2nd International Conference on Sustainable Solid Waste Management, Athens, June 12th-14th 2014.

Energy recovery from heat produced during aerobic treatment of organic waste through exploitation by micro organic Rankine cycle (ORC).

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Abstract

The study has investigated the possibility of recovering low heat available from the exhaust air coming from a full scale aerobic biological treatment section of an existing mechanical biological treatment plant for municipal solid waste supplemented by an Organic Rankine Cycle (ORC) section. Moreover the study analyzed the possibility of recovering heat also by exploiting the latent condensation heat of vapor, largely contained in the exhaust air. The analysis refers to an aerobic section treating about 32,000 $tyear^{-1}$. The inlet ORC exhaust air's mean temperature and composition are assumed to be, on the basis of experimental analysis performed on the basin, respectively of 341 K and 0.02%vol CH_4, 2.70%vol CO_2, 18.5%vol O_2 and78.7%vol N_2, in saturated conditions. The energetic analysis was performed for different ORC evaporator outlet temperatures ranging among 340 K and 316 K. Considering the operational conditions, the pressure ratio results to be among 1.9 and 3.5. A maximum net power production of about 19kW results for an outlet temperature of 321K and a pressure ratio of 2.17.The net cycle efficiency ranges among 3%-5%, instead the exergetic efficiency ranges among 1.5%-11%. The economic analysis, referred to the maximum net energy output, shows an ORC investment cost of 2,872.9€kW^{-1} and a total cost of 0.096 €kWh^{-1} considering an operative lifetime of 10 years and including the operation and maintenance costs.

Keywords

Aerobic treatment, Electrical energy, Exergetic analysis, Organic Rankine cycle, Organic waste.

Introduction

It is an established practice to exploit the biological processes to treat the organic fraction of municipal solid waste (MSW). The aim of the biological treatment facilities for organic fraction depends on its features and it could be to reduce the material's reactivity, to recover material or to produce renewable energy (Athanasoulia et al.,2012), (Cavinato et al., 2013), (Fricke et al.,2005), (Gomez, 2006),(Hessami et al.,1996), (Liu et al.2012), (Nguyenet al., 2007).

In fact the aerobic treatment facilities are normally exploited for the stabilization of organic matter before landfill disposal (Binner, Zach, 1999), (De Gioannis et al., 2009), (Komilis et al.,1999), to reduce the production of pollutants, but also for the production of organic fertilizers (Dlg.29/04/10 n75), (Eklind Y, Kirchmann H, 2000). The anaerobic digestion is a suitable way to produce a biomethane-rich gas exploitable as fuel in internal combustion engine, and so to produce renewable electrical energy (Salter RA, Frederickson J, 2001), (Bolonzella D, 2006), (De Baere L, Mattheeuws, 2010b), (Di Maria et al., 2012a), (Di Maria et. al, 2012b).

The aerobic biological process could also be exploited for the production of renewable electrical energy by recovering the exhaust air's heat (Di Maria et al., 2014a), (Di Maria et al., 2014b). In fact during the aerobic process the oxidation (Liwarska-Bizukojc, 2003) of the organic matter (OM) takes place through the aerobic bacteria, and that leads to the production of about 17,000-18,000kJ of heat per kg of OM (Themelis NJ et al., 2002). The aerobic bioconversion activity is higher during the first 2-4 weeks of the treatment, due to the initial high concentration of rapidly biodegradable material. So an high heat production rate and an increase in the OM mass temperature, and consequently in exhaust air temperature, occur. Maximum temperatures achieved during the aerobic process in full scale facilities range between

328 K to 348 K, depending by OM content, thermal loss, OF humidity content, and process air rate (Di Maria, 2012).

The possibility of exploiting the heat produced during the aerobic biological treatment was investigated in many previous studies. Di Maria et al.(2008) evaluated the possibility of recovering this heat for domestic use by heat pumps. Main results showed that the temperature of exhaust air ranges between 328 K and 343 K, depending mainly on the organic waste treated and the process air rate. The heat recoverable from the analyzed scenario ranged from about 120 to 350 kWh per treated tonne. A further possible solution to recover the low thermal energy from exhaust air coming from aerobic biological process is to exploit the Organic Rankine Cycle (ORC) systems. The ORC systems use the same components of the traditional steam power plants but with an organic fluid to extract low-grade thermal energy and so generate electricity. The ORC systems are mainly diffused in renewable energy production plants and industrial applications, such as biomass power (Drescher et al., 2007), solar power (Yamamoto et al., 2001), ocean energy conversion, geothermal power (Saleh et al., 2007), (Di Pippo R, 2005) and the waste heat recovery power (Kuo et al., 2001). Desideri and Di Maria (1998) report that the exploitation of ORC for recovering exhaust heat from humid air turbine cycle can lead to an overall cycle efficiency increase from 1.6% to 2.2%. Wang et al. (2011) analyse the effect of different working fluids on ORC efficiency for engine waste heat recovery. Similarly Hung et al. (2010) investigate the effect of different organic working fluids on ORC efficiency using heat generated by solar power and ocean thermal energy. Di Maria et al. (2014a) analyzed the possibility of recovering electrical energy from integrated aerobic-anaerobic treatment of organic waste. In this study the main results show that it is possible to produce from 1 kW to about 25 kW for a facility processing about 20,000 $tyear^{-1}$ of organic waste. Also the possibility of recovering thermal energy from exhaust air upgraded by the combustion of solid recovered fuel was investigated, with a power output ranging from 9 to 12 kW (Di Maria et al., 2014b).

In these previous studies only the sensible heat of exhaust air was recovered. Actually, the exhaust air, coming from the aerobic biological treatment of organic waste, contains a relevant amount of vapor ranging from about 100g per kg of dry air to 270g per kg of dry air, depending on temperature conditions. For this reason in the present study is investigated the possibility of recovering further renewable energy by exploiting also the latent condensation heat of vapor in exhaust air coming from a full scale aerobic biological treatment (BT) section of an existing mechanical biological treatment (MBT) plant supplemented by an ORC section. The study is performed by experimental, literature data and the aid of a numerical model, in order to evaluate the ORC efficiency and the system performances in different scenarios.

Materials and methods

Aerobic treatment experimental analysis

The experimental analysis was performed in an BT section of an existing MBT plant for MSW (Di Maria et al., 2012).The MSW is firstly conveyed to mechanical and physical screening devices that allow to recover the metals and to separate the inlet waste stream in two main outlet streams: a dry fraction, mainly rich in plastics, paper and cardboard, textile, wood, and a wet organic fraction (WOF), mainly rich in organics and fines (*i.e.* particle <20mm).Then the WOF is conveyed to the BT section to perform a mass and reactivity reduction before landfilling. In the considered BT facility are treated about 32,000 $tyear^{-1}$ of WOF, coming from the previous mechanical treatment section. The BT is performed into a concrete basin (Fig. 1), with an aerated floor through which electrical fans deliver the air rate of about 4,000Nm^3h^{-1} (Tab. 1) required by the aerobic biological process. On the basin a crane bridge with screws works, making the WOF stir and move from the inlet section to the outlet and creating a continuous stream of waste. During the process the humidity content is controlled and regulated to maintain the optimal process conditions.

[Figure 1 here]

In the first phase of the aerobic treatment process a large amount of the rapidly biodegradable OM has been oxidized and consequently an increase in WOF mass temperature occurs. During the biological process the waste mass temperature has been manually measured with a K-type thermocouple at the depth of about 1m from the waste mass surface. The measuring was performed at different times in different points of the basin. At least three measurement per each sampling point were performed. Contemporaneously an analysis of the exhaust air quality was performed. The composition of exhaust air quality (%vol) was investigated with the aid of a portable gas analyzer and a storage volume placed on waste mass surface. The CH_4(±1%) and CO_2(±1%) concentrations have been measured by infrared sensors whereas the O_2(±2%) concentration have been measured by electrochemical cells.

ORC section model

The heat produced during the biological process is mainly exchanged with the process air which heats up. A significant amount of heat is also absorbed by the water evaporation. In fact the exhaust air could be considered at saturated vapor condition because while passing through the waste mass it is enriched by the vapor. The vapor comes from the evaporation of the water produced by aerobic biological process, of the water contained in WOF and of the water which has been injected to maintain the optimal process conditions. To evaluate the exhaust air temperature (T_8) (K) in the outlet section of the BT section (Fig. 1), a difference with the WOF mass mean temperature of 5K was assumed (ΔT_{WOF}) (Tab.1) (Di Maria et al., 2008).On the basis of the experimental exhaust air investigations a mean exhaust air composition was assumed (Tab. 1).The N_2 concentration was evaluated assuming that the O_2 reacts only with organic carbon (Tab. 1). The exhaust air was assumed at atmospheric pressure conditions (P_8) (Tab.1) and with a relative humidity (Φ) of 100% (Tab.1).The exhaust air vapor mass rate ($\dot{m}_{VAP}$)(kgs^{-1}) was evaluated, according to equations (1) and (2), for different T_9 (K) scenarios (Fig. 1-2) considering the dry air mass rate ($\dot{m}_{AIR}$) (kgs^{-1}) and the T_8 temperature to be constant. The x ($kgkg^{-1}$) is the specific ratio between vapor mass (M_{VAP}) (kg) and air mass (M_{AIR}) (kg) (Eq. 1) depending on pressure and temperature conditions. The heat transferred from exhaust air, coming from BT (Q_{BT}) (kW), inside the evaporator, was evaluated in each T_9 (K) scenarios (Fig.1-2) and it is represented by equation (3). Q_{BT} consists of the sum of the heat coming from the dry air (Q_{AIR}) (kW) and the heat coming from the phase change of the vapor (Q_{VAP}) (kW) between the T_8 and T_9 temperatures. The Q_{AIR} and Q_{VAP} are described respectively by equation (4) and (5). c_p^{AIR} ($kJkg^{-1}K^{-1}$) represents the specific heat at constant pressure for dry air (Eq. 4). The Δh_{VAP}($kJkg^{-1}$) (Eq. 5) represents vapor's phase change enthalpy and $\Delta\dot{m}_{VAP}$ (kgs^{-1}) (Eq. 5) is the vapor's condensing mass during the cooling from T_8 to T_9.

$$x = \frac{M_{VAP}}{M_{AIR}} \qquad (kgkg^{-1}) \qquad (1)$$

$$\dot{m}_{VAP} = x \cdot \dot{m}_{AIR} \qquad (kgs^{-1}) \qquad (2)$$

$$Q_{BT} = Q_{AIR} + Q_{VAP} \qquad (kW) \qquad (3)$$

$$Q_{AIR} = \dot{m}_{AIR} \cdot c_p^{AIR} \cdot (T_9 - T_8) \qquad (kW) \qquad (4)$$

$$Q_{VAP} = (\Delta\dot{m}_{VAP} \cdot \Delta h_{VAP})_{T9-T8} \qquad (kW) \qquad (5)$$

The thermodynamic, economic and environmental properties of working fluids normally used in organic Rankine cycle (ORC) could be significantly different (Dongxiange et al., 2012). In the present study, in accordance with Wang et al.(2012), on the basis of the temperatures achieved by the waste mass, R-123 was chosen as working fluid (Tab. 2). Figure 2 represents a possible T-s diagram for the ORC. The condenser's temperature (T_c) (K) has been imposed (Tab. 2).The ambient temperature (T_{amb}) was imposed at 288 K (Tab. 2). The analysis has been performed for different T_9 temperature scenarios, and so a minimum temperature difference between T_9 and T_2 was imposed ($\Delta T_{9,2min}$) (K) (Tab. 2). The minimum temperature difference between the heating fluid and T_3, (ΔTpp) (Fig. 2-Tab. 2) was assumed to be 10K (Wang et al., 2012). The mathematical model used for simulating the ORC section is described by equations (6)-(10).The power absorbed by the pump (W_p)(kW) is described by equation (6) (Fig. 2) and the relation between the heat supplied to the working fluid (Q_{IN}) (kW) and the rate mass of the working fluid ($\dot{m}_{ORC}$) (kgs^{-1}) is represented by equation (7) (Fig. 2). The rate mass of working fluid depends on the heat transferred effectively into the cycle.

[Table1 here]

$$W_p = \dot{m}_{ORC}(h_2 - h_1) = \frac{\dot{m}_{ORC}(h_{2s} - h_1)}{\eta_p} \quad \text{(kW)} \quad (6)$$

$$Q_{IN} = \dot{m}_{ORC}(h_4 - h_2) \quad \text{(kW)} \quad (7)$$

$$W_{ex} = \dot{m}_{ORC}(h_4 - h_5) = \dot{m}_{ORC}(h_4 - h_{5s})\eta_{ex} \quad \text{(kW)} \quad (8)$$

$$Q_C = \dot{m}_{ORC}(h_5 - h_1) \quad \text{(kW)} \quad (9)$$

$$W_{net} = \eta_{eg}(W_{ex} - W_p) \quad \text{(kW)} \quad (10)$$

During the expansion step (Fig. 2) the power generated (W_{ex}) (kW) (Eq. 8) depends on the $\dot{m}_{ORC}$ (kgs^{-1}) and on condenser pressure and on evaporator pressure. The heat ejected at the condenser (Q_C) (kW) is described in equation (9). In the model only the global efficiency of pump (η_p) (%) and expander (η_{ex}) (%) (Tab. 2) has been considered and the heat losses and pressure drops were disregarded. The efficiency (η_{eg}) (%) assumed to evaluate the net electrical power generated (W_{net}) (kW) (Eq. 10) is 90% (Tab. 2).In order to evaluate the performances of the system for different T_9 scenarios the ORC net electrical efficiency (η_{net}) (%) has been analyzed by equation (11). The η_{net} (Eq.11) is the ratio between the net electrical energy produced by ORC and the amount of heat transferred to the cycle (Q_{IN}). Also the exergetic efficiency (η_{EXE})(%) of the ORC has been analyzed (Eq. 12). The η_{EXE} (Eq. 12) is the ratio between the power generated (W_{net}) (kW) and the exergy (EX_{IN})(kW) available from exhaust humid air from aerobic treatment section. The EX_{IN} is the sum of dry air's exergy (EX_{AIR})(kW) and vapor's exergy (EX_{VAP})(kW) (Eq. 13). The EX_{AIR}depends on the specific exergy (ex_8^{AIR},ex_{amb}^{AIR}) (kJkg^{-1}) at point 8 (Fig.

1-2) (Tab. 1) and ambient conditions (Tab. 2) and also on the $\dot{m}_{AIR}$ (Eq. 14). Similarly EX_{VAP} depends on specific exergy ($ex_8^{VAP}, ex_{amb}^{VAP}$) and also on vapor mass ($\dot{m}_8^{VAP}, \dot{m}_{amb}^{VAP}$) (Eq. 14). Also the pressure ratio (β) among the pump outlet (p2) and the condenser (p1) was evaluated (Eq. 15).

[Table2 here]

[Figure 2 here]

$$\eta_{net} = \frac{W_{net}}{Q_{IN}} \cdot 100 \qquad (\%) \qquad (11)$$

$$\eta_{EXE} = \frac{W_{net}}{EX_{IN}} \cdot 100 \qquad (\%) \qquad (12)$$

$$EX_{IN} = EX_{AIR} + EX_{VAP} \qquad (\text{kW}) \qquad (13)$$

$$EX_{IN} = [(ex_8^{AIR} - ex_{amb}^{AIR}) \cdot \dot{m}_{AIR}] + [(ex_8^{VAP} \cdot \dot{m}_8^{VAP}) - (ex_{amb}^{VAP} \cdot \dot{m}_{amb}^{VAP})] \quad (\text{kW}) \quad (14)$$

$$\beta = \frac{p_2}{p_1} \qquad (15)$$

Economic model

In order to evaluate the economic feasibility of the ORC cycle a preliminary analysis of the investment, operation and maintenance costs was evaluated. In order to obtain the total investment cost (€), a cost correlation is used for each component of the system and it is described in Table 3 (Quoilin et al., 2011) (Lecompte et al., 2013). The investment cost of the expander depends on the volumetric flow rate V_{exp} ($m^3 s^{-1}$) of the working fluid supplied to the expander, taking into account pressure and temperature conditions. The investment cost of the heat exchangers is related to the heat exchange area A (m^2) required. The investment cost of the working fluid pump and heat transfer fluid (HTF) pump depends respectively on the electrical power absorbed by the ORC pump W_p (W) and on the HTF pump W_{HTFp} (W). Also the cost of the liquid receiver and the piping's cost were evaluated. The capacity of the liquid receiver was evaluated considering a filling factor of about 33%. The pipe diameter d_{pipe} (mm) was evaluated imposing the fluid speed of 6ms^{-1}for the pump and the condenser, instead for the evaporator and the expander the fluid speed imposed is respectively of $10ms^{-1}$ and $12ms^{-1}$ (Quoilin et al., 2011). The labour cost was imposed to be the 30% of the total investment cost.

[Table3 here]

In order to evaluate also the total cost of the ORC cycle (€$year^{-1}$), an operation and maintenance cost of 15% of the total investment cost was considered. The total cost of the cycle (€$year^{-1}$) was referred to the electrical kWh produced considering a plant's lifetime of 10 years (Di Maria et al.,2012c) and a working time of 7,500 $hyear^{-1}$(Schuster et al.,2009).

Results and discussion

Energetic analysis

The maximum temperatures detected during experimental measures on the basin, turn out to be quite constant (Fig. 3), ranging among 343-349 K, with a mean temperature of about 346 K. The exhaust air composition detected by experimental analysis is represented in Figure 3. The exhaust air composition presents a content in O_2 ranging between about 15-21%vol and a CO_2 concentration ranging between about 5-0.15%vol. Also a not significant presence of CH_4 (less than 0,20%vol) was detected. The mean exhaust air composition assumed in the calculations is reported in Table 1.

[Figure 3 here]

Under the assumptions made (Table 1-2), the performances of the system at different T_9 scenarios have been investigated. The T_9 temperature has been changed from a maximum of 340 K to a minimum of 316K in variable intervals (Fig.4a-4b).

[Figure 4 here]

The electrical power generated in the minimum T_9 temperature scenario of 316 K results in about 17.7 kW (Fig. 4a); in the following scenario, in which a T_9 of 321 K has been considered, the electrical power generated achieves the maximum value of about 19.4 kW. In the other scenarios, in which temperature T_9 increases, the electrical energy production decreases rapidly until the minimum value of about 2.4 kW in correspondence with the maximum T_9 temperature of 340 K (Fig. 4a). Simultaneously the net efficiency shows an increasing trend (Fig. 4a) in correspondence with the increase of T_9. The η_{net} ranges among 2.7% for T_9 of 316 K and 3.3% for T_9 of 321 K (Fig. 4a); in the following scenarios, with T_9 of 326 K and 331K, the η_{net} is respectively of about 3.9% and 4.4%.The maximum value of η_{net}, of about 5.1% is reached in the scenario with T_9 of 340 K. Yamamoto et al.(2001) analyzed with experimental tests the performances of a radial expander and the global ORC efficiency. The study reported a cycle efficiency for the HCFC-123 as working fluid ranging between about 2% and 11% with a corresponding pressure ratio respectively of 1.5 and 5.Wang et al.(2011) analyzed the performances of different working fluids, reporting for R-123 a thermal efficiency ranging between about 9% and 10% with an evaporator and condenser temperature of 406 K and 320 K and about 10 kW of net power output. Hung et al.(2010) reported similar values for the ORC system applied to ocean thermal energy conversion and R-123 as working fluid operating at temperature of 278 K at condenser and 313K at evaporator. The β achieved in the present study turns out to be quite limited, due to the low temperature of the exhaust air from aerobic treatment section and to the $T_3+\Delta T_{pp}<T_8$ condition to respect. The β (Fig. 4-b) shows a quite linear increasing trend, in fact, T_8 being fixed (Tab. 1), the higher is T_9, the higher is β and the lower is the heat transferred to the cycle. The maximum value of β is of about 3.5 and corresponds to the T_9 of 340 K, the minimum value is of about 1.9 for T_9 of 316K.The higher electrical power production corresponds to a β of about 2.2 and a η_{net} of 3.4% (Fig. 4a-4b).

[Figure 5 here]

The exergetic efficiency depends on the electrical power produced (W_{net}) and on the exergy available from the exhaust air coming from the aerobic treatment section (EX_{IN}). The EX_{IN} refers to the exhaust air inlet conditions (T_8, P_8) (Tab. 1) and to the ambient conditions (T_{amb}, P_{amb})(Tab. 2) and remains constant in each scenario. Consequently η_{EXE} depends only on the electrical energy produced (Fig. 4-b) and achieves the maximum value of about 11.2% for T_9 of 321 K. The minimum value of η_{EXE} is of about 1.4% corresponding to the T_9 of 340 K and β of 3.5 (Fig. 4-b).

To explain the electrical energy production trend that shows a maximum for T_9 of 321 K, it is useful to consider the ratio between the ORC mass flow in the different scenarios ($\dot{m}_{ORC,T9}$) and the ORC mass flow at the scenario with T_9 of 340 K ($\dot{m}_{ORC,340K}$) in relation to the ratio between the h_4-h_5 enthalpy difference in the same scenarios (Fig. 5). As it is visible in Figure 5, the variation of the ratio in ORC flow mass, referred to the condition of T_9 of 340 K, shows a significant increase in ORC flow mass, becoming about 14 times higher for T_9 of 316K in comparison to ORC mass flow for T_9 of 340K scenario. The variation of enthalpy ratio (Fig. 5) instead is more limited, achieving a reduction of about 50% from the scenario with T_9 of 340 K to the scenario with T_9 of 316 K. In particular, from the scenario with T_9 of 316 K to the scenario with T_9 of 321 K, the ORC mass variation turns out to be lower than the variation in the other T_9 scenarios; in fact the the slope of the curve becomes more and more pronounced if T_9 increases. Simultaneously, instead, the increase in the enthalpy difference is quite linear and less pronounced. So the result of the two phenomena causes a maximum of power generated at expander in the scenario with a T_9 of 321K. Consequently also the electrical energy production is higher in the T_9 of 321K than in other T_9 scenarios (Fig. 5).

Economic analysis

A preliminary economic analysis was performed in order to evaluate the feasibility of the ORC system (Tab. 4).The analysis of the investment cost was performed considering a cost correlation for each component of the system (Tab. 3) related to the maximum power output and the best exergetic efficiency condition. The reference scenario for the economic analysis is T_9 of 321K and net power output of 19.4 kW. The main investment cost sources are represented by the expander and the exchangers with respectively an investment cost of about 16,200 € and 21,600 € (Tab. 4). The cost of the expander depends strictly on mass ORC mass flow. The use of organic working fluid with a low boiling point, in the low thermal heat recovery, involves inlet and outlet volume ratio that can be smaller if compared to water. This fact allows to use smaller and less expensive expanders. Lecompte et al.(2013) evaluated a thermo-economic analysis on ORC cycle for different working fluid. The investment cost for turbine ranges between 22% and 34% of the total investment cost. The investment cost for the exchangers instead ranges between 30% and 36% of the total investment cost. The total investment cost, reported for different working fluids at optimal conditions, ranges between 2,210 €kW^{-1} and 3,413 €kW^{-1}. Papadopuoulos et al.(2010) evaluated the exchangers cost for different working fluids. The investment cost was evaluated in dependence on the heat exchange area ranging about between 20,500 € and 26,500 € in correspondence respectively to a total exchange area of about 68m^2 and 95m^2, till a maximum of about 30,000 € for an heat exchange surface of about 156m^2. In these studies the external fluid used in the evaporator is normally water. In the present study the external fluid is the exhaust air coming from aerobic treatment section that represents a particular condition for the presence of air and vapor. The condensation of vapor during the heat exchange was taken into account to evaluate the heat transfer coefficient in order to estimate the heat exchange area. Quolin et al.(2011) evaluated for R-123 an investment cost of 2,916 €kW^{-1} and for other working fluids an investment cost ranging among about 2,100 €kW^{-1} and 4,260 €kW^{-1} for net power output and efficiency ranging respectively among 2.5 kW and 4.8kW and 3.6% and 7.9%. Schuster et al.(2009) assumed a specific investment cost of 3,755 €kW^{-1} for a power output of 35kW. The total investment cost of the proposed ORC cycle application (Tab. 4) turns out to be of 2,873 €kW^{-1} and the operation and maintenance annual costs are 8,360 €$year^{-1}$ (Tab. 4).

Considering the operational lifetime of the plant and the overall annual electrical energy production, the total cost of the ORC amounts to 0.096 €kWh^{-1} (Tab. 4). As a consequence, the exploitation of ORC to recover low thermal energy from exhaust air from an aerobic treatment section of a MBT facility could be a suitable way to enhance renewable and sustainable energy production.

[Table 4 here]

Conclusion

The micro Organic Rankine Cycle (ORC) is a suitable way to exploit the low grade heat available from the exhaust air coming from an existing aerobic biological treatment section of a mechanical biological treatment facility. Since the exhaust air contains a relevant amount of vapor, also if the exhaust air temperature is quite low, the amount of heat available for the exploitation inside the ORC can be consider to be satisfactory. The energetic analysis shows an increase of the power output and exergetic efficiency for lower pressure ratio corresponding to the lower outlet evaporator temperatures. Even if the power output is quite limited, the analyzed system seems to be an interesting solution for the production of renewable energy and is compatible with other micro-ORC similar applications. The economic analysis, related to the best energetic conditions, shows that the upgrading of the aerobic treatment section with an ORC section could be also an economically sustainable solution in line with other ORC similar applications.

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Legend:

Figure 1: Scheme of biological treatment section with Organic Rankine Cycle for heat recovery.
Figure 2: Example of a T-s diagram for the organic Rankine cycle (ORC) and of the heat exchange process.
Figure 3:Some temperature and exhaust air quality sampled at different point of the BT basin.
Figure 4: a-Electrical power generated and net efficiency at different T_9scenario; b-Exergetic efficiency and pressure ratio at different T_9scenario.
Figure5: ORC mass flow and h_4-h_5 variation vs T_9 temperature.

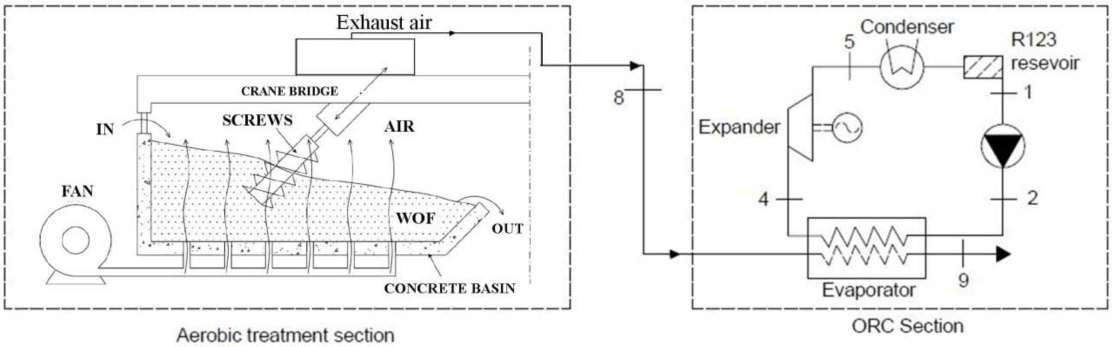

Figure 2: Scheme of biological treatment section with Organic Rankine Cycle for heat recovery.

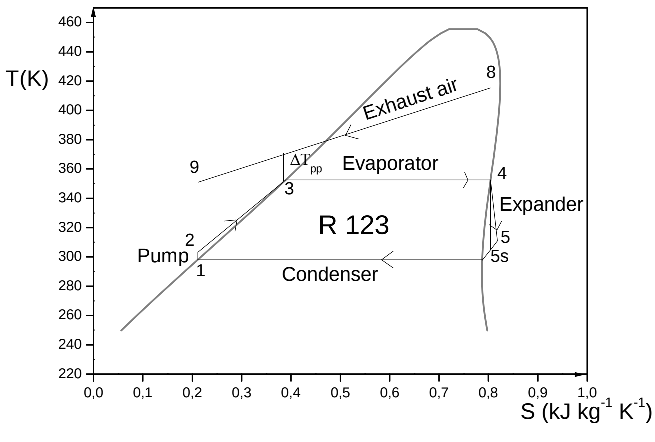

Figure 2: Example of a T-s diagram for the organic Rankine cycle (ORC) and of the heat exchange process.

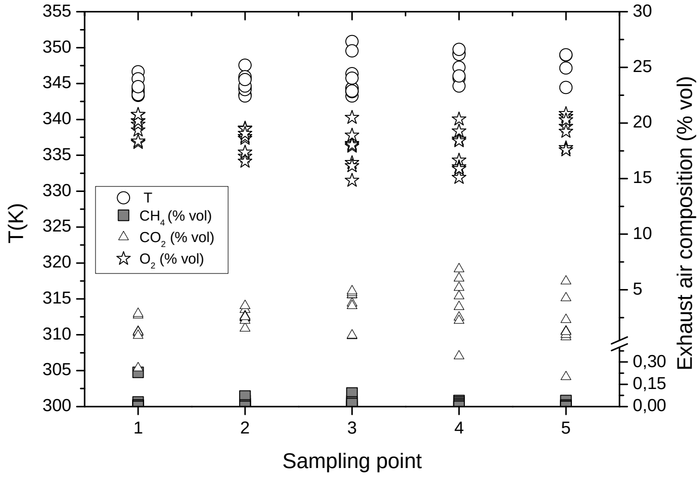

Figure 3:Some temperature and exhaust air quality sampled at different point of the BT basin.

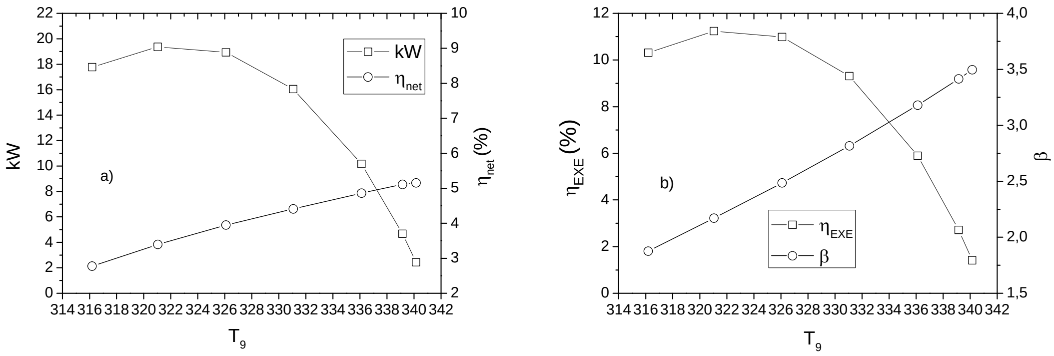

Figure 4: a-Electrical power generated and net efficiency at different T_9scenario; b-Exergetic efficiency and pressure ratio at different T_9scenario.

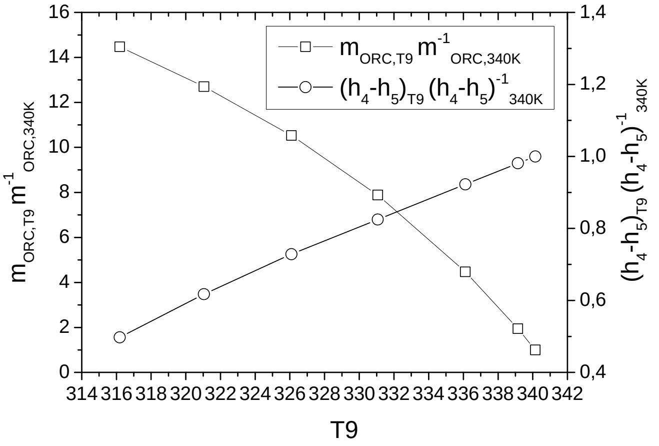

Figure5: ORC mass flow and h_4-h_5 variation vs T_9 temperature.

Legend:

Table1: Main parameters assumed for exhaust air from BT section .
Table2:ORC cycle main feature.
Table3: Component cost.
Table4: Main economic analysis results.

Parameter	Value	Unit
Air flow rate	4,000	Nm^3h^{-1}
p_8	101,325	Pa
T_8	341	K
Φ	100	%
ΔT_{WOF}	5	K
Exhaust air mean composition		
CH_4	0.02	%vol
CO_2	2.70	%vol
O_2	18.5	%vol
N_2	78.7	%vol

Table1: Main parameters assumed for exhaust air from BT section .

ORC features		
Parameter	**Value**	**Unit**
η_p	80	%
η_{ex}	55	%
η_{eg}	90	%
ΔT_{pp}	10	K
T_c	293	K
T_{amb}	288	K
p_{amb}	101,325	Pa
$\Delta T_{9,2min}$	10	K
Working fluid R123		
Molecular mass	152.93	g mol^{-1}
Boiling point	300.97	K
Critical Pressure	3.662	MPa

Table2:ORC cycle main feature.

Component	Dependent variable	Cost correlation
Expander (€kW^{-1})	Volume flow rate V_{exp} $(m^3 s^{-1})$	$1{,}5 \cdot (225 + 170 \cdot V_{exp})$
Heat exchangers (€)	Heat exchange area A (m^2)	190+(310· A)
Working fluid pump(€)	Electrical power W_p (W)	900· $(W_p \cdot 300^{-1})^{0.25}$
HTF pump(€)	Electrical power W_{HTFp} (W)	500· $(W_{HTFp} \cdot 300^{-1})^{0.25}$
Liquid receiver(€)	Volume V (l)	31,5+16 V
Piping(€)	Pipe diameterd_{pipe} (mm) and lenghtL_{pipe} (m)	$(0{,}897 + 0{,}21\ d_{pipe})L_{pipe}$
Working fluid(€)	Working fluid mass M_{ORC} (kg)	$20 \cdot M_{ORC}$
Hardware and control system(€)	-	800
Labour(€)	Total investment cost	30%
O&M(€year^{-1})	Total investment cost	15%

Table3: Component cost.

Component	Cost
Expander (€)	16,206
Heatexchangers (€)	21,671
Working fluid pump and HTF pump (€)	2,498
Liquid receiver and piping (€)	1,271
Workingfluid (€)	427
Control system and hardware (€)	800
Labor (€)	12,862
Investementcost (€kW^{-1})	2,873
O&M (€$year^{-1}$)	8,360
Total cost (€kWh^{-1})	0.096

Table 4: Main economic analysis results.