Analytical and Comparative Study of a Mini Solar-Powered Cogeneration Unit Based on Organic Rankine cycle for Low-Temperature Applications

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Abstract: In this paper, we analyze characteristics of a small Combined Heat and Power (CHP) system based mainly on Organic Rankine Cycle (ORC) and heating plant in actual series connection regarding the low-temperature heat carrier heated by purely solar flat collector field. Simultaneously and for specific power production, comparison of this layout with stand-alone ORC, and with the traditional ORC-CHP imposing gain of condenser heat for heating aims, in second step, has been conducted. For evaluation, energetic and design criteria have been determined opposite the heating effects and also temperatures of the heat source and sink. The simulations addressed interesting optimization ratios till 24 % for the power unit throughout this series CHP utility versus single power generation at the same conditions tested. Moreover, the high heat source temperatures and CHP ratios improve the performance of the overall series plant, while the high supply and return temperatures have negative effects. Finally, the ORC-CHP scheme handled here highlights distinctive exploitation aspects and more suitability in wide range of application in comparison to yielding the high-temperature condensation heat of ORC, especially at low ambient temperatures, high supply and heat source temperatures. So, it can be advised to be adopted instead of the two other strategies.

Keywords: Series Combined Heat and Power (CHP), Organic Rankine Cycle (ORC), Low-temperature solar heat, Parameters analysis and optimization.

1. INTRODUCTION

Generally, the systems which can simultaneously or even asynchronously provide more than useful output such as cogeneration and the tri-generation units, especially the small-size ones, find nowadays wide application. The latter may gain more importance and become more attractive when utilizing the low-temperature solar energy because of the current challenges such as high investment cost and great source wastage besides of the continuous increment of prices of the fossil energy carriers together with adopting several environmental measures. Recently and for different purposes, Organic Rankine Cycle (ORC) has been used as instrument for effectively converting this low-temperature solar heat into power either as stand-alone system or, but with little rates, in framework of the Combined Heat and Power (CHP) plants which are core of our study. The solar-driven ORC-CHP evolutions recently presented and analyzed have almost similar trends regarding the study methodology and goal [1-8]. Riffat and Zhao [1, 2] constructed and experimentally investigated a novel hybrid heat pipe solar collector/ORC-CHP system with n-pentane as working fluid in ORC. They indicated that their system could save the primary energy consumption and reduce CO2 emission up to 600 tons per annum compared to conventional electricity and heating supply. Yagoub *et al*. [3] developed and tested a hybrid solar-gas driven combined heat and power (CHP) system based on ORC along with testing two working fluids (n-pentane and HFE-301). It was found that HFE-301 is better than n-pentane regarding the electrical cycle efficiency, and the overall utilization efficiencies increase for both fluids to 17 % and 15 % for HFE-301 and n-pentane respectively and also CO2 emission could be reduced. Facão and Oliveira [4] analyzed a micro cogeneration system based on ORC and powered by solar energy and also supplemented by a natural gas boiler, where several working fluids such as n-pentane, HFE7100, methanol and cyclohexane were screened. Methanol presented the best performance within their study and the estimated system payback period was smaller than the system life time, and also the system could save 51 tons CO2 per year compared to the conventional situation. Facão *et al*. [5, 6] also simulated three solar-assisted thermodynamic cycles based on ORC for a micro cogeneration system with a power output of 5 kW and for three temperature ranges and also they evaluated performance of several working fluids. Among the fluids screened, Cyclohexane led to the best performance, and storage of the solar heat enabled the system to operate for longer periods. Mayere and Riffat [7] gave a short overview of the state-of-the-art in field of solar-

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driven micro ORC-CHP systems and associated current technologies, where all concentrating solar absorbers for micro CHP were described. Ziviani *et al*. [8] developed an advanced thermodynamic model for simulating ORC to evaluate the ORC capability to meet electric, thermal and cooling loads of a single residential building for typical temperatures of the hot water exiting from a solar collector. They focused on the electric efficiency and power output of ORC during screening some working fluids and testing the temperature of high reservoir (solar collector temperature). In framework of combining the solar with the geothermal energy for powering the ORC-CHP systems, Tempesti *et al*. carried out a thermodynamic analysis of two micro units in [9] and thermo-economic assessment of a micro one in [10] along with comparison three working fluids in the both. They stated that R245fa shows the best cycle efficiency and allows the lowest electricity price within scope of their studies, while R134a releases the highest heat, and the solar collectors have the highest exergy destruction. In principle, most of the present ORC-CHP studies, aforementioned, have the same proposal regarding the CHP concept, where the heat rejected by ORC condenser was used for space/water heating aiming to lower the heat source losses. Furthermore, the CHP part of the solar-powered tri-generation units based on ORC did not differ from the single ORC-CHP concepts, where they had almost the same principle regarding the heat gain from the overall installation [11-14]. For example, Al-Sulaiman *et al*. [11, 12] used the waste heat of ORC condenser for the heating through a heat exchanger and for cooling through a single-effect absorption chiller. Marrero *et al*. [13] proposed that ORC and a LiBr–H2O absorption chiller were parallel fueled by heat tank charged by solar collectors, while the waste heat in ORC condenser was utilized for water heating. Ozcan *et al*. [14] employed the exhaust gases from solid oxide fuel cell (IR-SOFC) along with parabolic trough solar collectors for leading a two-stage ORC and a LiBr–H_2O absorption chiller and also heating purposes. Differently, Freeman *et al*. [15] proposed another concept of the solar ORC-CHP combination, where the heat carrier heated by vacuum collectors drives ORC plant (with R245fa as working fluid) and then heats a domestic hot water cylinder, which is supplemented by auxiliary heater, in series manner. Moreover, a bypass technology was integrated to change the operation modus. They found that the average cost per unit power is 37-62 £/W_e compared to 20-30 £/W_e for solar-PV.

From the brief consequences derived from the literatures cited, it can be assumed that such mini solar ORC-CHP units need further investigations, where there are several points to be argued such as: The mutual influences of the parameters of ORC power and heating mechanism, impact the heating plant indicators on the performances, optimization potential of the power unit production through these technologies and extensive analysis of new integration way such as in [15] but for pure solar heat source. Therefore, this paper will extend the last studies, especially [15], and present a detailed and deep analysis of the series concept of integrating the mini ORC-CHP plant powered by low-temperature solar energy captured by simple flat collectors. Here, continuous operation modus for delivering the both power and heat output at the desired levels in the sunny days will be assumed and also no auxiliary energy sources will be integrated. Simultaneously and for displaying advantages of this method versus stand-alone ORC and the ORC-CHP way adopted in the references cited [1-14], the latter will be investigated under the same terms. Thus, the intended comparisons of this method with the two aforementioned cases will be easily argued and also individually conducted in two parts for the simplicity. For evaluation, energetic and design characteristics will be estimated under several eventual working conditions such as the heating plant parameters and heat source and sink temperatures probable to be available. As working fluid in ORC, the isentropic refrigerant R134a will be solely used because it is strongly advised for small-scale solar applications [16] and its physical properties can be obtained from the literatures [17-19]. Finally, to modulate and calculate all the systems, the software "Matlab" will be employed.

2. SYSTEM DESCRIPTION AND MODELING

Differently from the ORC-CHP presented in the referenced works [from 1 to 14] and somewhat similarly to [15], the ORC power and heating plant will be connected in series scheme regarding the heat carrier heated by flat solar collector, where it will be proposed that the condenser heat will be rejected to the ambient at as low temperature as possible (Figure **1**). This scheme is rather similar to the evolutions concerning the exploitation of the geothermal energy throughout CHP devices based on ORC [20-22], but the utilization principle of the solar energy is different, where the heat carrier in the latter circulates in closed loop. So, according to Figure **1**, the solar collector fluid is firstly cooled in the evaporator of ORC till specific temperature (T_m), which is considered as the correlation parameter between the both mechanisms

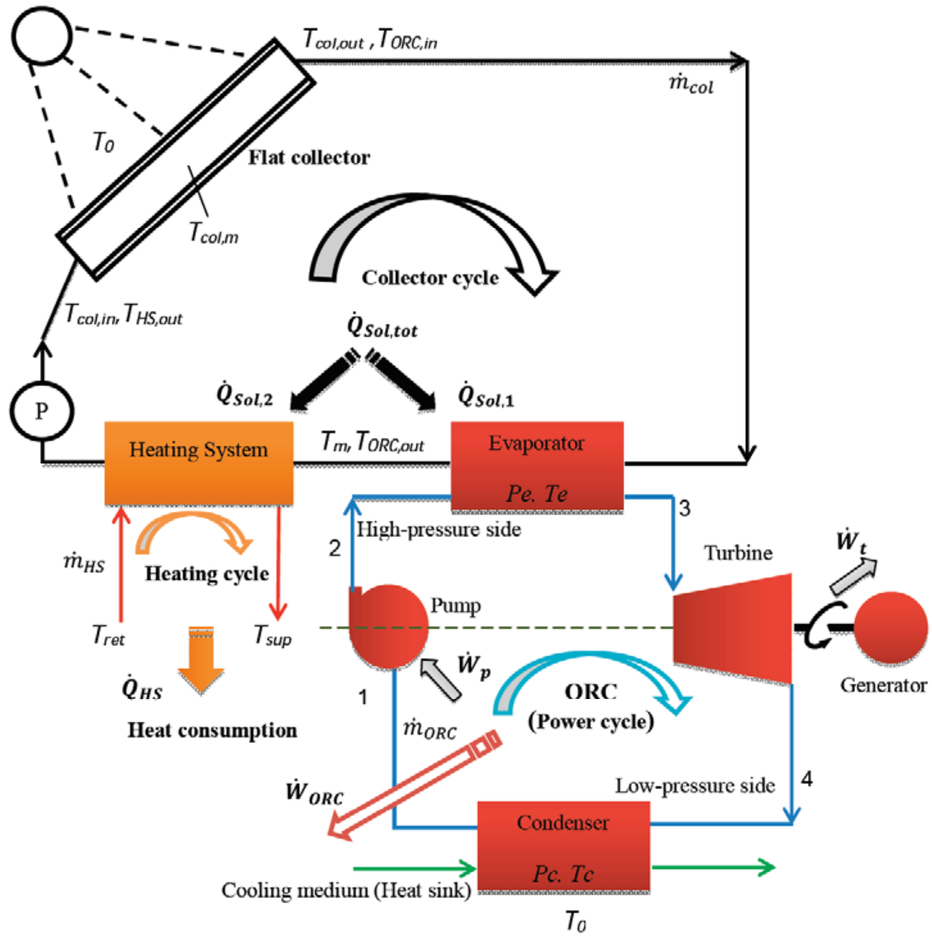

Figure 1: Schematic view of the proposed Collector –ORC-CHP Cycle.

and relates strongly to the heating effects, and then submits the heat load needed at heat exchanger of the heating system (heat consumer). Thus, this concept of configuration will allow having high temperatures ($T_{col,out}$, $T_{ORC,in}$) from the collector for ORC which its efficiency will be better at increasing heat source temperatures [23, 24]. At the same time, the medium temperature at collector input ($T_{col,in}$) will be shifted to lower levels (from $T_{ORC,out}$ to $T_{HS,out}$) throughout the extra cooling in the heating system located as intermediate circuit between ORC exit and the collector entrance. This procedure may somewhat promote the efficiencies of ORC and collector cycle expressed by the combined performance as we will see in the simulation results.

For modelling the overall unit under analysis, some assumptions will be included in the simulations. The first of all, the vapor quality at turbine inlet will be assumed to be saturated because the superheating, in case of the isentropic fluids, does not promote the efficiency and can be also dispensed whether it is requested to avoid the droplets after expansion in turbine in comparison to the wet fluids. Moreover, the vapor does not become superheated after expansion in turbine, in opposite of dry fluids [25-29]; thus, no superheater and desuperheater are needed. Secondly, the pressure loss in all cycles is ignored and the ORC-evaporator, ORC-condenser and the heating system are adiabatic heat exchangers (no heat losses during the heat transfer process). Furthermore, the temperature rise of the collector fluid through its pump is neglected. Finally, it is to be imposed that the heat load and the power output are proactively known as main aims of the solar installation. Thus, starting from the latter propositions, the overall series plant composed of the three cycles can be modulated as follow.

The thermal efficiency of ORC depends mainly on the working pressures and hence the related specific enthalpy changes in ORC ingredients

$$\eta_{th,ORC} = [(h_3(Pe) - h_4(Pc)) - (h_2(Pe) - h_1(Pc))] / (h_3(Pe) - h_2(Pe)) \quad (1)$$

where,

$$h_4 = h_3 - (h_3 - h_4') \cdot \eta_{is,t} \tag{2}$$

$$h_2 = h_1 - (h_2' - h_1) / \eta_{is,p} \tag{3}$$

The solar heat flow involved for producing the desired net output power $(\dot{W}_{ORC})$ can be consecutively computed by

$$\dot{Q}_{Sol,1} = \dot{W}_{ORC} / \eta_{th,ORC} = \dot{m}_{ORC} \cdot (h_3 - h_2) = \dot{m}_{col} \cdot (T_{col,out} - T_m) \tag{4}$$

Since the intermediate temperature of the heat carrier (collector medium) T_m represents the coupling parameter between the both power and heating plant, it can be determined in two ways as below

From ORC side

$$T_m = T_{col,out} - [\dot{W}_{ORC} / (\eta_{th,ORC} \cdot \dot{m}_{col} \cdot C_{p,col})] \tag{5}$$

From the heating plant side

$$T_m = [\dot{Q}_{HS} / (\dot{m}_{ORC} \cdot C_{p,col})] + T_{HS,out} = [(\dot{m}_{HS} \cdot C_{p,HS} \cdot T_{sup} - T_{ret})) / (\dot{m}_{col} \cdot C_{p,col})] + T_{HS,out} \tag{6}$$

When applying the last equation, the attention must be paid to the pinch point limitations (ΔT_{pp}), stated below, at the both sides of the heating plant (at the entrance and exit) for all thermal terms wished from the heat consumer.

$$T_m \geq T_{sup} + \Delta T_{pp} \;\&\; T_{HS,out} \geq T_{ret} + \Delta T_{pp} \tag{7}$$

Here, the supply (T_{sup}) and return (T_{ret}) temperatures of the heating cycle should be predefined beside the desired heating load $(\dot{Q}_{HS})$ in the heating system for defining, at least, one of the temperatures T_m or $T_{HS,out}$ throughout the pinch point equation (**7**). Thereby, the required mass flow rate of the heat carrier in the collector cycle $\dot{m}_{col}$ can be estimated throughout the equations **5** and **6** when neutralizing T_m from those equations and defining the temperatures $T_{HS,out}$ and $T_{HS,col}$.

$$\dot{m}_{col} = [(\dot{W}_{ORC} / \eta_{th,ORC}) + \dot{Q}_{HS}] / C_{p,col} \cdot (T_{col,out} - T_{HS,out})] \tag{8}$$

On the other side and in order to reveal impact of the working parameters on the combined performances of the three cycles (collector, ORC and HS), it is fundamental to determine the following efficiencies as described below.

The flat collector efficiency (η_{col}) is well known and assessed by the second approximation.

$$\eta_{col} = \eta_{op} - a \cdot x - b \cdot G \cdot x^2 + \cdots \tag{9}$$

$$x = (T_{col,m} - T_0) / G \tag{10}$$

The averaged collector temperature in case of stand-alone ORC

$$T_{col,m} = (T_{ORC,out} + T_{col,out}) / 2 \tag{11}$$

While, the averaged collector temperature in case of CHP is determined

$$T_{col,m} = (T_{HS,out} + T_{col,out}) / 2 \tag{12}$$

The combined thermal efficiency of the ORC-Collector cycle will be given

$$\eta_{com} = \eta_{th,ORC} \cdot \eta_{col} \tag{13}$$

The thermal combined heat and power (CHP) efficiency of the ORC-HS cycle

$$\eta_{CHP} = (\dot{W}_{ORC} + \dot{Q}_{HS}) / (\dot{Q}_{sol,1} + \dot{Q}_{sol,2}) \tag{14}$$

Consequently, the total efficiency of the combined three cycles ORC-HS-Collector

$$\eta_{tot} = \eta_{CHP} \cdot \eta_{col} \tag{15}$$

Furthermore, it is common for estimating the heat transfer capacities required for transferring the heat loads in ORC components (preheater, evaporator and condenser) that the product from the total heat transfer coefficient (k) and the heat exchanger area A is used and given as follow [22, 30-35].

$$k \cdot A = \dot{Q} / \Delta T_{m,log} \tag{16}$$

Similarly, the heat transfer capacities for heating system will be estimated for simplifying the calculations [22, 35]. While, the flat collector area needed for capturing the solar heat can be assessed depending on the efficiencies as follow.

For producing the desired power in both cases of ORC (ORC (SA) and ORC (CHP))

$$A_{col} = \dot{W}_{ORC} / (\eta_{com} \cdot G) \tag{17}$$

For delivering the both useful outputs in the total CHP plant

$$A_{col} = (\dot{W}_{ORC} + \dot{Q}_{HS}) / (\eta_{tot} \cdot G) \tag{18}$$

Furthermore, the following equivalences are always correct

$$\dot{Q}_{sol,2} = \dot{Q}_{HS} \tag{19}$$

$$T_{col,out} = T_{ORC,in} \;\&\; T_m = T_{ORC,out} \;\&\; T_{HS,out} = T_{col,in} \tag{20}$$

The heating capacity $\dot{Q}_{HS}$ will be always related to the net power $\dot{W}_{ORC}$ throughout the CHP ratio f which can be defined as ratio of the heat to the power.

$$\dot{Q}_{HS} = f \cdot \dot{W}_{ORC} \tag{21}$$

Finally, it is to be pointed out that the descriptive equations for simulating the single ORC are the same as the ones formulated above but without taking the heating plant parameters into account, where ORC can easily work at the optimums. The latter is also true and applicable for the classical ORC-CHP exploitation, but the condensation process must happen at temperatures matching the supply temperatures of the heat consumer.

where, $\dot{Q}_{Sol,1}$ and $\dot{Q}_{Sol,2}$ are the solar heat flows to be supplied into ORC and HS respectively; h_3 and h_4 are the specific enthalpies at inlet and outlet of the turbine respectively; h_2 and h_1 are the specific enthalpies at outlet and inlet of the working fluid pump respectively; $\eta_{is,t}$ and $\eta_{is,p}$ are the isentropic efficiencies of the turbine and working fluid pump of ORC respectively; h'_4 and h'_2 are the ideal specific enthalpies at outlet of the turbine and pump respectively; $\dot{m}_{ORC}$ is the mass flow rate of working fluid in ORC; $\dot{m}_{col}$ is the mass flow rate in the collector cycle; $\dot{m}_{HS}$ is the mass flow rate of the heating cycle; $T_{col,in}$ and $T_{col,out}$ are the input and output temperature of the collector fluid into and from the collector respectively; $T_{HS,out}$ is the collector fluid temperature after the heating system; ΔT_{pp} is the minimal temperature difference between the fluids (pinch point difference); $T_{ORC,in}$ is the temperature of the collector fluid entering the ORC evaporator; $T_{ORC,out}$ is the temperature of the collector fluid exiting from ORC; $\Delta T_{m,log}$ is the mean logarithmic temperature difference in a heat exchanger; $\dot{Q}$ is the heat flow to be transferred by a heat exchanger; η_{op} is the optical efficiency of the collector; a is the first order heat loss coefficient; b is the second order heat loss coefficient; G is the global irradiation; T_O is the ambient temperature.

3. BOUNDARY AND WORKING CONDITIONS, FURTHER ASSUMPTIONS

Table **1** shows all the nominal boundary conditions along with the average value of the ambient temperature and the global radiation and also the mostly awaited temperature from the flat collectors. The average ambient temperature and global radiation in this study will be adopted for the rather cooled regions such as Germany, where they are chosen according to the weather data of south Germany (Bayern region) as illustrated in Figure **2** which shows the monthly average temperatures over the year. Moreover, Table **1** also includes the nominal heating system parameters such as the supply and return temperature and also the heating load depending on the CHP ratio at predefined power output, where all the last parameters will be assessed for 1 kW power output. The condensation temperature and pressure of the ORC will be set as function of the ambient temperature for showing impact of the latter not only on the collector cycle but also on ORC performance, and thus on the combined cycle from the both last cycles. The evaporation pressure (working pressure) in ORC will be the controlling factor of the whole system and will be scaled according to all terms enforced, so this parameter will be always argued. For generalizing or extending the study, in addition to the working pressure in ORC along with the supply and return temperature of the heating system, some of the parameters, named previously, such as CHP ratio, ambient temperature and the possible temperature existing from the flat collector will be varied over a wide range as working parameters to meet other conditions available from the heat source or heat sink or desired at the heat consumer. Moreover, each working factor will be separately examined every time; while the others are consequently adjusted according to the circumstances enforced by the variables or will be kept constant. As evaluation criteria, the combined efficiencies, the heat transfer capacities of the ORC and heating system and solar collector areas will be characterized. Finally, for simulation of the proposed configurations, it will be assumed that they operate in steady state.

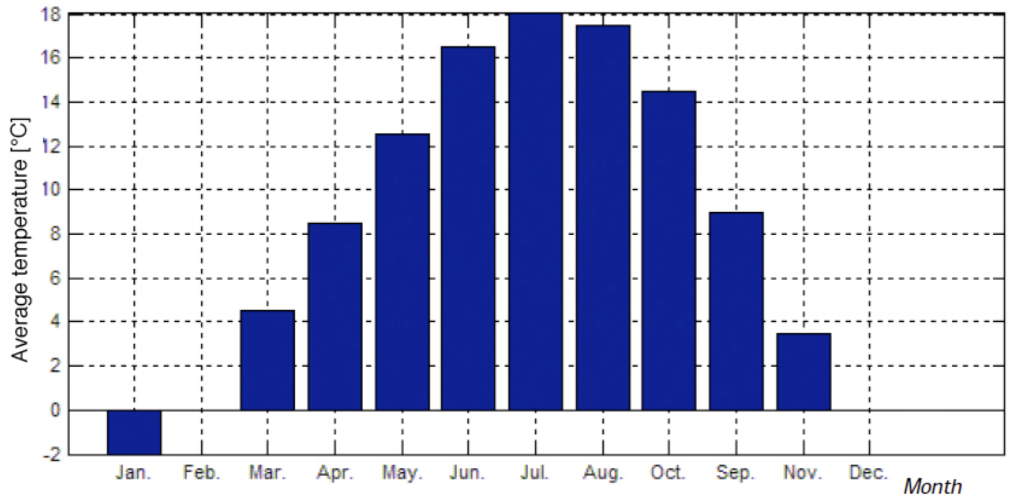

Figure 2: The average monthly temperatures in south of Germany (Bayern Region) over the year.

Table 1: The Boundary and the Nominal Conditions

Parameter	Symbol	Unit	Values
Ambient temperature	T_0	*K*	282 (9 °C)
Global irradiation	*G*	kW/m^2	0.7
Pinch point difference for all heat exchangers	ΔT_{pp}	°C	3
Isentropic efficiency of the ORC turbine	$\eta_{is,t}$	-	0,85
Isentropic efficiency of the ORC pump	$\eta_{is,p}$	-	0,8
Condensation temperature on ORC	T_c	°C	$T_c=T_0+8$
Nominal supply temperature of heating system	T_{sup}	°C	60
Nominal return temperature of heating system	T_{ret}	°C	35
The nominal ratio of the CHP	*f*	-	5
The nominal temperature at the collector outlet	$T_{col,out}$	°C	90
Optical collector efficiency	η_{op}	-	0.81
First order heat loss coefficient	*a*	$W/m^2.K$	3.2
Second order heat loss coefficient	*b*	$W/m^2.K^2$	0.015
Isobaric specific heat capacity of collector water	$C_{p,col}$	kJ/kg.K	4.2

4. SIMULATION IMPLEMENTATION AND RESULT DISCUSSION

4.1. Analysis of the Proposed Series ORC-CHP Along with Comparison with Single ORC

4.1.1. The Working Pressure (Evaporation Pressure) in ORC

When keeping all conditions at the nominal values adopted, impact of the evaporation pressure on the evaluative criteria appears in the Figure **3**, **a-f**. Here, it is visible that the efficiencies have optimums but at different pressures, where the typical pressure of the total efficiency (Figure **3**, **b**) will shift to lower pressure position compared to the combined efficiencies of ORC-Collector (Figure **3**, **a**). Occurrence of the latter optimums can be justified by the contradicted course of the ORC and collector efficiency, where with raising the pressure, ORC efficiency increases and thus the heat flow into evaporator for the same power output decreases (Figure **3**, **c**). Conversely, the scarcity of heat carrier (collector fluid) cooling, represented by progressive augmentation of its temperature at ORC outlet and caused by the compulsory pinch point difference in the evaporator, will associate the growing pressure (Figure **3**, **d**). Thereby, more mass flow rate of

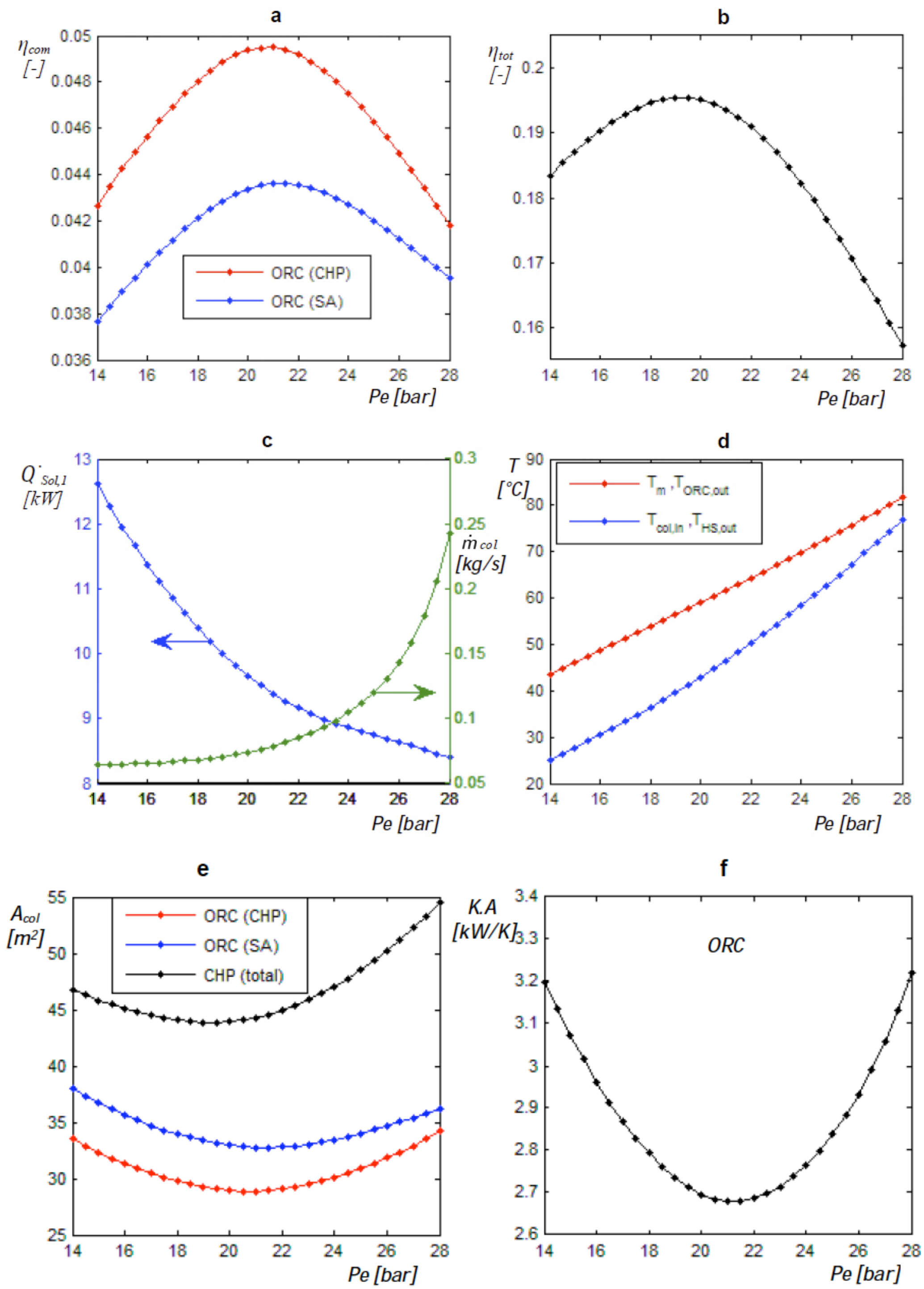

Figure 3: a, b, c, d, e and f: Influence of the evaporation pressure ***Pe*** of ORC (the working pressure) on the different performances of the considered cycles along with discussion of the impact of supply and return temperature $\boldsymbol{T_{sup}}$ and $\boldsymbol{T_{ret}}$ of the heating system at the nominal other conditions.

the heat carrier is involved for meeting this impact despite of running down the thermal energy necessary for ORC (Figure **3**, **c**). As result, destructing the collector efficiency versus augmenting the pressure will occur because, as known, more heat loss will accompany the increasing collector temperature caused, in turn, by the last phenomena. Concept of the series CHP reduces the last contradiction because the heating plant will take an amount of the collector fluid heat and hence lowering its temperature before returning to the collector (see Figure **3**, **d**). Consequently, the combined efficiency in case of the ORC (CHP) will be enhanced along the pressure interval investigated versus ORC (SA) for the same power output, where the optimization ratio at the optimum field reaches value till nearly 13.6 %. Furthermore, it can be seen that the optimization rate will weaken at the high pressures because of inability of the heating plant at the nominal CHP ratio to cool the heat carrier properly. This is, in turn, caused by jumping the mass flow rate towards high values due to decreasing the enhancement rate of the ORC efficiency at extremely high pressures. In addition, influence of the evaporation pressure on the collector areas required and the heat transfer capacities in ORC is shown in Figure **3**, **e** and **f**, where the latters have minimums but at dissimilar pressure values. Behavior of the flat collector areas can be derived from the efficiencies' curves, and similar optimization scope at same pressures (app. 13.6 %) can be registered. While, declining and thereafter progressing the ORC curve are caused by the counteractive influence of the heat flow transmitted and the temperatures' differences between the fluids exchanging the heat, where the both influencers decrease, especially in the evaporator, with raising the pressure and hence the related evaporation temperature (see Eq. 16).

As regards the desired supply and return temperatures of the heating system, the abovementioned diagrams and argumentations can give evident conceptions about them. At constant heat load, presence of an optimal pressure for the whole CHP unit means that there is a specified temperature of the heat carrier T_m available at entrance of the heating plant and hence reachable supply temperature when respecting the pinch point difference (see Figure **3**, **d**), where this case corresponds the values Pe=19.5 bar and T_m=58 °C, $T_{HS,out}$=41 °C and hence possibly T_{sup}=55 °C and T_{ret}=38 °C. Here, it is worth noting that all supply temperatures below this optimal one do not deviate the system performance from its optimum provided that the return temperatures have corresponding values according to Figure **3**, **d**, if this would be practically applicable. While raising the supply temperature above the typical value, mentioned above, leads to deteriorate the performances and the optimization possibility of power unit due to operating the CHP plant at pressures over the typical one, where the linear correlation between the pressure and the temperature T_m can explicitly interpret this reality. Likewise, the return temperature can affect the performances, where the high temperatures will enforce high values of T_m or higher mass flow rate of the heat carrier at heating plant inlet even at constant supply temperature and heating load (see Figure **3**, **d**), where the both cases make demands an higher pressures in ORC and thus reducing the overall effectiveness and improvement rates of the specific power. As final consequence for this, it is strongly recommended to keep the return temperatures as low as possible especially when holding the supply temperatures minimal.

4.1.2. Influence of the CHP Ratio and Hence the Heating Capacity of the Heating Plant

The analysis in term of this factor, while maintaining the rest of the conditions nominal, illustrates continual optimization rates for the power unit generation in case of ORC (CHP) in comparison to ORC (SA) (Figure **4**, **a** and **e**), where a maximal enhancement till app. 24 % for the combined efficiency and collector area required can be accomplished at the highest CHP ratio (10). While, ORC (CHP) indicators will follow a constant trend for a wide range till the ratio 8.5, because the thermal energy carried by the collector medium remains appropriate within this scope of heating plant parameters (Figure **4**, **c**, **d** and **f**). Inadequacy of the heat carrier energy for the heating effects after the named ratio (8.5), which results from pinch point restriction on the return temperature side, obliges the sudden augment at pressure and the related mass flow rate (Figure **4**, **c** and **d**), but this has not much to do with the overall behaviors. Generally, the total plant will be more effective at high heat demands, where the total efficiency progresses increasingly (Figure **4**, **b**), but more collector and heating system areas will be assigned in order to meet these high thermal loads (Figure **4**, **e** and **f**). On the other side and according to the Figure **4**, **c**, **d** and **f**, it is obvious that versus the last optimizations ORC (CHP) will not noticeably differ from the optimal operation modus contemplated by stand-alone state. Only, the pressure will be conditioned by small increase above the typical value to meet the supply temperature set, while the heat transfer capacities have almost no changes especially before the ratio 8.5.Another observation is to be pointed out that the inconsistent or non-sustained course of the heat transfer capacities of the heating system versus the heat load represented by the named ratio is due to shifting the pinch point constraint after the CHP ratio 8.5 from the supply temperature side to return temperature side in the heating system exchanger. Finally, it can be summarized for this evolution that the higher the CHP ratio and lower the return temperature at the same supply temperature, the cheaper or the more advantageous the producing the power unit and the more preferable the total plant.

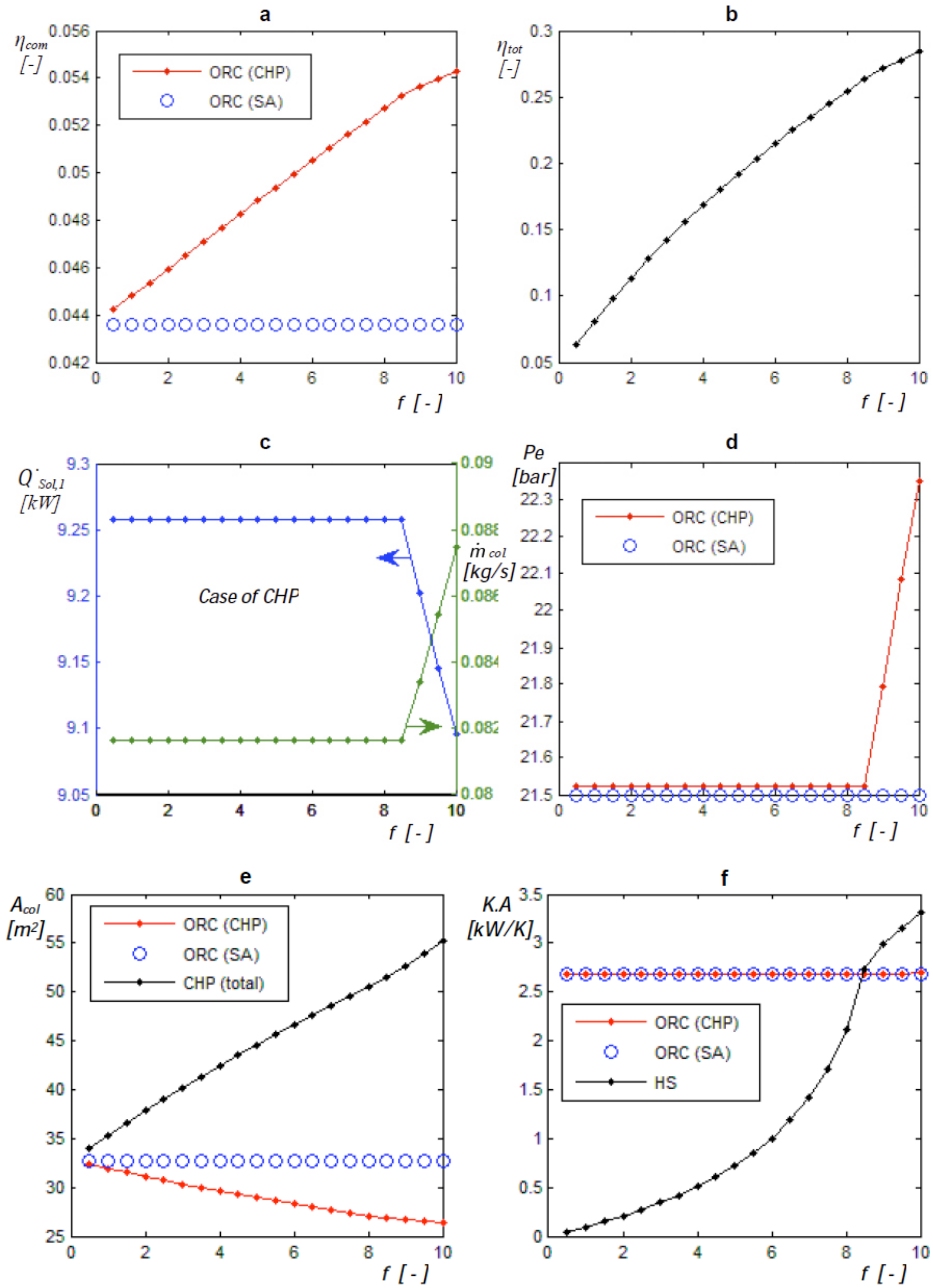

Figure 4: a, b, c, d, e and f: Influence of CHP ratio ***f*** and hence the heat demand in the heating system on the different performances of the considered cycles at T_{sup} = 60 °C and T_{ret} = 35 °C and at the other nominal conditions.

4.1.3. Influence of the Available Collector Temperature at the Outlet

Solving the cycles with variation of the fluid temperature exiting from flat collector when holding the other parameters invariable is represented by Figure **5**, **a-f**. Within scope of this temperature variation, increasing the latter will permanently deteriorate the performance of ORC (SA) and improve the optimization possibility in ORC (CHP), where the efficiency will be enhanced till app. 20 % at 100 °C (Figure **5**, **a**). Similarly, the collector area needed for the output power unit will be reduced till the last percentage (Figure **5**, **e**). Moreover, the heat transfer capacities needed for ORC in both cases become lower at high heat source temperature (Figure **5**, **f**) due to shortage the heat flow involved for the same power outcome

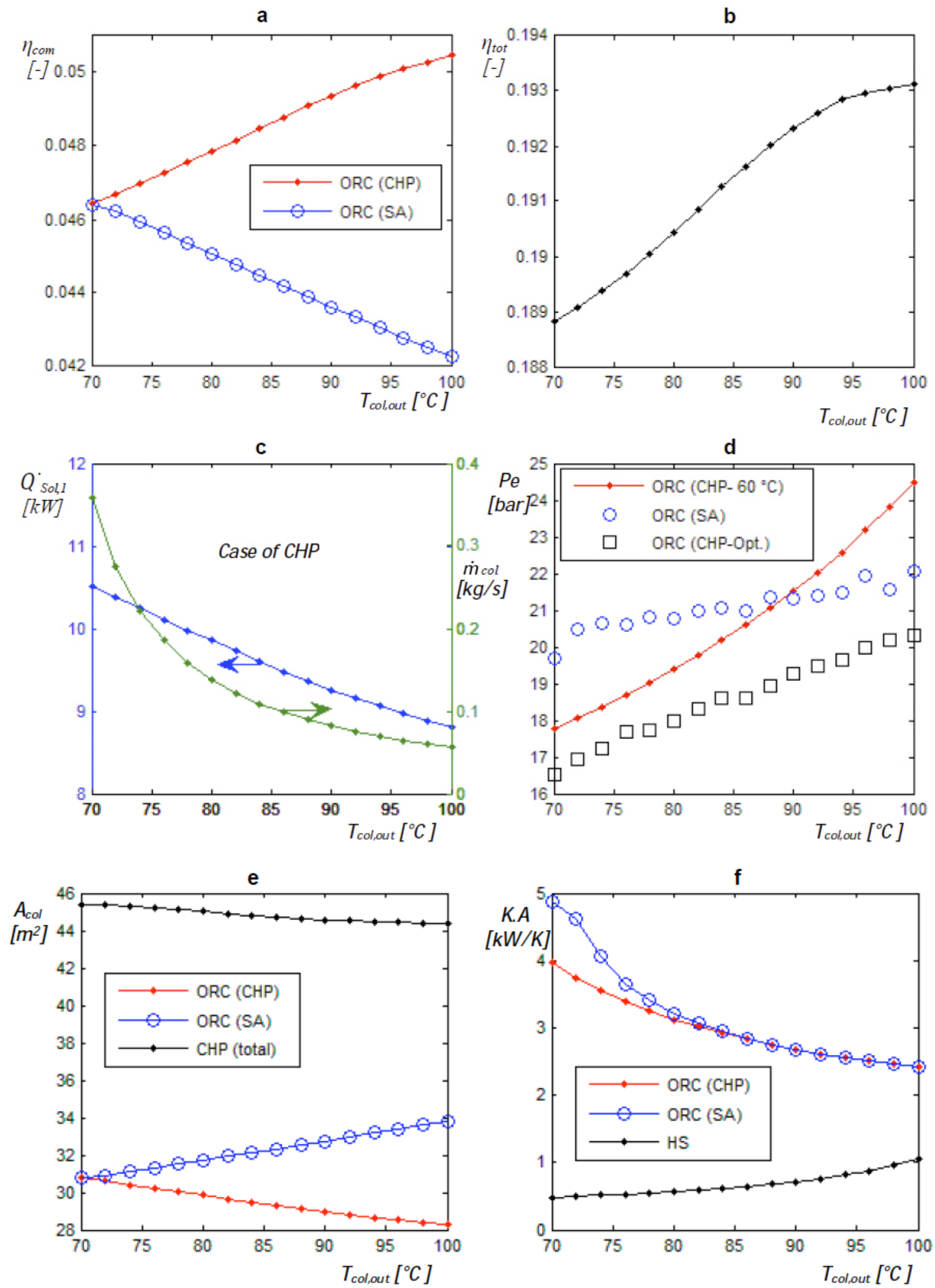

Figure 5: a, b, c, d, e and f: Influence of the temperature available at flat collector outlet $\boldsymbol{T_{col,out}}$ on the different performances of the considered cycles at heating system terms of $\boldsymbol{f}$ = 5,$\boldsymbol{T_{sup}}$ = 60 °C and $\boldsymbol{T_{ret}}$ = 35 °C and at the other nominal conditions.

(Figure **5**, **c**). Therefore, it can be recognized that when operating the ORC in case of sole power production, the low driving temperatures have positive aspects regarding the energetic performance and collectors' areas and negative ones regarding the ORC dimensioning. While, high driving temperatures are preferable and have purely positive repercussion with respect to all aspects in case of ORC (CHP). As regards the entire CHP plant, the study puts in evidence that the total efficiency exhibits slight but progressive improvement and the related total collectors' areas display also modest decrement at the same heating system parameters versus raising this temperature (Figure **5**, **b** and **e**). The lone disadvantage in this case is increment of the heating system heat transfer capacities due to shortage of the mass flow rate of the heat carrier involved for the power unit at high heat source temperatures (Figure **5**, **c**), where the temperatures' differences between the fluids in heating system become smaller. Nevertheless, large heating system remains to calculate the lower expenses compared to the collectors and ORC components' prices. On the other hand, influence of the supply temperature appears clearly, where according to the (Figure **5**, **d**) the CHP system should work at higher pressures than the optimal ones to ensure reaching the supply temperatures intended, because at the optimal pressures the heat carrier will not enable temperatures corresponding the desired ones. Also, it can be noticed that the optimal and required pressures in all cases increase with raising the heat source temperatures; this means that these pressures are functions to this factor. Consequently, integrating ORC in CHP plant such ours significantly removes the performance contradiction between ORC and collector cycles, in particular at high heat source temperatures, when comparing to stand-alone power production. Thus, the high temperatures served by the flat collector are definitely desirable for fuelling such establishments including ORC within scope of these in- and outcomes.

4.1.4. Influence of the Ambient Temperature and the Related Condensation Temperature

Figure **6**, **a-f** highlights the evaluative characteristics under investigation versus the ambient temperature to show the optimization potential for different regions or weathers. Variety of ambient temperature along with the related condensation one in ORC leads to an optimum in the efficiencies and the collector areas, but at different locations in these curves due to similar reasons discussed previously (Figure **6**, **a**, **b** and **e**). The highest improvement ratio for generating the power unit will be accomplished in the so cold regions till app 16.3 % (at outset of the range), where the collector efficiency is minimal and ORC efficiency is maximal. Subsequently, this ratio will run short for the hot regions till app. 5 % (at end of the range), where the collector efficiency becomes maximal and ORC efficiency will be just the contrary within scope of our propositions. Furthermore, the simulations refer to converse tendency for the heat transfer capacities in ORC (CHP), where they will continuously increase due to increasing the heat flow through ORC involved for the power unit (Figure **6**, **c**) and have almost equal values to the optimal ORC (SA) despite of deviation of the influential pressures (Figure **6**, **d** and **f**). This means that improving the performance and abbreviating the collector field do not add extra expenditures on ORC (CHP) versus the stand-alone ORC scaled at the optimal status. Also, it can be concluded that this application exhibits advantages in the so cold regions more than the hot ones. Taking the total CHP plant into consideration, superiority of this utility turns up at averaged ambient temperatures corresponding somewhat the weather data adopted in this article (Germany weather), where these lie between 8-12 °C (Figure **6**, **b** and **e**).

Differently, the demand at heat transfer capacities for the same heat load in heating plant will always go smaller with ambient temperatures owing to the progressive augmentation of the mass flow rate of the collector fluid and thus the thermal energy carried (Figure **6**, **c** and **f**), where the temperatures' differences between the fluids will be greater especially at the return temperature side. Furthermore, it deserves to be noticed that delivering the heat load at desired supply temperature in CHP plant will not have any negative impacts at high ambient temperatures, where the actual and typical pressures are identical (Figure **6**, **d**). While, a remarkable divergence between these pressures' curves in range of the low temperature field of ambient is observed. Also, there exists an absolute interconnection between the two factors, where the typical pressures will go after a direct relationship with this temperature.

4.2. Analytical Comparison of the Series Proposed ORC-CHP Method (SM) and Common Method (CM) of using the Heat of ORC Condenser

For precise comparison, it is quite essential to be assumed that the same head load obtained by ORC condenser is to be gained from the heating system integrated after ORC as in our case at the same supply and return temperatures.

4.2.1. Supply Temperature Effect of the Heating Plant

Table **2** contains the evaluation indicators at their optimums at different supply temperatures for the both methodologies of yielding the heat load. Here, it is evident that the heating capacity available cannot be controlled when utilizing the condenser heat (CM), where it poses mounting trend with supply temperatures. This is associated with increasingly extreme request at the collector areas and ORC heat transfer capacities and slight deterioration of the total efficiency. While the series method (SM) seems to be more attractive regarding all evaluation parameters, where it shows lower cycles' requirements for the same

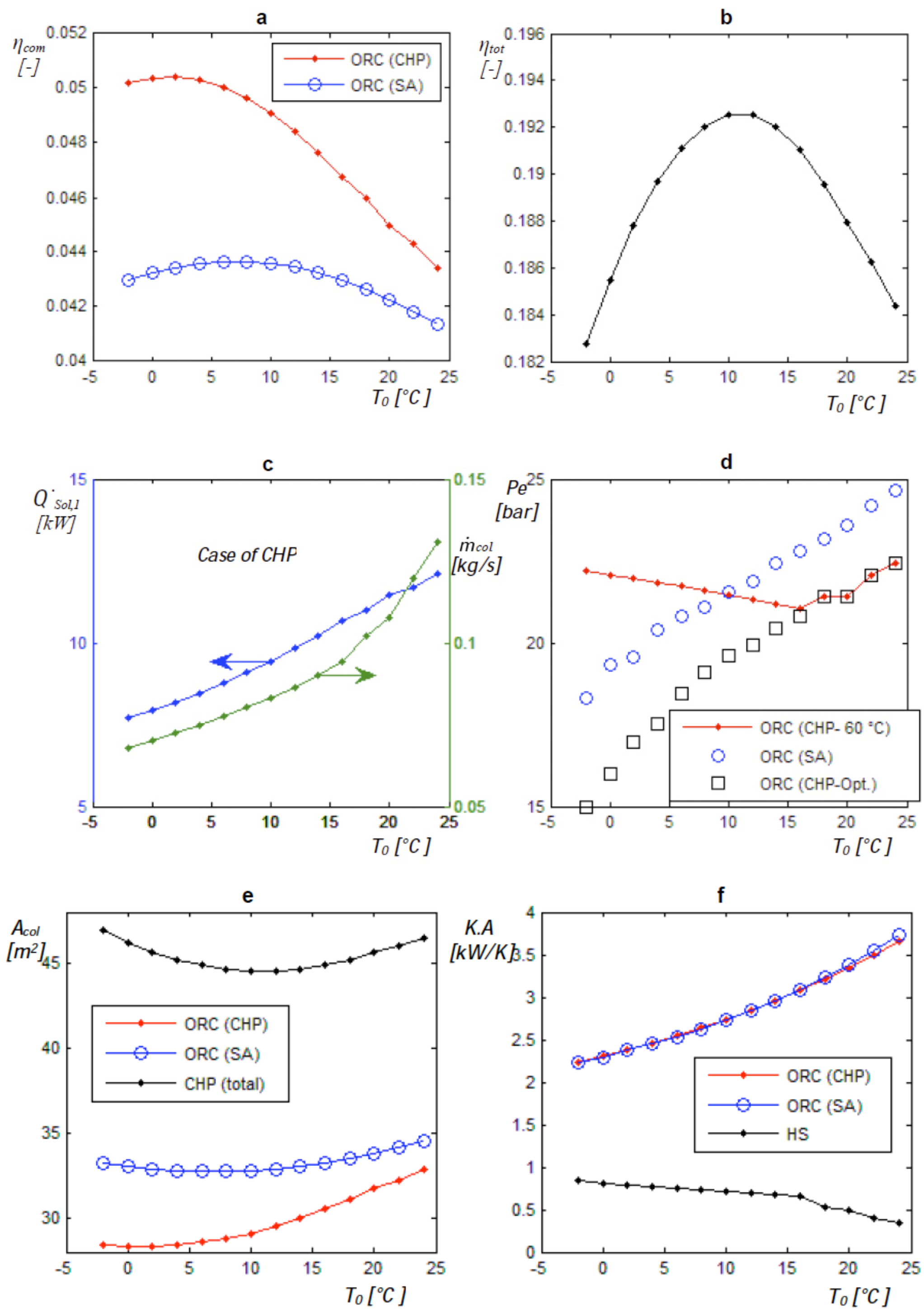

Figure 6: a, b, c, d, e and f: Influence of the ambient temperature available T_0 along with condensation temperature T_c on the different performances of the considered cycles at heating system terms of f = 5, T_{sup} = 60 °C and T_{ret} = 35 °C and at the other nominal conditions.

outputs and the other nominal conditions except for at the lowest temperature (40 °C), where only the collector areas are a bit bigger. As numerical example, at supply temperature 45 °C a reduction about 3.61 % for the collector area and app. 33.5 % for the total heat transfer capacities and also improvement app. 4 % for the efficiency can be registered for the same inputs and outputs in favor of SM versus CM. Considering the maximal supply temperature possible to be caught also makes the method (SM) more favorable in the whole range of the supply temperatures screened, where for example at the same conditions a supply temperature of 64. 8 °C instead of 40 °C can be set but this requires only extra heating plant areas (see Table **2**).

Table 2: Comparison of the both Methods for Gaining the Heat Load in ORC-CHP Plant at Variable Supply Temperature and Related CHP Ratio for Constant Return Temperature of T_{ret}=35 °C with Keeping the other Thermal Conditions Nominal According to Table 1

Indicator	Common Method (CM) (using the heat of ORC condenser)					Series proposed ORC-CHP Method (SM) (Series ORC-HS regarding the collector cycle)				
T_{sup} [°C]	40	45	50	55	60	40	45	50	55	60
A_{col} [m^2]	58.897	66.202	75.687	88.563	106.77	60.937	63.809	69.506	77.066	88.115
η_{tot}[-]	0.3155	0.3044	0.3042	0.3039	0.3037	0.3063	0.3166	0.3325	0.3500	0.3692
T_m, $T_{ORC,out}$ [°C]	85.894	88.665	88.728	88.792	88.859	67.819	68.989	70.852	72.923	75.190
$T_{sup,max}$ [°C]	40	45	50	55	60	64.819	65.989	67.852	69.923	72.190
$K.A^1_{ORC+HS}$ [kW/K]	5.539	6.098	6.575	7.394	8.656	3.817	4.053	4.442	4.988	5.814
$K.A^2_{ORC+HS}$ [kW/K]	-	-	-	-	-	6.760	7.134	7.858	8.811	10.184
Pe_{opt} [bar]	29	30	30	30	30	23.29	23.71	24.37	25.09	25.86
Q_{HS} [kW]	12.00	13.10	15.11	17.84	21.70	12.00	13.10	15.11	17.84	21.70
Tc in ORC [°C]	$T_{sup}+\Delta T_{pp}$					T_0+8				

$K.A^1_{ORC+HS}$ (at T_{sup}); $K.A^2_{ORC+HS}$ (at $T_{sup,max}= T_m - \Delta T_{pp}$).

Furthermore, enhancing the operation characteristics through the method (SM) compared to the other (CM) will be more significant at high supply temperatures, where the parametric optimization will be greater.

4.2.2. Collector Outlet Temperature Effect

Selecting the supply and return temperature constant at rather low levels to avoid the high CHP ratio along with keeping the other parameters fixed, analytically comparing the two methods at the optimal status versus variable solar heat temperature is tabulated in Table **3**. This table confirms a fact that because of improving the thermal efficiency of ORC with the heat source temperature, the condenser heat used for heating purposes (CM) will be lower at the high temperatures existed by collector. Therefore, all the indicators become lower with increasing this factor. This description can somewhat generalize on the series configuration (SM) at the same conditions. However, the comparative studies show that the system (SM) is totally superior in range of the high heat source temperatures within scope of this study, while the other way (CM) is more reasonable regarding only the collector area and the total efficiency within the low temperatures till 85 °C. For example, at the temperature 85 °C it is noted that the collectors will

Table 3: Comparison of the both Methods for Gaining the Heat Load in ORC-CHP Plant at Variable Collector Outlet Temperature and Related CHP Ratio for Constant Supply and Return Temperature T_{sup}=45 °C / T_{ret}=35 °C with Keeping the other Thermal Conditions Nominal According to Table 1

Indicator	CommonMethod (CM) (usingthe heat of ORC condenser)							Series proposed ORC-CHP Method (SM) (Series ORC-HS regarding the collector cycle)						
$T_{col,out}$ [°C]	70	75	80	85	90	95	100	70	75	80	85	90	95	100
$A_{col,tot}$ [m^2]	74.22	66.42	65.02	65.97	66.21	67.42	68.52	85.38	75.38	70.35	67.57	63.81	64.41	64.27
η_{tot}[-]	0.452	0.416	0.379	0.343	0.304	0.293	0.279	0.393	0.367	0.350	0.336	0.316	0.307	0.298
T_m, $T_{ORC,out}$ [°C]	69.77	74.67	79.50	83.77	88.66	86.37	84.78	59.49	61.66	64.31	67.03	68.98	72.18	75.35
$T_{sup,max}$ [°C]	45	45	45	45	45	45	45	56.49	58.66	61.31	64.03	65.98	69.18	72.35
$K.A^1_{ORC+HS}$[kW/K]	10.90	8.88	7.82	6.86	6.09	4.60	3.95	6.91	5.74	4.99	4.48	4.05	3.82	3.58
$K.A^2_{ORC+HS}$[kW/K]	-	-	-	-	-	-	-	11.33	9.56	8.56	7.90	7.13	6.93	6.67
Pe_{opt} [bar]	19.7	22	24.5	27	30	30.5	31.5	16.71	18.1	19.82	21.77	23.71	26.39	29.53
Q_{HS} [kW]	22.49	18.35	16.26	14.86	13.11	12.85	12.41	22.49	18.35	16.26	14.86	13.11	12.85	12.41
Tc in ORC [°C]	$T_{sup}+\Delta T_{pp}$							T_0+8						

$K.A^1_{ORC+HS}$ (at T_{sup}); $K.A^2_{ORC+HS}$ (at $T_{sup,max}= T_m - \Delta T_{pp}$).

have a small difference with only 2.4 % and the total efficiency only 2% in favor of CM, while the concept (SM) remains to have advantages versus the other at this temperature (85 °C), where heat transfer capacities of ORC and HS will be shortened up to 29.5 % confronted to CM. Moreover, the opportunity for high supply temperatures stands always for availability at unchanged solar collector areas and ORC parameters, where supply temperature till 64 °C can be reached, but with attention to the heating system sizing as mentioned previously.

4.2.3. Ambient Temperature Effect

When changing the ambient temperature at the same nominal terms adopted previously, it is apparent that gaining the heat load through ORC condenser (CM) is not so feasible at the so cold weathers, whereas the series principle (SM) seems to be completely efficient till ambient temperature 10 °C (suitable for Germany weather) Table **4**. For example at 5 °C, saving ratios of 11.19 % in collectors and 36.45 % in ORC and HS can be obtained in favor of the latter (SM) versus the former (CM). While running the system CM in hot regions will make it profitable versus the plant SM with regard to the efficiency and collectors, while the latter (SM) still preserve, at these conditions, many preferences such as lower demand an ORC and HS sizes along with offering high supply temperatures, when necessity, (between 60.9 and 64.5 °C) without extra dimensions for collector and ORC. Another important observation is to be mentioned that the ambient temperature does not affect the operation modus of ORC, integrated in the plant CM, from the temperature 5 °C due to constancy of the working pressure, where there no longer exists a correlation between the condensation in ORC and the surroundings on the contrary of the other system (SM).

Based on the last three tables, it is apparent that it is so difficult to create or calibrate a constant CHP ratio in case of employing the condenser heat for heating targets even when aiming to reduce the heat source losses because the latter enforces extra requirements. The real problem arises when the heat rejected in condenser is not entirely used at the desired supply temperatures. The absolute dependence of the heat delivery to the heat consumer on ORC is the main reason for inflexibility of this kind of plants because the investment costs such as the collector field and condensation unit, which works as heating plant, correlate exclusively to ORC. While, these negative appearances are less severity in the series evolution, where the relation between CHP ratio and the other conditions either in ORC or in heating plant is almost absent at reasonable CHP ratios and supply temperatures lower than the one recorded at the optimal pressure (see the last figures). Furthermore, outstanding performances in a wide range of the terms required for the heat consumer or presented by the heat source or sink turn up in this development in comparison to the other. In addition to that, it enables far higher supply temperatures without adding any changes or prerequisites on the solar field or on ORC evaporator and condenser at all conditions investigated. Thus, the latter grants it a priority for application especially for industrial targets which

Table 4: Comparison of the both Methods for Gaining the Heat Load in ORC-CHP Plant at Variable Ambient Temperature and Related CHP Ratio for Constant Supply and Return Temperature T_{sup}=45 °C / T_{ret}=35 °C with Keeping the other Thermal Conditions Nominal According to Table 1

Indicator	Common Method (CM) (using the heat of ORC condenser)							Series proposed ORC-CHP Method (SM) (Series ORC-HS regarding the collector cycle)						
T_0 [°C]	-5	0	5	10	15	20	25	-5	0	5	10	15	20	25
$A_{col,tot}$ [m^2]	100.89	86.36	74.08	64.50	57.29	51.67	47.18	86.36	77.69	65.79	63.38	61.86	61.45	62.86
η_{tot}[-]	0.249	0.276	0.272	0.312	0.351	0.390	0.427	0.291	0.307	0.306	0.318	0.326	0.328	0.321
T_m, $T_{ORC,out}$ [°C]	74.25	77.62	88.66	88.66	88.66	88.66	88.66	74.42	73.00	69.85	68.76	67.48	65.95	63.90
$T_{sup,max}$ [°C]	45	45	45	45	45	45	45	71.42	70.00	66.85	65.76	64.48	62.95	60.90
$K.A^1_{ORC+HS}$[kW/K]	4.86	4.94	6.09	6.09	6.09	6.09	6.09	3.82	3.90	3.87	4.10	4.38	4.76	5.34
$K.A^2_{ORC+HS}$[kW/K]	-	-	-	-	-	-	-	7.91	7.74	6.97	7.17	7.40	7.71	8.20
Pe_{opt} [bar]	24	25.5	30	30	30	30	30	26.04	25.44	24.2	23.58	22.83	21.9	20.65
Q_{HS} [kW]	16.58	15.70	13.10	13.10	13.10	13.10	13.10	16.58	15.70	13.10	13.10	13.10	13.10	13.10
Tc in ORC [°C]	$T_{sup}+\Delta T_{pp}$							T_0+8						

$K.A^1_{ORC+HS}$ (at T_{sup}); $K.A^2_{ORC+HS}$ (at $T_{sup,max}= T_m - \Delta T_{pp}$).

consume hot water. Therefore, it is advisable that the concept of utilizing the condenser heat for heating purposes is to be replaced with this series concept from last points of views, where it is proven by the last simulations that expanding the working fluid as much as possible in ORC turbine and extra cooling the heat carrier after ORC is more feasible in case of exploiting the low-temperature solar heat collected by flat collectors.

5. CONCLUSIONS

In this paper, parametric study of a series concept of ORC-CHP unit fuelled by solar energy captured by the simple flat collectors for the low-temperature applications has been conducted. Simultaneously, an analytical comparison between this concept and the single ORC, and also with the common ORC-CHP principle, which proposes benefit of the condenser heat, has been also carried out. Within scope of this study, the simulation results showed that for the power unit produced, an evident downsize of the contradiction between the performances of ORC and collector cycle could be achieved through the series ORC-CHP configuration, where improvement rates between app. 13.6- 24 % were obtained. Moreover, meeting the thermal consumptions at the heat users did not extremely keep away the ORC heat exchange capacities from the nominal dimension estimated at the single case. The notably attractive features of this ORC-CHP evolution versus the sole ORC appeared at low ambient temperatures and high heat source temperatures and CHP ratios, while high supply and return temperatures affect negatively. On the other hand, distinctive performances of this unit versus the usualutilization of the condenser heat were registered for a wide range of terms, especially when considering the need at high supply temperatures without extra demand at collector areas and ORC plant.

Finally, it is strongly recommended to install this technology instead of the single power production or the technique of using the condenser heat when exploiting the low-temperature solar heat captured by flat collector and employing ORC as power unit. As future work, other working fluids will be screened in this system for optimization potentials along with carrying out detailed economic calculations.

NOMENCLATURE

ACRONYMS

CHP = Combined Heat and Power

ORC = Organic Rankine Cycle

SA = Stand-Alone

ORC (CHP) = Organic Rankine Cycle operating in the Combined Heat and Power plant

ORC (SA) = Organic Rankine Cycle operating as Stand-Alone plant

HS = Heating System

Opt. = Optimum

CM = Common Method

SM = Series Method

Symbols

T = Temperature *[°C]*

P = Pressure *[bar]*

$\dot{W}$ = Power *[kW]*

$\dot{m}$ = Mass flow rate *[kg/s]*

$\dot{Q}$ = Heat flux *[kW]*

h = Specific enthalpy *[kJ/kg]*

A = Area *[m²]*

C_p = Isobaric, specific heat capacity *[kJ/(kg.K)]*

k = Heat transfer coefficient *[kW/m².K]*

f = Ratio [-]

G = Global irradiation *[kW/m²] or [W/m²]*

a& b = Heat loss coefficients *[W/m².K] & [W/m².K²]*

Greek Letters

Δ = Difference [-]

η = Efficiency [%]

Subscripts

Sol = solar

c = condensation or condenser

e	=	evaporation or evaporator, electric
com	=	combined
t	=	turbine
p	=	pump
is	=	isentropic
th	=	thermal
tot	=	total
op	=	optical
col	=	collector
sup	=	supply
ret	=	return
in	=	inlet
out	=	outlet
pp	=	pinch point
m	=	mean
0	=	reference for ambient
1..4	=	state points
max	=	maximal
log	=	logarithmic

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