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Selection Maps For ORC And CO_2 Systems For Low-Medium Temperature Heat Sources

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Abstract

Low-medium temperature heat sources in the range 5 - 50 MW_{th} are made available by many industrial fields but they may also be of interest for biomass and solar energy applications. ORC has been proposed in the last 20 years as a reliable solution for the exploitation of these energy sources since the alternative represented by steam cycles leads to an inefficient conversion of such small available thermal powers. However, the use of organic fluids involves a number of safety and environmental issues, either related to fluid flammability (for hydrocarbons) or to their high-Global Warming Potential (for halogenated fluids), and of limitations to the achievable cycle maximum temperature, due to fluids thermal decomposition. To overcome these limitations, CO_2-based transcritical and supercritical cycles have been proposed, in recent years, as a viable option for waste heat recovery applications. The present work aims to present a fair comparison between CO_2 and ORC power plants for waste heat recovery applications.

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Keywords: ORC, CO_2, carbon dioxide, waste heat recovery

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1. Introduction

Nowadays, ORC is the most reliable option available on the market for the exploitation of low-medium temperature heat sources in a large range of power outputs. In the last 20 years, the ORC technology has been able to penetrate the market in a more effective way with respect to other technologies like Kalina cycle and Goswami cycle [1] or thermoacoustic Stirling engine [2] reaching more than 2.8 GW of installed power and more than 1700 installed plants. ORC field spans from renewable energy sources like geothermal, biomass and solar applications to waste heat recovery from industrial processes or engines flue gases [3]. Such energy sources are characterized by either a nearly constant or a variable temperature profile. In recent years, the use of CO_2 cycles for power production has gained a large interest from both the Industry and the Scientific Community. Supercritical CO_2 cycles are typically envisaged for large and high-temperature power plants coupled with solar tower technology and nuclear field [4,5,6]. In these fields of application, in fact, CO_2 plants can compete with conventional steam cycles thanks to their smaller investment cost, more compact turbomachines, simpler plant arrangement, higher flexibility. A number of experimental plants have been design and tested in recent years with a focus on solar applications [7] and nuclear energy field [8,9]. A 25MW plants is in construction in Texas with a turbine manufactured by Toshiba [10]. The applications where ORC can compete with steam cycles has been named “grey zone” by the ORC community underling that the choice between the two power systems is by no means self-evident. Besides the already attested application of high-temperature CO_2power cycles, this technology may be also considered as a viable solution for the exploitation of medium temperature heat sources, competing with ORC. In the USA this concept is investigated by Echogen which manufactures supercritical CO_2 cycles for waste heat recovery applications. Accounting for CO_2power cycles within this technological comparison, a new “grey zone” actually emerges. With that respect, the purpose of this work is to present performance maps to enable the straightforward thermodynamic comparison and easier selection between ORC and CO_2cycles, in a wide range of applications where they may compete. The analysis is thus carried out considering both constant and variable-temperature heat sources, with a maximum heat source temperature ranging from 200 to 600°C. Each point of these maps provides the optimal performance of both ORC and CO_2 power cycles, considering their most suitable configurations

As regards ORC, they are modelled as subcritical Rankine cycles investigating the use of 47 different pure working fluids. For CO_2 cycles, two plant layouts are investigated: simple recuperated and re-compressed regenerative configurations. In all cases, the expander efficiency is evaluated with a correlation which accounts for the effect of volume ratio and last stage size parameter [11]. The analysis is performed considering both high- and low-temperature heat sinks, representative of ambient air and water. In the first case, heat is rejected to the ambient with an air-cooled condenser, limiting the CO_2 cycle to a non-condensing Brayton configuration while, in the second case, the availability of water enables condensation of CO_2 thus allowing the less-power-consuming compression of highly-dense cool CO_2 as well as low temperature heat rejection.

Nomenclature and acronyms

η	Efficiency	SP	Turbine size parameter, m
T	Temperature	Vr	Turbine volume ratio
P	Pressure	HTF	Heat Transfer Fluid
Q	Heat	SCO2	supercritical CO2
HE	Heat Exchanger	CSP	Concentrated Solar Power
HR	Heat Rejection	ECO	Economizer
PHE	Primary Heat Exchanger	EVA	Evaporator
Rec	Recuperator	SH	Superheating

2. Methodology

The comparison between ORC and CO_2 power cycles is performed in this work considering a heat sources of 30 MW_{th}, characterized both by different maximum temperatures $T_{s,max}$(comprised between 200°Cand 600°C) and by different cooling grades ΔT% (varying from 0 to 100%). The cooling grade is defined as the ratio between the maximum allowable temperature variation of the heat source and the maximum temperature difference given by $T_{s,max} - T_0$. Minimum temperature of the heat source can be thus calculated as:

$$T_{s,min} = T_{s,max} - \Delta T_{\%}(T_{s,max} - T_0)$$

This approach allows to investigate a wide number of cases, from completely isothermal heat sources ($\Delta T_{\%}$= 0 %) to hot streams undergoing complete cooling ($\Delta T_{\%}$=100%). Results provided by this work aim to cover all the low-medium temperature heat sources exploitable in waste heat recovery, biomass and solar applications. Figure 1-a shows the minimum temperature of the heat source, as a function of $T_{s,max}$and ΔT% while Figure 1-b reports the power output attainable with a reversible process, namely using a Carnot cycle for isothermal heat sources and trapezoidal or triangular Lorenz cycles for the other cases. These minimum temperature values should be necessarily considered to fix the inferior limit to the cooling grade when the available heat source cannot be cooled down to very low temperatures. This is the case, for example, of combustion flue gases, which cooling limit is imposed in order to avoid the condensation of acid compounds; similar limitations to the cooling grade of the heat source also regard CSP and biomass applications, where HTF fluid minimum temperatures are limited by the necessity to avoid the excessive increasing of fluid viscosity or pouring issues.

a. Heat source maximum temperature,°C

$T_{min,s}$ °C — Heat source cooling grade	200	250	300	350	400	450	500	550	600
0%	200	250	300	350	400	450	500	550	600
20%	163	203	243	283	323	363	403	443	483
40%	126	156	186	216	246	276	306	336	366
60%	89	109	129	149	169	189	209	229	249
80%	52	62	72	82	92	102	112	122	132
100%	15	15	15	15	15	15	15	15	15

b. Maximum reversible power, kW

W_{rev} MW	200	250	300	350	400	450	500	550	600
0%	11.7	13.5	14.9	16.1	17.2	18.0	18.8	19.5	20.1
20%	11.0	12.7	14.1	15.3	16.4	17.3	18.1	18.8	19.4
40%	10.1	11.8	13.2	14.4	15.4	16.3	17.1	17.8	18.5
60%	9.2	10.7	12.1	13.3	14.3	15.2	16.0	16.7	17.3
80%	8.1	9.5	10.8	11.9	12.8	13.7	14.5	15.2	15.8
100%	6.8	8.1	9.1	10.1	10.9	11.7	12.4	13.0	13.6

Figure 1 - a. Minimum temperature of the heat source and b. maximum reversible power attainable from the heat source, as a function of the maximum temperature of the heat source and of the cooling grade for water cooled applications .

It is worth observing that, depending on the power cycle, the heat entering the cycle itself may be lower than the maximum one made available by the heat source (30 MW_{th}) because of the presence of the recuperator (CO_2 cycles) or because of an evaporation temperature considerably higher than the minimum temperature of the heat source (ORC cycles). In such cases, the heat source is cooled up to a minimum temperature higher than the one reported in Figure 1 and, in particular, dependent both on the fluid temperature in inlet to the Primary Heat Exchanger ($T_{in,PHE}$) and on the PHE approach point temperature difference ($\Delta T_{ap,PHE}$).For each heat source condition defined by $T_{s,max}$ and ΔT% (i.e. for each condition in the grid of Figure 1), ORC and CO_2 power cycles are optimised and compared, obtaining a *performance map* that presents the resulting optimal cycle, among the ones considered in this work.Each performance map refers to a specific heat sink condition and in this work the analysis is repeated for two different heat sinks, either available at 15 °C (as representative of cooling water from a borehole, a river or a lake) or at 30 °C (representative of ambient air), thus resulting in the production of two performance grids. The selection of two different heat sink temperatures allows comparing ORC power systems with both supercritical and transcriticalCO_2configurations respectively resulting from the availability of either high-temperature (T_0 = 30°C) or low-temperature (T_0 = 15°C) heat sinks. Heat rejection unit consumption has been accounted differently according to the cooling medium: fan consumption for the air cooled condensed ORC is calculated as described in [12] while for CO_2 the fan power is set equal to 0.5% of thermal power rejected to the environment. In case of cooling water, the water pump consumption is determined assuming a fixed temperature rise in the unit equal to 7°C and an overall pressure drop on the water loop equal to 1.5 bar. Table 1 summarizes the assumptions common to ORC and CO_2 configurations, regarding their interaction with heat source and heat sink and the design of the main components.In the following sections 2.1 and 2.2 we describe in more details the configurations of ORC and CO_2 power cycles, providing the applied specific assumptions.

Table 1. General assumptions for ORC and CO_2 configurations.

	CO_2		ORC	
Heat sink temperature	15°C water cooled	25°C air cooled	15°C water cooled	25°C air cooled
Minimum working fluid temperature	25°C	40°C	25°C	40°C
$\Delta T_{ap,PHE} \Delta T_{pp,\ PHE} \Delta T_{pprec}$	10°C		5°C	
$\Delta T_{subcooling}$	-		5°C	
Δp(or, whether relative, $\Delta p/p_{in}$) PHE	2%		50kPa (ECO), ΔT=1°C (EVA), 2% (SH)	
Δp (or, whether relative, $\Delta p/p_{in}$) REC	2% (hot side), 2% (cold side)		2% (hot side), 50 kPa (cold side)	
Δp(or, whether relative, $\Delta p/p_{in}$) HR	2%		2% (desuperheating), ΔT=0.5°C (condensation)	
Compressor/pump hydraulic efficiency	0.85		0.75	
Generator electrical efficiency	0.97			
Mechanical efficiency	0.97			
Pump electrical motor efficiency	0.97		0.97	
Auxiliaries consumption loss	2%			

2.1. CO_2 plants

CO_2 plants are based on a cycle where the working fluid is compressed, heated, expanded and eventually cooled down in a closed loop configuration. Differently from open gas cycles (Combustion Gas Turbines) [13], the use of the recuperator is always profitable from a thermodynamic perspective since it allows to increase the plant efficiency by recovering a relevant fraction of the thermal power available at turbine discharge for the pre-heat of the compressed fluid. We thus studied only CO_2 cycles provided with a recuperator, which configuration is reported in Figure 2-a. However, the design of a highly-efficient recuperative CO_2 plant is limited by the presence of marked real gas effects at low temperature and high pressure which enhance the difference between the specific heat capacity of the cold and the hot streams, leading to a less efficient heat transfer process characterized by higher temperature differences. In order to mitigate this penalization, the recompressed cycle proposed by Angelino [14] has been considered in this work; it is show in Figure 2-b. Differently from the simple cycle configuration, the internal recovery process of recompressed cycles takes place in two heat exchangers, where the cold side of the low-temperature recuperator is characterized by a reduced CO_2 mass flow to better balance thermal heat capacity of hot and cold regenerator sides.

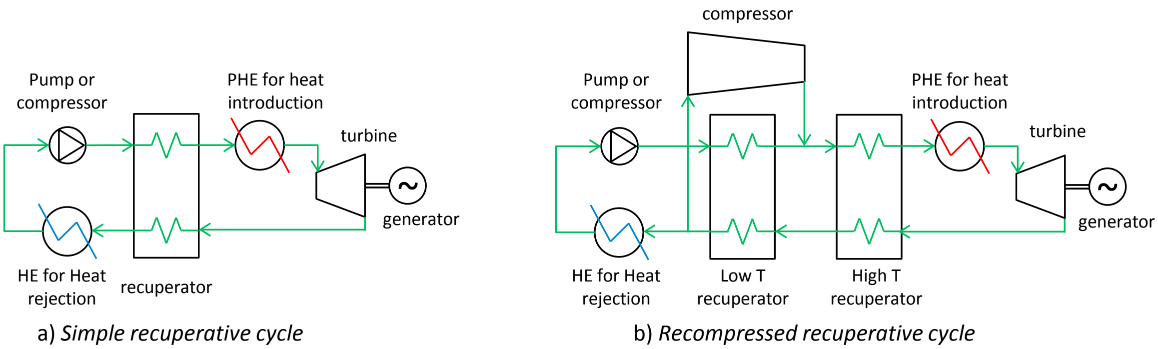

Figure 2. CO_2 cycle configurations considered in this work. In these figures, representations regard the water-cooled cases (with pumps, besides compressors, for pressure increase); air-cooled cases are the same but compressors are always present instead of pumps.

In case of water-cooled cycle, the CO_2 plant is condensative, while, for the air-cooled configuration, the cycle minimum temperature is above the critical temperature of CO_2, thus resulting in a supercritical configuration. The optimal design for both configurations is defined by maximising the power output. The set of optimizing variables changes depending on the cycle configuration and the available heat sink. Maximum pressure of the cycle is optimized at all times while the minimum pressure is varied only for air-cooled systems. It is also specified that the superior limit fixed for maximum pressure is 300 bar. In addition, also the split ratio (equal to $m_1/m_9$with reference to points indicated in Figure 2-b) is optimized for recompressed cycles. Turbine efficiency is computed with the equation presented in [11] as function of the turbine Size Parameter (SP) and of the volume ratio (V_r), considering a three-stage turbine. For the calculation of plants power output (simple and recompressed configurations) we considered the compressors connected to the turbine shaft and the pump, which is present in case of water-cooled cases, connected to an electrical motor.

2.2. ORC plants

For each considered heat source and heat sink condition, the best ORC plant is found screening 47 potential working fluids and investigating four different cycle configurations (namely superheated and saturated cycles, with or without recuperator) [15]. Refprop 9.1 [16] is used for the calculation of working fluids thermodynamic properties. Among the 47 working fluids there are 15 alkanes, other 8 hydrocarbons, 16 halogenated fluids (with hydrogen atoms partially or totally substituted by fluorine atoms) and 8 siloxanes. The set of working fluid candidates spans from low critical temperature and relatively low complexity fluids (R143a, propane, R134a) up to heavy, complex and high critical temperature compounds (benzene, MDM, decane). The complete list of considered fluids is reported in Figure 3 with their critical parameters and the operating maximum temperature. This last parameter is strongly related to the fluid thermal stability limit above which molecules decompose in lighter compounds that change the fluid thermodynamic properties and they also can form solid particles that could damage the turbine blades and increase fouling on the heat exchangers surfaces. For each specific fluid, we consider this limit as the maximum value of temperature of the experimental dataset used to calibrate the equation of state.

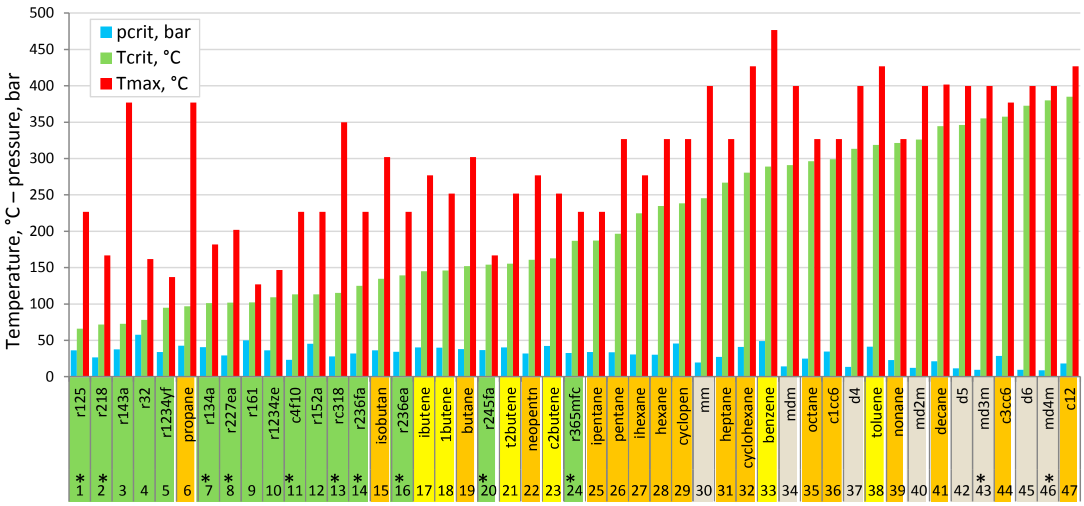

Figure 3.List of the 47 working fluid candidates for ORC applications sorted from lower to higher critical temperature fluid and divided in 4 groups and fluids labeled with (*) are non-flammable.

In this research, we limit the analysis to subcritical cycles since the vast majority of the installed ORC plants are based on this cycle layout. Supercritical ORC may also be advantageous, however this configuration has not been considered here since it has only been used in a limited number of plants: geothermal installations by TAS in USA and in the experimental activity carried out by ENEL in Livorno or, for instance, in waste heat recovery applications. We decided to focus this work on configurations being well-established in the current state-of-the-art of ORC technology, based on either superheated or saturated cycles possibly provided with a recuperator. Each plant is optimized from a thermodynamic perspective varying the evaporation and condensation temperature and the superheating degree, for superheated cycles. Turbine efficiency is computed with the correlation presented in [11] as function of SP and V_r considering a three stages turbine, as for CO_2-cycles. For complex molecules having an overhanging saturation vapor line, the maximum evaporation temperature for saturated cycles coincides with the temperature corresponding to the maximum saturated vapor entropy, in order to avoid two-phase flow expansion in the first turbine stage. Differently, for superheated cycles the maximum evaporation temperature is set 10°C below the critical temperature and a minimum value of superheating equal to 5°C is considered. In case of simpler molecules, characterized by non-overhanging saturation vapor lines, we consider a penalization of turbine efficiency due to the possible presence of liquid droplets: for vapor quality below 0.95 we accounted for 1 percentage point of penalization for each 1% of liquid fraction. To prevent from air leakages resulting from the achievement of sub-atmospheric minimum pressures by the organic working fluid, the minimum condenser pressure is limited to 1 bar. It is recalled that in ORC, in fact, the removal of non-condensable gases by venting them in the environment is not possible by using a deaerator like in steam cycles, for both safety and environmental as well as economic reasons. This constraint penalizes especially high-critical temperature fluids. To quantify the efficiency lost by the imposition of such a limit, the effect of lower condensing pressures lower bound on ORC efficiency is discussed.

3. Results and discussion

Results are presented in *performance maps* for an easy comparison of the maximum efficiency attainable either with ORC power plants or with CO_2 cycles, for each heat sink and heat source condition considered. For ORC systems, the maximum power, the optimal plant and the best fluid are presented while, for CO_2 systems, the maps propose the use of either simple regenerative or recompressed cycle, depending on which of the two resulted to be the optimal solution.

3.1. Water-cooled condensed cycles

The first comparison is carried out comparing water-cooled $CO_2$and ORC cycles. In the following, we describe the results obtained for these two cycles, in the mentioned order. In water-cooled CO_2cycles, CO_2 condensates, leading to a reduced pump consumption and to an increased plant efficiency. Figure 4-ashows the power output of optimal CO_2 cycles from temperatures of the heat source higher than 250°C. Power output increases with the maximum temperature of the heat source, thanks to the resulting increased difference between turbine production and pump consumption. Moreover, their design is favored by heat sources with a lower temperature variation. This is justified by the small CO_2 temperature drops along the expansion, on the one side, and the efficient internal heat recovery within the recuperative process, on the other, that make the exploitation of low thermodynamic quality heat sources the less convenient option. In this case, in fact, the heat recovered by recuperative process is reduced, as well as the input heat power exploited by the available heat source, leading to a penalization of the power output. From our calculations, it results that the optimal maximum cycle pressure coincides with the upper bound value (300 bar) at all times leading to a non-trivial design of heat exchangers that must withstand to a large pressure difference. Recompressed cycles (R) are more efficient for heat source temperatures higher than 500°C and $\Delta T_{\%}$ higher than 0.6; however, the attainable increase of power output with respect to the simple recuperative cycle (S) is often limited. In figure 4-a we highlight, with a line pattern, the region where the performance of the two plants differs by less than 2%. Figure 4-b depicts the maximum power output attainable with ORC power plants: net power ranges from 3.8 MW to 9.5 MW respectively for low-temperature heat sources with high temperature grades and for isothermal heat sources with temperatures higher than 500°C. The best cycle layout is always the superheated one and the use of the recuperator results to be convenient in all cases, except those marked with the asterisk (*) in Figure 4-b, namely low-temperature heat sources with very low-minimum temperatures. A small optimal superheating degree is obtained in particular for low-temperature heat sources with small $\Delta T_{\%}$ meaning that also saturated cycles are appropriate in these cases. The resulting optimal fluid depends on heat source characteristics: the higher the maximum average temperature of the heat source (namely high-$T_{s,max}$ and low-$\Delta T_{\%}$), the higher the critical temperature of the optimal fluid. On the contrary, for heat sources with higher temperature variations, it is preferable to adopt fluids with a lower critical temperature because they allow to condense at low temperatures while maintaining limited volume ratios and high turbine efficiency. For maximum temperatures of the heat source higher than 500 °C the optimal ORC plant does not change since the maximum temperature of the cycle is bounded by the thermal stability limit of the fluid, rather than by the heat source temperature profile. Figure 4-c represents the regions where it is preferable to adopt a CO_2 cycle instead of an ORC. Also in this case, the shaded pattern highlights the area where the relative power difference between the two systems is below 2%. ORCs are able to produce higher outputs than CO_2 cycles in a large range of heat sources and $\Delta T_{\%}$ and are particularly suggested for heat sources having temperatures below 400°C and low minimum temperatures.

3.2. Air-cooled condensed cycles

The same analysis is repeated for air-cooled cycles. For both CO_2 and ORC cycles the plant power output is reduced. The penalization is larger for CO_2 cycles since they cannot be anymore condensative, in fact the minimum CO_2 temperature is higher than its critical value (which is approximately 31°C). Results are similar to the previous case. Nevertheless, it is possible to observe, for CO_2 cycles, a more extended region where the use of simple recuperative cycles is preferable to recompressed recuperative cycles. As regards ORC, optimal fluids remain almost the same; however, for lower average temperature heat sources, it is convenient to adopt saturated non-recuperative cycles because the higher condensing temperature of these air-cooled configurations limits the use of the recuperator. As result, CO_2 is still competitive in some case but only for higher heat source temperatures (550°C and 600°C) and limited $\Delta T_{\%}$.

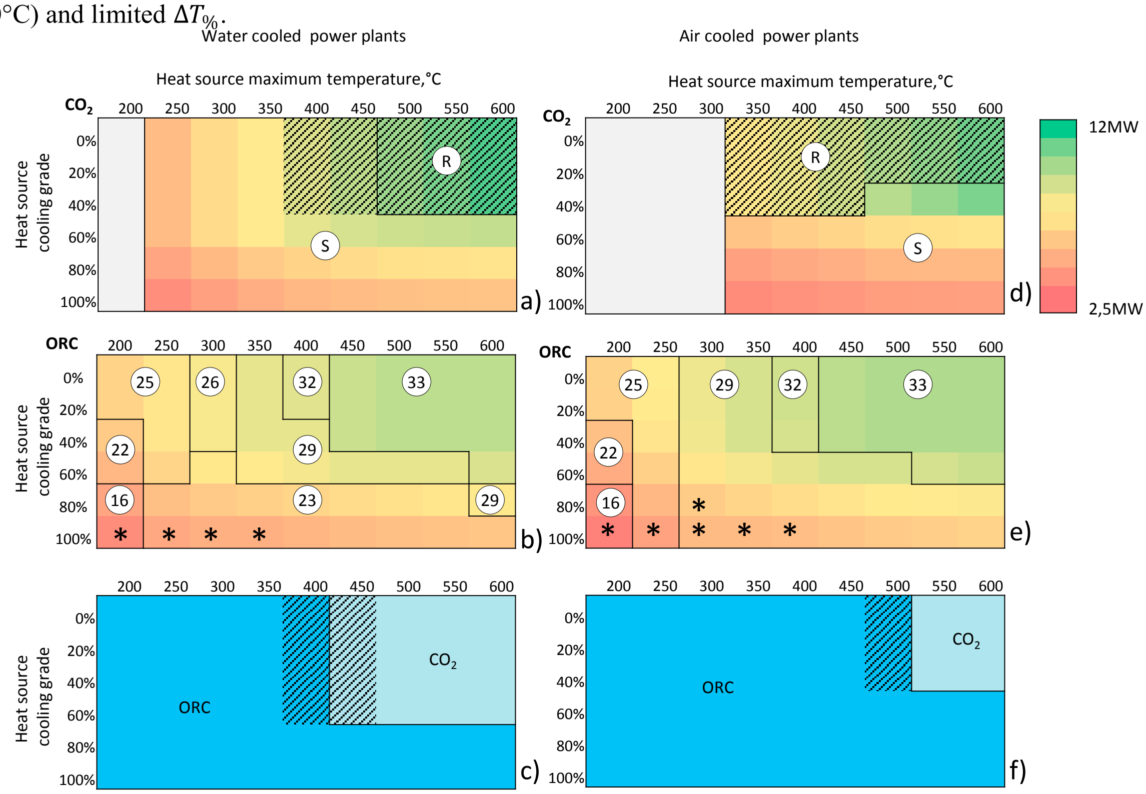

Figure 4. Performance maps of CO_2 (figures (a), (d)), of ORC (figures (b), (e)), of CO_2 versus ORC (figures (c), (f)) considering power plants for both water cooled (figures (a-c)) and air cooled systems (figures(d-f)). Labels (S) and (R) refer to simple recuperative and recompressed recuperative CO_2 cycles respectively. For ORC maps the numbered labels refer to the optimal working fluid with reference to figure 3. Cells with (*) marker refer to optimal solution without recuperator.

4. Conclusions

This work aims to compare performances of CO_2-power cycles and ORC for waste heat recovery applications. Performance maps are presented for both solutions, considering (1) heat sources characterized by different maximum temperatures ($T_{s,max}$ = 200°C – 600°C) and cooling grades ($\Delta T_{\%}$ = 0 – 100), (2) water-cooled (T_c = 15°C) and air-cooled (T_c = 30°C) cycles. The key-points of the analysis and main outcomes are briefly described below.

- In this work, CO_2 cycles have been designed both as simple recuperative and recompressed recuperative configurations, since they are recognized as the most simple and possibly the most suitable configurations for small to medium size plants. From the analysis it resulted that increasing cycle maximum pressure and temperature is profitable at all times while the cycle minimum pressure must be optimized considering the effect on both turbine efficiency and compressor consumption. The attested increase of power output, attainable with recompressed rather than simple recuperative cycles, is always below 2%.

- ORC plants can be designed according to different plant layouts and they can adopt a large number of working fluids. Optimization of both aspects is crucial, to obtain high-conversion efficiency and maximize power output. From the analysis reported in this paper, superheated recuperative cycles turn out to be the best solution for almost all the investigated cases with the exception of heat sources having a very high cooling grade. On the other hand, the optimal fluid critical temperature results to be strictly linked to the thermodynamic quality of the heat source.
- If water is available as cooling medium, ORC proves to be preferable to CO_2-cycles for low temperature heat sources (namely below 400°C) and for heat sources characterized by a high-cooling grade. These cases are representative of biomass and solar power plants where a loop of synthetic oil is used within the furnace or the solar field, with temperatures usually ranging from 150°C to 400°C. On the contrary, CO_2 cycles are promising for high-temperature heat sources characterized by a limited cooling grade. Those can found application in waste heat recovery form industrial processes or from Gas Turbines flue gases. Furthermore, both ORC and CO_2 air-cooled plants are characterized by reduced performances with respect to water-cooled solutions. However, the necessary use of supercritical CO_2-cycles, when air is available as cooling medium, enhances their penalization with respect to the one afflicting ORC, confining CO_2 plants in a very narrow region of the provided performance maps and highlighting the large advantages in adopting transcritical CO_2 cycles instead of supercritical ones.
- Two other aspects are highlighted: (i) considering a higher vacuum at ORC condenser allows to increase the performance of these cycles, weakening even more the competitiveness of CO_2 systems, (ii) excluding flammable fluids from the optimization of ORC systems makes CO_2-cycles the preferable solution even for heat source temperature higher than 350°C.

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