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Exergetic and economic analysis of energy recovery from the exhaust air of organic waste aerobic bioconversion by organic Rankine cycle

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Abstract

The amount of heat rejected by the exhaust air generated by the aerobic treatment of organic waste (OW) was investigated with the aim of evaluating the amount of electrical energy recoverable by a micro organic Rankine cycle (micro-ORC). Both an energetic and exergetic analysis were performed along with an evaluation of the investment costs. The investigation of the heat content and composition of the exhaust air was experimentally performed on a full scale facility processing 32,000 tonnes/year of OW. Results shows that the average exhaust air rate is of about 4,000 Nm^3/h with a temperature of 341 K and a relative humidity of 100%. By cooling thi gaseous stream up to 316 K the net power output of the micro-ORC ranges from about 2 kW to about 20 kW. Contemporary the net electrical efficiency decreases from 5% to about 2% whereas the exergetic efficiency ranges in parallel with the net power output from 11% to 1%. Specific investment ranges from about 2,800 €/kW to about 3,900 €/kW and the cost of the electrical energy results of about 0.1 €/kWh to about 0,13 €/kWh.

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Keywords: Aerobic treatment, Economic analysis, Electrical energy, Exergetic analysis, Organic Rankine Cycle, Organic Waste;

1. Introduction

Anaerobic and aerobic biological treatments are widely exploited in processing OW both for energy production and for biological reactivity reduction before final recovery and/or disposal [1,2,3,4,5,6,7]. In particular AD can lead to the production from about 80 Nm^3 up to 210 Nm^3 of biogas per tonne of processed OW. The methane concentration usually ranges from 50 to 70% v/v [8,9,10,11,12,13,14], whereas the other main component is CO_2. The corresponding lower heating value (LHV) varies from 18,000 kJ/Nm^3 to 24,000 kJ/Nm^3 and biogas can be exploited as fuel in internal combustion engines for renewable energy production.

The viability of AD is greatly influenced by plant size and by the variation in the rate and composition of OW during the year [11,12,15]. Aerobic treatments are used to reduce both OW and AD digestate residual biological reactivity before disposal or for the production of organic fertilizer, depending on OW quality [16]. As extensively demonstrated [17,18,19], aerobic treatment can lead to long-term emission reduction in landfills, up to 90%. If OW quality is compatible with the characteristics of organic fertilizer [20,21,22], aerobic bioconversion is generally used to convert the OW to substances exploitable for agricultural use. During the aerobic process, bacteria oxidize the organic matter [23], generating about 17,000-18,000 kJ/kg OM [24] of heat. Due to the initial concentration of OM, heat release is particularly high in the first 2-4 weeks, causing an increase in the OW mass and

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consequently in process air temperatures. Maximum temperatures achieved in full-scale facilities range from 55°C to 75°C, depending mainly on thermal loss, OW moisture content OM content and process air rate [25,26]. In a previous study, Di Maria et al. [26] evaluated the possibility of recovering this heat for civil use by heat pumps. Results showed that the process exhaust air temperature ranged from about 55°C to 70°C and the amount of heat ejected daily ranged from about 120 to about 350kWh/tonne depending mainly on the amount of OW treated and the process air rate.

Another solution proposed by Di Maria et al. [27,28], was to exploit the sensible heat (*i.e.* without humidity condensation) of the exhaust air to generate electrical energy by micro-ORC. Results shows that, for an aerobic facility processing about 20,000 tonnes/year of OW, the power output ranges from 400 W to 700W. The ORC uses the same components as a conventional steam power plant, but uses an organic fluid to extract low-grade thermal energy to generate electricity. ORC is commonly used in practical industrial applications such as biomass power [29], [30] solar power [31] also aimed at water production [32], ocean thermal energy conversion, geothermal power [33], [34], and waste heat recovery power [35]. Bidini et al. [36] analyzed the exploitation of ORC in an integrated gas turbine-geothermal power plant for recovering low-grade heat ejected from gas turbine exhaust after geothermal fluid heating.Gewald et al [37] showed that ORC can improve the efficiency of landfill gas-fired power plants by about 12%. Desideri and Di Maria [38] reported that the exploitation of ORC for recovering exhaust heat from a humid air turbine system can lead to an overall cycle efficiency increase from 1.6 to 2.2%. Wang et al. [39] analyzed the effect of different working fluids on ORC efficiency for engine waste heat recovery. Similarly Hung et al. [40] investigated the effect of different organic working fluids on ORC efficiency using heat generated by solar pond and ocean thermal energy. ORC is a promising solution for decentralized, small- (*i.e.*<100kW) and micro- (*i.e.*<15kW) scale combined heat and power generation [41], [42], [43] and for this reason it is particularly used in biomass-fired plants. Even if its efficiency is low, between 6% and 17%, ORC has low maintenance and personnel costs [43]. Dong et al. [44] reported that costs are comparable with gasification in the same small- and micro-scale range. Furthermore, among the small number of commercially viable biomass gasification systems, only a few have been shown to be economical [41]. On the contrary, several small-scale ORC systems are operating and their viability has been fully proven [41], [42], [43]. Anyway there is a lack of investigation about the possibility of exploiting ORC for electrical energy production from the heat produced during the bioconversion of OW. On the basis of previous study of Di Maria et al. [27], [28], the performances of an ORC for energy recovery from an existing 32,000 tonnes/year aerobic facility of OW were investigated including the contribution of heat released by the humidity condensation of exhaust air. An exergetic and economic analysis of the proposed system were also performed.

Nomenclature

AD	Anaerobic Digestion	T	Temperature
AT	Aerobic Treatment	V	Volumetric flow rate
β	Compression ratio	W	Power
c_p	Specific heat at constant pressure	*Subscripts*	
Δh	enthalpy difference	AIR	Air
ΔT	Temperature difference	amb	ambient
ex	specific exergy	AT	Aerobic Treatment
EX	Exergy	eg	electrical generator
Φ	Relative humidity	ex	expander
h	specific enthalpy	EXE	exergetic
η	Efficiency	IN	Inlet
HFT	Heat Fluid Transfer	min	minimum
LHV	Low Heating Value	net	net
$\dot{m}$	mass rate	OW	Organic Waste
M	Mass	ORC	Organic Rankyne Cycle
M	Organic Matter	p	pump
ORC	Organic Rankyne Cycle	pp	pinch point
OW	Organic Waste	s	isoentropic
p	Pressure	VAP	vapour
Q	Thermal power		

2. Material and Methods

2.1 Existing aerobic facility and experimental measures

The evaluation of the features of the exhaust air generated by the aerobic process performed on the OW in an existing full-scale aerobic facility were performed by on site experimental measures. The aerobic facility consist of a large, continuous flow, concrete basin with an aerated floor on which moves a crane bridge with screws [25]. The screws stirs and moves from the inlet to the outlet section the OW (Fig. 1a) for ensuring a mean residence time at least of about 15 days, whereas the process air was supplied by electrical fans.The OW arises from a previous mechanical and dimensional screening of the MSW performed in another facility were metals, plastics, paper and other not biodegradable and bulky materials were largely removed.

The aim of the considered aerobic treatment is to reduce the OWmass and biological reactivity before landfilling. The amount of OW processed in this facility is of about 32,000 tonnes/year whereas the electrical fans supplies about 4,000 Nm^3/h of process air through the aerated floor. During the process the humidity content is controlled and regulated to maintain theoptimal process conditions.

The exhaust air features where measured in double line of five different points, A and B, along the basin width (Fig. 1b) in different periods of the year. Temperature was detected by a portable K-type thermocouple whereas exhaust air composition (% vol.) in terms of CH_4, CO_2, H_2S and O_2 were evaluated by a portable gas analyzer. CH_4 and CO_2 concentration were detected by infrared sensors (±1%). H_2S and O_2 concentration were detected by electrochemical sensors (±2%). Considering the path during the process the exhaust air relative humidity was assumed to be 100%.

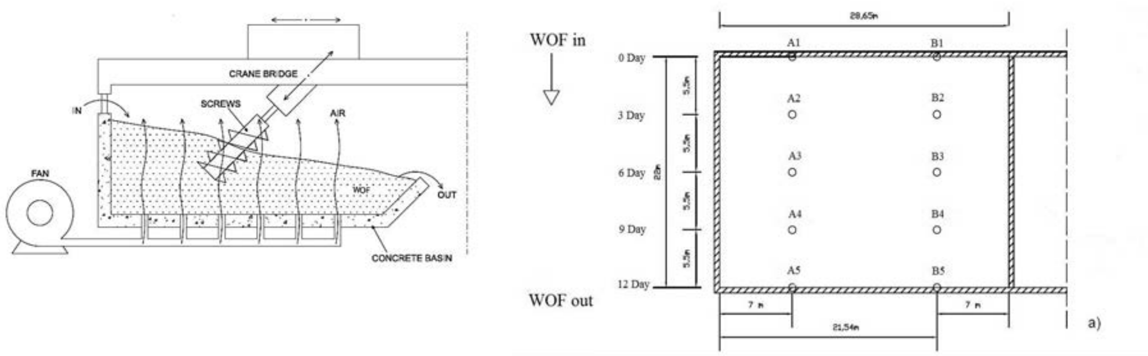

Fig. 1. Aerobic basin scheme representing the section (a) and plant with sampling points (b).

2.2 Proposed system and ORC model

ORC is fuelled by the heated air stream generated from the aerobic treatment of the OW (Fig. 2). In the evaporator the heat released by the humid air is transferred to the ORC working fluid R123 (Tab. 1) (Fig. 3). Once evaporated the working fluids expands in the expander and successively is condensed in the condenser. By the condenser extraction pump the R123 pressure is increased until the maximum value before entering again the evaporator. In the following the heat exchange and ORC model are described and discussed.

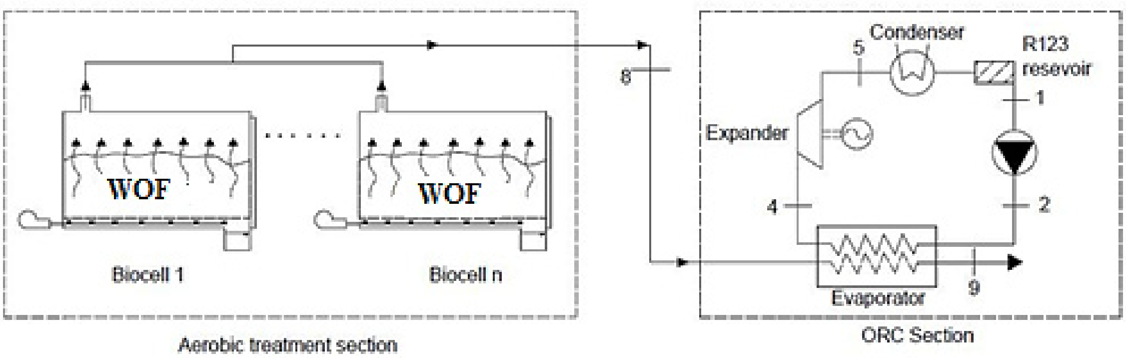

Fig. 2. Scheme of the system with the ORC fuelled by exhaust air from aerobic treatment.

During the aerobic biological process the air pass through the OW heap causing complex mass and heat exchange phenomena. From the thermal point of view, these phenomena can be assimilated to the ones occurring in direct contact heat exchanger. In fact, due to the heat generated by the biological activity, the OW temperature rises causing a heat flow toward the process air. As the air temperature increase the relative humidity results reduced. Consequently, due to the high water content of the OW, usually around 50% on wet basis, a given amount of water evaporates and saturates the process air. The amount of water that passes from the OW to the air depends on the air temperature and pressure.

To evaluate the exhaust air temperature (T_8) (K) in the outlet section of theaerobic treatment section section (Fig. 2-3), a difference with the OW mass mean temperature of 5K was assumed (ΔT_{OF}) according with [26].The exhaust air vapor mass rate ($\dot{m}_{VAP}$)(kg/s) was evaluated, according to equations (1) and (2), for different T_9(K).

Specific humidity x (kg/kg) of exhaust air was evaluated according to Eq. (1) on the basis of the T_9 and p_8 values and consequently the vapor mass rate (kg/s) was evaluated according to Eq. (2). The thermal power exchanged in the evaporator Q_{AT} (kW) was evaluated according to Eqs. (3), (4) and (5) were Δm_{VAP} (kg/s) and Δh_{VAP} (kJ/kg) represents respectively the amount of vapor condensed and the correspondent specific enthalpy difference from the temperature of point 8 and point 9 (Fig. 3).

$$x = \frac{M_{VAP}}{M_{AIR}}(\text{kg/kg}) \quad (1)$$

$$\dot{m}_{VAP} = x \cdot \dot{m}_{AIR}(\text{kg/s}) \quad (2)$$

$$Q_{AT} = Q_{AIR} + Q_{VAP}(\text{kW}) \quad (3)$$

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

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

The thermodynamic, economic and environmental properties of working fluids normally used in organic Rankinecycle (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.1).

Figure 2 represents a possible T-s diagram for the ORC. The condenser's temperature (T_c)(K) and the ambient temperature (T_{amb})were assumed constant (Tab.1).Refferring to Figure 3, two minimum temperature differences values were assumed respectively in correspondence of the point 3, ΔT_{pp} (K), and of point 2, $\Delta T_{\min,9,2}$ (K). For both these temperature differences was assumed a value of 10K according to [49].

The ORC performances were evaluated accordin to the Eq. from (6) to (11) assuming the parameters reported in Table 1. The exergetic efficiency was defined as the ratio between the ORC net power output (kW) and the system inlet exergy evaluated at the point 8 of the system (kW) (Figs. 2-3) Eq. (12). EX_{IN} results the sum of the exergy of the dry exhaust air ad of the vapor exiting the aerobic treatment section Eq. (13) and was evaluated according to Eq. (14) considering as reference temperature T_{amb}.

Table 1. ORC cycle main feature.

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
ΔT_{OF}	5	K
$\Delta T_{9,2min}$	10	K
Working fluid R123		
Molecular mass	152.93	g /mol
Boiling point	300.97	K
Critical Pressure	3.662	MPa

The power generated by the expander (W_{ex}) (kW) Eq.(8) depends on the $\dot{m}_{ORC}$(kg/s) and on pressure difference between point 4 and 5. In the model only the global efficiency of pump (η_p) (%) and expander (η_{ex}) (%) (Tab.1) were considered whereas the heat losses and pressure drops were disregarded. The electrical generator efficiency (η_{eg}) (%) was assumed to of 90% (Tab.1).The pressure ratio of the pump from point 1 to point 2 of the cycle (Fig. 3) is expressed by Eq. (15).

2.3 Economic Model

In order to evaluate the economic feasibility of the ORC cycle a preliminary analysis of the investment, operation and maintenance costs was performed. The total investment cost was obtained as the sum of the costs of the single components

In order to obtain the total investment cost (€), a cost correlation is used for each component of the system according to

Quoilin [53] and Lecompte [54] (Table 2). The investment cost of the expander depends on the volumetric flow rateV_{ex} (m^3/s) of the working fluid at the inlet;the investment cost of the heat exchangers is related to the heat exchange surfaceA (m^2); $t$he investment cost of the working fluid pump and of the heat transfer fluid (HTF) pump, that supplies the cooling fluid to the condenser, depends respectively on the electrical power absorbed by the ORC pump W_p (W) and on the HTF pumpW_{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 6m/sfor the pump and the condenser, instead for the evaporator and the expander the fluid speed imposed is respectively of 10m/sand 12m/s[53]. The labour cost was assumed to be the 30% of the total investment cost.
In order to evaluate also the cost of the electrical energy generated by thesystem (€/kWh), an O&M cost of 15% of the total investment cost was considered whereas the investment period and the operation h per year were assumed respectively of 10 years and 7,500 h/year [43].

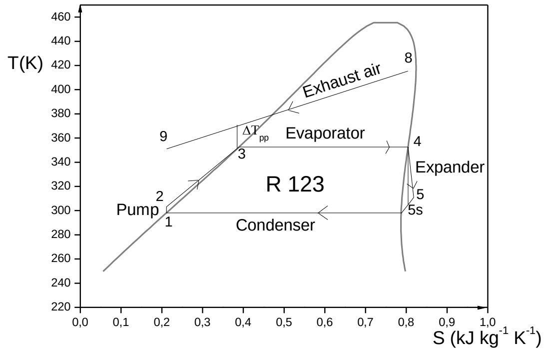

Fig. 3. Example of a T-s diagram for the organic Rankine cycle (ORC) and of the heat exchange process.

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

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

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

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

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

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

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

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

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

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

3. Results and discussion

3.1 Energetic analysis

Table 3 reports the average values measured for the exhaust air temperature (T_8) and composition on dry basis expressed as % by volume. The average inlet air flow rate corresponds to 4,000 Nm^3/h whereas exhaust air pressure and relative humidity were assumed to be respectively 101,325 Pa and 100%.

Experimental values of these parameters per each sampling point (Fig. 1b) are reported in Figure 4.Maximum temperature values are achieved in correspondence of the sampling point n°3. Lower values are achieved both in sampling point n°1 and n°5 corresponding to basin inlet and outlet section (Fig. 1a). On contrary the exhaust air composition shows a maximum concentration of O_2 in sampling points n°1 and 5 and minimum in point n°3. This trend results in accordance with the evolution of the biological process activity. In fact in point 1 the material has a limited residence time (about 1-2 days) and the aerobic micro-organism has not achieved its maximum activity. In point n°3 the residence time is of about 7-8 days and the aerobic micro-organisms shows their maximum activity generating the higher amount of heat and consuming the larger amount of free oxygen of the process air. Consequently the CO_2 concentration achieves its maximum value. Methane concentration is always lower than 0.03% by vol. indicating that there are no relevant anaerobic zones in the processed OW and that the biological process can be considered fully aerobic. As the OW is moved toward the outlet section (*i.e.* point 4 and 5) the amount of OM remaining the OW results reduced and consequently the biological activity of the micro-organisms decreases leading to a reduction of the T and of the CO_2concentration and to an increase of the oxygen one.

Table 2. Economic analysis correlations.

Component	Dependent variable	Cost correlation	u.m.
Expander	Volume flow rate V_{ex} $(m^3/s\ )$	$1{,}5\cdot(225+170\cdot V_{ex})$	€/kW
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	Total investment cost	15%	€/year

Table 3. Average values of the exhaust air.

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

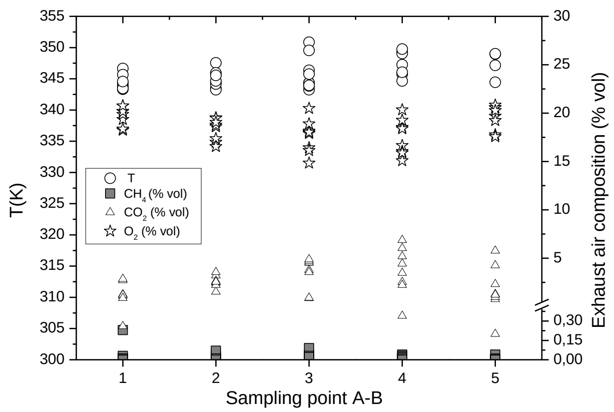

Fig. 4. Temperature and exhaust air compositionexpressed as average values between the A and B series.

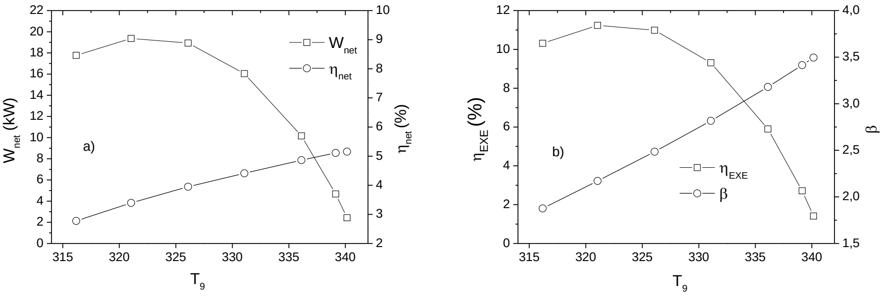

Fig. 5. Electrical power generated and net efficiency (a) andexergetic efficiency and pressure ratio (b) for different T_9 values.

By varying the T_9 from 316K up to 340K the W_{net} achieve a maximum value of about 20 kW in correspondence of 321K (Fig. 5a). For higher T_9 values the W_{net} decreases constantly. On contrary the η_{net} rises constantly ranging from about 3% to about 5%. The constant increase of the efficiency is a direct consequence of the increase of the β (Fig. 5b) as T_9 rises. Due to the constance of the EX_{IN} the η_{ex} show a trend similar to the one of the W_{net}.

Yamamoto et al. [31] analyzed with experimental teststhe 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. [39] 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 406K and 320K and about 10 kW of net power output. Hung et al. [40] reported similar values for the ORC system applied to ocean thermal energy conversion and R-123 as working fluid operating at temperature of 278K 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 $T_3+\Delta Tpp<T_8$ condition to respect. This leads to a maximum value of β of about 3.5 in correspondence of T_9=340K, and a minimum value of about 1.9 for T_9=316K.. The maximum W_{net} corresponds to a β of about 2.2, a $\eta_{\mathbf{net}}$ of 3.4% and a η_{exe} of about 11% (Fig.5a,b).

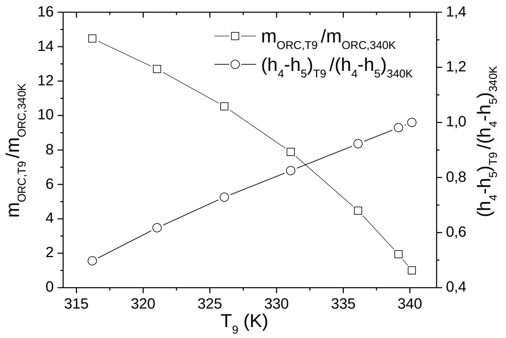

Fig. 6. Ratio of ORC mass flow and h_4-h_5 vs T_9 temperature related to the values assumed for 340K.

To explain the maximum achieved by W_{net}at T_9=321K it is useful to consider the ratio between the ORC mass flow in the different scenarios ($\dot{m}_{ORC,T9}$) andthe ORC mass flow atthe scenario with T_9=340K ($\dot{m}_{ORC,340K}$) in relation tothe ratio between the h_4-h_5 enthalpy difference in the same scenarios (Fig.6). As shown, $\dot{m}_{ORC}$ rises significantly as T_9 decreases becoming up to 14 times higher than $\dot{m}_{ORC,340K}$ when T_9=316K. The relation between $\dot{m}_{ORC}$ ratio and T_9 results strongly not linear with an increased reduction of the $\dot{m}_{ORC}$ ratio as T_9 rises. In parallel enthalpy difference ratio reduction decreases with a quite linear trend becoming about 0.5 when T_9=316K. The combination of the two effects described leads to a maximum value of the product of$\dot{m}_{ORC}$ and (h_4-h_5) (*i.e.* expander power) for T_9 values close to the minimum ones (Fig. 5a).

3.2 Economic analysis

An economic analysis was performed in order to evaluate the feasibility of the ORC system (Tab. 4).The reference scenario for the economic analysis is T_9 of 321K at which corresponds the maximum net power output of 19.4 kW and the maxiumiexergetic efficiency of about 11%. The main investment cost sources are represented by the expander and the heat exchangers with respectively an investment cost of about 16,200 € and 21,600€ (Tab. 4) representing respectively the 29% and the 39% of the total investment cost. The cost of the expander depends strictly on the ORC volumetric flow rate. The use of organic working fluid with a low boiling point,in the low termperature 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. [54] 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 and 3,413€/kW. Papadopuouloset al. [55]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 hot water. The heating fluidused in this study 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. [53] evaluated for R-123 an investment cost of 2,916 €/kW and for other working fluids an investment cost ranging among about 2,100€/kW and 4,260€/kW for net power output and efficiency ranging respectively among 2.5 kW and 4.8kW and 3.6% and 7.9%. Schuster et al. [43] assumed a specific investment cost of 3,755 €/kW for a power output of 35kW. The specific investment cost of the proposed ORC cycle application (Tab. 4) turns out to be of 2,873 €/kW whereas the operation and maintenance annual costs results of 8,360 €/year. Considering the operational lifetime of the plant and the overall annual electrical energy production, the cost of the electrical energy generated by the proposed ORC system amounts to 0.096€/kWh. As highlighted in the previous section (3.1) a compression ratio of 2.2 results quite low if compared to the one proposed by other authors. For this reason another economic evaluation of the proposed system was performed assuming a β=3. In these conditions (Fig. 5) the net power output of the ORC is of about 10 kW. Results shows that the specific investment cost is of about 3,900 €/kW (Tab. 5) and the single unit of electrical energy costs about 0.13 €/kWh. The high reduction in W_{net} (>50%) is not adequately compensated by the reduction in investment (<30%) and in O&M (<30%) leading to an increase of both specific investment and electrical energy costs.

Finally the cost of the kWh generated by the proposed system was also compared by the cost of the energy €/kWh related to other renewable source (Tab. 6). Results shows that the proposed system has comparable values with the ones of biogas and wind energy whereas results higher than the one of landfill gas and lower than the one of photovoltaic.

Table 4. Main economic analysis results for T_9=321K and β=2.2.

Component	Cost (€)	Incidence (%)
Expander	16,206	29.1
Heat exchangers	21,671	38.9
Working fluid and HTF pump	2,498	4.48
Liquid receiver and piping	1,271	2.28
Working fluid	427	<1
Control system and hardware	800	1.44
Labour	12,862	23.1
Total Investment	55,735	100
Specific Investement cost (€/kW)	2,873	-
O&M (€/year)	8,360	-
Electricity cost (€/kWh)	0.096	-

Table 5. Main economic analysis results for T_9=335K and β=3.

Component	Cost (€)	Incidence (%)
Expander	7,338	18.17
Heat exchangers	19,339	47.88
Working fluid and HTF pump	2,547	6.31
Liquid receiver and piping	723	1.79
Working fluid	319	0.79
Control system and hardware	800	1.98
Labour	9,320	23.08
Total Investment	40,386	100
Specific Investment cost (€/kW)	3,975	-
O&M (€/year)	6,057	-
Electricity cost (€/kWh)	0.132	-

Table 6. Comparison with the cost of the single kWh generated by different renewable sources.

Energy source	kW	€/kWh	Reference
Heat from aerobic treatment	10-20	0.096-0.132	This study
Biogas from organic waste	500-1,900	0.11-0.28	Di Maria et al.,2012
Landfill gas	500	0.051-0.057	www.autorita.energia.it
Wind	30-150	0.117-0.286	www.autorita.energia.it
PV	3-20	0.318-0.567	www.autorita.energia.it

4. Conclusions

The aerobic treatment of the organic waste generates a relevant amount of heat that is released as a mixture of heated exhaust air and water vapor. Micro organic Rankine cycle (ORC) shows suitable features for allowing energy recovery from this low grade heat with a quite acceptable efficiency and economic investment. The cost of the single kWh of electrical energy generated by this system results in line with the one of other renewable energy systems. The compression ratio at which the system achieve the best performance in terms of net power output, thermal and exergetic efficiency appears quite low if compared to the one proposed by other authors. Assuming compression ratio in line with the typical one of micro-ORC the power output results in any case sufficiently high even if the thermodynamic and exergetic efficiencies were practically halved. The cost of the kWh generated in these conditions remains in line with the one of similar ORC systems.

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