Potential of ORC Systems to Retrofit CHP Plants in Wastewater Treatment Stations

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ABSTRACT

Wastewater treatment stations take advantage of the biogas produced from sludge in anaerobic digesters to generate electricity (reciprocating gas engines) and heat (cooling water and engine exhaust gases). A fraction of this electricity is used to operate the plant while the remaining is sold to the grid. Heat is almost entirely used to support the endothermic anaerobic digestion and a minimum fraction of it is rejected to the environment at a set of fan coolers. This generic description is applicable to on-design conditions. Nevertheless, the operating conditions of the plant present a large seasonal variation so it is commonly found that the fraction of heat rejected to the atmosphere increases significantly at certain times of the year. Moreover, the heat available in the exhaust gases of the reciprocating engine is at a very high temperature (around 650 ºC), which is far from the temperature at which heat is needed for the digestion of sludge (around 40 ºC in the digesters). This temperature difference offers an opportunity to introduce an intermediate system between the engines and the digesters that makes use of a fraction of the available heat to convert it into electricity. An Organic Rankine Cycle (ORC) with an appropriate working fluid is an adequate candidate for these hot/cold temperature sources. In this paper, the techno-economic effect of adding an Organic Rankine Cycle as the intermediate system of an existing wastewater treatment station is analysed. On this purpose, different working fluids and system layouts have been studied for a reference wastewater treatment station giving rise to optimal systems configurations. The proposed systems yield very promising results with regard to global efficiency and electricity production (thermodynamically and economically).

KEYWORDS

CHP, ORC, Organic Rankine Cycle, Wastewater treatment station, Cogeneration

INTRODUCTION

Wastewater treatment stations, which produce biogas by processing sludge (organic matter) in anaerobic digesters, constitute an important source of biofuels worldwide as confirmed by the fact that 15% of the total production of biogas in Europe in 2007 was expected to be produced at such facilities [1]. This biogas fuel is usually employed to generate electricity in reciprocating engines (spark-ignition engines specifically adapted for biogas operation) in order to supply the plant's internal demand of electricity and, if possible, export the excess energy to the grid. Then, the heat rejected from the engine

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(either in the form of hot cooling water or flue gases) is used to support the anaerobic digestion of organic matter [2].

The singular features of the fuel and the simultaneous production of heat and power pose two main challenges to these installations: (i) to be qualified as high efficiency co-generators in order to have access to the beneficial electricity sale conditions [3], and (ii) to eliminate certain impurities that are inevitably found in the biogas yield, compromising the mechanical integrity of the system (engines and others equipment). Volatile sulphur compounds and siloxanes are some of the species whose concentrations must be controlled [4, 5].

From an energy management standpoint, it is observed that the amount of heat required for sludge digestion and the operating conditions of the CHP plant present substantial seasonal variations that make it common that the fraction of heat rejected to the atmosphere (i.e. excess heat) increases significantly at certain times of the year. On top of this mismatch between available waste heat from the engine and heat demand from the digesters, a large temperature difference is found between the temperature at which heat is available in the engine exhaust gases (in the order of 650 ºC) and the operating temperature of the digestion process (around 40 ºC).

Such temperature difference makes it possible to introduce an intermediate system between the engines and the digesters to make use of a fraction of the available heat and convert it into electricity, therefore increasing the efficiency of the system.

An Organic Rankine Cycle (ORC) with an adequate fluid selection is a candidate for the hot/cold temperature source/sink of the reference CHP and WWT plant [6, 7]. These ORC systems are used in a variety of low and medium temperature applications such as geothermal [8, 9], waste heat recovery [10-12] and biomass [13, 14], where the performance of the cycle mainly depends on the working fluid of choice [10, 15]. In this regard, a number of working fluids as hydrocarbons and refrigerants have been studied [12, 16] even if only a few of them are commercially available nowadays.

Further to the previous works referred above and for the particular case of waste heat recovery from internal combustion engines, an ample discussion of the different technologies available can be found in reference [17] for the automotive industry and in reference [18] for the case of heavy-duty stationary diesel generators. In the latter group, and concerning system integration, Vaja and Gambarotta [19] discuss different integration schemes between stationary diesel engines and bottoming Organic Rankine Cycles: recuperative versus non-recuperative cycles that recover heat from the exhaust gases only or also from the cooling water are explored.

The afore listed works consider bottoming Rankine cycles working on organic species, on steam or on even more original working fluids like in the Kalina cycle (water-ammonia mixture). Concerning the former group (organic), a very complete analysis is provided by Maizza and Maizza [20], who screen twenty different species (mostly refrigerants) in order to select the candidate with the most interesting properties from the cycle efficiency standpoint. In the same sense, Quoilin et al. [21] provide a review of the many different organic fluids studied to date, along with the application and temperature ranges in which they have been considered. Nevertheless, amongst the works that concentrate on characterising the behaviour of organic working fluids, that by Angelino and Colonna must be acknowledged for its completeness and technical consistency [22].

Finally, the work by Gewald et al. [23] on the assessment of waste heat recovery from landfill-gas engines must be noted as it has many similarities with the present article. This very recently published research work discusses the interest of recuperating the waste heat carried by flue gases at a power plant comprising fifteen reciprocating engines operating on landfill gas. This recuperated energy is then employed by an organic Rankine cycle to produce additional electricity. Nevertheless, in spite of both plants being fired by biogas coming from municipal waste (solid waste in ref. [23] and wastewater in the present article) there are substantial differences between both facilities that make it quite difficult to exchange the conclusions between one another.

In this work, the incorporation of an Organic Rankine Cycle into an existing wastewater treatment station, a combination that has not been previously studied in detail to the authors' best knowledge, is analysed. To this aim, the retrofit of the CHP plant is thoroughly studied by considering new operating modes and adaptation of certain components with the goal of maintaining the digesters' optimal conditions for most of the operating range of the plant. The analysis yields promising results with regard to energy and economic performance. It is hence clear that the most notable singularity of the plant analysed in this work and reference [23] is the fact that the exhaust gases from the engine are released to the atmosphere at the solid waste plant whereas they supply heat to the process at a wastewater treatment plant. In other words, the incorporation of an additional bottoming system is unconstrained in the first case and constrained in the second one. These issues will be covered throughout the article.

FACILITY DESCRIPTION

Description

The reference wastewater treatment station is located in Andalusia, South of Spain. A general overview showing how the processes are organised is shown in Figure 1. In the primary sedimentation stage, wastewater is conveyed from the collecting well to the primary sedimentation tanks where heavy solids settle by gravity whilst oil, grease and lighter solids float on the surface. In the secondary treatment, biological processes eliminate the matter suspended or dissolved in the wastewater. A final tertiary treatment provides the effluent with the specifications required by the water quality regulations applied to the market under consideration.

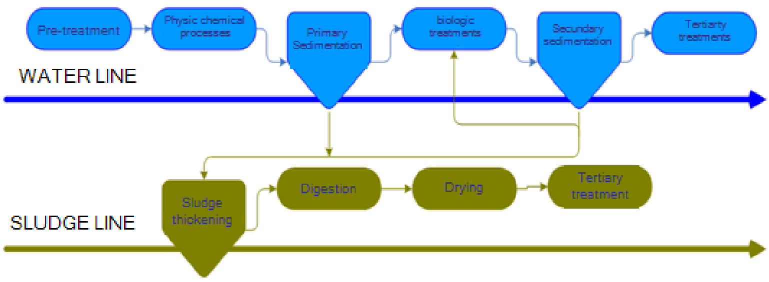

Figure 1. Wastewater treatment processes

In the wastewater station under study, sludge generated in the primary and secondary treatments is homogenised prior to the injection of ferric chloride to abate sulphur compounds by precipitation/scavenging. Then, sludge is conveyed to the anaerobic digesters where they are stored for twenty eight days at temperatures within a very narrow range in order to ensure that the rate at which the microorganisms break down the biodegradable matter is highest. Finally, sludge is dried by centrifugation and sent for final storage/disposal.

In the reference plant, the anaerobic digestion of sludge takes place in three reactors (digesters) with approximated volumes 7,400, 10,500 and 21,000 m^3. Within these vessels, biogas is generated with an average composition of 60% methane and 35% CO_2 (volume percentage) and a mean Lower Heating Value (LHV) of 23 MJ/Nm^3. Nevertheless, special attention must be paid to the siloxanes and volatile sulphur compounds (VSCs) that are also present in the biogas inasmuch as they can seriously damage the engines and heat exchangers even if found in very small quantities [4, 24]. As indicated before, there are different technologies for the abatement of VSCs and siloxanes [4, 25] though, in the context of this work, a treatment with ferric chloride ensures that the concentration of hydrogen sulphur in the clean biogas is below 20 ppm.

The CHP power plant comprises three engine generator sets that typically operate on clean biogas even though biogas/natural gas blends can also be used. Tables 1 and 2 present the main specification of the engines and their part-load performance as provided by the manufacturer.

Table 1. Engine specifications at rated conditions

Model	GUASCOR FGLD 480/80
Configuration	V-16
Shaft speed [rpm]	1,500 rpm
Brake/ Electrical Power (ISO conditions)	690/667 kW
Heat input [kW]	1,860
Heat rejected by cooling water [kW]	548
Heat rejected by oil cooler [kW]	81
Heat rejected by intercooler [kW]	61
Heat rejected by radiation [kW]	43
Intake air flow [kg/h]	3,670
Exhaust gas flow [kg/h]	3,810
Electric efficiency [%]	35.1

Table 2. Part-load performance of the engine

Load	100%	75%	50%
Electrical power [kW_{el}]	667	495	330
Exhaust temperature [ºC]	404	379	354
Efficiency [-]	0.351	0.308	0.264
Heat from cooling water [kW_t]	548	496	445

The CHP plant, shown in Figure 2, includes the following equipment and subsystems in addition to the generator sets:

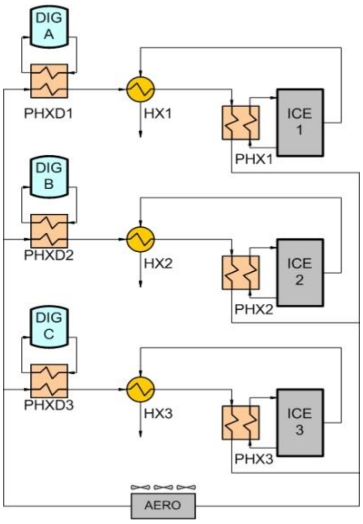

Figure 2. Layout of the reference CHP plant

- An intermediate closed water loop to convey heat from engines to digesters.
- Plate heat exchangers for low temperature heat recovery from engine cooling jackets and lube oil system (PHXi).
- Shell and tube heat exchangers for high temperature heat recovery from the exhaust gases (HXi).
- Plate heat exchangers for low temperature heat supply to the anaerobic digesters (PHXDi)
- Pressurised storage and supply of biogas for the engines.
- Air coolers to reject excess heat from the water loop

CHP performance

The design goals of the current CHP installation are as follows:

- Highest utilisation of biogas.
- Temperature of digesters close to optimum value (for highest biogas production rate) across the entire range of operation.
- Highest net electricity delivered to the grid during peak hours.

Under these considerations, the default operating mode of the CHP plant is based on running two engines only in ordinary market conditions and starting-up the third one during peak price periods. Table 3 presents some relevant performance parameters of the CHP plant. The “design” values correspond to the predicted operating conditions of the original project (i.e. the rated operating conditions) whereas “operation 2007” stands for mean deviation from the former values as obtained from the operation of the digesters in year 2007 (there were no data available for the rest of the equipment).

Table 3. Rated vs. real performances of the CHP installation

		Design		Operation 2007	
		Hot side	Cold side	Hot side	Cold side
Engine cooling jackets	Flow [m^3/h]	67.2	59.9	-	-
	Inlet Temperature [ºC]	85	70	-	-
	Exhaust Temperature [ºC]	78	77.8	-	-
Digester A	Flow [m^3/h]	61	25.5	-26.2%	96.1%
	Inlet Temperature [ºC]	81.8	59	-18.1%	3.4%
	Exhaust Temperature [ºC]	78	68	-17.3%	-5.9%
Digester B	Flow [m^3/h]	59.9	48.8	-24.9%	-9.8%
	Inlet Temperature [ºC]	81.8	59	-18.1%	2.5%
	Exhaust Temperature [ºC]	74.4	68	-16.7%	-5.7%
Digester C	Flow [m^3/h]	59.8	90.7	-24.7%	34.5%
	Inlet Temperature [ºC]	81.8	59	-18.1%	2.5%
	Exhaust Temperature [ºC]	68	68	-7.2%	-5.6%

The values in Table 3 confirm that the performances obtained in real operating conditions differ substantially from the corresponding design values, both in flow and temperature. As a general rule of thumb, it is observed that there is a trend to reduce the mass flow rate of water in the hot side of the water loop and also to maintain the temperature drop of this hot water though at a lower average temperature level. In other words, the temperatures of hot water entering and leaving the heat exchangers of digesters A, B and C are lower than the rated values even though the temperature drops are at their design values (just a slight deviation with respect to this general pattern is observed in digester C).

With respect to the cold side, the general pattern shows that the mass flow rate of water is largely increased with respect to the rated value whereas the temperature drop decreases: higher inlet temperature and lower outlet temperature at the cold side of heat exchangers A, B and C. In conclusion, the differences observed in Table 3 suggest that the rated conditions are probably wrongly selected (the plant is not sized correctly) or, conversely, that there is room for increasing the efficiency of the plant by taking advantage of the unused waste heat from the biogas engines. This is further assessed below.

The annual energy balance for year 2007 is presented in Figures 3 and 4 where the main energy consumptions of the wastewater treatment station, the energy balances in the CHP plant and some indicators of the cogeneration power block are given [26]. The global electricity demand was of 15.56 GWh_{el}, the annual electricity production 10.99 GWh_{el} and the biogas generated in terms of energy capacity 32.74 GWh_t.

Figure 3 presents the monthly variation of the energy balance in the existing wastewater treatment station (key symbols are *W* for the production of electricity, *Q* for the recuperated heat energy from the engine and *F* for the primary/chemical energy in the biogas fuel produced by the anaerobic digesters).

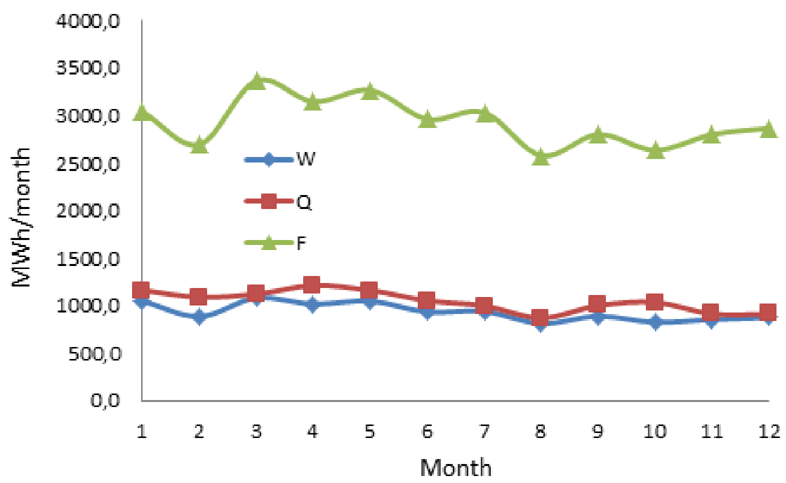

Figure 3. Monthly variation of energy parameters in the wastewater treatment station (2007)

The indicators used to characterize the cogeneration are equivalent electric efficiency (*EEE*) [3] and energy utilization factor (*EUF*) [26], defined as follows:

$$EUF = \frac{(W+Q)}{F} \tag{1}$$

$$EEE = \eta_{EEl} = \frac{W}{(F-Q/\eta_{Q,Ref})} \tag{2}$$

Where *W*, *F* and *Q* stand for electricity, fuel primary energy (heat input) and used waste heat (useful process heat) respectively. The global values of *EEE* and *EUF* were 65.3% and 67.4% respectively. The current cogeneration facility complies with the requirements to be certified as a highly efficient cogeneration plant, which is set by the minimum *EEE* specified by the applicable legislation depending on the characteristics of the CHP unit [3]; for wastewater biogas with internal combustion engines this value is 0.5 as defined in the Royal Decree RD 661/2007 [3]. The classification is not dependent on the current moratorium of electricity generation for special regime in Spain. Nevertheless, in spite of this reasonably good performance, there is still room for performance enhancement as suggested by the aforementioned analysis of the operating modes and data. Amongst the potential actions to be taken, the following can be cited:

- Operate with three engines running at 95% continuous rating. This is possible given the current production of biogas.
- Implement a control system to monitor the heat exchangers and, ultimately, the flow of hot water to each digester in order to maintain temperatures closer to the design values.
- Upgrade the air coolers to avoid excessive cooling of water (it currently works 10 ºC below the design point with an outlet temperature of 60 ºC).
- Add an intermediate system to take advantage of the temperature difference between the engine's exhaust gas temperature and the operating temperature of the digesters.

The incorporation of new operating strategies and a new intermediate system based on Organic Rankine Cycles to recuperate surplus energy from the engine exhaust gases is analysed in the next sections.

ORC PLANT

The previous section has presented the most relevant characteristics of the existing CHP plant along with a discussion on the potential for efficiency improvement if changes in the operational modes and/or in the plant layout were incorporated. In the latter regard, the integration of an Organic Rankine Cycle (ORC) into the existing CHP plant is now proposed. Such a cycle is suitable for facilities operating at moderate to low temperatures (at turbine inlet), to which they adapt by an appropriate selection of the working fluid.

Model description

The model of the CHP plant is implemented in Engineering Equation Solver (EES) [27]. It consists of a lumped volume approach where mass and energy conservation equations are applied to each component of the plant. The latter are presented below:

Heat exchangers: $$\dot{m}_{CF}\Delta h_{CF} = \dot{m}_{HF}\Delta h_{HF} \tag{3}$$

Turbine (ORC): $$\dot{W}_t = \dot{m}\Delta h_s \eta_t \tag{4}$$

Pumps: $$\dot{W}_p = \dot{m}\frac{\Delta h_s}{\eta_p} \tag{5}$$

In addition to heat and mass balances, the performance of heat exchangers is modelled by means of the *ε-NTU* method:

$$\dot{Q} = \varepsilon \dot{Q}_{max} \tag{6}$$

$$\varepsilon = \varepsilon(C_{CF}, C_{HF}, NTU) \tag{7}$$

$$NTU = \frac{UA}{min(C_{CF}, C_{HF})} \tag{8}$$

where C is the heat capacity (specific heat at constant pressure times mass flow rate), NTU the number of transfer units and U is a global heat transfer coefficient evaluated with Gnielinski's correlation [28]:

$$Nu_D = \frac{h \cdot D}{k} = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{9}$$

$$f = (0.790 \cdot ln\, Re_D - 1.64)^{-2} \tag{10}$$

$$\dot{Q} = U \cdot A \cdot LMTD \tag{11}$$

$$LMTD = \frac{(T_{HF,in} - T_{CF,out}) - (T_{HF,out} - T_{CF,in})}{\ln\left(\frac{T_{HF,in} - T_{CF,out}}{T_{HF,out} - T_{CF,in}}\right)} \tag{12}$$

where f is the friction factor coefficient that can be determined by Eq. (10) for smooth surfaces and $LMTD$ is the logarithmic mean temperature difference. Fouling effects within the heat exchangers are also accounted for as indicated by the TEMA standard code, Table 4 [29].

Table 4. Fouling resistances for the fluids in the plant [29]

Fluid	Fouling resistance [m^2K/W]
Engine exhaust gases	0.001-0.002
Water (non-demineralised)	0.0001-0.0002
Industrial organic heat transfer media	0.0001-0.0002

The performances of the engine at rated and partial loads are summarized in Tables 1 and 2, whose data are later corrected to account for variable ambient conditions as per the following equations taken from the ISO 3046 standards [30]:

$$P_x = \alpha\, P_r \tag{13}$$

$$\alpha = \beta - 0.175(1 - \beta) \tag{14}$$

$$\beta = [(P_x - \varphi_x P_{sx})/(P_r - \varphi_r P_{sr})][T_r/T_x]^{0.75} \tag{15}$$

$$b_x = (\beta/\alpha)\, b_r \tag{16}$$

where P_s stands for saturation pressure at the corresponding temperature. The reference conditions for pressure P, temperature T and relative humidity φ in Eqs. (13-16) are 100 kPa, 298 K and 30% respectively [30].

ORC plant performance

In the course of the analysis, which is performed with a lumped volume model of the type described in Section 3.1, recuperative and non-recuperative cycle layouts are considered and the selection of the most appropriate working fluid is done on the basis of previous works by the authors: in combined cycles with gas turbines [6, 11], new concepts of solar plants [10] or compound systems with high temperature fuel cells [31]. In effect, the performance of these systems is strongly dependent on the thermo physical properties of the working fluid [32] and, following this rationale, the fluids considered in this work are Toluene, Cyclohexane, R245fa and Isobutane, whose critical conditions and stability limits are presented in Table 5 [10, 15]. It gives the critical pressure and temperature, the maximum operating pressure, the maximum temperature as saturated vapour and the maximum temperature for the fluid stability.

Table 5. Temperature and pressure limits for the fluids under analysis

Fluid	T_{crit} [K]	P_{crit} [bar]	T_{maxsat} [K]	P_{max} [bar]	T_{lim} [K]
Toluene	591.75	41.3	569.05	31.2	671.95
Cyclohexane	553.65	40.8	536.05	32.7	560.75
R245fa	427.25	36.5	403.05	23.3	433.65
Isobutene	407.85	36.4	380.05	25.5	413.35

Figure 4 presents the ORC recuperative layout and the T-s diagram. The recuperative layout is feasible when organic (dry) fluids are used since they are found in superheated state at the turbine exhaust section. Hence, as a general rule of thumb, it can be considered that ORC recuperative cycles are of interest when the temperature difference between the turbine exhaust and the pump impulse section is above 50 ºC (if not, the added cost of this device is not compensated by the increased efficiency) [10, 11].

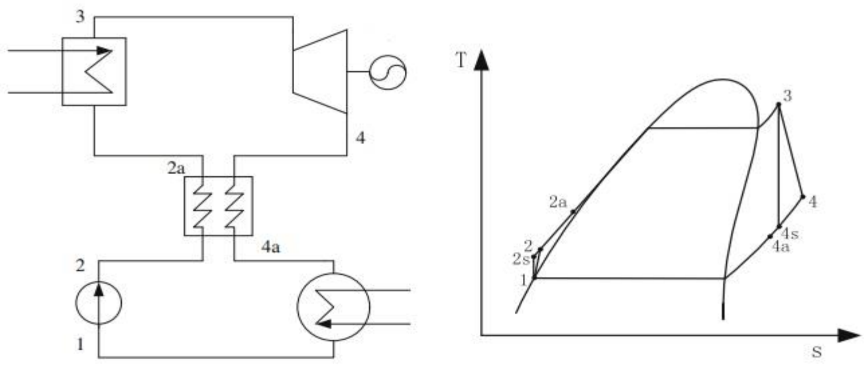

Figure 4. ORC recuperative layout and *T-s* diagram

Table 6 presents the specifications of the reference ORC cycle used for comparison (ΔT_{rec} is the pinch point in the recuperator).

Table 6. ORC cycle parameters

	Value
η_t	0.85
η_p	0.8
T_i [ºC]	Depends on the ORC fluid
T_c [ºC]	Depends on the month (Reference June 30)
ΔT_{rec} [ºC]	10

The bottoming cycle global efficiency (BC) results of the combined effects on the efficiencies of the Heat Recovery Vapour Generator (HRVG) and the ORC cycle:

$$\eta_{BC} = \eta_{HRVG} \cdot \eta_{ORC} \tag{17}$$

In Fig. 5 is represented the bottoming cycle efficiency as function of the turbine inlet temperature (TIT) for a toluene recuperative cycle at different live vapour pressures. It shows that in function of the vapour pressure the optimum TIT can be at saturated vapour or superheated vapour. It will depend on the characteristics of the hot gas stream, the lower limit for the stack temperature and the heat exchanger design. The bottoming cycle efficiency evolution can be explained from the HRVG and ORC cycle efficiencies, Fig.6. The HRVG efficiency is constrained by the lower limit for the stack exhaust temperature (130 ºC for this plant) and by the minimum temperature difference between hot and cold streams in the economizer. Fig. 6 shows that the ORC cycle efficiency is increased with the turbine inlet temperature. On the other side the HRVG efficiency value remains constant in the TIT range where the minimum stack temperature is the major operation constraint and it decreases when the minimum temperature difference between hot and cold streams in the economizer controls the stack temperature.

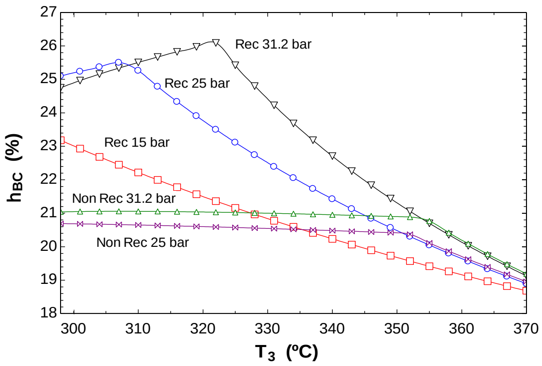

Figure 5. Bottoming cycle efficiency as function of the toluene ORC turbine inlet temperature at different pressures

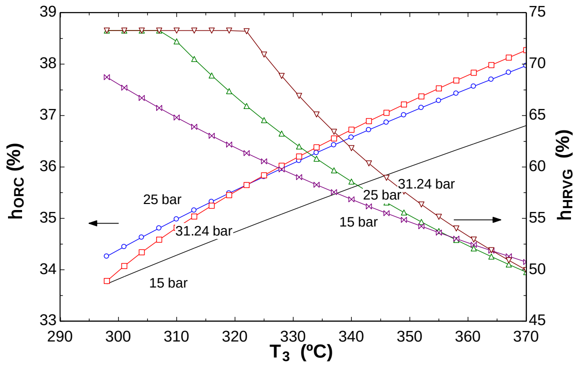

Figure 6. ORC and HRVG efficiencies of the toluene recuperative ORC as function of turbine inlet temperature for different pressures

Table 7 presents the performance of recuperative and non-recuperative standalone ORC cycles for both saturated and superheated vapour under the conditions and limits given in Tables 5 and 6. It presents the bottoming cycle and ORC efficiencies and the relative power referenced to the value of the superheated toluene. Superheated values are given at the maximum efficiency values.

Table 7. Performance of standalone ORC cycles

Recuperative/ Non-Recuperative	**Saturated**				**Superheated**			
Fluid	T_3/P_3 [°C]/[bar]	η_{ORC} [%]	η_{BC} [%]	Power [kW]	$T_{3rec}/T_{3non\ rec}/P_3$ [°C]/[bar]	η_{ORC} [%]	η_{BC} [%]	Power [kW]
Toluene	295.9 / 31.2	33.5/ 28.7	24.6/ 21	355.1/ 304	320.6/307 31.2	35.6/ 28.7	26.1/ 21.1	376.7/ 304.5
Cyclohexane	262.9 / 32.3	25.5/ 31	18.7/ 22.7	270.2/ 328.2	281.3/262.1 / 32.3	32.8/ 25.5	24.1/ 18.7	347.7/ 270
Isobutane	106.9/ 22.5	NF/ 13.6	NF / 9.9	NF / 142.6	140.2/140.2 22.5	16.6/ 13.7	12.2/ 10	175.4/ 143.8
R245fa	129.9 / 23.3	NF / 16.2	NF / 11.8	NF / 170.5	153.4/159 / 23.3	19.6/ 16.3	14.4/ 11.9	207.2/ 172.5

From the results presented in Table 7 two cases are most interesting: recuperative cycle with toluene and non-recuperative cycle with R245fa. The first configuration is selected because of the good cycle performance due to its high maximum temperature in comparison with other fluids. The non-recuperative cycle with R245fa is also considered for its stability and condenser pressure (above atmospheric), which make it a good candidate even though its power production and efficiency are lower. The remaining cases are disregarded.

The following conclusions are drawn (some of which are to be expected from the fundamentals of Rankine cycles):

- Higher evaporation pressures result in performance improvement. Thus, for the ORC integration, the evaporation pressure shall be taken as high as possible within the limits set in Table 4.
- Lower condensation pressures result in better performance. Hence, this pressure shall be selected as low as possible for a given cooling fluid temperature, even if it must be noted that working at vacuum pressure brings about higher costs and complexity due to the need to deaerate (evacuate non-condensing gases from the system). For the water temperature in the wastewater treatment station, toluene is assumed to condense at atmospheric pressure whereas the R245fa's condenser operates above atmospheric pressure.
- Working with superheated vapour is irrelevant (R245fa) or even slightly detrimental (toluene) in the non-recuperative cycle.
- For the recuperative cycle, whether or not the limited improvement in cycle performance pays off the added cost to the system due to the new component in the vapour generator needs to be assessed in a global context where not only capital costs but also revenues are taken into account. As a rule of thumb a minimum temperature difference of 50 ºC must exist between the hot and cold streams to justify the additional cost of introducing the recuperator.

The following sections focus on the two last bullet points. First, the integration of the ORC system into the reference CHP island is studied in Section 4, aiming to determine the global efficiency that can be attained by the combined plant. This global efficiency consists of accounts for the performance of various subsystems: ORC (already studied in this section), heat recovery vapour generator (which is affected by the ORC live vapour and condenser parameters) and gas engine. Finally, the performance of the updated CHP plant is studied from an economic/financial standpoint in order to see how it compares to the reference plant. This is presented in Section 5.

CYCLE INTEGRATION AND RESULTS

The analysis of stand-alone organic cycles presented in the preceding section yields the following conclusions with respect to integrating an ORC system into the reference CHP facility. For the Toluene recuperative ORC at rated conditions, superheated live vapour at 320 ºC and 31.24 bar is employed and the condenser pressure is set to 0.2 bar absolute. A deaerator is therefore needed and special safety measures (ventilation) must be implemented accordingly due to the flammable characteristics of toluene vapours. To take advantage of the higher operating temperature of this cycle, the Heat Recovery Vapour Generator (HRVG) is placed just downstream the engines' exhausts, before any heat exchange with the digesters takes place, Figure 7. The air coolers reject the excess heat from the water loop of the digesters and from the cooling loop to the ORC condenser (not represented in figure). Therefore the condenser pressure is conditioned by the air coolers' performance and through them, linked to the water loop of the digesters.

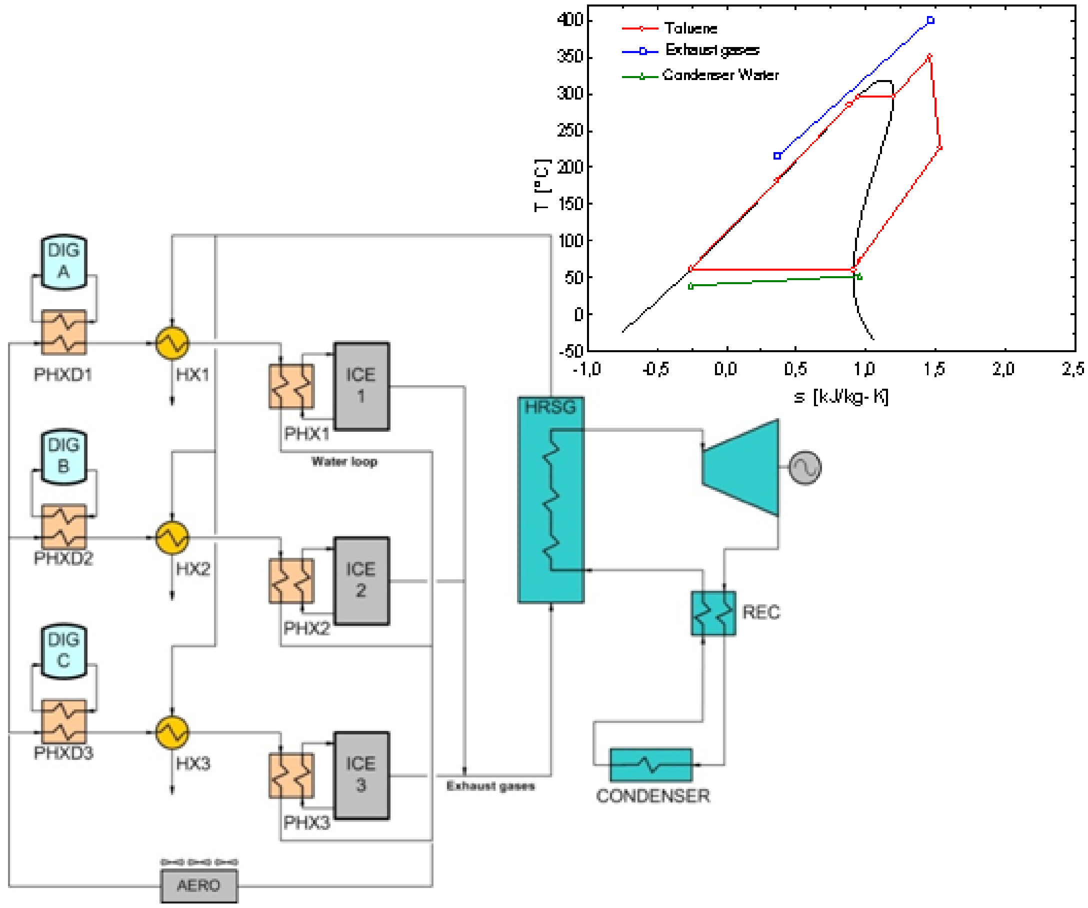

Figure 7. Layout 1. Integration of the toluene recuperative cycle.

The R245fa non-recuperative plant works with saturated vapour at a maximum pressure of 23.3 bar (129.9 ºC) and condensing at 3 bar so there is no need to incorporate a deaerator. Due to the lower maximum temperature of this cycle, the HRGV is placed after the digesters' heat exchangers, using the bypass valves at the engine's exhaust to increase the inlet temperature if needed. Such layout adds less complexity to the reference plant, Figure 8.

As aforementioned, the ORC integration in the CHP plant requires changes in the operating modes to take maximum advantage of the new layout. Hence, a new operating mode based on maintaining the digesters' temperatures as steady as possible at their design values under different conditions is adopted. To this aim, the three engines operate simultaneously at 95% load, delivering hot water at 80 ºC from the heat exchangers located in the exhaust gas streams. With this solution and the air cooler temperature switching on at 70 ºC (design temperature), the power consumption in the cooler is largely reduced due to the heat absorbed by the ORC system. The stack temperature is designed to be well above 130 ºC in order to avoid acidic condensate. During off-design operation, the power delivered by the ORC system is controlled by acting slightly on the throttle control valve† and, under certain operating conditions where the heat demand from the digesters is very high and the output of the ORC drops below the minimum stable rating, a diverter can be used to bypass the HRVG.

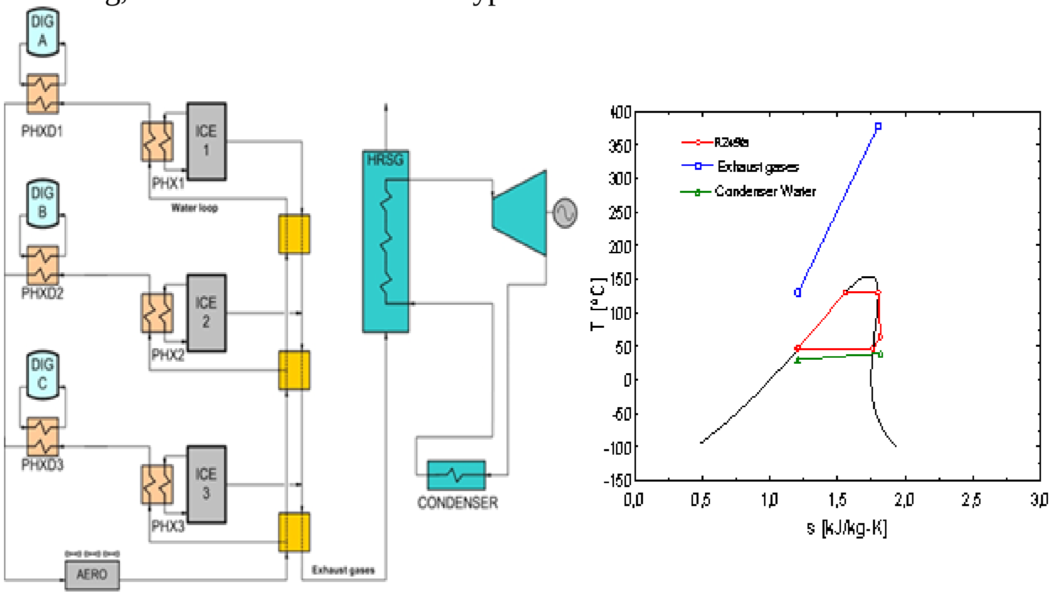

Figure 8. Layout 2. Integration of the non-recuperative cycle with R245fa

Heat Recovery Vapour Generator

The critical component in the integration of the ORC system is its Heat Recovery Vapour Generator. This component is sized in order to obtain the highest output from the turbine whilst, at the same time, fulfilling the heat demand of the wastewater station. This thermal power transferred to the digesters is primarily controlled by varying the flow of water in the intermediate loop though, if necessary, it can also be modified by lowering the amount of heat recuperated from the engine's cooling water and exhaust gases. This is accomplished with a set of by-pass valves installed accordingly (not represented in the layouts for the sake of clarity).

Designing the vapour generator involves selecting a certain configuration/technology and applying mass and energy balances. To this aim, the HRVG is assumed to be composed of cross-flow heat exchangers due to their lower area requirements (and the associated cost savings) for a certain duty in comparison with other technologies. For the energy balances, the properties of the exhaust gases are evaluated as those of a perfect gas mixture with the given composition at the engine's exhaust conditions. Finally, a 5% pressure loss is also assumed to take place within the HRVG's liquid and vapour paths.

Tables 8 and 9 present the predicted performance of the plant at design conditions and the average performance of the retrofitted plants (both layouts) during year 2007 respectively.

Table 8. Surface of heat exchangers and CHP performance at design conditions

	A_{HRVG} [m^2]	A_{reg} [m^2]	W_{ORC} [kW]	η_{ORC} [%]	W_{CC} [kW]	η_{CC} [%]
Layout 1	370	40	241.2	0.35	2131	0.387
Layout 2	48	-	120.1	0.1398	2010	0.365

† A more detailed description of the control strategy implemented in an Organic Rankine Cycle for part-load operation is given in reference [11] by the authors, where an analysis of gas turbine and ORC bottoming systems is presented.

Table 9. Annual performance of the retrofitted plants

	W [MWh]	*Q* [MWh]	*F* [MWh]	*EEE* [%]	*EUF* [%]
Layout1	18281.38	18860.68	47580.4	0.8738	0.7764
Layout2	17215	20404.59	47580.4	0.9340	0.7907

In the existing wastewater station, the annual electricity consumption is 15590 MWh, 10400.6 MWh of which are produced by the CHP plant. In contrast, Table 9 reports that surplus electricity can be exported to the grid in both cases, providing additional incomes to the wastewater treatment station. In particular, 2691.4 MWh are generated in excess annually when the combined system using a toluene ORC (layout 1) is selected; when layout 2 is used (R245fa), this excess energy is lower (1625 MWh) but still largely enough to cover the annual demand. Furthermore, if the energy deficit of the reference plant is added to the calculations (5189.4 MWh), it is found that the equivalent electricity savings totalize 7880.8 MWh for layout 1 with toluene and 6811.4 MWh for layout 2 with R245fa. This is shown in Figure 9 where the annual electricity production, the CHP efficiency and the electricity/biogas ratio (right) for the retrofitted plants and the original are compared.

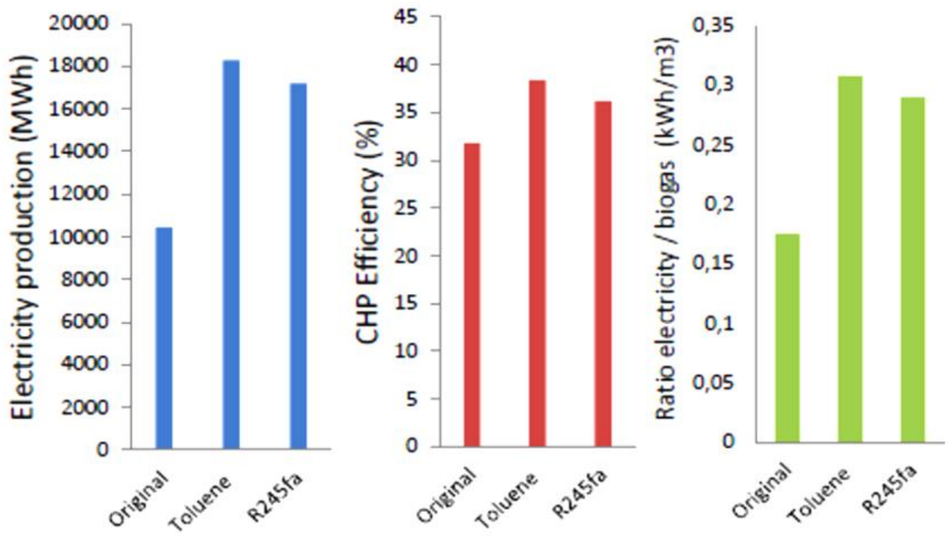

Figure 9. Comparison of the annual electricity production (left), CHP efficiency (middle) and the electricity/biogas ratio (right) for the retrofitted plants

The analysis presented so far suggest that layout 2 using R245fa is the best choice for the following reasons:

- Only minor modifications in the existing power plant are required to implement layout 2.
- *EEE* is high enough for the plant to qualify as a high efficiency energy cogenerator even for future regulations of highly strict conditions.
- R245fa has higher stability and lower toxicity and flammability than toluene [12, 33].
- The ORC with R245fa is simpler: absence of deaerator and recuperator, reduced investment and simpler plant operation.

- There is ORC technology currently operating with R245fa; i.e. it is close to be off-the-shelf technology (for instance, Capstone's Clean Cycle [34] or Infinity's IT10 [35]).

An economic assessment is presented in the subsequent section in order to asses if there are economic considerations that off-set this preliminary decision in favour of R245fa.

ECONOMIC ANALYSIS

The steady-state performance of the proposed CHP facilities has been characterized in previous sections. Now, a deterministic economic analysis of these plants is presented, based on the appraisal of Net Present Value (*NPV*) and Internal Rate of Return (*IRR*).

$$NPV(i,N) = \sum_{t=1}^{N} \frac{R_t}{(1+i)^t} \tag{18}$$

$$\sum_{t=1}^{N} \frac{R_t}{(1+IRR)^t} = 0 \tag{19}$$

$$LCOE = \frac{I+L_e+M+R+F}{E_1 \times \sum_{i=1}^{n} \frac{1}{(1+r)^i}} \tag{20}$$

The first step towards the economic appraisal of the retrofitted plant is estimating the specific cost of the Organic Rankine Cycle to be added to the original CHP plant. This difficulty comes about because of the little commercial deployment of the technology which, even if already competitive, still needs to become more mature from an economic standpoint. Based on this statement, a thorough literature review has been performed, yielding the results shown in Table 10 where the specific capital cost of the organic system (including vapour generator, turbine, heat exchangers) [€/kW$_{el}$] depends on the power output of the unit [kW$_{el}$]. This dependence aims to account for the economies of scale that are to be expected from power systems though, as observed in the table, the lack of commercial information results in important discrepancies between references (for instance, note the specific costs for the megawatt range).

Table 10. Specific investment cost vs. capacity (power output) of the system

Specific cost ORC [€/kW$_{el}$]	Capacity [kW$_{el}$]	Ref.
3,755	35	[13]
5,775	2	[36]
3,034	50	[36]
3,000	50	[37]
2,500	100	[37]
3,832	538	[38]
1,600	1,000	[39]
2,765	1,000	[40]&[41]
2,116	1,155	[38]
1,947	2,079	[38]
2,023	2,310	[42]&[43]

The information presented in Table 10 is shown graphically in Figure 10 where the scattering of specific costs coming from different references is more visible. As indicated by Walsh and Thornley [42], logarithmic functions replicate the behaviour of specific cost vs. power output for these power systems (i.e. provide the best curve fitting to the data in Table 10). Thus, the fitting given by the solid line and equation shown in Figure 10 is used henceforth.

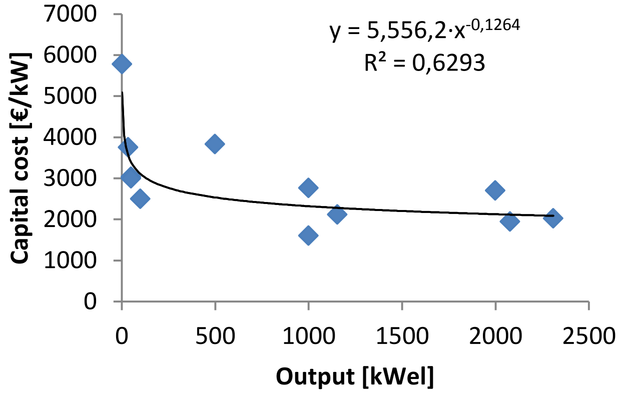

Figure 10. Specific capital cost vs. capacity (power output) of the system. Curve fitting.

Due to the inevitable uncertainty associated with the specific cost, the economic appraisal of the plant is based on a sensitivity analysis of this variable. To this aim, the following economic assumptions are defined:

- Financial scenario presented in Table 11.
- Market prices for the electricity exported to the grid are determined from the applicable Spanish legislation. The recent RDL 9/2013, 12th of July of 2013 [44], changes drastically the special regimen electricity production although it maintains electricity tariffs for a transient period.

Table 11. Main assumptions of the financial and economic analysis

Installed capacity ORC unit [kW_{el}]	120.1
Lifetime [years]	15
Discount rate [%]	10
Inflation [%]	2.5
Taxes [%]	30
Interest rate (loan) [%]	10
Loan payback time [years]	15
Debt share (loan/total cost) [%]	50
Amortisation [%]	100
Electricity sale price for cogeneration, [€/kWh]	0.16061
O&M cost annual [% initial cost]	1.5
Capacity factor [%]	90

Figure 11 presents the effect of varying the cost of the ORC skid on the plant's *NPV* and *IRR*. It is shown that, in spite of the expected negative effect of increasing the ORC cost, high Net Present Values are still obtained even for high variations of this parameter (i.e. high uncertainty). Therefore, the benefits derived from retrofitting the conventional reference plant are ensured even if strong deviations from the normal economic scenario were to take place.

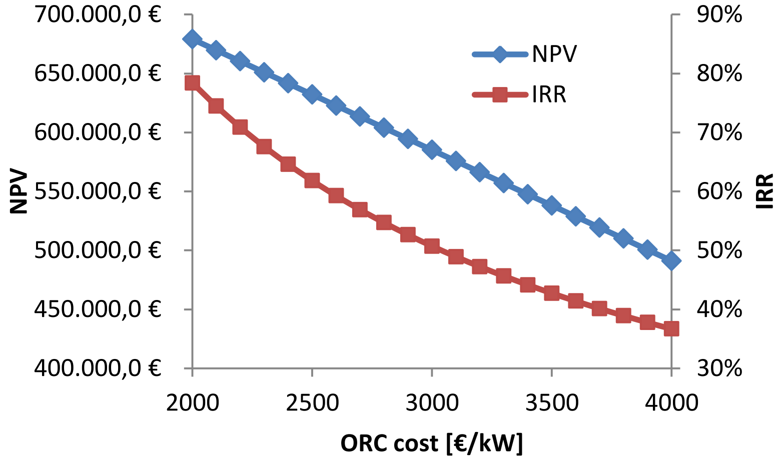

Figure 11. Sensitivity analysis to the capital cost of the ORC system

In addition to the specific cost of the ORC system, there are other parameters in Table 11 that affect the economic performance of the retrofitted plant and whose values are difficult to estimate. To account for this, the authors have deemed it convenient to perform a "Monte Carlo" analysis incorporating the uncertainties shown in Table 12 below.

Table 12. Sensitivity parameters for the Monte Carlo analysis

Parameter	Mean value	Uncertainty
Specific cost ORC [€/kW$_{el}$]	3033	± 750
Capacity factor [%]	90	± 5
Sale price of electricity, [€/kWh]	0.16061	± 0.025
Inflation [%]	2.5	± 0.5
Interest rate [%]	10	± 2.5

The results obtained from running ten thousand cases with the restrictions of Table 12 are shown in Figure 12. This figure shows the number of cases (percentage) for which the *NPV* and *IRR* take the values given horizontally. For the Net Present Values, the highest probability corresponds to the interval from 550 k€ to 625 k€ (cumulative probability in the order of 50%), whereas this most probable Internal Rate of Return is in the order of 45 to 60% (cumulative probability over 50%).

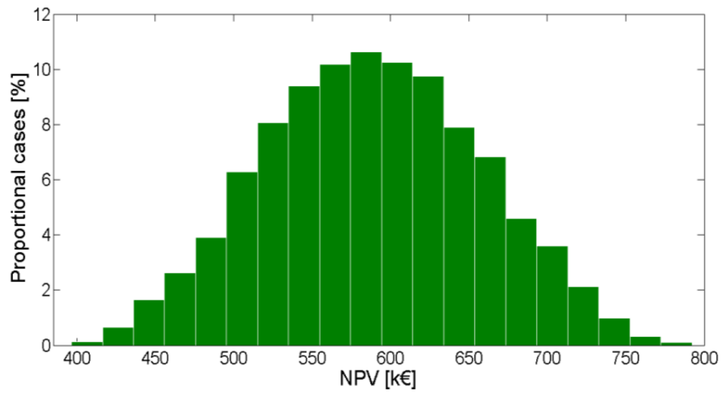

Figure 12. Net Present Value resulting from the Monte Carlo analysis

CONCLUSIONS

This work analyses the retrofitting of a CHP in a wastewater treatment plant with an Organic Rankine Cycle. The following considerations should be highlighted from this analysis:

- The introduction of an adequate Organic Rankine Cycle to take advantage of the temperature difference between the available heat and the anaerobic digestion process allows an increase of the recuperated heat in the CHP plant.
- For the reference wastewater treatment station the retrofitting, combined with an adequate change in the operation mode of the existing engines, increases the power production while maintains the high efficiency cogeneration qualification that brings about complementary incentives and trading conditions.
- Two different CHP- ORC integrations were selected for the retrofitting: a recuperative ORC plant with toluene working with superheated vapour (31.3 bar, 320 ºC) and a non-recuperative cycle with R245fa working with saturated vapour (23.3 bar, 129.9 ºC). The toluene ORC plant was selected because its high performance characteristics while the R245fa ORC was selected because its commercial development, higher stability, lower toxicity and flammability and condenser pressure above the atmospheric.
- The integration of both layouts was designed with the objective of maintaining the temperatures of the digesters at their rated values. For this reason in layout 1, with toluene, the Heat Recovery Vapour Generator (HRVG) is placed at the exhaust of the engines and a set of air coolers maintain the temperature in the digesters while in the layout 2, with R245fa, the HRVG is installed downstream of the digesters heat exchangers.
- The proposed retrofitted CHP & ORC plants yield an average efficiency for the reference year 2007 of 38.7% for the layout1 and 36.5% for the layout 2. The change in the operating mode for the retrofitted CHP plant produces more electricity from the existing engines, changing the net electricity balance of the plant from purchase to sale. The equivalent annual electricity savings totalize 7880.8 MWh for the layout 1 with toluene and 6811.4 MWh for the layout 2 with R245fa.
- The R245 layout would be the preferred option because only minor modifications in the existing power plant are required. The plant with R245fa is simpler (absence of deaerator and recuperator) with reduced investment and probability of problems in the plant operation. Regarding the working fluid, it has higher stability and lower toxicity and flammability. In addition there is ORC technology currently operating with R245fa; i.e. it is off-the-shelf technology.
- The economic analysis confirms the interest of the proposed retrofitting. The effect of the ORC cost is relatively small due to the little economic weight of the investment compared with the change in the electricity production of the whole CHP plant. Therefore, the benefits derived from retrofitting the conventional reference plant are ensured even if strong deviations from the normal economic scenario were to take place. A Monte Carlo analysis accounting for uncertainty in

determining the financial and economic scenario confirms the robustness (generality) of this conclusion. For the Net Present Values, the highest probability corresponds to the interval from 550 k€ to 625 k€ (cumulative probability in the order of 50%), whereas the most probable Internal Rate of Return is in the order of 45 to 60% (cumulative probability over 50%).

NOMENCLATURE

C - heat capacity, kW/ºC
EEE - equivalent electric efficiency, %
EUF - energy utilisation factor, %
F - Fuel Heat Input, kWh
H - enthalpy, kJ/kg
LCOE - levelized cost of electricity
LHV - Lower Heating Value, kJ/kg
m - mass flow, kg/s
NTU - number of transfer units
P - pressure, kPa
Q - heat, kWh
T - temperature, ºC
U - global heat transfer coefficient, kW/ºC
W - electricity, kWh

Greek symbols

ε - effectiveness, %
η_{th} - thermal efficiency, %

Subscripts

c - condenser
e - evaporator
el - electric
CF - cold flow
HF - hot flow
i - turbine inlet
max - maximum
p - pump
r - reference ambient conditions
rec - recuperative cycle
s - isentropic
t - turbine or thermal
x - current ambient conditions

Abbreviations

AERO - aero-refrigerator
CHP - Combined Heat and Power plant
DIG - digester
EES - Engineering Equation Solver
HRVG - Heat Recovery Vapour Generator
HXi - shell and tube heat exchangers for heat recovery at engine exhaust
ICE - Internal Combustion Engine

ORC - Organic Rankine Cycle
PHXi - plate heat exchangers (cooling jackets)
PHXDi - plate heat exchangers (digesters)
VSC - volatile sulphur compounds

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