM) and dimethyl carbonate (DMC). Those are dry fluids, which means that expansion in turbine takes place only in the superheated region of the temperature-entropy property plot. On the one hand this is beneficial to thermal efficiency of expansion devices and the turbine working conditions (less erosion due to lack of condensation in turbine), on the other hand dry fluids have larger areas of heat transfer compared to wet and isotropic fluids [11].

Toluene is recommended as one of best performing working fluids that can be used in ORC [10, 12]. Moreover it is cheap, thermally stable and it is used in commercial ORC applications [10]. Nonetheless toluene is flammable and hazardous to human health (respiratory sensitization and carcinogenicity) [10]. MM is a siloxane that is also regarded as promising working fluid due to its good thermal stability, good material compatibility, high cycle efficiency and being not hazardous to human health [10, 13]. According to [10, 14] it has already been used in ORC cycles applied for mobile and heat recovery applications. From the other side it is hazardous to the environment and similarly to toluene is flammable [10, 13] (although it is considered to be much safer than hydrocarbons). Dimethyl carbonate was found as very promising in regard of cycle efficiency for industrial waste heat applications [10, 15]. In comparison to toluene and MM it is not recognized as hazardous to human health and environment [16], nonetheless it is still flammable [10].

Even though some organic fluids can be hazardous to human health and environment, it can be mitigated by proper design of ORC unit (by ensuring proper sealing) [10]. Therminol66 was chosen as a heat transfer oil due to wide application in industry, including ORC units applied for waste heat recovery in glass industry [17]. As a coolant 40% glycol-water mixture was used.

Parameters of waste heat stream, which have been used for the analysis are gathered in Table 1. The source of waste heat is the flue gas from the glass melting furnace fired with high-methane content natural gas. Basic data on the heat source (temperature, mass flow and composition of exhaust gas) were provided by the employees of the glassworks. This data comes from the continuous process monitoring system and additionally verified was temperature of flue gas has been. In agreement with the employees of the glassworks, the minimum temperature of exhaust gas at the outlet of the economizer was set at 120°C, to prevent water condensation in the further part of the exhaust system.

Table 1: Heat source parameters adopted for analysis.

Parameter	Unit	Value
Mass fraction CO_2	%	11.93
Mass fraction N_2	%	72.59
Mass fraction O_2	%	11.09
Mass fraction H_2O	%	4.40
Flue gas temperature, T_{s1}	°C	400
Flue gas temperature limit, $T_{s2,\min}$	°C	120
Flue gas mass flow, $\dot{m}_s$	kg/s	9.426

Basing on Table 1 the available thermal power of the source was calculated as:

$$\dot{Q}_{HS} = \dot{m}_{s1} \left(h_{s1} - h_{s2,\min}\right) , \tag{1}$$

$$\dot{m}_{s1} = \dot{m}_{s2} , \tag{2}$$

$$p_{s1} = p_{s2} . \tag{3}$$

Mass specific enthalpy of flue gas is calculated as

$$h_{si} = h_{\mathrm{CO_2}}^{T_{si},p_{si}} \mathrm{CO_2} + h_{\mathrm{N_2}}^{T_{si},p_{si}} \mathrm{N_2} + h_{\mathrm{O_2}}^{T_{si},p_{si}} \mathrm{O_2} + h_{\mathrm{H_2O}}^{T_{si},p_{si}} \mathrm{H_2O} . \tag{4}$$

The calculations were held for two working modes, namely purely for power generation and for cogeneration. The assumptions for both modes are gathered in Table 2. In case of cogeneration mode, assumptions presented are refined to ensure temperature of coolant at the condenser output, T_{c2}, is appropriate for heating system at industrial plant. It is done by increasing minimal condensation temperature, $T_{condens,\min}$, up to 85°C.

Assumed layout is the same for both modes, but in case of cogeneration mode, cooling tower is off and heat that was removed from cycle in condenser is further used for heating purposes at the industrial plant.

The process of heat exchange in heat exchangers is assumed to be isobaric and without any heat losses to the environment:

$$p_{\min} < p_1 \leq p_{\max} , \tag{5}$$

$$p_1 = p_5 = p_6 , \tag{6}$$

$$p_{\min} = p_2 = p_3 = p_4 , \tag{7}$$

$$p_o = p_{o1} = p_{o2} = p_{o3} , \tag{8}$$

$$p_c = p_{c1} = p_{c2} . \tag{9}$$

Table 2: Assumptions for ORC layout in different working scenarios.

Parameter	Unit	Power-only	Cogeneration
Minimum temperature difference in economiser, $\Delta T_{ECO,\mathrm{min}}$	°C	45.0	30.0
Minimum temperature difference in evaporator, $\Delta T_{EVAP,\mathrm{min}}$	°C	50.0	30.0
Minimum temperature difference in regenerator, $\Delta T_{REG,\mathrm{min}}$	°C	30.0	30.0
Minimum temperature difference in condenser, $\Delta T_{COND,\mathrm{min}}$	°C	15.0	15.0
Overheating in evaporator ΔT_{sup}	°C	5.0	5.0
Overcooling in condenser, ΔT_{sub}	°C	5.0	5.0
Working medium pump efficiency, η_{MP}	–	0.35	0.35
Oil pump efficiency, η_{OP}	–	0.6	0.6
Coolant pump efficiency, η_{CP}	–	0.3	0.3
Cooling water temperature, T_{c1}	°C	25.0	45.0
Hot oil temperature, T_{o1}	°C	325.0	330.0
Minimum condensing temperature, $T_{condens,\mathrm{min}}$	°C	55.0	85.0
Maximum pressure in ORC loop, p_{max}	kPa(a)	1500.0	1500.0
Minimum condensing pressure, p_{min}	kPa(a)	20.0	20.0
Oil loop pressure, p_o	kPa(a)	300.0	300
Oil loop pressure, p_c	kPa(a)	300.0	300.0
Head of oil pump, ΔH_o	m	20	20
Head of coolant pump, ΔH_c	m	10	10

If for tested fluid temperature at minimal pressure p_{min} is lower than allowed $T_{condens,\mathrm{min}}$, then p_{min} is increased to met this limitation.

The cycle was optimized by changing pressure at turbine inlet, p_1. For each tested value of pressure p_1, the cycle was solved by changing three crucial temperatures: T_{s2}, T_{o3}, T_{c2}, and to meet assumed minimal temperature differences in the economiser, $\Delta T_{ECO,\mathrm{min}}$, evaporator, $T_{EVAP,\mathrm{min}}$, and condenser, $\Delta T_{COND,\mathrm{min}}$. Mass flows of oil, working fluid and coolant depend on iterated temperatures and are calculated basing on heat exchangers balance equations (10)–(18):

$$\dot{Q}_{ECO} = \dot{m}_{o1} \left(h_{o1} - h_{o3} \right) = \dot{m}_{s1} \left(h_{s1} - h_{s2} \right) , \tag{10}$$

$$\dot{m}_{o1} = \dot{m}_{o2} = \dot{m}_{o3} , \tag{11}$$

$$\dot{Q}_{EVAP} = \dot{m}_{o1} \left(h_{o1} - h_{o2} \right) = [\dot{m}_1 \left(h_1 - h_6 \right) , \tag{12}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_4 = \dot{m}_6 \,, \tag{13}$$

$$\dot{Q}_{COND} = \dot{m}_3 (h_3 - h_4) = \dot{m}_{c1} (h_{c2} - h_{c1}) \,, \tag{14}$$

$$\dot{m}_{c1} = \dot{m}_{c2} \,. \tag{15}$$

In case of regenerator it was assumed that minimal temperature difference occurs at the cold end of regenerator and temperature at its hot outlet was calculated. This allowed to calculate temperature of working fluid at evaporators inlet, T_6:

$$T_3 = T_5 + \Delta T_{REG,\min} \,, \tag{16}$$

$$\dot{Q}_{REG} = \dot{m}_2 (h_2 - h_3) = \dot{m}_5 (h_6 - h_5) \,, \tag{17}$$

$$T_6 = f (h_6, p_6) \,. \tag{18}$$

Net electrical power was chosen as the optimization criterion, which is given as

$$N_{net} = N_T - N_{OP} - N_{MP} - N_{CP} - N_{CT} \,. \tag{19}$$

In case of cogeneration mode consumption of energy by cooling tower, N_{CT}, was omitted. The turbine power is given as:

$$N_T = \dot{m}_1 (h_1 - h_{2s}) \eta_{isT} = \dot{m}_1 (h_1 - h_2) \,, \tag{20}$$

$$h_{2s} = f (s_1, p_2) \,, \tag{21}$$

$$s_1 = f (T_1, p_1) \,. \tag{22}$$

It was assumed that single stage axial turbine will be used for analysis. According to [18] axial-flow turbines are widely used in power generation sector, approximately contributing of 90% of worldwide power generation. It is due to the fact that axial turbines can be designed to fit a wide range of parameters including thermodynamic properties of fluids, pressure ratios, rotational speed and dimensions. They are successfully used as wind, hydro and gas turbines, in classic water-based Rankine cycle plants (using chemical energy of fossil fuels and nuclear energy) as well in organic Rankine cycle driven mostly by biomass, geothermal and waste energy [4,18]. Axial turbines are used in most of commercial ORC plants, especially with power output from 100 kW to several megawatts, with one or more stages and can reach efficiency from 80% to 90% [6]. The efficiency of axial turbine for ORC was estimated for each case using correlation (23) obtained from [18]

$$\eta_{isT} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \sum_{j=0}^{15} A_j F_j \,, \tag{23}$$

where A_j and F_j are the coefficients for the efficiency correlations and terms of the efficiency correlations, respectively, and summation is performed for 16 elements (number from 0 to 15) of the correlation enlisted in Table 3.

Table 3: Single stage axial turbine coefficients for isentropic efficiency estimation [18].

j	F_j	A_j
0	1	0.90831500
1	$\ln \mathrm{SP}$	−0.05248690
2	$\ln \mathrm{SP}^2$	−0.04799080
3	$\ln \mathrm{SP}^3$	−0.01710380
4	$\ln \mathrm{SP}^4$	−0.00244002
5	V_r	–
6	$\ln V_r$	0.04961780
7	$\ln V_r^2$	−0.04894860
8	$\ln V_r^3$	0.01171650
9	$\ln V_r^4$	−0.00100473
10	$\ln V_r \ln \mathrm{SP}$	0.05645970
11	$\ln V_r^2 \ln \mathrm{SP}$	−0.01859440
12	$\ln V_r \ln \mathrm{SP}^2$	0.01288860
13	$\ln V_r^3 \ln \mathrm{SP}$	0.00178187
14	$\ln V_r^3 \ln \mathrm{SP}^2$	−0.00021196
15	$\ln V_r^2 \ln \mathrm{SP}^3$	0.00078667

Two physical quantities: SP and V_r are independent dimensionless variables used for analysis, where SP represents the size parameter and V_r represents the volume ratio:

$$\mathrm{SP} = \frac{\dot{V}_{out,is}^{0.5}}{\Delta h_{is}^{0.25}}\,, \tag{24}$$

$$V_r = \frac{\dot{V}_{out,is}}{\dot{V}_{in}}\,, \tag{25}$$

where $\dot{V}_{out,is}$ is volume flow rate at turbine outlet, Δh_{is} is difference between enthalpy at turbine inlet and outlet assuming ideal expansion in turbine and $\dot{V}_{in}$ is volume flow rate at the turbine inlet.

The temperature at the turbine inlet, T_1, and outlet, T_2, was calculated as:

$$T_1 = T_{p_1,X=1} + \Delta T_{sup}\,, \tag{26}$$

$$T_2 = f\left(p_2, h_2\right), \tag{27}$$

$$h_2 = h_1 - \eta_{isT}\left(h_1 - h_{2s}\right), \tag{28}$$

where $T_{p_1,X=1}$ is the evaporation temperature at pressure p_1 and ΔT_{sup} is the superheating in evaporator. The temperature at the condenser outlet was calculated as

$$T_4 = T_{p_2,X=0} - \Delta T_{sub}, \tag{29}$$

where $T_{p_2,X=0}$ is the condensation temperature at p_2 and ΔT_{sup} is the subcooling in condenser.

The power consumed by the pump can be calculated as:

$$N_{MP} = \dot{m}_4\left(h_5 - h_4\right) = \dot{m}_4 \frac{h_{5s} - h_4}{\eta_{MP}}, \tag{30}$$

$$\eta_{MP} = \frac{h_{5s} - h_4}{h_5 - h_4}. \tag{31}$$

In order to estimate the power of the cooling tower, a polynomial function of two variables (the coolant mass stream, $\dot{m}_{c1}$, and the discharged rate of heat, $\dot{Q}_{COND}$) was developed, which was the result of adjusting the parameters of various cooling tower models offered by one of the manufacturers of such cooling systems [19]:

$$\begin{aligned} N_{CT} = {} & 0.2898 + 0.2407\dot{m}_{c1} + 0.007028\dot{Q}_{COND} - 0.00356\dot{m}_{c1}^2 \\ & + 7.599 \times 10^{-5}\dot{m}_{c1}\dot{Q}_{COND} - 2.976 \times 10^{-5}\dot{Q}_{COND}^2. \end{aligned} \tag{32}$$

Efficiency of electricity generation in ORC was calculated as

$$\eta_{ORC} = \frac{N_{net}}{\dot{Q}_{ECO}}. \tag{33}$$

Efficiency of ORC in cogeneration mode was calculated as

$$\eta_{COG} = \frac{N_{net} + \dot{Q}_{COND}}{\dot{Q}_{ECO}}. \tag{34}$$

Efficiencies of waste heat usage η_{HS} were defined as relation between useful products of ORC and available rate of waste heat $\dot{Q}_{HS}$. In case of power-only mode it is defined as

$$\eta_{HS} = \frac{N_{net}}{\dot{Q}_{HS}}. \tag{35}$$

In case of cogeneration it is defined as

$$\eta_{HS} = \frac{N_{net} + \dot{Q}_{COND}}{\dot{Q}_{HS}} \,. \tag{36}$$

Heat disposed in condenser $\dot{Q}_{COND}$ in cogenerative mode is used for heating purposes and is an additional useful product of ORC.

All calculations were executed in Python programming language [20] involving additional packages, majorly the Coolprop [21] for obtaining thermodynamic properties and SciPy [22] for optimisation and equation solving functions.

4 Results

For analysed heat carrier there is available $\dot{Q}_{HS} = 2884.\ 126$ kW of recoverable waste heat, according to assumptions shown in Section 3. Results for the best performing fluid in power-only mode are presented in Fig. 2 and results for best performing fluid in cogeneration are presented in Fig. 3. Detailed results of the analysis for all cases are presented in Table 4.

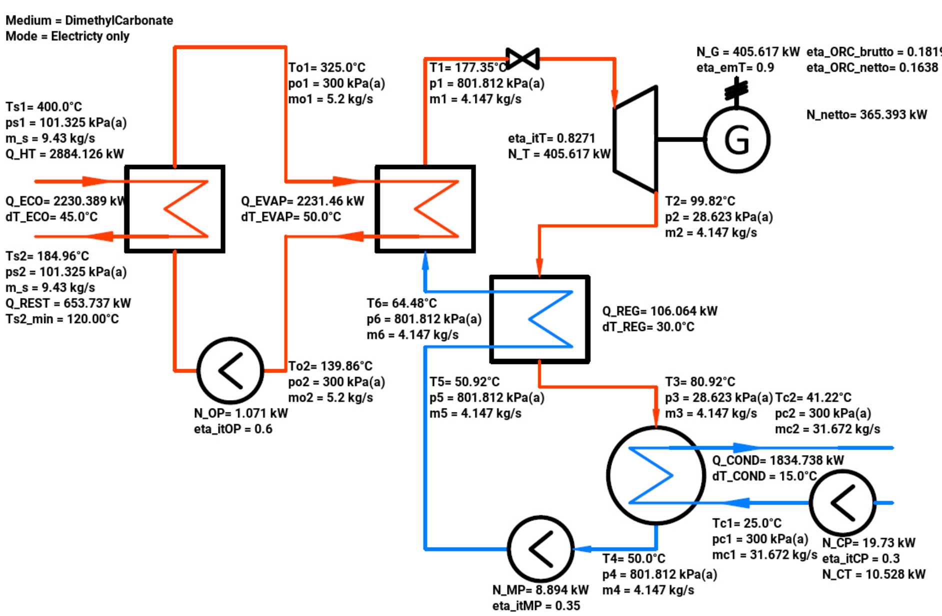

Figure 2: Detailed results of ORC using DMC as working fluid – power-only mode.

Table 4: Selected results of analysis.

Parameter	Unit	Mode					
		Power-only			Cogeneration		
		Fluids			Fluids		
		Toluene	MM	DMC	Toluene	MM	DMC
$\dot{m}_{o1}$	kg/s	5.214	5.226	5.204	5.414	5.452	5.399
$\dot{m}_1$	kg/s	3.959	6.125	4.147	4.375	6.872	4.550
$\dot{m}_{c1}$	kg/s	20.172	25.748	31.672	18.479	15.843	19.338
p_1	kPa(a)	484.037	829.931	801.812	807.446	1236.582	1439.253
p_2	kPa(a)	20.0	21.522	28.623	46.062	63.083	85.872
T_{s2}	°C	199.05	208.05	184.96	184.65	199.18	176.44
T_{o2}	°C	153.94	162.94	139.86	154.65	169.18	146.44
T_{o3}	°C	154.05	163.05	139.96	154.65	169.18	146.44
T_1	°C	181.63	199.07	177.35	209.19	222.56	208.82
T_2	°C	112.21	154.06	99.82	142.57	178.92	135.64
T_3	°C	87.61	81.31	80.92	142.57	178.92	135.64
$T_{condens}$	°C	61.92	55.00	55.00	85.00	85.00	85.00
T_4	°C	56.92	50.00	50.00	111.14	111.99	111.67
T_5	°C	57.61	51.31	50.92	81.14	81.99	81.67
T_6	°C	76.63	112.94	64.48	105.96	139.08	99.38
T_{c2}	°C	48.78	42.79	41.22	72.386	75.388	72.345
$\dot{Q}_{ECO}$	kW	2087.690	1996.245	2230.389	2233.559	2086.278	2316.570
$\dot{Q}_{EVAP}$	kW	2088.794	1997.372	2231.460	2234.707	2087.468	2317.695
$\dot{Q}_{REG}$	kW	138.723	776.494	106.064	210.435	839.605	157.648
$\dot{Q}_{COND}$	kW	1719.445	1637.282	1834.738	1854.398	1766.534	1937.616
N_T	kW	375.656	379.381	405.617	392.057	353.830	397.869
N_{OP}	kW	1.104	1.127	1.071	1.148	1.191	1.126
N_{MP}	kW	6.307	19.291	8.894	11.748	32.896	17.790
N_{CP}	kW	12.566	16.039	19.73	11.633	9.974	12.174
N_{net}	kW	346.774	332.968	365.393	367.528	309.770	366.780
N_{CT}	kW	8.906	9.955	10.528	0.000	0.000	0.000
η_{isT}	–	0.8354	0.8075	0.8271	0.8378	0.8178	0.8315
η_{ORC}	–	0.1661	0.1668	0.1638	0.1645	0.1485	0.1583
η_{COG}	–	–	–	–	0.9948	0.9952	0.9947
η_{HS}	–	0.1202	0.1154	0.1267	0.7744	0.7234	0.8032

In all cases electric output of ORC (Fig. 5) exceeded 300.0 kW, reaching the highest value for toluene in cogeneration mode (367.528 kW) and the lowest for MM in cogeneration mode (309.770 kW). In case of DMC the difference between power-mode (Fig. 3) and cogeneration mode (Fig. 4) is

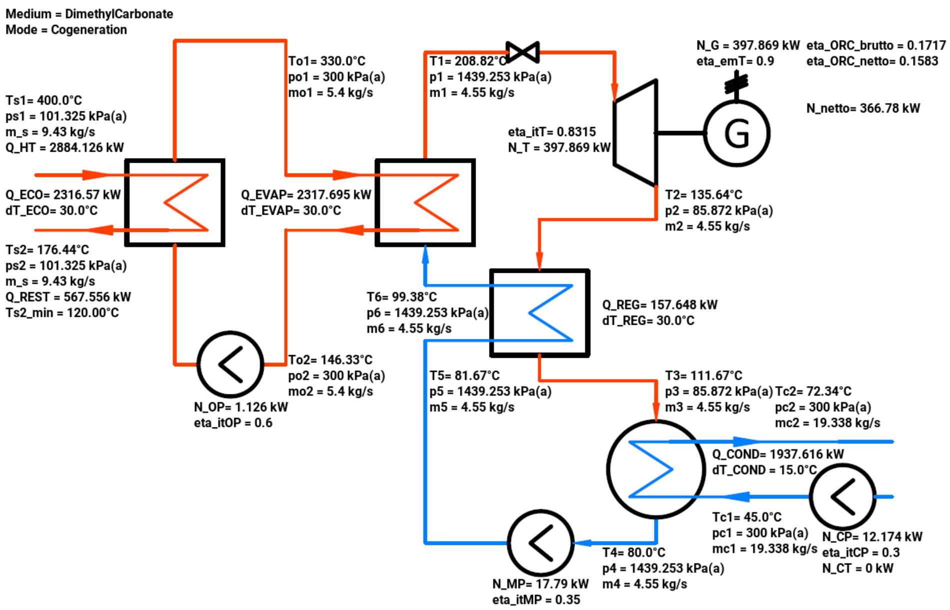

Figure 3: Detailed result of ORC using DMC as working fluid – cogeneration mode.

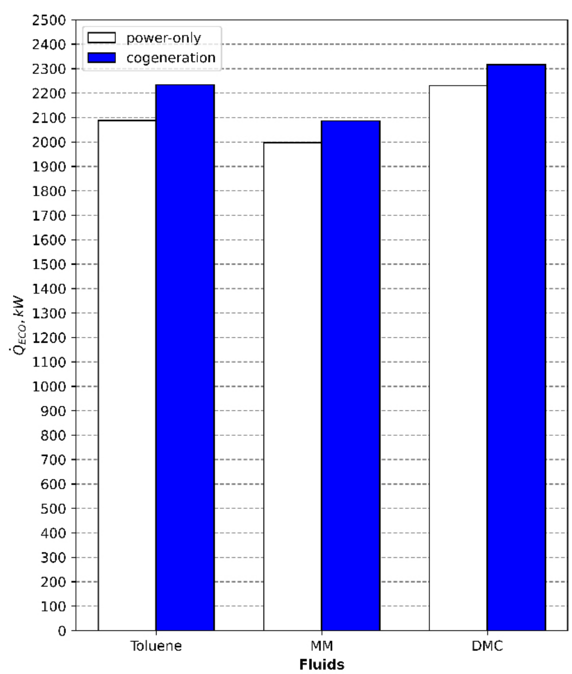

Figure 4: Heat harvested in economiser.

less than 2.0 kW. In case of toluene the power output is even higher in case of cogeneration mode for more than 20.0 kW. Only in case of MM cogeneration ORC mode generates 23.198 kW less power than in power-only mode.

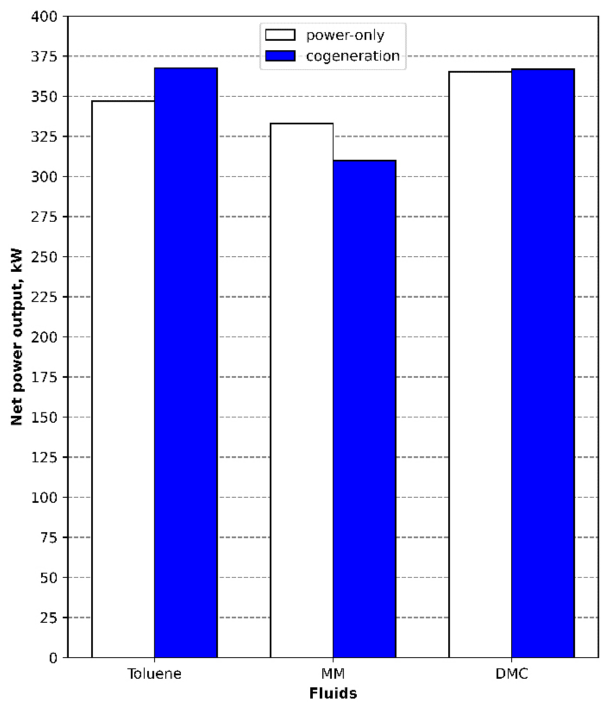

Figure 5: Net power output.

Amount of energy harvested through economiser (Fig. 4) is higher for each analysed fluid in case of cogeneration mode.

Efficiency of turbine (Fig. 6) reached more than 80% for each case, with the highest value (83.78%) for toluene in cogeneration mode and the lowest for MM in power-only mode (80.75%). The efficiency of ORC is presented for each case in Fig. 7. Except for MM and DMC in cogeneration mode, all other scenarios reached efficiency higher than 16% with the highest result for MM in power-only mode (16.68%) and also the lowest for MM in cogeneration mode (14.85%). Nonetheless in every case efficiency of power generation was always higher in power-only scenario, while in case of cogeneration mode the highest efficiency was reached for toluene and slightly less for DMC.

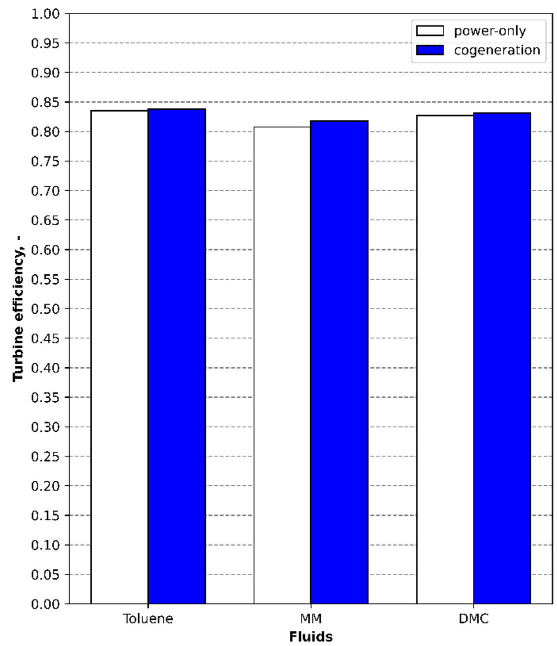

Figure 6: Turbine efficiency.

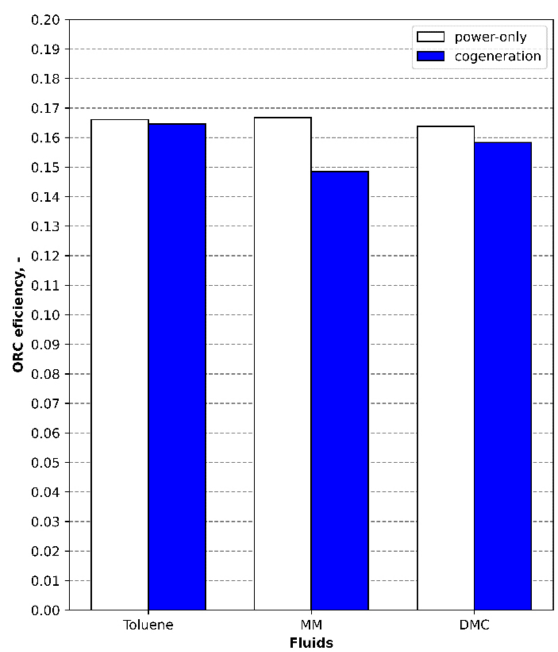

Figure 7: Net efficiency of ORC.

For all fluids in cogeneration mode the temperature of coolant leaving condenser was higher than 70°C. In all cogenerative cases efficiency of ORC unit including heat recovered from condenser was higher than 99% when referred to heat harvested in economiser and more than 70% when referred to all heat considered as recoverable from heat carrier (Fig. 8) according to assumptions presented in Section 3, with the highest value for DMC (79.90%) and the lowest (71.99%) for MM. In case of power-only scenario it is significantly lower, because in this case electricity is considered as only useful product of ORC.

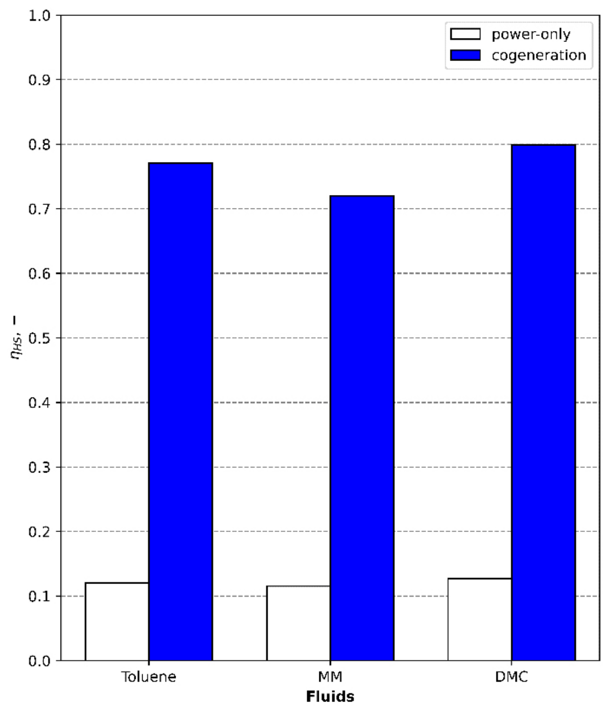

Figure 8: Efficiency of ORC referred to technically available rate of heat $\dot{Q}_{HS}$.

5 Conclusion

Results shows that even in cogeneration mode ORC unit can have similar or even higher net power output in comparison to units dedicated purely to power generation. Cogeneration unit can be taken into account when the waste heat source is already used for central heating to enable production of additional electricity or on request of potential user. According to analysis presented in [4] (small units and large units with power output higher than 1 MW), planned unit with calculated power output in all scenarios can be classified as small ORC unit and according to assumptions presented in [3] all calculated cycles are subcritical. Comparing power outputs obtained in section 4 with data gathered in [5] it can be noticed that planned unit will be one of the smallest available in this area of application (glass industry). Units with lower power output are applied by DeVeTec (250 kW) and under construction and validation by Enerbasque (100 kW). Other manufacturers like Turboden, Ormat and Exergy have applied units with power ranging form 500 kW to 6200 kW, with more than half of applied units with power higher than 1 MW. In 2018 summarised power output of 10 projects for glass industry was 25.25 MW worldwide (including one project under construction), with 8.65 MW in Europe. Comparing this to a possible electricity production in ORC system applied only in flat glass manufactures estimated to be 78.5 MW in EU27 it shows that there is still a great potential for implementing ORC technology in this area of industry.

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