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Energetic efficiency of the combined ORC and gas turbine installation powered by the anaerobic sewage sludge stabilization system

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Abstract The present paper describes a cycle, which may be applied in sewage treatment plants as a system to convert biological waste into process heat and electricity. In sludge stabilization processes anaerobic fermentation acts as the source of methane, which can be used then to generate heat and electric current in gas turbines. Products of high-temperature oxidation can be utilized in organic Rankine cycles to generate electric power. Waste heat is used for heating the fermenting biomass. Energy balance equations mentioned in the thesis: organic Rankine cycle, regenerative gas turbine engine, anaerobic sludge stabilization system.

Keywords: ORC; Sludge; Biogas; Gas turbine

Nomenclature

B	–	calorific value
c_p	–	specific heat
En_{el}	–	energy of the electricity
h	–	enthalpy
$\dot{m}$	–	mass flow
p	–	pressure
T	–	temperature
$\dot{Q}$	–	rate of heat
$\dot{W}$	–	electric power

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Greek symbols

ξ	–	corrective coefficient for fluid-flow machines, strictly associated with their internal efficiency
η	–	efficiency

Subscripts

a	–	air
b	–	biogas
c	–	compressor
con	–	condenser
$evap$	–	evaporator
g	–	gas
gt	–	gas turbine
n	–	net power
ks	–	combustion chamber
th	–	theoretical
$turb$	–	turbine

1 Introduction

An extensive literature exists regarding the organic Rankine cycles (ORC) and, in particular, fluid selection for waste heat recovery applications [1,2]. Proposal of combined cycle with a topping 1.3 MW gas turbine fuelled by gasified biomass and a bottoming ORC plant can be found in [3]. Muńoz *et al.* [4] examined part-load performance of gas turbine and ORC combined cycles using Thermoflow's GTPro [5] software in order to simulate the performance of gas turbines. Rowen [6] and Camporeale *et al.* [7] developed mathematical models that scrutinise part-load performance of gas turbines. Cao *et al.* [8] showed how to estimate the off-design performance of gas turbines by characteristic curves. A review on potential innovations in production, conditioning and reprocessing of biogas can be found in [9] – the authors show all possible technologies for obtaining power and heat from biomass fermentation. Issues similar to those discussed in this paper are described in [10]. Some general and economic considerations related to making choice between a steam cycle and ORC are discussed in [11].

Sewage treatment plants reprocess biosolids and other waste products. Biosolids can be processed in order to raise the energetic efficiency of the sewage treatment plant, which is possible due to the newest energetic technologies. The efficiency can be improved through installation of a gas turbine with a regenerative air heater that raises the air temperature in gas turbine combustion chamber.

This paper presents an exemplary thermal model of a gas turbine installation with the regenerative air heater (Fig. 1), in which hot combustion gases power the ORC cycle and raise the temperature in the digestion chamber. The paper includes also a comparison of a regenerative gas turbine installation and a gas turbine installation without the regenerative heater.

2 Concept of the thermal cycle

In gas turbine systems, products of chemical reactions oxidize at high temperatures. Products of high-temperature oxidation can be used to generate electric power in the ORC cycles and, through releasing waste heat, can support other systems of the sewage treatment plant.

Main elements of the juxtaposed installations are as follows:

a) Digestion chamber
The process of anaerobic sewage sludge stabilization takes place in the digestion chamber, which usually consists of a power system, heating system, blending system, fermented mash drainage system and a biogas drainage system.

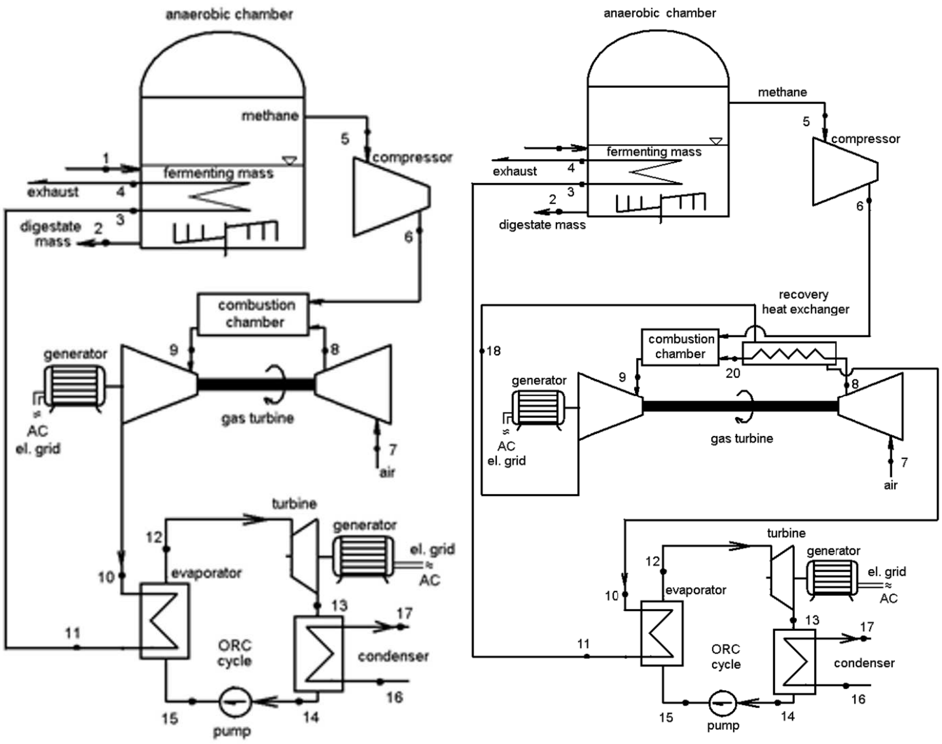
b) Gas turbine (Fig. 1a)
Gas turbine with air compressor, combustion chamber and turbine with an electric generator, in which the combustion products are decompressed. There is also an additional system, which compresses the biogas so that its pressure becomes equal to the pressure in the combustion chamber.

c) Gas turbine (Fig. 1b)
Gas turbine with air condenser, combustion chamber, additional regenerative exchanger and turbine.

d) Organic Rankine cycle
ORC cycle with boiler, condenser, circulation pump and gas turbine with electric generator.

It is presumed that in the digestion chamber biogas with low concentrations of methane and carbon dioxide is released (40% and 55%, respectively). The biogas produced in the digestion chamber amounts to approximately 60% of the total batch. It is produced by the thermophilic bacteria. Calorific value of the biogas is assumed to be about 14.350 kJ/nm^3 [12,13,14]. Toluene

– operating medium used in the ORC cycle – enables the utilisation of efficient heat exchange processes, which occur during the boiling phase change [9,15,16].



a) b)

Figure 1: Concept of a system that can improve energetic efficiency of sewage treatment plants: a) gas turbine installation without the regenerative system, b) gas turbine installation with the regenerative system.

3 Assumptions and energy balance for the proposed thermal cycle

Figure 1a shows a setup discussed in detail therein [17], whereas Fig. 1b presents a setup designed for the proposed thermal cycle.

Assumptions and simplifications:

- The model describes phenomena taking place in steady state.
- Internal energy losses of the condenser and gas turbine are presumed to be at a constant level (Tab. 1).
- Heat exchange processes are ideal.
- Pressure loss in pipelines was neglected.
- There are specified operating conditions for the gas turbine system and the digestion chamber.
- Operating conditions for the ORC cycle are variable.
- The ORC cycle pump work was omitted in the presented model because it was assumed that temperature increase in the ORC cycle pump equals zero. Electric power delivered to the ORC cycle pump that operates with toluene as working fluid is negligible in comparison with power generated in the ORC cycle turbine.
- Toluene was chosen as the working fluid in the ORC cycle due to its beneficial thermodynamic properties and relatively high critical point temperature which allows it to be used in ORC cycles operating at higher temperatures and lower pressures (318 °C, 4.1 MPa) than cycles based on explosive hydrocarbon fluids (acetone – 234 °C, 4.7 MPa, ethanol – 241 °C, 6.2 MPa).

The following equation shows mass balance in a gas turbine.

$$\dot{m}_6 + \dot{m}_8 = \dot{m}_9 \, . \tag{1}$$

Other equations depicting the mass balance at certain points result from the processes specified in Fig. 2. The equations below describe the proposed thermal cycle.

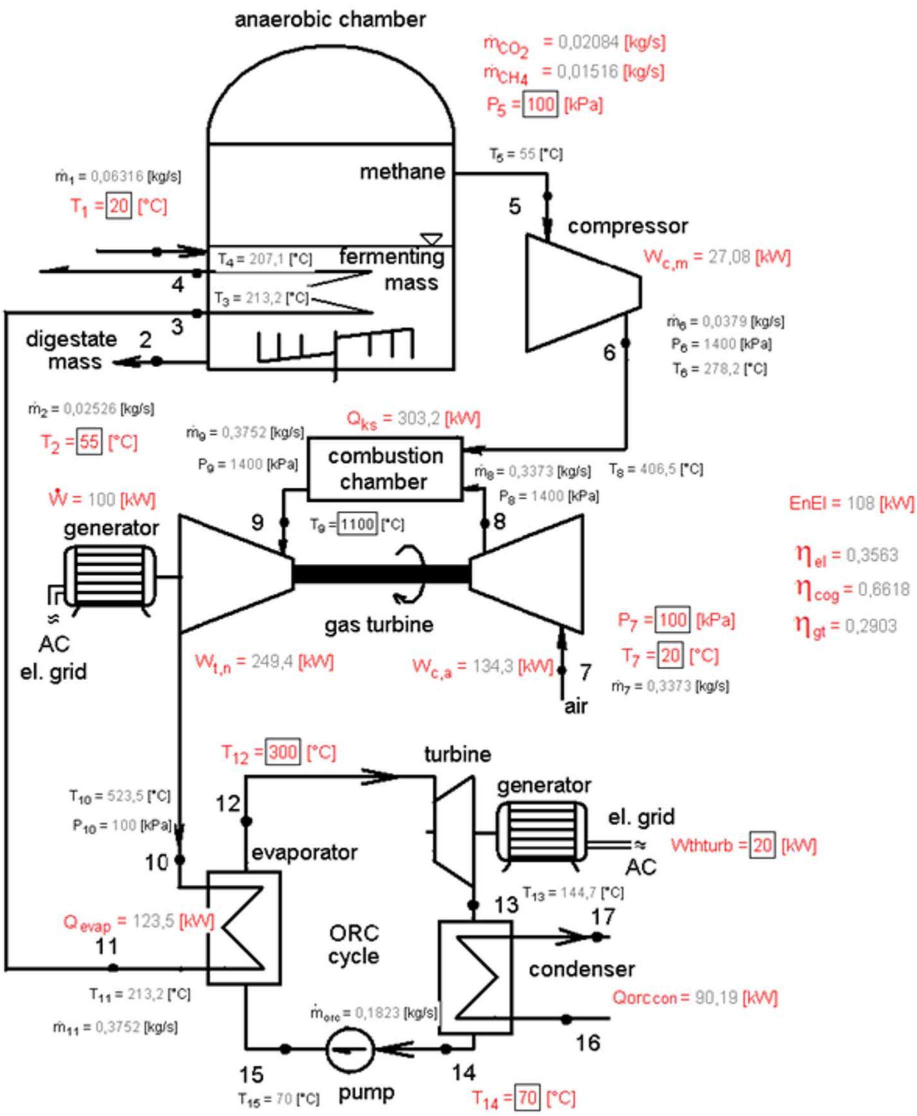


Figure 2: Screenshot of the software for thermodynamic parameters calculation in gas turbine thermal cycle without a regenerative system.

- Total energy balance of the fermentation chamber:

$$\dot{m}_1 T_1 c_{p,1} - \dot{m}_5 T_5 c_{p,5} - \dot{m}_2 T_2 c_{p,2} + \dot{m}_3 T_3 c_{p,3} - \dot{m}_4 T_4 c_{p,4} = 0 \,. \tag{2}$$

- Heat delivered to the fermentation chamber:

$$Q_b = \dot{m}_3 T_3 c_{p,3} - \dot{m}_4 T_4 c_{p,4} \,. \tag{3}$$

- Power delivered to the biogas compressor:

$$W_{c,m} = \xi_{c,m} (h_6 - h_5) \, \dot{m}_6 \,. \tag{4}$$

- Power delivered to the gas turbine air compressor:

$$W_{c,a} = \xi_{ca} (h_8 - h_7) \, \dot{m}_8 \,. \tag{5}$$

- Power obtained from the gas turbine system:

$$W_{t,n} = \xi_{tg} (h_9 - h_{10}) \, \dot{m}_9 \,. \tag{6}$$

- Energy balance of the gas turbine regenerative exchanger:

$$\dot{m}_8 \, (h_{20} - h_8) - \dot{m}_9 \, (h_{18} - h_{10}) = 0 \,. \tag{7}$$

- Thermal balance of the gas turbine combustion chamber:

$$\dot{m}_5 B + \dot{m}_5 h_6 - h_9 \dot{m}_9 + h_{20} \dot{m}_{20} = 0 \,, \tag{8}$$

  where B is the calorific value.

- Heat delivered to the gas turbine combustion chamber:

$$\dot{Q}_{ks} = \dot{m}_5 B \,. \tag{9}$$

- Thermal balance of the boiler:

$$(h_{10} - h_{11}) \, \dot{m}_9 - (h_{12} - h_{14}) \, \dot{m}_{ORC} = 0 \,. \tag{10}$$

- Energy balance in the boiler:

$$\dot{Q}_{evap} = (h_{10} - h_{11}) \, \dot{m}_9 \,. \tag{11}$$

- Power generated in the steam turbine of the ORC cycle:

$$W_{th,turb} = \xi_{tp}\left(h_{12} - h_{13}\right)\dot{m}_{ORC}\ . \tag{12}$$

- Thermal power received by the ORC cycle condenser:

$$\dot{Q}_{orc,con} = \left(h_{13} - h_{14}\right)\dot{m}_{ORC}\ . \tag{13}$$

- Total power generated by the given thermal cycle:

$$En_{el} = W_{t,n} - W_{c,m} + W_{th,turb} - W_{c,a}\ . \tag{14}$$

- Gas turbine electric efficiency:

$$\eta_t = \frac{W_{t,n} - W_{c,m} - W_{c,a}}{Q_{ks}}\ . \tag{15}$$

- Electrical efficiency of the gas turbine and the ORC cycle:

$$\eta_c = \frac{W_{t,n} - W_{c,m} + W_{th,turb} - W_{c,a}}{Q_{ks}}\ . \tag{16}$$

- Cogenerative efficiency:

$$\eta_k = \frac{En_{el} + Q_b + Q_{orc}}{Q_{ks}}. \tag{17}$$

The equations mentioned above allow to determine total energy balance of the whole thermal cycle.

4 Results of the calculations

Calculations were done for the assumed loading of the ORC cycle turbine that changes from 20 to 30 kW. The analysed range of electric power generated in the ORC cycle turbine is also present in the commercially available devices manufactured by V-EnerTek [17]. Table 1 gathers results of the calculations made in accordance with the equations given in Sec. 4. There are also data connected with modelled subsystems of the presented thermal cycle.

Table 1: Results of calculations for presented thermal cycles.

$W_{th,turb}$	Q_{evap}	$Q_{orc,con}$	En_{el}	η_c	η_k
kW	kW	kW	kW	–	–
Gas turbine system without regeneration					
30	185.3	135.3	118	38.92	84.35
29	179.1	130.8	117	38.59	82.53
28	172.9	126.3	116	38.26	80.72
27	166.8	121.8	115	37.93	78.90
26	160.6	117.2	114	37.60	77.08
25	154.4	112.7	113	37.27	75.27
24	148.2	108.2	112	36.94	73.45
23	142.1	103.7	111	36.61	71.63
22	135.9	99.21	110	36.29	69.81
21	129.7	94.70	109	35.96	68.00
20	123.5	90.19	108	35.36	66.18
Gas turbine system with regeneration					
30	185.3	135.30	119.7	46.10	98.99
29	179.1	130.80	118.7	45.71	96.87
28	172.9	126.30	117.7	45.33	94.75
27	166.8	121.80	116.7	44.94	92.63
26	160.6	117.20	115.7	44.56	90.51
25	154.4	112.70	114.7	44.17	88.38
24	148.2	108.20	113.7	43.79	86.26
23	142.1	103.70	112.7	43.40	84.14
22	135.9	99.21	111.7	43.02	82.02
21	129.7	94.70	110.7	42.63	79.9
20	123.5	90.19	109.7	42.25	77.78

Additional data		
Parameter	Unit	Magnitude
$W_{t,n}$	kW	249.40
$W_{c,a}$	kW	134.30
$W_{c,m}$ (Fig. 1a)	kW	27.80
$W_{c,m}$ (Fig. 1b)	kW	23.20
Q_{ks} (Fig. 1a)	kW	303.20
Q_{ks} (Fig. 1b)	kW	259.70
Q_{reg} (Fig. 1b)	kW	44.41
ξ_{cm}	–	1.200
ξ_{ca}	–	1.205
ξ_{tg}	–	0.870
ξ_{tp}	–	0.600

Figures 2 and 3 depict screenshots of software designed in Enginering Equation Solver (EES) in order to determine appropriate thermal parameters for thermodynamic cycles presented in this paper. The developed software makes use of the first principle of thermodynamics so as to determine the efficiency of the systems presented in Figs. 1 and 2.

Table 2 shows how the efficiency changes together with gas turbine load modifications. It can be noticed that application of the regenerative exchanger resulted in medium increase in the efficiency of electric current production (by 7%) and medium increase in the cogenerative efficiency (by 13%).

Results of the calculations presented in Figs. 2 and 3 correspond to the turbine power $W_{th,turb}$ = 20 kW (Tab. 1). These results prove that the efficiency of the whole system increase together with the electric power produced in the ORC cycle. This tendency applies also to the cogenerative efficiency of the whole system.

Table 2: Electric and cogenerative efficiency increase in the regenerative gas turbine system.

	Electric efficiency			Cogenerative efficiency		
$W_{th,turb}$, kW	η_c	$\eta_{c,reg}$	$\Delta\eta_{c,reg}$	η_k	$\eta_{k,reg}$	$\Delta\eta_{k,reg}$
	Thermal cycle without a regenerative heat exchanger	Thermal cycle with a regenerative heat exchanger	Increase in the efficiency	Thermal cycle without a regenerative heat exchanger	Thermal cycle with a regenerative heat exchanger	Increase in the efficiency
30	38.92	46.1	7.18	84.35	98.99	14.64
29	38.59	45.71	7.12	82.53	96.87	14.34
28	38.26	45.33	7.07	80.72	94.75	14.03
27	37.93	44.94	7.01	78.90	92.63	13.73
26	37.60	44.56	6.96	77.08	90.51	13.43
25	37.27	44.17	6.90	75.27	88.38	13.11
24	36.94	43.79	6.85	73.45	86.26	12.81
23	36.61	43.40	6.79	71.63	84.14	12.51
22	36.29	43.02	6.73	69.81	82.02	12.21
21	35.96	42.63	6.67	68.00	79.90	11.90
20	35.36	42.25	6.89	66.18	77.78	11.60
			Average increase			Average increase
			6.92			13.11

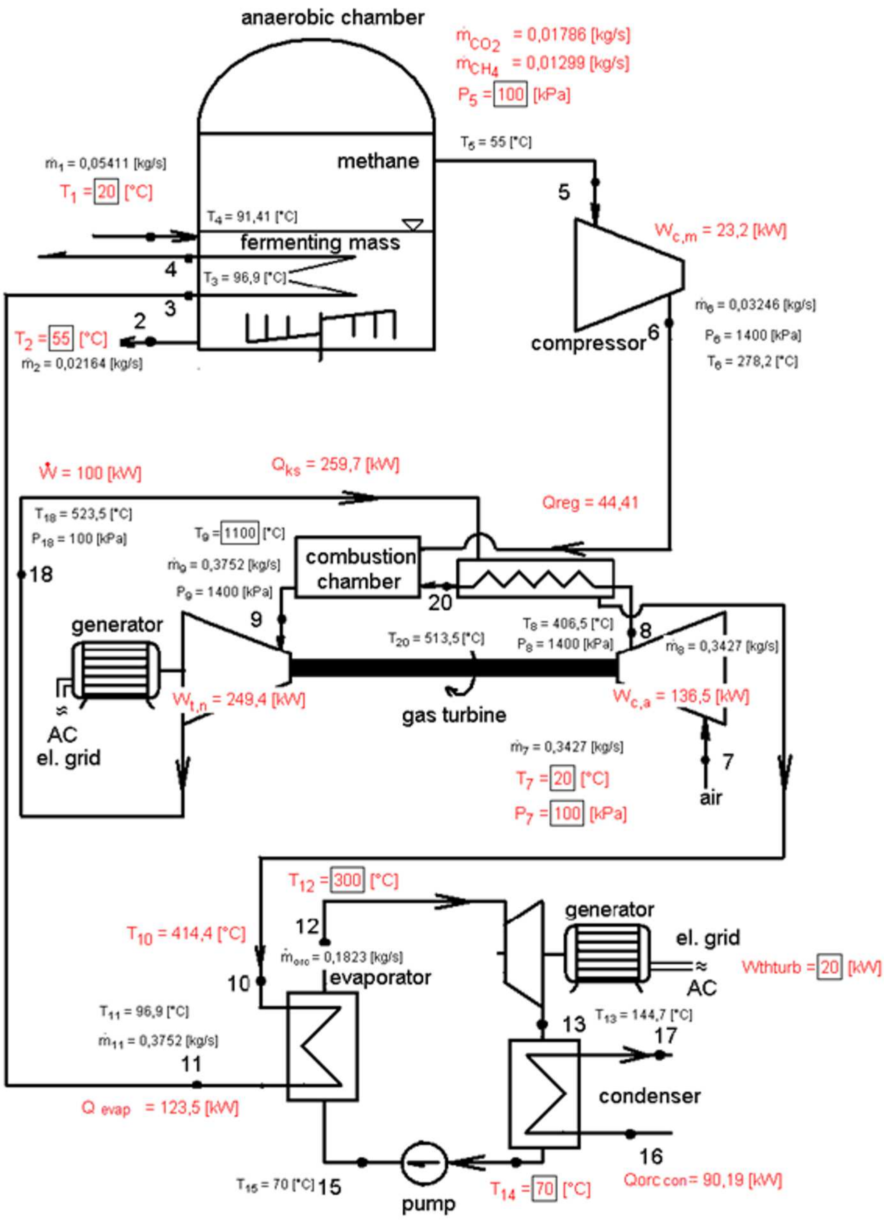


Figure 3: Screenshot of the software for thermodynamic parameters calculation in gas turbine thermal cycle with a regenerative system.

The model juxtaposes two types of gas turbines. Exhaust fumes from big capacity gas turbines attain nearly 500 °C. This enables one to work with water-based cycles (in which water is the operating medium). Small gas turbines attain the temperature of 275 °C (e.g., Capstone C30). These can improve either the efficiency of the electric current production if equipped with an additional micro ORC cycle or total efficiency of the whole system if equipped with regenerative heat exchangers.

5 Summary

An increasing number of investments in the environment protection systems (e.g., sewage treatment plants) should encourage the investors to devote some attention to their energetic efficiency. While introducing new energetic installations to the sewage treatment plants, it is necessary to make efficiency analysis of applicable energetic systems in order to choose the most suitable one. Introducing gas turbines to the sewage treatment plants, especially those with regenerative systems, may improve their total energetic efficiency and provide financial benefits.

This paper describes two different gas turbine systems, one out of which works with a regenerative system. The ORC cycle improves energetic efficiency in both systems powered by biogas obtained from the sludge stabilization processes. Heat obtained in the ORC cycle condenser can cause an increase in total cogenerative efficiency of the system (at 70 °C it can be used in absorption refrigeration cycle or for heating purposes). Electric efficiency of a system without both the ORC cycle and the gas turbine regenerative exchanger amounts to 29.03%, whereas the electric efficiency of a gas turbine with the regenerative heat exchanger totals up to 34.55%. In the case of higher efficiency of the combined system, a faster return on investment may be expected than in the case of the system of lower efficiency.

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