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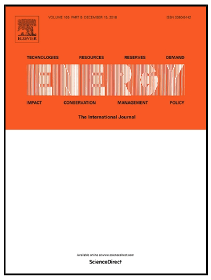

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Improving the Power Share of Waste-Driven CHP Plants via Parallelization with a Small-Scale Rankine Cycle, a Thermodynamic Analysis

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Abstract

Waste-fired CHP plants are largely contributing to the base load supply of heat and electricity sectors of several European countries. In such systems, about two-third of the production is heat and about one-third is electricity. In this study, the utilization of a certain portion of the heat production of a waste-fired CHP plant as the heat source of an Organic Rankine Cycle (ORC) is proposed. The main objective is to maximize the share of electricity production of waste-CHP plants instead of a higher heat production rate in a cost-effective way. The inspiration for this idea is that not only electricity is extremely more valued in most of the European energy markets, but also their heat sectors are to be occupied with efficient electricity-driven technologies, e.g. heat pumps. This work presents a detailed thermodynamic analysis of the proposed combined system and the best share of heat supply for driving the ORC unit is investigated. The results show that an electricity efficiency improvement of up to 25% is achievable via this integration while the heat production capacity of the power plant still remains significant. It is shown that the integration picks up of exergy efficiency of the power cycle.

Keywords: Waste incineration; Waste-driven CHP; Organic Rankine cycle; District heating; Power grid; Thermodynamic modelling.

1. Introduction

The trend of the global energy matrix change is toward a smart energy system [1]. A smart energy system has a number of features such as a high share of renewable energy systems, concrete integration of energy systems, active interactions of different energy sectors, the utilization of the most advanced clean energy technologies, the lowest rates of losses and the highest possible rate of the utilization of waste or freely available energy flows, etc. [2]. The transition process from the existing energy system to the future smart energy system is, however, somewhat challenging and requires special attention of the experts of this field for addressing the gaps.

Among all the energy sectors (including electricity, heat, cold, and gas), the electricity sector will be of an especial attention in the future energy systems. This is mainly because: a) there has been a very strong advancement in the state-of-the-art and –practice of renewable based-electricity production technologies [3]; b) the transportation sector goes fast toward the very high penetration of electrical vehicles [4]; c) electrical-driven cold and heat production machines, e.g. heat pumps, are highly efficient and are being more popular for the integration into the district heating and cooling networks in large-scale applications [5]. Naturally, solar and wind power systems are the most important renewable sources in the global energy systems. These two sources, however, suffer from the irregular profiles of availability whereas the accurate long-term forecast of their fluctuations is yet impossible [6]. Therefore, for being able to have a reliable energy system, other clean energy sources with controllable availability and supply should come along with solar- and wind-based power systems [7]. Biomass, biogas and waste are of the sources that can be highly beneficial for the stabilization of future 100% renewable-based energy systems [8].

For biomass and waste sources, there is an argument if they can be considered as sustainable sources of energy because the combustion production of such fuels contains greenhouse gases. This, especially, is more of a challenge for waste sources. However, the clear response is that such sources can of course be considered as sustainable energy resources. The justification for this is that the emissions resulting from the combustion of biomass and biogas resources is way lower than the amount of carbon dioxide they absorb in the environment while growing and the emission from the waste incineration process is far less than that released if the alternative method of waste disposal (landfilling) is used [9]. Today, waste incineration is a popular technology for municipal waste disposal where the energy released out of the incineration process may be used for heat or power production. Such systems are especially of much interest in a few of the Northern European countries, such as Denmark, Germany, etc. [10]. Munster and Meibom [11] presented a study aiming at the optimization of the use of waste in the future energy system. Erikson et al. [12] evaluated the performance of heat-only and heat-power production waste incineration technologies in Danish district heating system. Tobiasen and Kamuk [13] highlighted how a waste-fired CHP plant can substantially increase the overall energy efficiency of district heating systems and optimized the performance of such a system for increasing the electrical output of the plant. Tomic et al. [14] assessed the operation of waste-CHP plants under the new coming legislations and regulations in the energy markets within the European Union. Waste-fired CHP plants are one of the key elements of the Danish energy system supplying a portion of the base load of heat and electricity grids [15]. In such systems, though in various designs, the fuel-to-heat efficiency is far higher than the fuel-to-power efficiency [16]. However, based on the facts presented, regarding the importance of the electricity grids in the future energy systems and the fact that electricity is more valued than heat in the Danish energy market, the interest is to increase the fuel-to-electricity efficiency of these units as much as possible in an economically rational way [17].

An ORC is a Rankine-cycle based power plant that employs an organic working fluid, e.g. ammonia etc. [18]. The main advantage of an ORC is that a lower temperature (than a regular Rankine cycle) is required for producing an evaporated working fluid. Therefore, low temperature heat sources can come into service for producing electricity [19]. One of the main applications of ORC technology is the utilization of any waste or surplus heat flow to produce free electricity. A thorough review of this application of the ORC may be found in [20]. There is a quite extensive literature about the ORC technology, state-of-the-art, various working fluids, and applications [19–24]. Thus, no further information about this technology and literature is presented here.

This work presents a feasibility study of integrating a small-scale ORC with a waste-fired CHP plant. For this, first, the study presents a thorough thermodynamic assessment of the performance of the conventional configuration of the power plant in both heat and electricity markets. Then, the proposed smart integration method is introduced and finally, a comprehensive thermodynamic analysis of the combined waste-fired CHP-ORC technology is accomplished. The main novelty of this work is on the objective of this hybridization, which is to maximize the electricity generation of the plant rather than simply increasing its net energy output and based on which the combined plant is designed and sized, and the selection of the low-temperature heat source of the power plant, which is going to be the local underground water sources (this will discussed in the next section).

2. Waste-Fired CHP Plant and the Proposed Combined System

This section provides background information about the conventional configuration of waste-fired CHP plants is presented. Then, the schematic of a regular Rankine cycle is discussed and finally, the proposed combined waste-fired CHP-ORC system is introduced and discussed.

2.1. Waste-fired CHP plant

In a waste-fired CHP plant, specifically those of focus of this work, the system comprises a waste-fired boiler and a Rankine cycle power block. In the Rankine cycle, the condenser is practically a phase change heat exchanger attached to the local district heating system to supply the released heat from the steam while being condensed [27]. Fig. 1 shows the schematic of a typical waste-fired grate boiler (i.e. the waste incinerator) with the name of its different components marked on that. For such an incinerator, the energy conversion efficiency might be in the range of 70-82% [28]. There is a comprehensive information about this type of boiler which is used for both waste and biomass sources in [29].

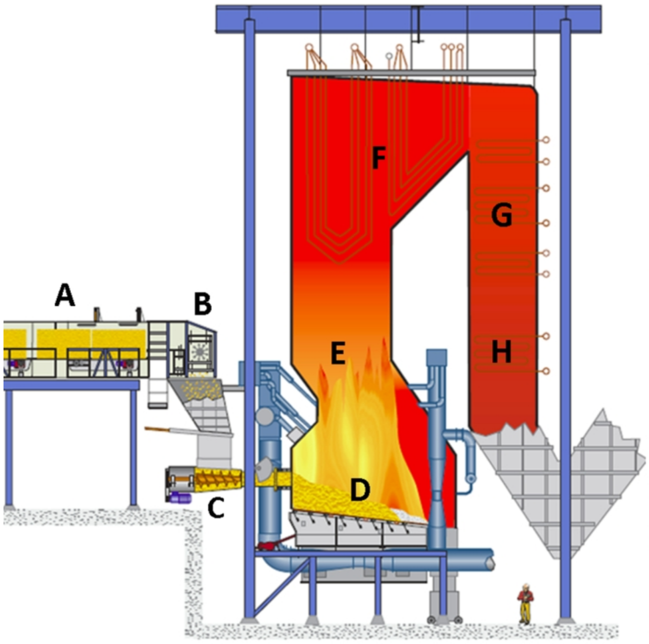

Fig. 1: Waste-fired grate boiler configuration; A: fuel transport belt, B: the rotatory rake, C: stoker screw, D: vibrating grate, E: freeboard, F: second and third super heater, G: first superheater, H: economizer [29].

Table 1 presents information about the features of the waste source and the incinerator considered in this project. It is noteworthy that the heating value of the waste source may vary depending on the compositions of the waste, on its moisture and ash contents. Here, based on the given composition for the waste source adopted from [30], the lower heating value is estimated as 12,500 kJ/kg.

Table 1 – The waste incineration unit main features.

Item	Information/value
Type of waste	Municipal solid waste
Waste compositions (weight percent)	5.91% Ash
	47.18% Carbon
	6.25% Hydrogen
	39.57% Oxygen
	0.91% Nitrogen
	0.18% Sulphur
Lower heating value of the waste (kJ/kg)	12,500
Excess air in the incineration process	80% [28]
Combustion product temperature (°C)	1100
Stack flue gas temperature (°C)	165

Fig. 2 illustrates a simplified schematic of a typical Rankine cycle based CHP plant. According to the figure, the energy released from the incineration of the waste source is absorbed by the Rankine-cycle working fluid (water/steam) to make a pressurized superheat steam. This steam is expanded through a three-stage turbine to derive an electricity generator. The dead steam after the low-pressure turbine passes through the condenser to be condensed again. Indeed, although the pressure and temperature of the steam flow are to be almost constant across the condenser, there is much heat that must be released from the steam flow to let it be condensed (the latent heat of the steam). This heat flow is collected to supply the local district heating system. In such a system, depending on the design of the system, the fuel-to-heat and fuel-to-power efficiencies vary significantly. For the presented configuration, the former is 50% and the latter is 35% [31].

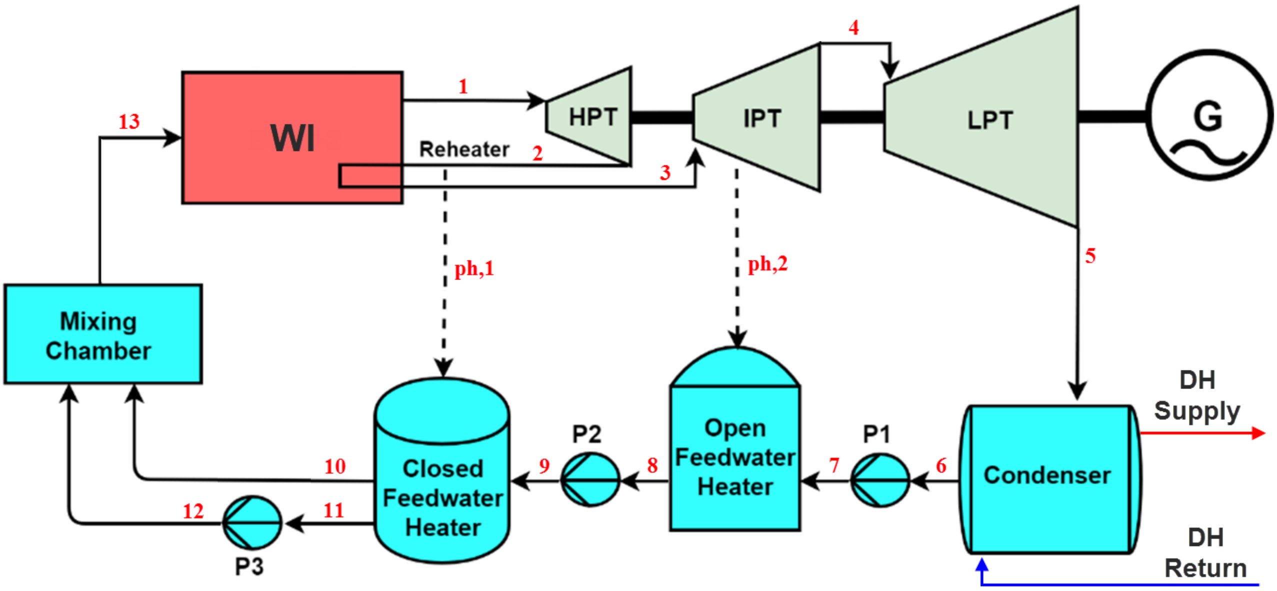

Fig. 2 Simplified schematic of the waste-fired CHP plant connected to district heating; WI: waste incinerator, HPT: high-pressure turbine, IPT: medium-pressure turbine, LPT: low-pressure turbine, P: pump, G: electricity generator, ph: preheating line, DH: district heating.

Table 2 gives detailed information about the thermodynamic properties of the CHP working fluid at various points.

Table 2 – The Rankine cycle technical information.

Point	Temperature / Pressure (℃ / MPa)
1	550 / 10
ph,1	350 / 3
2	350 / 3
3	500 / 3
ph,2	350 / 1
4	300 / 0.25
5	90 / 0.07 (vapor)
6	90 / 0.07 (condensed)
7	90 / 0.25
8	172 / 0.25
9	177 / 10

10	279 / 10
11	279 / 3
12	281 / 10
13	292 / 10
DH Supply	80 / 1.2
DH Return	40 / 1.2

The isentropic efficiency of the turbines and the pumps are all equal to 85% while the electricity generator net efficiency is assumed as 95% [32].

2.2. The proposed combined waste-fired CHP-ORC

As discussed, inspired by the facts presented about the importance of increasing the electricity share in the European energy matrix, specifically Denmark, this work proposes the utilization of a portion (up to all) of the heat output of a waste-driven CHP plant as the heat source of an ORC unit. Fig. 3 shows the configuration of the proposed combined waste-fired CHP-ORC system. As seen, the heat withdrawn from the working fluid of the CHP plant through the condenser is utilized for the cogeneration of heat and electricity by employing a small-scale ORC unit. The main objective for this is to increase the share of the electricity production of the CHP plant rather than a higher heat output in an economically feasible way. Therefore, the ratio of heat and electricity production prices is a key parameter for this design.

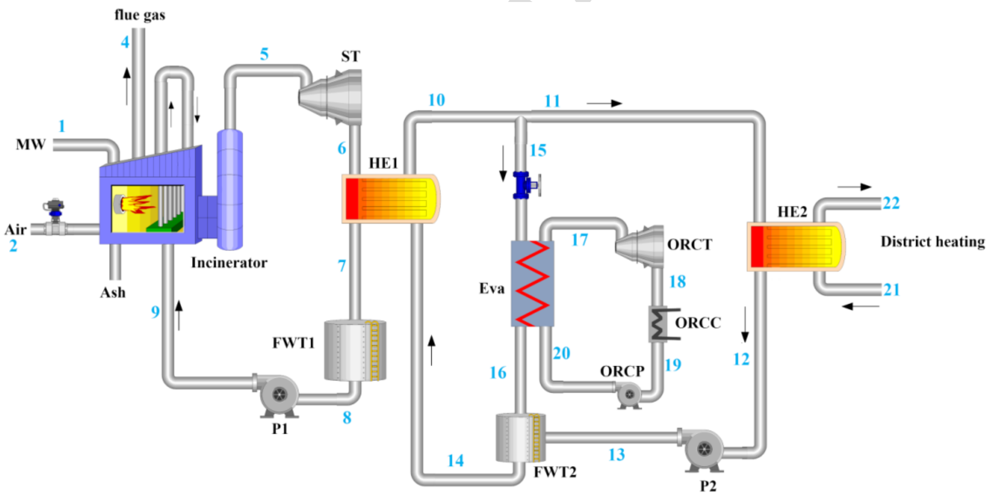

Fig. 3 Configuration of the waste-fired CHP plant accompanied with a small-scale ORC unit; ST: steam turbine, HE1: heat exchanger 1 (condenser), P1: pump 1, FWT1: feed water tank 1, Eva: evaporator, ORCT: ORC unit turbine, ORCC: ORC unit condenser, ORCP: ORC unit pump, FWT2: feed water tank 2, P2: pump 2, HE2: heat exchanger 2 (district heating supply).

Naturally, for a heat engine, the bigger the temperature difference of the hot and cold heat sources, the higher rate of work may be producible. Therefore, the cold source of energy for the ORC unit is a key factor in this work. As a smart choice, the cold source of the ORC unit in this work is proposed to be the local groundwater reservoir, from which the water demand of people in Denmark is supplied, from a depth of about 25 m underground. The temperature of this source is in the range of 5-7 °C [33].

Table 3 presents information about the technical features of the ORC unit based on the heat flow availability from the conventional configuration of the CHP plant [34].

Table 3 – The ORC unit main features [34].

Parameter	Information/value
Working fluid	R123
Turbine inlet pressure (MPa)	0.2-0.75
Turbine inlet condition (^oC)	Saturated vapor
Condenser temperature (^oC)	10
Turbine isentropic efficiency (%)	0.9
Pump isentropic efficiency (%)	0.85
Evaporator pinch temperature difference (^oC)	5
Condenser pinch temperature difference (^oC)	3
Condenser secondary fluid inlet/outlet pressure (MPa)	0.12/0.12
Condenser secondary fluid inlet temperature (^oC)	5-7

3. Thermodynamic Model

In this section, a detailed thermodynamic model, including both energy and exergy models, of the proposed combined system is presented.

3.1. Energy analysis

For fulfilling the modelling of the system, each of the components is considered as a control volume, and the energy conservation equation is developed for each. The overall energy conservation equation, while neglecting the change of kinetic and potential terms, is as follow [35,36]:

$$\dot{Q} + \sum \dot{m}_i h_i + m_I u_I = \dot{W} + \sum \dot{m}_e h_e + m_{II} u_{II} \tag{1}$$

where, $\dot{Q}, \dot{W}, \dot{m}$, h, m and u are the rate of heat transfer, the rate of mechanical work, the mass flow rate, the specific enthalpy, the mass of the control volume, and the internal energy of the control volume, respectively. The subscripts i and e denote the inlet and outlet flows into/from the control volume and the subscripts I and II refer to the conditions for the control volume before and after any process. The variation of the internal energy of the control volume over time will be zero if the process takes place under steady state conditions.

As such, the mass conservation law for a given control volume could be written as:

$$\sum \dot{m}_i + m_1 = \sum \dot{m}_e + m_2 \tag{2}$$

Based on the above two principal correlations and taking into account the rules governing on various components of the combined system (see [37]), one could develop the formulations applied to each of these components. Table 4 lists the applied equations derived for the main components of the configuration presented in Fig. 3.

Table 4 – The governing energy equations on the components of the combined CHP-ORC plant.

Component	Equation	Number
Incinerator	$\dot{m}_1 LHV_{waste} + \dot{m}_2 h_2 + \dot{m}_9 h_9 = \dot{m}_4 h_4 + \dot{m}_5 h_5$	(3)
ST	$\dot{W}_{ST} = \dot{m}_5 (h_5 - h_6)$, $\eta_{is,ST} = \frac{\dot{W}_{ST}}{\dot{W}_{is,ST}}$	(4)
HE1	$\dot{m}_6 (h_6 - h_7) = \dot{m}_{14} (h_{10} - h_{14})$	(5)
P1	$\dot{W}_{P1} = \dot{m}_8 (h_9 - h_8)$, $\eta_{is,P1} = \frac{\dot{W}_{is,P1}}{\dot{W}_{P1}}$	(6)
FWT2	$\dot{m}_{13} h_{13} + \dot{m}_{16} h_{16} = \dot{m}_{14} h_{14}$	(7)
Eva	$\dot{m}_{15} (h_{15} - h_{16}) = \dot{m}_{20} (h_{17} - h_{20})$	(8)
P2	$\dot{W}_{P2} = \dot{m}_{12} (h_{13} - h_{12})$, $\eta_{is,P2} = \frac{\dot{W}_{is,P2}}{\dot{W}_{P2}}$	(9)
ORCT	$\dot{W}_{ORCT} = \dot{m}_{17} (h_{17} - h_{18})$, $\eta_{is,ORCT} = \frac{\dot{W}_{ORCT}}{\dot{W}_{is,ORCT}}$	(10)
ORCP	$\dot{W}_{ORCP} = \dot{m}_{19} (h_{20} - h_{19})$, $\eta_{is,ORCP} = \frac{\dot{W}_{is,ORCP}}{\dot{W}_{ORCP}}$	(11)
HE2	$\dot{m}_{11} (h_{11} - h_{12}) = \dot{m}_{21} (h_{22} - h_{21})$	(12)

In this table, η_{is} is the isentropic efficiency, $\dot{W}_{is}$ is the rate of work in an isentropic process, and LHV is the lower heating value of the waste source. The numeric subscripts refer to the points marked on Fig. 3.

Having the above formulations, one could analyze the energy performance of the entire cycle. For the given waste-fired CHP-ORC plant, the fuel-to-heat efficiency (η_{fth}), the fuel-to-power efficiency (η_{ftp}) and the overall energy efficiency (η_t) are respectively defined as below:

$$\eta_{fth} = \frac{\overbrace{\dot{m}_{21}(h_{22} - h_{21})}^{Heat\ Output}}{\dot{m}_{waste} LHV_{waste}} \tag{13}$$

$$\eta_{ftp} = \frac{\overbrace{\dot{W}_{ST}\eta_G + \dot{W}_{ORCT}\eta_{ORCG} - \sum \dot{W}_P/\eta_P}^{Total\ Electricity\ Output}}{\dot{m}_{waste} LHV_{waste}} \tag{14}$$

$$\eta_t = \frac{\overbrace{\dot{W}_{ST}\eta_G + \dot{W}_{ORCT}\eta_{ORCG} - \sum \dot{W}_P/\eta_P + \dot{m}_{21}(h_{22} - h_{21})}^{Total\ Energy\ Output}}{\dot{m}_{waste} LHV_{waste}} \tag{15}$$

In these correlations, η_G, η_{ORCG} and η_P are the energy conversion efficiency of the electricity generators in the steam cycle and the ORC cycle, and the efficiency of the pumps, respectively.

3.2. Exergy analysis

Exergy can be defined as the maximum useful work that could be achieved from the system at a given state in an identified environment when there is an interaction between the system and the environment only [38]. Again, ignoring the kinetic and potential exergies, the exergy balance equation under steady state can be expressed as [39,40]:

$$\sum_{in} \dot{E}_i = \sum_{out} \dot{E}_j + \dot{E}_D \tag{16}$$

where, $\sum_{in} \dot{E}_i$ and $\sum_{out} \dot{E}_j$ are, respectively, the rates of inlet and outlet exergy flows and $\dot{E}_D$ refers to the rate of exergy destruction through each of the components. Part of the destroyed exergy is because of internal irreversibilities of the component, while some can be the useless exergy discharged to the environment, e.g. hot exhaust gas [34].

The specific exergy of each stream is divided into the two main categories of specific physical exergy (e^{ph}) and specific chemical exergy (e^{ch}):

$$e = e^{ph} + e^{ch} \tag{17}$$

The specific physical exergy of a system at an identified state is a function of the environmental conditions (so-called the dead state) and the properties of the system. In other words, physical exergy is a system–environment dependent property, and is expressed as follow:

$$e_i = h_i - h_0 - T_0(s_i - s_0) \tag{18}$$

In which, *o* represents the dead state (or ambient) condition.

The specific chemical exergy for a mixture of ideal gases can be written as:

$$e^{ch}_{mixture} = \sum_i n_i e^{ch}_{0,i} + \overline{R}T_0 \sum n_i \ln x_i \tag{19}$$

where, x_i and $e^{ch}_{0,i}$ symbolize the molar fraction and the standard chemical exergy of the i^{th} component of the mixture.

Finally, defining the total exergy rate of each stream ($\dot{E}$) as the specific exergy multiplied by the mass flow rate of the stream, and considering the general rules on the exergy formulation of each of the components and the energy exchange mechanisms, i.e. heat and work, [41], one simply can derive the exergy balance equations for the components of the CP-ORC system. Determining the rate of supplied exergy ($\dot{E}_s$) and product exergy ($\dot{E}_p$) for each component may give a clear understanding of the exergy destruction ($\dot{E}_D = \dot{E}_s - \dot{E}_p$) and exergetic efficiency ($\varepsilon = \dot{E}_p/\dot{E}_s$) of the given components [41]. Table 5 presents a brief exergy modelling of the combined cycle components.

Table 5 - Exergy balance equations for each of the components

Components	$\dot{E}_s$	$\dot{E}_p$	Number
Incinerator	$\dot{E}_1 + \dot{E}_2 + \dot{E}_9$	$\dot{E}_4 + \dot{E}_5$	(20)
ST	$\dot{E}_5 - \dot{E}_6$	$\dot{W}_{ST}$	(21)
HE1	$\dot{E}_6 - \dot{E}_7$	$\dot{E}_{10} - \dot{E}_{14}$	(22)
P1	$\dot{W}_{P1}$	$\dot{E}_9 - \dot{E}_8$	(23)
FWT2	$\dot{E}_{13} + \dot{E}_{16}$	$\dot{E}_{14}$	(24)
Eva	$\dot{E}_{15} - \dot{E}_{16}$	$\dot{E}_{17} - \dot{E}_{20}$	(25)
P2	$\dot{W}_{P2}$	$\dot{E}_{13} - \dot{E}_{12}$	(26)
ORCT	$\dot{E}_{17} - \dot{E}_{18}$	$\dot{W}_{ORCT}$	(27)
ORCP	$\dot{W}_{ORCP}$	$\dot{E}_{20} - \dot{E}_{19}$	(28)
HE2	$\dot{E}_{11} - \dot{E}_{12}$	$\dot{E}_{22} - \dot{E}_{21}$	(29)

In the end, the overall exergetic (or second law) efficiency of the proposed cogeneration system (ε_t) can be defined as the ratio of the net produced exergy (in the form of electricity and heat) to the supplied fuel exergy into the incinerator. Thus, one has:

$$\varepsilon_t = \frac{\overbrace{\dot{W}_{ST}\eta_G + \dot{W}_{ORCT}\eta_{ORCG} - \sum\dot{W}_P/\eta_P + \left(\dot{E}_{22} - \dot{E}_{21}\right)}^{Total\ Exergy\ Output}}{\dot{E}_{waste}} \tag{30}$$

4. Results and Discussion

In this section, the results of the simulations carried out on the proposed combined system are presented and discussed.

In the beginning, Table 6 reports the values of the technical specifications of the waste-fired CHP plant when working at full-load in the conventional configuration, resulting from the energy analysis carried out. Note that, since the power plant maximum capacity does not affect the technical performance indices of the cycle, just as a sample case, the capacity of the power plant is considered equal to the power output of the CHP plant when incinerating 1 kg/s of the waste source (resulting in 2.9 MWe). Besides, the effects of the off-design operation of the power plant have not been taken into account. Although this makes an uncertainty of the simulations compared to the real-life operation of power plants, it still might be reasonable for waste-fired CHP plants. This is because firstly, such power plants are mainly employed for base-load coverage and as a result they usually work at full load [42], and secondly, the performance degradation a power plant is not a linear function of the operation load and the effects are much less when working on a load above 70-75% of the nominal capacity [43].

Table 6 – Technical parameters values in the conventional system (T_{10} = 90 °C,).

Parameter (Unit)	Value
Mass flow rate of fuel (kg/s)	1
Net electricity generated (MW)	2.9
Heat released into the boiler (MW)	12.5
The heat successfully transferred within the boiler (MW)	9.25
Heat wasted from the exhaust (MW)	3.25
Mass flow rate of steam (kg/s)	3.08
Heat released from the condenser (or supplied to district heating, MW)	6.4
Work consumed by the pumps (MW)	0.05
Mass flow rate of pressurized hot water through HE2 line (kg/s)	35.9
Energy efficiency (%)	74.33
Electrical efficiency (%)	23.19

Table 7 gives the results of the exergy performance analysis on the conventional power plant. The information includes the rate of inlet and outlet exergies as well as the rate of exergy destruction of the main components when the CHP plant is working at full-load.

Table 7 – Results of the exergy analysis accomplished on the conventional system at full-load (T_{10} = 90 °C).

Parameter (Unit)	Values
Boiler inlet / outlet / destructed exergy (MW)	14.4 / 4.6 / 9.8
Steam turbine inlet / outlet / destructed exergy (MW)	3.15 / 2.9 / 0.25
Condenser inlet / outlet / destructed exergy (MW)	1.47 / 1.09 / 0.38
HE1 inlet / outlet / destructed exergy (MW)	1.09 / 0.95 / 0.14
Exergy efficiency (%)	26.71

Fig. 4 presents relative heat transfer in the evaporator for both hot and cold streams. The main aim is to describe the heating and phase change processes of the organic working fluid via pressurized hot water. According to the figure, the pinch temperature occurs at the end of economizer; however, this matter was checked for all hot water temperatures to make sure that temperature cross between the hot and cold streams has not taken place.

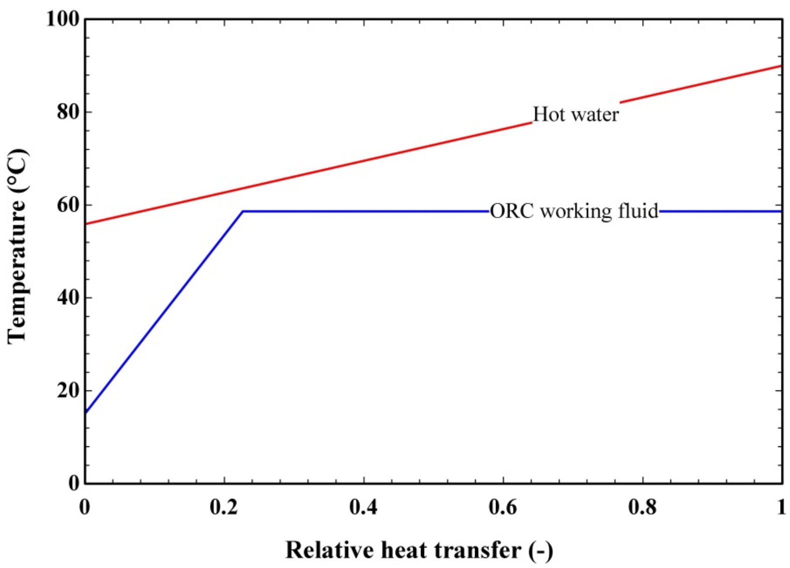

Fig. 4 Temperature versus relative heat transfer in the evaporator with a source temperature of 90 °C.

A primary parameter to study in the combined system is the effect of various outlet temperature/pressure of the low-pressure turbine of the main cycle on the net power and heat producible. In the conventional system, the temperature output is set at 90 °C because the temperature should be enough for supplying hot water at 80 °C for district heating application. Besides, this can be highly effective on the performance of the ORC system because the higher temperature of the evaporator, the better efficiency would be obtained in this system. Fig. 5 shows the effect of increasing the outlet temperature on the rate of heat and power production in the conventional cycle. As can be seen, a change in the steam turbine outlet temperature from 363.15 K (90 °C) to 383.15 K (110 °C) reduces the produced power for about 200 kW, while this leads to an increase in the rate of delivered heat to the district heating system for almost the same amount.

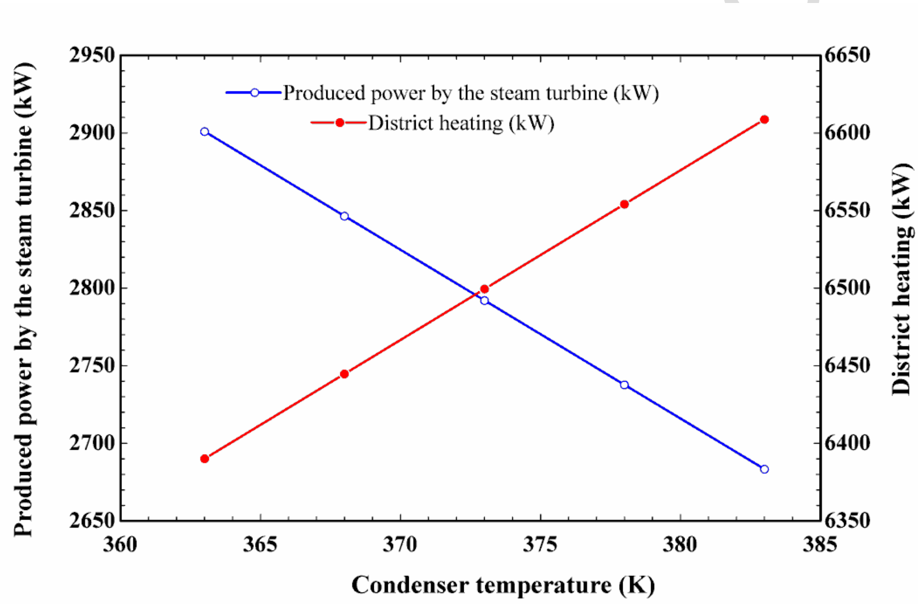

Fig. 5 The effect of the condenser temperature on the net heat and power output of the conventional CHP plant.

Fig. 6 investigates how much electricity might be generated by the ORC unit for different temperature outlets of the LPT in the main CHP plant incinerating 1 kg/s municipal waste. Naturally, this item is a direct function of the portion of the heat (collected through the condenser) that is used for driving the ORC unit (and the rest is used for district heating). Hereafter, the ratio of the heat used for district heating to that collected from the condenser is addressed with R_q. In Fig. 6, the x-axis is the ratio of the mass flow rate used for supplying district heating to the total pressurized water heated in the condenser (note that $\frac{\dot{m}_{11}}{\dot{m}_{10}} = R_q$). According to the figure, expectedly, as the temperature of supply increases, the producible power by the ORC cycle increases. As such, the more heat (or mass flow rate) is used for deriving the ORC evaporator, the more power is generated. In case all the heat gained through the condenser is used for the ORC unit, a total of 920 kW more power might be supplied if the supply temperature is 90 °C. This can increase to just below 1,100 kW if the supply temperature increases to 110 °C. This, however, is a trade-off because as the higher temperature is supplied to the ORC unit, a lower electricity output of the main CHP cycle is resulted (see Fig. 5).

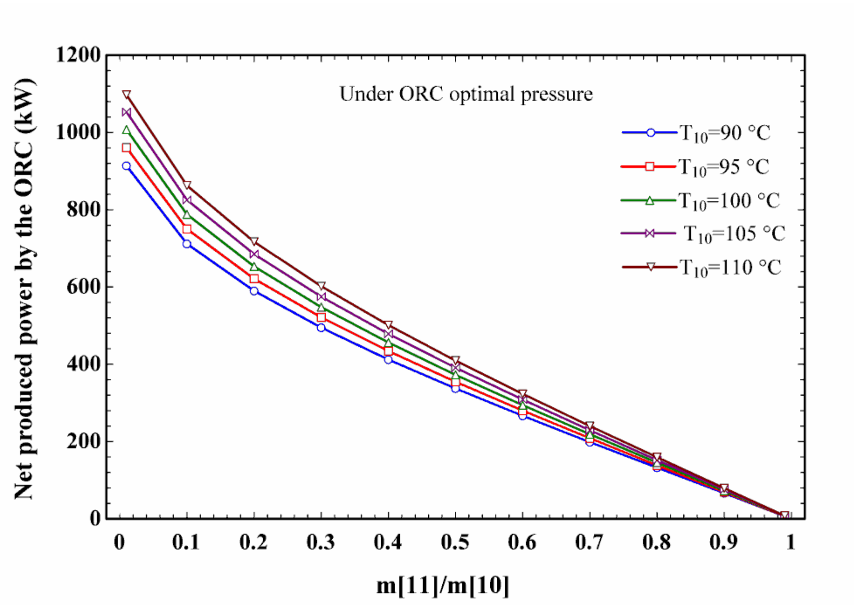

Fig. 6 the effect of various deriving temperatures and R_qs on the net power output of the ORC unit.

Note that the above figure is based on the optimal evaporator pressure in the ORC cycle in different supply temperatures, corresponding to the maximum ORC output power. Fig. 7 shows what the optimal evaporator pressure will be for various ORC evaporator temperatures.

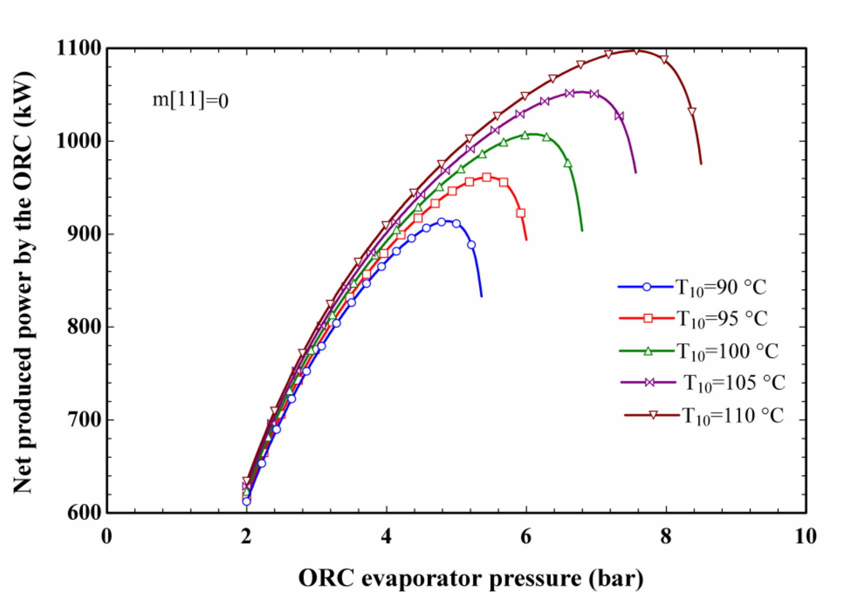

Fig. 7 The optimal pressure of the evaporator in the ORC cycle for various evaporator pressures.

The above figure is indeed for a full supply of the ORC (no heat supply for district heating, $R_q = 0$). Fig. 8 shows the effect of the change of the R_q value on the optimal pressure of the evaporator of the ORC unit. As seen, as the value of R_q increases (less heat to the evaporator), the optimal pressure decreases. This reduction is sharper for the small ratios and the pace of reduction in the optimal pressure value gets milder as the R_q value increases. As such, the lower the supply heat temperature, the lower pressure would be required at the evaporator. The effect of this item can be estimated to be linear.

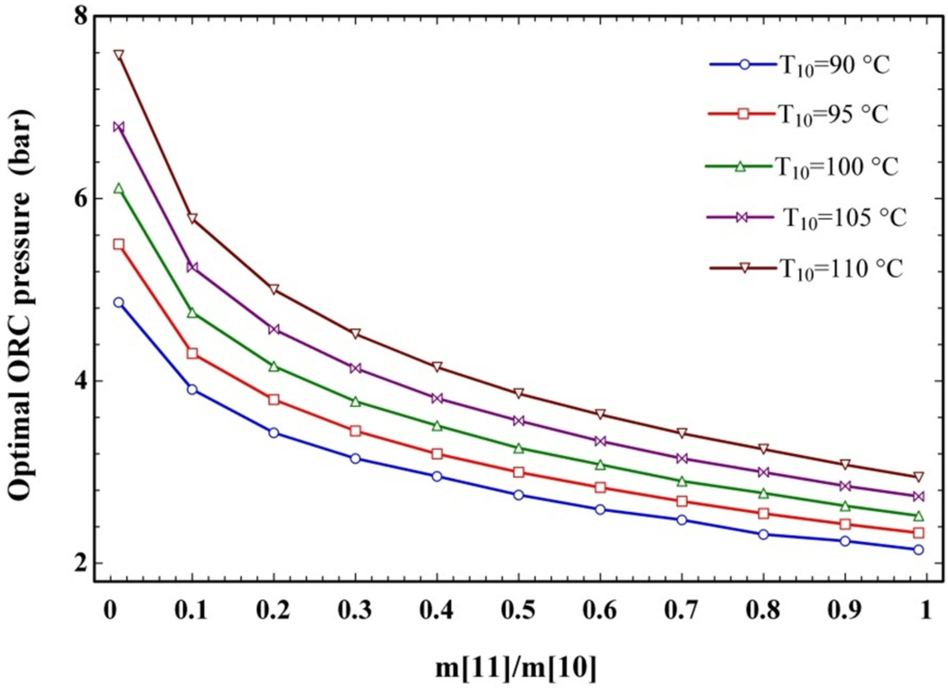

Fig. 8 The effect of R_q value change on the optimal pressure of the evaporator in the ORC unit.

Fig. 9 shows how the rate of heat production of the combined cycle might be affected as the rate of supplied heat to the district heating heat exchanger changes. As seen, the heat output of the system goes up almost linearly as a higher mass flow rate of the heated water through the condenser goes through district heating heat exchanger (HE2). This figure is based on a condenser temperature of 90 ^{o}C. Naturally, based on simple thermodynamic rules (just consider a typical T-s diagram of water), as the temperature of condenser increases, higher heat rates must be rejected from the steam flow to condense it. However, since the range of temperature considered (only up to 110 ^{o}C), the change in the value of the condenser heat rejection is so small that all the graphs almost lie on each other.

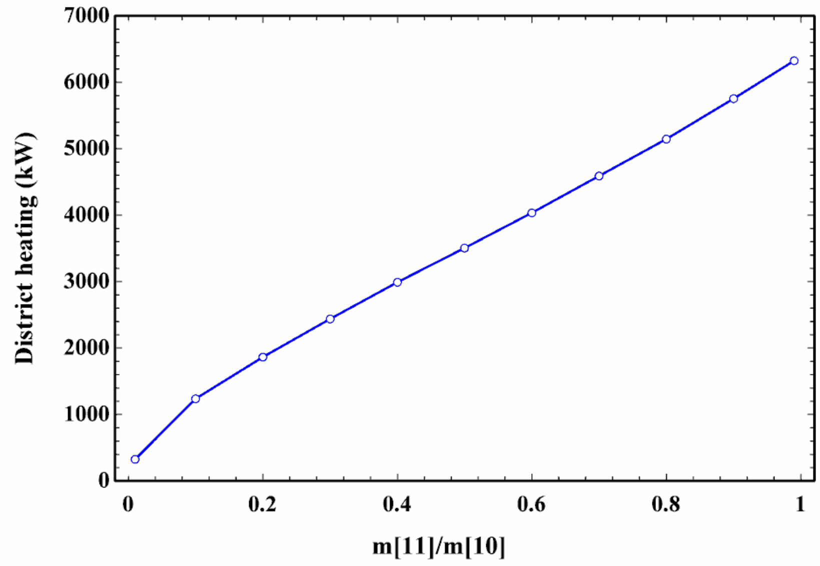

Fig. 9 The rate of heat producible by the district heating heat exchanger in various values of R_q.

Having the information presented so far, one could evaluate the effect of the LPT outlet temperature and various R_q values on the total producible power by the combined cycle (CHP + ORC). Fig. 10 shows the results of this investigation. As seen, and expectedly, the less heat goes for district heating supply, the more power can be produced. The interesting point, however, is that, the higher temperature of the condenser, the less total power can be produced by the entire combined system. This means that the optimal condenser temperature would be the minimum temperature (just the same temperature as the conventional system) because the reduction of the power output of the main CHP cycle caused by increasing the pressure of the condenser is stronger than the improvement made in the net power output of the ORC unit.

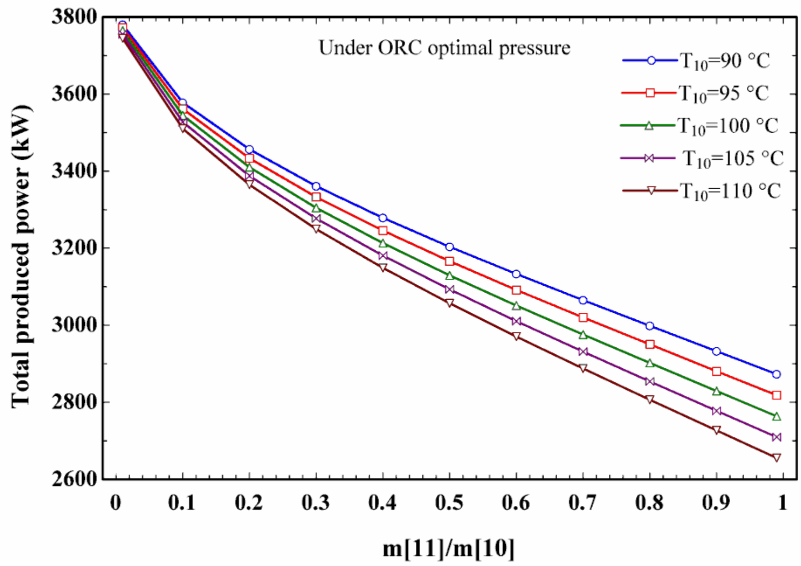

Fig. 10 the total power producible at various condenser temperatures and R_qs.

Fig. 11 shows the overall electricity efficiency of the combined CHP-ORC plant.

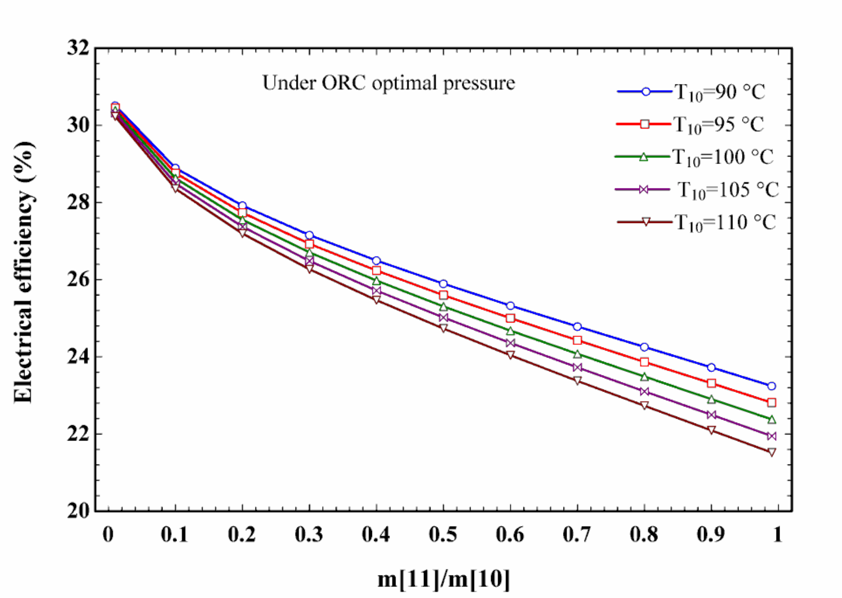

Fig. 11 The electrical efficiency of the combined cycle.

According to the figure, just the same trend as the previous figure is presented because the electricity efficiency of the system is a direct linear function of the total produced power (i.e. produced power divided by the inlet energy as fuel). As seen, the maximum possible electricity efficiency of the system is just below 31% and is achieved at the condenser temperature of 90 ^{o}C. It should be highlighted that the electrical efficiency of the conventional CHP system was 23.19% (see Table 6). Then, the improvement in this efficiency is almost more than 25 percent compared with the conventional systems. When the heat collected from the condenser is fully used to run the ORC unit, not much reduction of the total power output is observed with the increase of the condenser temperature, but this temperature growth results in a less and less electricity efficiency as the rate of heat used for district heating increases.

Fig. 12 shows the total energy efficiency of the waste-fired CHP-ORC plant for various R_qs. Like the change in the district heating values, altering condenser temperature (in the range of 90-110 °C) did not changed the energy efficiency significantly.

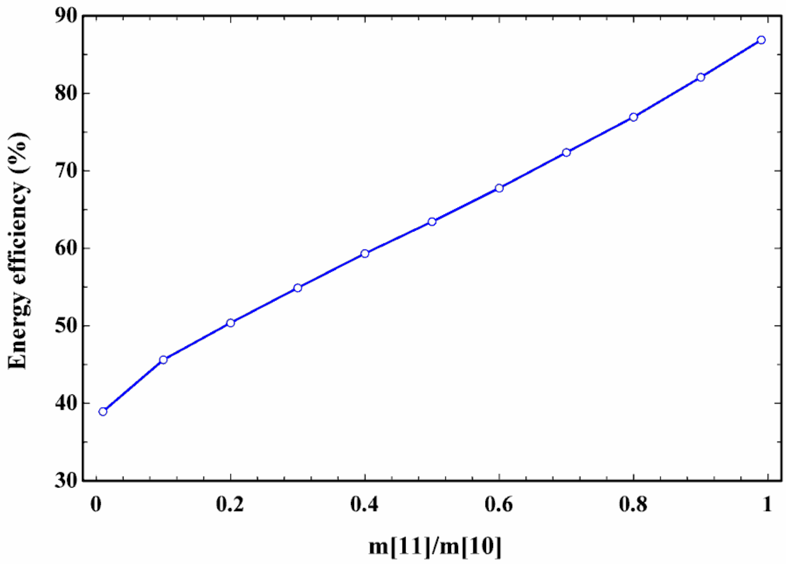

Fig. 12 The total energy efficiency of the combined waste-fired CHP-ORC plant.

After the thorough results presented regarding the energy performance of the conventional CHP plant and the proposed combined system, Table 8 outlines information about the rate of exergy destruction within the system at various condenser temperatures, optimal ORC evaporator pressures, full-load operation of the cycle, and when half of the heat is supplied to the ORC and the rest is used for district heating. As the rate of heat supplied to the ORC unit and district heating changes, the rate of exergy destruction in these two components alters proportionally.

Table 8 – The exergy destruction rate of the system at various condenser temperatures.

Condenser Temperature (°C)	Steam Turbine (kW)	ORC Unit (kW)	Condenser (kW)	District heating Heat Exchanger (kW)
90	247.8	182.7	353.8	76.16
95	239.9	190.7	376.5	100.8
100	232.2	199	399.3	125.3
105	224.7	206.8	421.2	149.9
110	217.4	215.3	444	173.9

Naturally, the change of the operation condition at the district heating heat exchanger and the ORC unit does not affect the incinerator exergy efficiency. For this unit, the rate of exergy destruction is fixed and equal to 9,832 kW.

Finally, Fig. 13 shows the net exergy efficiency of the combined CHP-ORC system for various heat supplied ratios and condenser temperatures. It is seen here that there is a low extremum point for the exergy efficiency of the system when the rate of the heat supplied to the ORC grows. It means that for various condenser temperatures, at full ORC supply state, the exergy efficiency is at a moderate level. As the value of R_q increases, the exergy efficiency drops and somewhere around R_q=30-60% this parameter is minimized. Thereafter, the increase in the exergy efficiency value is observed and the best exergy efficiency is obtained when the entire generated heat utilized to run the ORC. This trend is almost similar for all condenser temperatures while the level of the exergy efficiency value falls with the increase of the condenser temperature. However, according to the figure, the exergy efficiency change for a specific temperature is not that significant. For example, for the condenser temperature of 110 °C, the exergy efficiency hits its maximum value of 26.54% when all the heat of the condenser is used to derive the ORC unit. This value decreases to the minimum value of 25.17% when 40% of the heat is used for the ORC supply and it hits the value of 25.41% when all the heat is used for district heating.

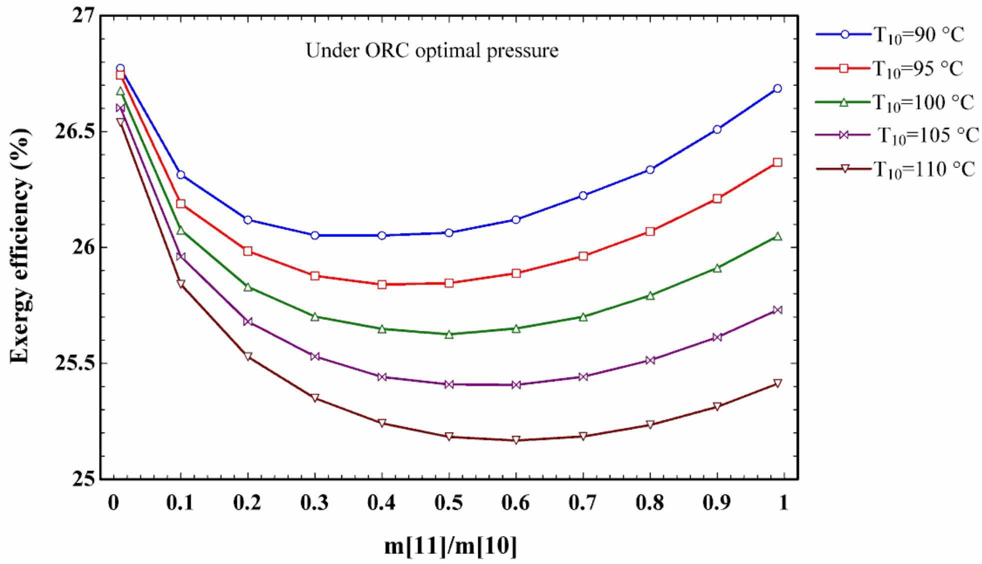

Fig. 13 the exergy efficiency of the system for various condenser temperatures and R_qs.

Therefore, in contrast with the energy analysis section in which remarkable changes in the energy output and energy efficiency of the system would be observed for various cases, the exergy performance indices of the system does not change considerably. This is a good indication because one could conclude that attaching the ORC unit to the waste-fired CHP plant in order to maximize the electricity output of the plant, not only does not make a negative impact on the exergy performance of the system, but also slightly improves the exergetic performance.

Taking into account the results of the energy analysis section, one could conclude that the hybrid system makes a less overall energy efficiency than the conventional system whereas there is even an investment required to attach the ORC unit to the plant. This, however, as explained, is not only about the efficiencies, rather it is all about the importance of the fact that the energy systems are moving toward a much stronger dependency on the electricity grids where even district heating systems will be dominated with electrical driven heat production systems, e.g. heat pumps. In the future energy systems, where advanced transcritical heat pumps and other highly efficient heat pump technologies come into operation, and where district heating goes to much lower temperatures (i.e. about 40-45 °C at supply line [44]), a very high coefficient of performance for the heat pumps (≥6.5) will be achievable [45]. The factors heating power versus the inlet temperature to the evaporator, the coefficient of performance versus the temperature of the lower and upper temperature heat sources and the obtained heat energy price have been designated for advanced heat pumps in [46]. Therefore, it will be interesting to investigate which rate of power-to-heat value in the future energy market makes the proposed combined system a logical investment. For this, the factor equivalent power ratio ($P_{eq,ratio}$) is defined as:

$$P_{eq,ratio} = \frac{P_{CHP-ORC} + {Q_{CHP-ORC}}/{X_{pth}}}{\eta_{CHP-ORC,max}}) \tag{31}$$

in which, $P_{CHP-ORC}$ and $Q_{CHP-ORC}$ are respectively the total electricity and heat produced by the combined cycle, X_{pth} is the power-to-heat worthiness factor (how much power might be values compared to heat in the future energy systems) and $\eta_{CHP-ORC,max}$ is the maximum obtainable energy efficiency from the combined cycle (was calculated as 87%, obtained in a case all the heat of the condenser is used for district heating). Fig. 14 shows how the value of $P_{eq,ratio}$ changes for various X_{pth} values and different R_q values. As seen, when R_q is zero (all the heat to district heating) and X_{pth} is 1, will result to the best performance of the system. However, evidently, this will not be the governing conditions to the future energy markets. For X_{pth} = 1, as the higher values of R_q, the less value of $P_{eq,ratio}$ is obtained. The difference between the various R_q cases get smaller as the X_{pth} value grows and all of a sudden at $X_{pth} = 4.8$, the trend gets revers and a case with lower R_q value makes a better value of $P_{eq,ratio}$. This means that, the proposed combined system, which aims at converting a heat flow (used to be employed for district heating) to electricity would be a smart measure provided that electricity is more values than heat, at least 4.8 more. The more power is valued compared to heat, the more this configuration will make sense techno-economically.

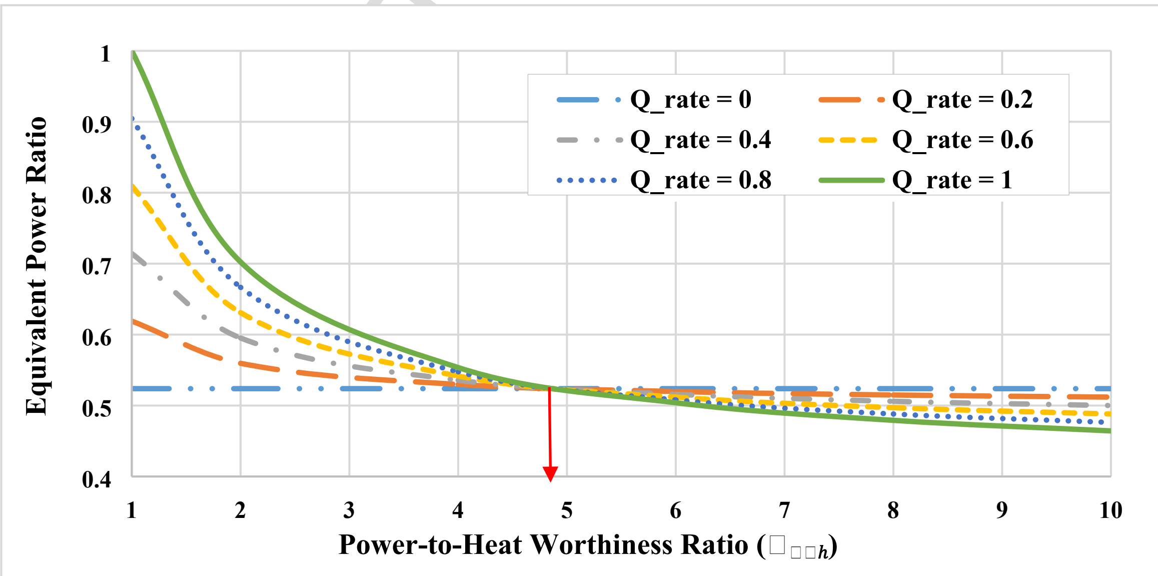

Fig. 14 variation of the equivalent power output of the system versus various R_qs and power-to-heat worthiness ratios.

5. Conclusion

In this study, the parallelization of a small-scale ORC unit with a waste-fired CHP plant was investigated in order to increase the share of electricity output of the waste-fired CHP plant. This idea is an inspiration from the fact that electricity sector is becoming more and more important in the global energy systems.

This work proposes a configuration for the aimed hybrid power plant and accomplished a detailed energy and exergy analysis to dig into the very deep thermodynamic performance details of the systems. The thermodynamic assessments show that an electrical efficiency of up to 44% might be achieved from the combined system while the exergy efficiency is not affected considerably. The main points of exergy destruction in the system are addressed and the rates of irreversibilities in each part are quantified.

It is found out that the increasing the condenser temperature for getting more power from the ORC unit is not a good idea because the power output degradation in the main CHP cycle is more than what is achieved from the ORC in this case. Thus, the regular condenser temperature of the CHP plant which is set at 90 ^{o}C for being able to supply the local district heating system is the best heat flow to be employed for running the ORC unit. It was concluded that, at a given condenser temperature, as the rate of heat used for the ORC unit increases (i.e. less heat for district heating), the overall energy efficiency of the combined system decreases.

Indeed, the system is proposed because electricity is more valuable than heat, and this superiority will be extremely higher in the future energy systems. Therefore, introducing the equivalent power ratio for the system, it was investigated when and at which power-to-heat worthiness ratio the combined system would make sense techno-economically. It was realized that a ratio of 4.8 is enough to make the system feasible, where there are already electrical-driven systems, e.g. heat pumps, which offer higher energy conversion efficiencies.

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- The combination of a waste-CHP plant with an Organic Rankine Cycle is proposed.
- The aim is to maximize power output of the system instead of a higher heat output.
- The combined configuration is designed and analyzed thermodynamically.
- The power output may increase up to 25% via the proposed integration.
- A power-to-heat value of 4.8 is needed to make the system feasible economically.