Energy analysis of a hybrid top/bottoming ORC-based CHP configuration for residential applications

João S. Pereira[*] and José B. Ribeiro

Univ Coimbra, ADAI, Department of Mechanical Engineering, Coimbra Portugal,
joao.pereira@dem.uc.pt

Abstract:

The development of ORC-based micro-CHP systems to retrofit the current combi-boilers is receiving noticeable attention from research centres and companies due to the huge dimension of the potential market. Recognizing the importance that the evaporator has on the technological/ commercial success of these systems, and instead of avoiding the problem through the adoption of alternative restricting solutions (i.e.: indirect vaporization process), it seems reasonable to face the challenges associated with the development of a specifically designed ORC-evaporator capable to directly use the high-temperature combustion gases to perform the vaporization of the working fluid. In an attempt to overcome those challenges, that are believed to be preventing the widespread use of these systems, this paper presents and discusses some of its main design principles. From those principles emerged a hybrid (top/bottoming) CHP configuration in which the thermal energy is produced stepwise: firstly in the ORC condenser and then in a post-heater, that is integrated into the ORC-evaporator. A model of this configuration was developed to determine the fraction of the CHP water heating process performed in the post-heater that maximizes the primary energy savings and ORC net power output for a wide range of CHP operating conditions. When compared to a standard CHP configuration, this solution show benefits for the greater part of those conditions. Besides the performance benefits shown and solving the safety issue posed by the ORC-evaporator requirements, this configuration has an additional positive side effect: the decrease of the combustion gases' temperature before they reach the organic fluid heat-exchanger section in the ORC-evaporator that leads to a reduction of the risk its thermal degradation.

Keywords:

Organic Rankine Cycle, Hybrid CHP configuration, ORC-evaporator, Direct vaporization arrangement.

1. Introduction

The combined production of heat and power (CHP) is one of the major alternatives to traditional energy production systems in terms of energy savings and environmental conservation [1–3]. The most promising target for the micro-CHP systems lies in the residential sector given the huge dimension of the market [4,5]. At this scale, and for solutions attempting to retrofit the wall-mounted combi-boilers that are currently applied in residential dwellings, where the noise, vibrations, weight, dimension and reliability are crucial features, the Organic Rankine Cycle (ORC) based technology appears to be the most promising [6,7]. Even if it may seem surprising, in its basic configuration and from a thermodynamic point of view, an ORC is almost identical to a reverse refrigeration cycle in which the throttling valve is replaced by a pump, the condenser works as an evaporator, the compressor works as an expander and the evaporator as a condenser. From a technological point of view, the ORC-based micro-CHP system may share with the refrigeration devices their main component, the compressor (converted to an expander in the micro-CHP [8,9], while the pump and the condenser are off-the-shelf components of, e.g., coffee machines and HVAC systems, respectively [10]. This share of technology presents obvious benefits from the point of view of reliability, maintenance and cost that, partially due to the existence of alternative moving parts in Stirling Cycle and spark-ignited internal combustion systems, are envisaged to present less noise, vibrations and maintenance requirements [11,12].

Among the ORC-based micro-CHP main components, the evaporator is the only one that, due to its specificities (e.g. working with a vaporizing fluid at relatively high pressures, typically above 10 bar), cannot be found directly on the market or easily adapted from a mass production part of other appliances and needs to be specially designed and analyzed [13]. To partially avoid this question, the overwhelming majority of manufacturers and research centres choose to perform indirect vaporization of the working fluid. In those cases, an intermediate circuit with thermal oil, or slightly pressurized water, is commonly used to transfer the

energy from the high-temperature combustion gases to the organic fluid, being its vaporization performed in a plate-type heat exchanger [14,15]. In this way, it is not only possible to reduce the pressure but also to ensure the existence of a continuous liquid phase in the heat exchanger that is directly exposed to the high-temperature combustion gases, from which a significant reduction in the design requirements is obtained. Moreover, such a solution is expected to perform better control of the temperature of the working fluid vaporization process that can reduce the risk of its thermal degradation [16,17]. In comparison with direct vaporization, in which the phase change of the organic fluid occurs in the part of the evaporator directly exposed to the combustion gases, the aforementioned CHP configuration demands the inclusion of, at least, an additional heat-exchanger, a circulating pump, an expansion vessel, and the necessary tubes and accessories from which an increase of the system dimension and thermal inertial are expected. As an obvious consequence, such micro-CHP strongly reduces its ability to face the intermittence of the hot-water demands, with multiple short-time requests, that characterize one of the operation modes of the boilers [18].

Recognizing the importance that the evaporator has on the technological success of ORC-based micro-CHP systems, and instead of avoiding the problem through the adoption of restricting solutions (indirect vaporization [13]), it seems reasonable to face the challenges associated with the development of a specifically designed ORC-evaporator capable to directly use the combustion gases to perform the vaporization of the working fluid [19,20]. In an attempt to overcome those challenges and fulfil this technological gap, that is believed to be preventing the widespread use of these systems, this paper presents a solution for such evaporators. This solution leads to a hybrid (topping/ bottoming) CHP configuration in which the useful thermal energy is produced stepwise, firstly in the ORC condenser and then in a post-heater directly with hot combustion gases [21]. The benefits arising from its use, in what refers to the primary energy savings, cycle efficiency, net power output and risk of the organic fluid thermal degradation, are illustrated and discussed as a function of several CHP operating (design) conditions.

2. Direct vaporization ORC-evaporator

The gas burner with the characteristics required for the ORC-evaporator was promptly found in the market and is shown in Figure 1-a. Contrarily, the heat exchanger needs to be specifically designed. A schematic representation of its configuration can be seen in Figure 1-b. In this, the hot combustion gases flow around the tubes of the heat exchanger, within which is the organic fluid, in what can be described as a mixed counter flow/cross-flow arrangement. The use of high-temperature insulation material sleeves (presented in brown in Figure 1-b) prevents the energy transfer along the radial direction from the central combustion chamber to the helical coils to keep the counter-flow arrangement nature. Besides the mentioned characteristics, this configuration contains a water-cooled baffled sleeve to be used as a cold surrounding of the gas-burner head to ensure its proper and safe operation. In this, the water flows through the sleeve after being pre-heated in the ORC condenser, reducing the organic fluid condensing temperature in this component, which is beneficial for the cycle efficiency. In addition, this baffle sleeve can also reduce the risk of organic fluid thermal degradation because it decreases the combustion gases' temperature before they reach the ORC heat-exchanger part.
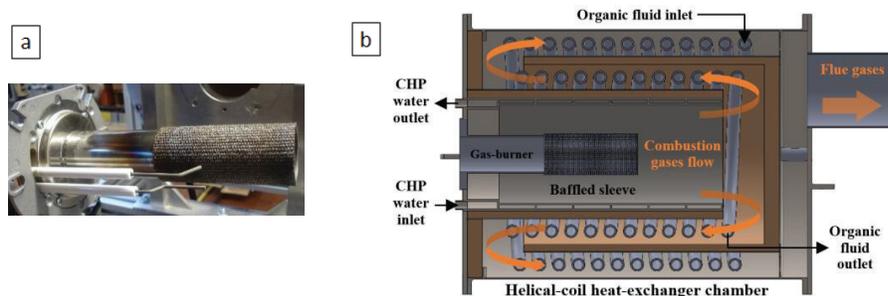
Figure 1: Photograph of the premix gas-burner head with the ignition spark and flame detector rod (a) and the schematic representation of the ORC-evaporator design with the water baffled sleeve (b).
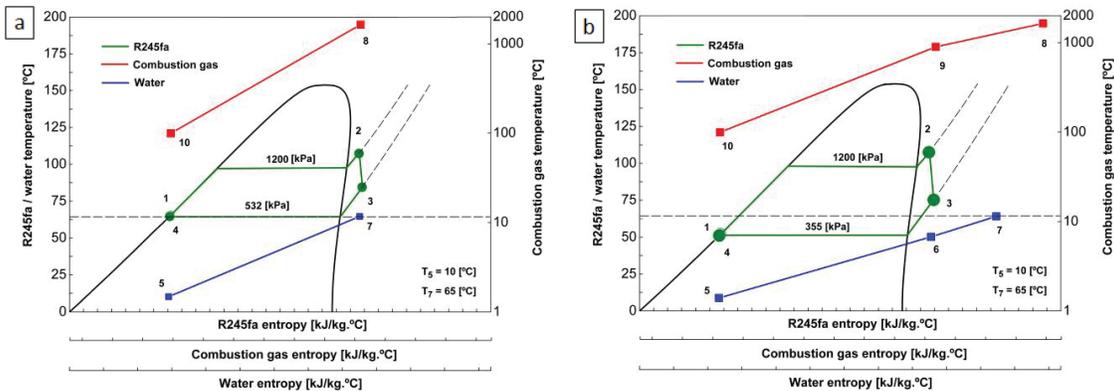
3. Hybrid ORC-based CHP configuration

The schematic configuration of the micro-CHP system, derived from the integration of the newly designed ORC-evaporator, is shown in Figure 2. This can be described as a hybrid (topping/bottoming) CHP configuration [21] since the useful thermal energy is transferred to the water stepwise - before and after the production of work in the prime mover (expander).
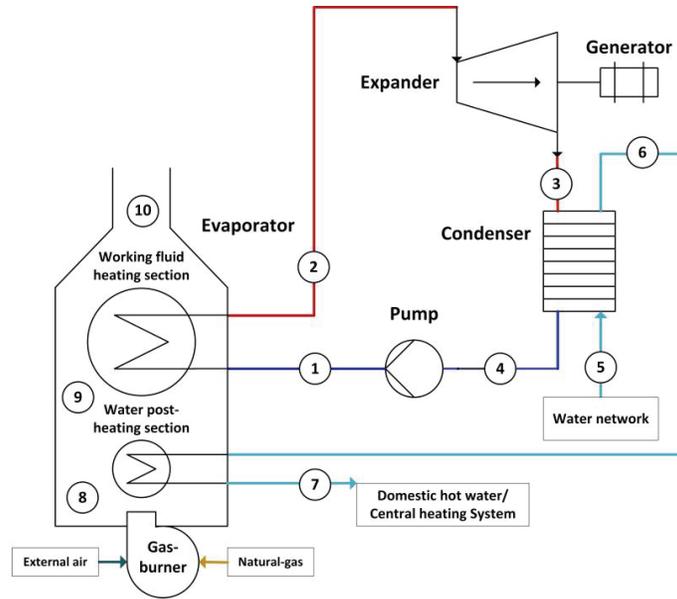
Figure 2: Schematic diagram of the hybrid CHP configuration with the new ORC-evaporator

The T-s diagrams of the hybrid and a standard ORC-based CHP configuration, including the combustion gases and water streams, can be found in Figure 3 using the organic fluid r245fa. For the same CHP outlet water temperature, the condensing pressure (and the temperature) of the ORC in this hybrid configuration may be significantly lower. The well-known advantages of such a decrease in the condensing pressure are an increase in the ORC specific power (net power per unit of working fluid mass flow rate) and an increase in the cycle efficiency. However, these advantages need to be considered against an inevitable reduction of the ORC working fluid mass flow rate that can partially, or totally, hinder them.

Figure 3: T-s diagrams of R245fa ORC-based CHP configurations: a) standard and b) hybrid

In this configuration, the amount of energy transferred to the water in the post-heating section is an additional design parameter that needs to be analyzed. This parameter can be defined by equation (1) where θ can be seen as the fraction of the water heating process that is done in the water post-heating section of the ORC-evaporator. To perform such analysis, a physical model, where all the main ORC components are considered working on their nominal conditions, was developed using Matlab® and REFPROP thermodynamic database [22]. The model assumes no heat or pressure losses as well as neglects the electro-mechanical inefficiencies for both the pump and expander generator.

$$\theta = \frac{T_7 - T_6}{T_7 - T_5} \times 100 \tag{1}$$

The input parameters of this model, with the values used or the intervals within they are allowed to vary (in case of a parametric analysis), are listed in Table 1. Within the list of the input parameters is the water post-heating fraction (θ), defined by equation (1), which can be seen as the fraction of the water heating process that is done in the ORC-evaporator water post-heating section. A value of $\theta = 0\%$ refers to a standard CHP system in which all the water heating process occurs in the ORC condenser, while a value of $\theta = 100\%$ refers to a situation in which there is no cogeneration, or the ORC system is not working.

Table 1. Input parameters of the CHP physical model (based on [21]).

Model parameter	Symbol	Units	Value
Working fluid	-	-	R245fa
Expander isentropic efficiency	η_T	%	0,75
Pump isentropic efficiency	η_P	%	0,5
Condenser efficiency	η_{CHE}	%	0,98
Evaporator efficiency	η_{EHE}	%	0,9
Maximum ORC pressure	p_{max}	kPa	1200
Working fluid superheating degree	ΔT_2	°C	10
Water pressure	p_w	kPa	300
CHP inlet water temperature	T_5	°C	10 [a]
CHP outlet water temperature	T_7	°C	65 [a]
Water post-heating fraction	θ	%	[0, 50]
End-user thermal power demand	Q_w	kW	25
Atmospheric pressure	p_{atm}	kPa	101,325
Combustion gases adiabatic flame temperature	T_{flame}	°C	1540 [b]
Combustion gases' mass flow rate	$\dot{m}_{cg}$	kg/s	0,0137 [b]
Combustion products' mass fraction	m'_{cg_j} [c]	kg_j/kg_{fuel} [c]	[b, d]

[a] Isolated values were used to simulate a specific CHP operating condition (in accordance with the standard [23]) while the intervals were used to perform a system's parametric analysis. [b] The presented values were obtained for the complete combustion of natural gas with 30% of the excess air (according to the gas-burner manufacturer recommendation as shown in section 2.3). [c] $j = \{CO_2; H_2O; O_2; N_2\}$. [d] $\{m'_{cg,CO_2} = 0,123;\ m'_{cg,H_2O} = 0,093\ ;m'_{cg,N_2} = 0,734; m'_{cg,O_2} = 0,051\}$.

The way how the properties of the working fluid, the water or the combustion gases are evaluated at each of the CHP key points is presented in Table 2. To avoid problems related to the pinch-point in the ORC-condenser, the model assumes that the condensing temperature is limited (inferiorly) by the water temperature at the ORC-condenser exit (see Table 2 where, for point 3, at the expander exit, the pressure is equal to the saturation value defined by the water temperature at the condenser exit, point 6). The knowledge of the enthalpy at those key points allows solving the energy balance equations as they are shown, for each of the CHP main components, in Table 3 from which the CHP behaviour can be inferred and the usual performance indicators can be obtained.

Table 2. Evaluation of the thermodynamic properties at different CHP key points.

# [a]	p [kPa]	T [°C]	h [kJ/kg]	s [kJ/kg]	State [b]
1_s [c]	$p_{1i} = p_{max}$	-	$h_{1i} = h(p_{1i}, s_{1i})$	$s_{1i} = s_4$	−
1	$p_1 = p_{max}$	$T_1 = T(p_1, h_1)$	$h_1 = h_4 + \left({h_{1i} - h_4}/{\eta_P} \right)$	$s_1 = s(T_1, h_1)$	CL
2	$p_2 = p_{max}$	$T_2 = T_{sat}(p_2) + \Delta T_2$	$h_2 = h(T_2, p_2)$	$s_2 = s(T_2, p_2)$	SH
3_s [c]	$p_{3i} = p_3$	-	$h_{3i} = h(p_{3i}, s_{3i})$	$s_{3i} = s_2$	−
3	$p_3 = p_{sat}(T_6)$	$T_3 = T(p_3, h_3)$	$h_3 = h_2 - \eta_T \times (h_2 - h_{3i})$	$s_3 = s(h_3, p_3)$	SH
4	$p_4 = p_3$	$T_4 = T(p_4, x_4)$	$h_4 = h(T_4, p_4)$	$s_4 = s(T_4, x_4)$	SL
5	$p_5 = p_w$	T_5	$h_5 = h(T_5, p_5)$	−	−
6	$p_6 = p_w$	$T_6 = T_7 - \theta \times (T_7 - T_5)$	$h_6 = h(T_6, p_6)$	−	−
7	$p_7 = p_w$	T_7	$h_7 = h(T_7, p_7)$	−	−
8	$p_8 = p_{atm}$	$T_8 = T_{flame}$	$h_8 = h(T_8, p_8)$	−	−
9	$p_9 = p_{atm}$	$T_9 = T(p_9, h_9)$	$h_9 = h_8 - \dot{Q}_{w_{PH}}/\dot{m}_{cg}$	−	−
10	$p_{10} = p_{atm}$	$T_{10} = T(p_{10}, h_{10})$	$h_{10} = h_9 - \dot{Q}_{in}/\dot{m}_{cg}$	−	−

[a] In accordance with Figure 2. [b] CL – Compressed liquid; SL - Saturated liquid; SH – Superheated vapour. [c] Correspond to an intermediate calculation for the isentropic thermodynamic condition.

Table 3. Power balance equations of the CHP system.

CHP component [a]	Stream	Power balance
Pump	ORC working fluid	$\dot{W}_{in} = \dot{m}_f \times (h_1 - h_4)$
Working fluid heating section (ORC-evaporator)	ORC working fluid	$\dot{Q}_{in} = \dot{m}_f \times (h_2 - h_1)$
	Combustion gases	$\dot{Q}_{cg_f} = \left(\dot{m}_{cg} \times (h_9 - h_{10}) \right)/\eta_{EHE}$
Water post-heating section (ORC-evaporator)	Water	$\dot{Q}_{w_{PH}} = \dot{m}_w \times (h_7 - h_6)$
	Combustion gases	$\dot{Q}_{cg_{PH}} = \left(\dot{m}_{cg} \times (h_8 - h_9) \right)/\eta_{EHE}$
Expander	ORC working fluid	$\dot{W}_{out} = \dot{m}_f \times (h_2 - h_3)$
Condenser	ORC working fluid	$\dot{Q}_{out} = \dot{Q}_{w_{CHE}}/\eta_{CHE}$
	Water	$\dot{Q}_{w_{CHE}} = \dot{m}_w \times (h_6 - h_5)$
Hybrid CHP system	Natural gas/air mixture	$\dot{Q}_{CHP} = \dot{Q}_{cg_f} + \dot{Q}_{cg_{PH}}$
	Water	$\dot{m}_w = \dot{Q}_w/(h_7 - h_5)$
	ORC working fluid	$\dot{m}_f = \dot{Q}_{out}/(h_3 - h_4)$

[a] In accordance with Figure 2.

Among these indicators are the ORC efficiency (η_{ORC}), defined as shown in equation (2), the ORC net power output ($\dot{W}_{net}$), see equation (3), and the Primary Energy Savings (PES), as defined by the EU directive [24], that can be calculated using equation (4). In this, η_{REF_H} and η_{REF_E} are the harmonized efficiency reference value for separate production of heat (for the purpose of this paper was assumed 0,9 considering that the thermal energy produced is in the form of hot water from a natural-gas boiler manufactured before 2016 [25]) and the harmonized efficiency reference value for separate production of electricity (for this paper was assumed 0,445, considering that the electrical energy is produced in a natural-gas fueled power plant built before 2012 and an aggregated correction factor that includes the climatic specificities and the grid losses for low-voltage level end-users [25]). The values of η_{CHP_H} and η_{CHP_E}, that represents the thermal and electrical efficiencies of the CHP systems, respectively, are calculated by equation (5). Besides that, two non-dimensional parameters, the $rPES$ and the $r\dot{W}_{net}$, that relate the values obtained for the hybrid configuration (CHP system with $\theta \neq 0\%$) with those obtained for the standard configuration (CHP system with $\theta = 0\%$), see equation (6), and the reduction of the combustion gases temperature associated with the energy transfer in the water post-heating section of the ORC-evaporator ($T_8 - T_9$), are also retrieved from the model.

$$\eta_{ORC} = \frac{\dot{W}_{out} - \dot{W}_{in}}{\dot{Q}_{in}} \times 100 \tag{2}$$

$$\dot{W}_{net} = \dot{W}_{out} - \dot{W}_{in} \tag{3}$$

$$PES = \left(1 - \left(\frac{1}{\left(\frac{\eta_{CHP_H}}{\eta_{REF_H}} + \frac{\eta_{CHP_E}}{\eta_{REF_E}}\right)}\right)\right) \times 100 \tag{4}$$

$$\eta_{CHP_H} = \frac{\dot{Q}_{w_{PH}} + \dot{Q}_{w_{CHE}}}{\dot{Q}_{CHP}}, \qquad \eta_{CHP_E} = \frac{\dot{W}_{out} - \dot{W}_{in}}{\dot{Q}_{CHP}} \tag{5}$$

$$rPES = \frac{PES\,(\theta)}{PES\,(\theta = 0)}, \qquad r\dot{W}_{net} = \frac{\dot{W}_{net}(\theta)}{\dot{W}_{net}(\theta = 0)} \tag{6}$$

4. Model results

The variation's effect of θ over the net specific work, the efficiency (η_{ORC}) and the organic fluid mass flow rate ($\dot{m}_f$) is shown in Figure 4-a. As expected, mainly due to the increase of the pressure difference in the working fluid when passing through the expander, the net specific work and the efficiency increased with θ. However, since an increasing part of the water heating process is shifted from the ORC condenser to the post-heating section, an inevitable reduction of the working fluid mass flow rate is observed. The non-despicable result of these antagonistic variations is presented in Figure 4-b, where the values of the primary energy savings (PES) and the ORC net power output ($\dot{W}_{net}$) are shown as a function of θ. The maximum positive variations of PES and $\dot{W}_{net}$ are 23% and 7% for values of θ equal to 33% and 21%, respectively. The shifting of part of the water heating process from the ORC condenser to the post-heating section of the ORC-evaporator allows for solving a safety issue that will permit the adoption of a compact ORC-evaporator design and also induces positive effects in the CHP performance increasing not only the efficiency (an increase of PES) and economic (an increase of $\dot{W}_{net}$) figures in comparison with the standard configuration (for which $\theta = 0\%$).
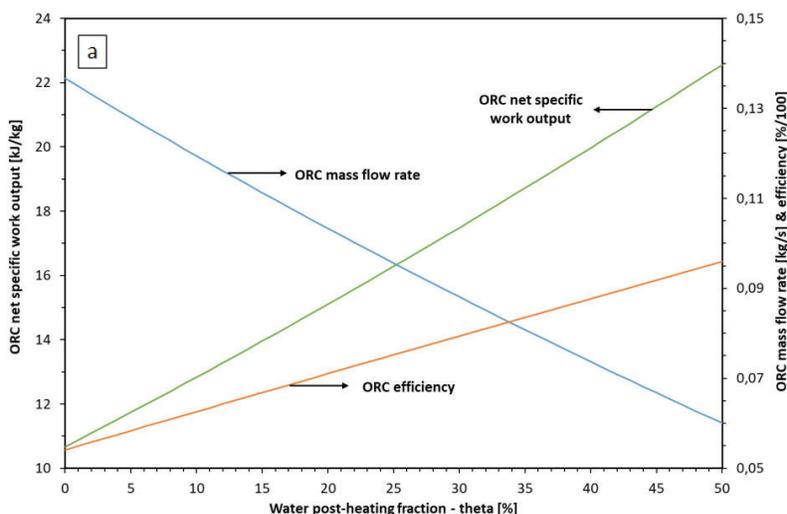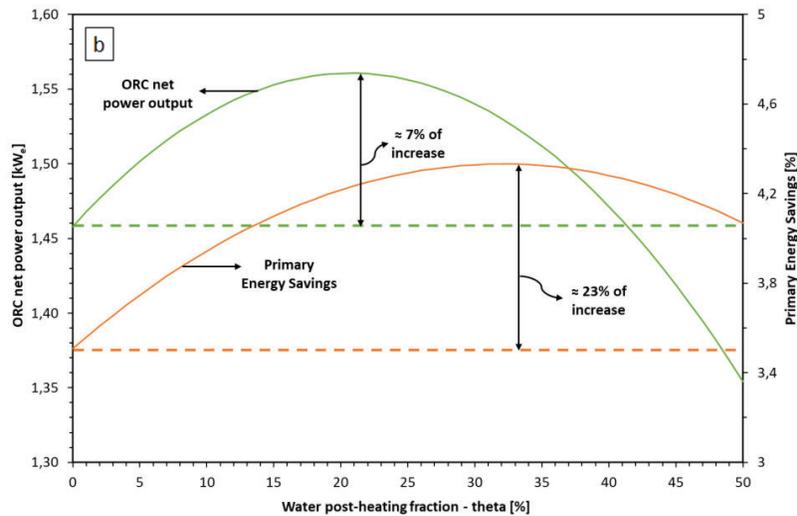
Figure 4: a) ORC net specific work, $\dot{m}_f$ and η_{ORC} as function of θ, b) PES and $\dot{W}_{net}$ as function of θ.

Moreover, the proposed hybrid configuration presents an additional advantage of reducing the risk of organic fluid thermal degradation that arises from a significant temperature reduction of the combustion gases on their passage through the water post-heating section of the ORC-evaporator. The temperature reduction for the value of θ that maximizes PES is about 420 ºC and for the value of θ that maximizes $\dot{W}_{net}$ is about 260 ºC. It is also important to mention that this value can be increased without any efficiency or net power output losses regarding the standard CHP configuration ($\theta = 0\%$) to more than 500 ºC if θ is increased to around 41%, as shown in Figure 5.
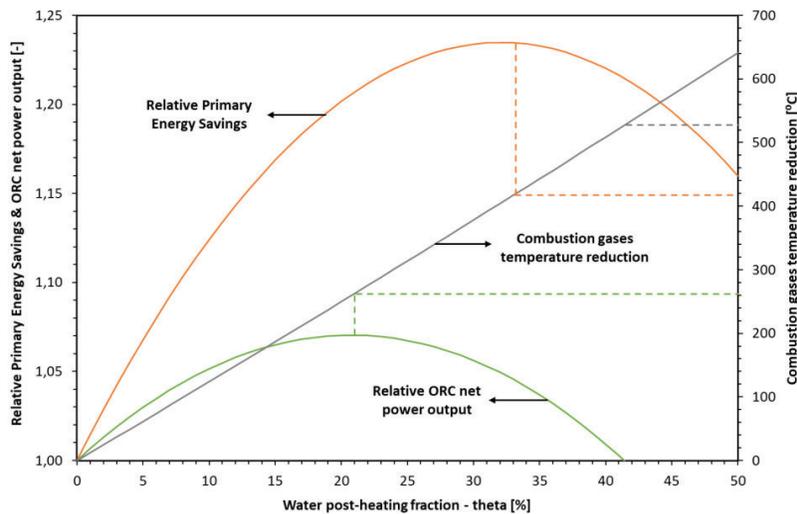

Figure 5: Relative value of PES / $\dot{W}_{net}$ and combustion gases temperature reduction as a function of θ.

5. Conclusion

The development of ORC-based micro-CHP systems aiming to retrofit the current combi-boilers demands small dimensions, high efficiency and high turn-down ratio evaporators. To achieve such requirements, the design principles of those ORC-evaporators should include: i) direct vaporization, ii) counter-flow arrangement and iii) pre-mixed gas burners. A solution for an ORC-evaporator accomplishing these specifications was presented. That solution showed the need for a cold surrounding on the gas-burner head. This last design principle led to the development of an ORC-evaporator where part of the energy contained in the combustion gases is transferred to the water before they reach the ORC working fluid heat exchanger. The integration of such ORC-evaporator in the CHP system gave origin to a hybrid configuration in which the useful thermal energy transferred to the water is done stepwise: firstly, in the ORC-condenser and then in

the water post-heating section of the ORC-evaporator. Such configuration, since it reduces the average temperature at the ORC-condenser, increases the cycle efficiency and the net specific work mainly because it increases the pressure ratio of the ORC cycle. Even taking into consideration the negative effect of the associated working fluid mass flow rate reduction, for a significant part of the CHP typical operating conditions, an increase of the primary energy savings by 23% and of the ORC net power output by 7% for values of θ equal to 33% and 21%, respectively, are shown for one of the most characteristic CHP operating condition (T_5 = 10 ºC, T_7 = 65 ºC). Furthermore, and for the same operating condition, the temperature reduction of the combustion gases in the water post-heating section of the ORC-evaporator can be as high as 520 ºC without no losses in the primary energy savings or the ORC net power output. This temperature reduction can help prevent, in a significant way, the occurrence of one of the major concerns associated with the direct vaporization of the ORC working fluid: the risk of thermal degradation. Therefore, the fraction of the CHP water heating process performed in the post-heater becomes a new design parameter that needs to be considered not only for the ORC-evaporator but for all of the CHP components which must be accordingly selected or designed.

The integration of part of the CHP water heating process in the ORC-evaporator led to a hybrid (topping/bottoming) configuration that not only solves a safety issue related to the gas-burner head overheating but also brings several (non-despicable) performance benefits and important reliability improvements for systems attempting to retrofit the current combi-boilers. Besides, this hybrid configuration applies to all the situations where the difference between the temperature of hot and cold sources is smaller as when water from non-pressurized biomass (or other fuels) boilers is used to power those CHP systems.

Nomenclature

h	Enthalpy, kJ/kg	EHE	ORC-Evaporator
θ	Tetha, %	f	ORC working fluid
η	Efficiency, %	flame	Combustion flame conditions
$\dot{m}$	Mass flow rate, kg/s	fuel	Related to the fuel used
$\dot{Q}$	Thermal power, kW	H	Thermal reference
$\dot{W}$	ORC power output, kW_e	s	Isentropic
m'	Mass fraction, kg_i/kg_{fuel}	in	Inlet conditions
PES	Primary Energy Savings, %	max	Maximum value
T	Temperature, ℃	Net	Net value
p	Pressure, kPa	ORC	Organic Rankine cycle system
$r\dot{W}_{net}$	Relative ORC net power output -	out	Outlet conditions
$rPES$	Relative Primary Energy Savings, -	P	ORC-Pump
s	Entropy, $kJ/(kg.℃)$	PH	Post heater
x	Quality, -	REF	Reference value
$\Delta T2$	Superheating temperature, ℃	T	ORC-Expander
		w	End-user water circuit
Subscripts and superscripts			
{1 ... 10}	CHP locations	**Acronym**	
atm	Atmospheric conditions	CHP	Combined heat and power
cg	Combustion gases	EU	European Union
CHE	ORC-Condenser	GHG	Greenhouse gases
CHP	Combined heat and power system	HVAC	Heating, ventilation and air conditioning
E	Electrical reference	ORC	Organic Rankine Cycle

References

[1] M. Bianchi, L. Branchini, A. De Pascale, A. Peretto, Application of environmental performance assessment of CHP systems with local and global approaches, Appl. Energy. 130 (2014) 774–782. https://doi.org/10.1016/j.apenergy.2014.04.017.

[2]   Y. kuang Chen, I.G. Jensen, J.G. Kirkerud, T.F. Bolkesjø, Impact of fossil-free decentralized heating on northern European renewable energy deployment and the power system, Energy. 219 (2021) 119576. https://doi.org/10.1016/j.energy.2020.119576.

[3]   International Energy Agency, Global CHP/DHC Data - Current Baseline, (n.d.). https://www.iea.org/chp/data/globalchpdhcdata-currentbaseline/ (accessed October 26, 2018).

[4]   CODE 2 - Cogeneration Observatory and Dissemination Europe, Micro-CHP potential analysis - European level report, (2014). http://www.code2-project.eu/wp-content/uploads/D2.5-2014-12-micro-CHP-potential-analysis_final1.pdf.

[5]   S. Pezzutto, S. Croce, S. Zambotti, L. Kranzl, A. Novelli, P. Zambelli, Assessment of the space heating and domestic hot water market in Europe—open data and results, Energies. 12 (2019). https://doi.org/10.3390/en12091760.

[6]   B. Peris, J. Navarro-Esbrí, F. Molés, A. Mota-Babiloni, Experimental study of an ORC (organic Rankine cycle) for low grade waste heat recovery in a ceramic industry, Energy. 85 (2015) 534–542. https://doi.org/10.1016/j.energy.2015.03.065.

[7]   E. Galloni, G. Fontana, S. Staccone, Design and experimental analysis of a mini ORC (organic Rankine cycle) power plant based on R245fa working fluid, Energy. 90 (2015) 768–775. https://doi.org/10.1016/j.energy.2015.07.104.

[8]   M. Cambi, R. Tascioni, L. Cioccolanti, E. Bocci, Converting a commercial scroll compressor into an expander: Experimental and analytical performance evaluation, Energy Procedia. 129 (2017) 363–370. https://doi.org/10.1016/j.egypro.2017.09.234.

[9]   S. Declaye, S. Quoilin, L. Guillaume, V. Lemort, Experimental study on an open-drive scroll expander integrated into an ORC (Organic Rankine Cycle) system with R245fa as working fluid, Energy. 55 (2013) 173–183. https://doi.org/10.1016/j.energy.2013.04.003.

[10]   S. Quoilin, V. Lemort, J. Lebrun, Experimental study and modeling of an Organic Rankine Cycle using scroll expander, Appl. Energy. 87 (2010) 1260–1268. https://doi.org/10.1016/j.apenergy.2009.06.026.

[11]   A. Algieri, P. Morrone, Energetic analysis of biomass-fired ORC systems for micro-scale combined heat and power (CHP) generation. A possible application to the Italian residential sector, Appl. Therm. Eng. 71 (2014) 751–759. https://doi.org/10.1016/j.applthermaleng.2013.11.024.

[12]   R. Bracco, D. Micheli, R. Petrella, M. Reini, R. Taccani, G. Toniato, Micro-Organic Rankine Cycle systems for domestic cogeneration, Elsevier Ltd, 2016. https://doi.org/10.1016/B978-0-08-100510-1.00018-1.

[13]   J.S. Pereira, J.B. Ribeiro, R. Mendes, G.C. Vaz, J.C. André, Development of a direct concept helical-coil evaporator for an ORC based micro-CHP system, Energy Procedia. 129 (2017) 474–478. https://doi.org/10.1016/j.egypro.2017.09.162.

[14]   J.S. Pereira, J.B. Ribeiro, R. Mendes, G.C. Vaz, J.C. André, ORC based micro-cogeneration systems for residential application – A state of the art review and current challenges, Renew. Sustain. Energy Rev. 92 (2018) 728–743. https://doi.org/10.1016/j.rser.2018.04.039.

[15]   M. Farrokhi, S.H. Noie, A.A. Akbarzadeh, Preliminary experimental investigation of a natural gas-fired ORC-based micro-CHP system for residential buildings, Appl. Therm. Eng. 69 (2014) 221–229. https://doi.org/10.1016/j.applthermaleng.2013.11.060.

[16]   X. Dai, L. Shi, Q. An, W. Qian, Thermal stability of some hydrofluorocarbons as supercritical ORCs working fluids, Appl. Therm. Eng. 128 (2018) 1095–1101. https://doi.org/10.1016/j.applthermaleng.2017.09.046.

[17]   C.M. Invernizzi, D. Bonalumi, Thermal stability of organic fluids for Organic Rankine Cycle systems, Elsevier Ltd, 2016. https://doi.org/10.1016/B978-0-08-100510-1.00005-3.

[18]   M.O. Bamgbopa, E. Uzgoren, Quasi-dynamic model for an organic Rankine cycle, Energy Convers. Manag. 72 (2013) 117–124. https://doi.org/10.1016/j.enconman.2013.01.040.

[19]   J.S. Pereira, J. Almeida, J.C. André, R. Mendes, J.B. Ribeiro, Modelling and experimental validation of the heat-transfer processes of a direct vaporization micro-scale ORC-evaporator for thermal degradation risk assessment, Energy Convers. Manag. 238 (2021) 114130. https://doi.org/10.1016/j.enconman.2021.114130.

[20]   J.S. Pereira, M. Santos, R. Mendes, J.C. André, J.B. Ribeiro, Thermal degradation assessment study of a direct vaporization ORC based micro-CHP system under close-to-real operating conditions, Appl. Therm. Eng. 214 (2022). https://doi.org/10.1016/j.applthermaleng.2022.118878.

[21]   J.S. Pereira, J.B. Ribeiro, R. Mendes, J.C. André, Analysis of a hybrid (topping / bottoming) ORC based CHP configuration integrating a new evaporator design concept for residential applications, Appl. Therm. Eng. 160 (2019) 113984. https://doi.org/10.1016/j.applthermaleng.2019.113984.

[22] M.O. Lemmon, E.W., Huber, M.L., McLinden, NIST Reference Fluid Thermodynamic and Transport Properties (REFPROP) - Version 9.1, Natl. Inst. Stand. Technol. (2013).

[23] European Committee for Standardization, EN 13203: Gas-fired domestic appliances producing hot water, 2016.

[24] European Parliament, European Council, Directive 2012/27/EU, Off. J. Eur. Union. L315/1 (2012) 1–56. https://doi.org/10.3000/19770677.L_2012.315.eng.

[25] European Commision, COMMISSION DELEGATED REGULATION (EU) 2015/2402 of 12 October 2015 reviewing harmonised efficiency reference values for separate production of electricity and heat in application of Directive 2012/27/EU of the European Parliament and of the Council and rep, Off. J. Eur. Union. L 333/54 (2015) 54–61.

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