COMPARISON BETWEEN DIRECT AND INDIRECT ORCS FOR CSP APPLICATIONS

Damiano Perusi[1], Marco Astolfi[2]*

[1]Zuccato Energia
Via della Consortia, 2,
37127 Verona, Italy
d.perusi@zuccatoenergia.it

[2]Politecnico di Milano, Energy Department,
via Lambruschini 4
20156 Milano, Italy
*marco.astolfi@polimi.it

ABSTRACT

Concentrating solar power is still a niche market for ORCs counting several small applications but only few large scale plants. The main limit of solar ORCs with respect to PV systems is related to the complexity of this technology and the consequent higher specific cost. However, the possibility to adopt a thermal energy storage (TES) allows solar ORCs to provide dispatchable solar energy and to be possibly competitive with large PV systems integrated with electrochemical storage. The aim of this work is to define the best solar ORC power plant configuration for different maximum solar field temperatures by comparing conventional indirect power plants against non-conventional direct systems. Conventional indirect cycles consists of a solar field where the heat transfer fluid is heated up to maximum around 400°C, hot oil can be stored in the TES or used directly as heat input for the ORC generally designed as a subcritical saturated cycle. On the contrary, in direct cycles the working fluid flows directly into the solar field and stored in liquid phase. Hot working fluid is then throttled (partially or completely) and vapor fraction is expanded in a turbine. This configuration allows avoiding the HTF/working fluid heat exchanger possibly leading to a lower investment cost and higher efficiency. A Matlab code have been implemented in order to carry out the thermodynamic optimization of both indirect and direct solar ORCs. Indirect systems are optimized varying the evaporation and condensation pressures and the turbine inlet temperature while direct systems by varying both the condensation and the flash pressures. Both recuperative and non-recuperative configurations are investigated considering a large number of working fluids from Refprop database. The optimal working fluid is selected considering the system efficiency as figure of merit but optimal power plant selection considers also the size of the storage system. Respect to the scientific literature some novel aspects are implemented: (i) a model of the solar field in order to link pressure drops to fluid properties, (ii) a turbine efficiency dependent on the expansion volume ratio and (iii) an extensive sensibility analysis on different assumptions like minimum pressure and components performance.

1. INTRODUCTION

The exploitation of solar energy for power generation has been extensively studied in the last decades (Einav, 2004) (Galvez, 2010) making CSP a mature technology for the energy production providing renewable and dispatchable energy thanks to the use of thermal storage (Casati et al., 2013). Linear parabolic trough collectors and central towers are the two main adopted solutions and generally uses a proper heat transfer fluid (HTF) into the collectors/receiver and a steam cycle power plant as power unit. Only recently, for large scale and high temperature (mainly solar towers) applications the use of supercritical CO_2 as working fluid (and possibly as HTF) is investigated (Binotti et al., 2017) (Zhanga et al., 2008) thanks to the possibility to reduce turbomachinery dimensions and improve cycle flexibility. On the contrary, for small-medium scale CSP systems (from few hundreds of kW to some MW) the adoption of an Organic Rankine Cyle (ORC) is attractive thanks to high efficiency of this technology for this class of applications. Scientific literature is abundant on this topic with more than

twenty papers published in the last 10 years. Moreover, a consistent number of real size systems have been built and operated (Goswami et al., 2013) (Dickes et al., 2014) (Chambers et al., 2014).
Two main cycle configurations are reported in literature: the first one is a conventional indirect cycle where a loop of diathermic oil (or water+glicole for low temperature applications) is used as heat transfer fluid in the solar field while in the second one, so called direct cycle, the organic working fluid directly flows int the solar field collectors tubes (Li Zhi et al., 2017) (Casati et al., 2013). Both configurations are promoted as a viable solution for the exploitation of solar energy (although all the installations are based only on the indirect one) but a fair comparison among the two, based on consistent set of assumptions and methodology is still missing.

2. NUMERICAL CODE DESCRIPTION

In order to assess a fair comparison among direct and indirect ORC for CSP applications a dedicated numerical tool has been developed in Matlab and integrated with REFPROP 9.1 database (Lemmon et al. 2013). With respect to the studies available in scientific literature this numerical code allows to optimize the performance of different cycle configurations while considering the effect of expander performance and solar field pressure drop allowing for a fair comparison between different plant layouts. The next sections are focused on the description of the code and its features.

2.1 Investigated power plants configurations and assumptions

Considering the literature review two main classes of solar ORC cycles are considered: namely the indirect and direct configurations. Indirect configuration (Figure 1.a plant scheme, Figure 1.d T-s diagram) is the most studied and the only one adopted in real size power plants: it is based on a loop of thermal oil (Therminol VP-1 in this case) flowing into the solar field collectors and heated by concentrated solar power. Hot heat transfer fluid can be stored in a thermal energy storage or directly exploited by an ORC. ORC can be designed as a subcritical (saturated or superheated) or a supercritical cycle. Moreover, it may feature an internal recuperator allowing to partially pre-heat the liquid cooling down the hot vapors released by the turbine. In this work only subcritical cycles are considered since supercritical cycles in spite of their higher attainable performances are scarcely used on the market. The main advantage of this cycle configuration is the limited amount of organic fluid which inventory may represent a relevant cost if the working fluid is a safe compound. On the other hand, main drawbacks are represented by the need of the HTF loop and the cost of the additional heat exchanger (HTF/working fluid). Direct cycle configuration is based on the concept that the organic working fluid is also the HTF into the solar field that can be stored in liquid phase. Hot working fluid is then throttled in order to produce saturated vapor that can be eventually expanded by a turbine. Two different configurations can be distinguished: total flash configuration involves a throttling process that terminates in saturated vapor condition (Figure 1.b plant scheme, Figure 1.e T-s diagram) while partial flash configuration (Figure 1.c plant scheme, Figure 1.f T-s diagram) throttling process terminates in the saturation dome and a vapor/liquid separator is required in order to prevent turbine blades from liquid droplets erosion. In this case, liquid fraction from the separator is mixed to the condensate (possibly preheated) before being pumped to the solar field. The advantage of direct cycles is the lack of the thermal oil loop that simplifies the plant layout but, on the other side, it involves a larger fluid inventory and additional issues related to working fluid decomposition in hot spot of solar loops. Air cooled condenser is considered in both cases according to the scarcity of water in desertic locations. Turbine efficiency can be assumed equal to a nominal value independently of the expansion characteristics (namely assuming a proper number of stages and optimized velocity) or it can be calculated as function of expansion volume ratio and turbine size parameter as proposed by Macchi et al., 2017 for a three stages axial turbine at optimal rotational speed. Nominal ambient temperature is equal to 30°C, the reference size of the system is 1MW electric power output and the design of the different plants relies on a common set of assumptions as reported in Table 1.

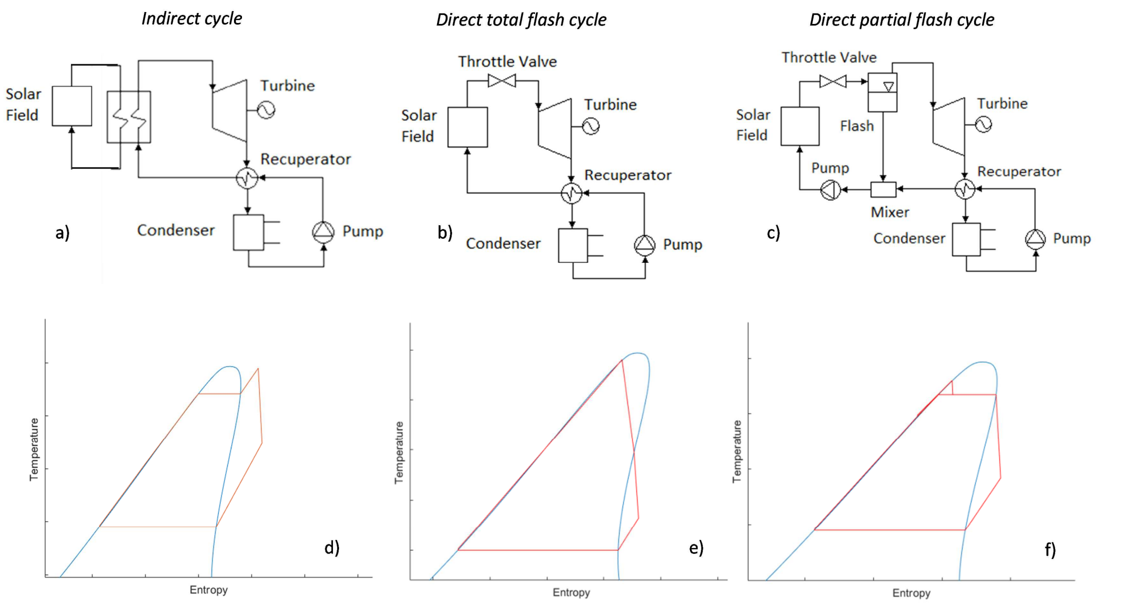

Figure 1: plant schemes and T-s Diagrams for the three different cycle configurations investigated: Indirect cycles (a-d), direct partial flash cycles (b-e) and direct total flash cycles (c-f)

Components efficiency		Pressure drops		Temperature differences	
η_{pumps}	75%	Δp_{eco}	0.3 bar	$\Delta T_{pp,rec}$	5°C
η_{turb}*	85%	$\Delta p_{rec,liq}$	0.3 bar	$\Delta T_{pp,PHE}$	5°C
$\eta_{mecc-el}$	97%	$\Delta p_{rec,vap}$	2%	$\Delta T_{min,ap,cond}$	15°C
η_{aux}	95%	Δp_{de-sh}	1%	ΔT_{sc}	5°C
*if not calculated by correlation		Δp_{sh}	1%		

Table 1: main assumptions adopted for the simulation of both direct and indirect solar ORC

2.2 Solar field model

The correct evaluation of the performance of a solar power systems must consider also the characterization of the solar field and how its design affects the overall system efficiency. In particular, two quantities should be calculated instead of assumed: (i) Solar field thermal efficiency that is affected mainly by tube receiver type and HTF mean temperature and (ii) solar field pressure drop that depends on solar field arrangement, loop length, minor losses and HTF thermophysical properties. In this study solar field is made by different loops of Andasol type concentrating parabolic trough collectors and Shott PTR70 receiver tubes (Schott Solar, 2013) arranged in different solar field sections in order to limit the overall pressure drop. Fluid is distributed and collected by headers having a variable diameter and a constant fluid velocity. A simplified model for the collectors, based on Forristal model (Forristal, 2003) and including receiver spatial 1D discretization has been implemented allowing to calculate distributed pressure losses and thermal efficiency of the solar field. A similar model has been adopted for the headers assuming a perfect insulation and neglecting heat losses. In this work minor losses (due to ball joints in collectors loops and diameter changes and curves for the headers) have been included in a simplified way as incremental factor on the distributed pressure drops evaluated for collectors loop and headers. The two incremental factors (eq.1 and eq.2) used to evaluate the total pressure drops in the solar field have been calibrated on design data available in (Kelly and Kearney, 2006) and referring to SEGS VI (NREL, 2015) and Andasol-3 solar fields (NREL,2013).

$$\Delta p_{Loop}^{Tot} = \Delta p_{Loop}(1 + 0.6504 + 0.0114\ N_{coll}) \qquad \text{eq.1}$$

$$\Delta p_{Header}^{Tot} = \Delta p_{Header}(1.67) \qquad \text{eq.2}$$

Where distributed pressure drops Δp_{Loop} and Δp_{Header} are calculated by the simplified models and the incremental term accounts for minor losses: in example the correction factor for loops pressure drop is a function of the number of collectors (N_{coll}) in a single loop representative of the number of ball joints. Table 2 reports SEGS VI and Andasol – 3 solar field specifications and the comparison between reference data and results from the numerical model. In order to provide the plant by a sufficient number of yearly operating hours and allowing for solar energy dispatchability and peak shaving, solar field must be oversized. In this work a solar multiple (SM) equal to 2 is adopted. Thermal heat storage can be designed as two tanks storage or a thermocline but this choice do not affect the simplified calculations reported in this work. In design condition a DNI equal to 800 W/m^2 is adopted.

	SEGS VI		Andasol - 3	
Solar field configuration	I		H	
Solar field Area, m^2	188000		510120	
Collector length, m	51		150	
Mirror aperture, m	5		5.76	
Number of loop	50		156	
Loop length, m	816		600	
Number of collector in a loop	16		4	
Pressure drops	Data	Model	Data	Model
Δp_{Loop}, bar	7.52	7.34	6.30	6.15
Δp_{Loop}^{Tot}, bar	13.45	13.45	10.43	10.43
$\Delta p_{receiving\ Header}$, bar	1.59	1.63	3.15	2.71
$\Delta p_{distributing\ Header}$, bar	1.44	1.42	2.00	2.34
Δp_{SF}^{Tot}, bar	16.48	16.50	15.58	15.47

Table 2: Main data and pressure drop validation for solar field model

3. METHODOLOGY

The comparison among direct and indirect ORC for solar applications is carried out analyzing with an enumerative approach different combinations of solar field maximum temperature (150-400°C), different working fluids (46 candidates fluids has been selected from Refprop database) and different cycle configurations (indirect subcritical saturated/superheated, direct partial/total flash, all of them recuperative/non recuperative). For each combination the plant design has been optimized maximizing the system efficiency namely the ratio among the net power output and the heat collected by the solar field. This figure of merit is strictly related to the solar field size and thus it provides a good estimation of the system specific cost since the capital cost of the solar field is usually the largest fraction of a solar power plant capital cost. Another quantity that is considered of interest for solar applications is the Equivalent Electricity Energy Density (EEED) of the storage system calculated as reported in eq. 3 (Casati et al., 2013).

$$EEED = \frac{W_{net}}{m_{HTF,hot}} \frac{\rho_{HTF,hot}}{3600};\ [kWhel/m^3 storage] \qquad \text{eq.3}$$

EEED represents the amount of energy that can be produced by the exploitation of one cubic meter of storage volume and it gives a direct estimation of the specific cost of the storage. Each plant have been optimized varying different design parameters as reported in Table 3 while respecting some constraints related to minimum fluid pressure and maximum evaporation temperature with respect to critical point. The optimization of the condensation temperature is of interest only if the turbine efficiency is calculated instead of fixed, otherwise the minimum value (according to pressure and temperature lower bound) is adopted independently of the actual feasibility of turbine design.

Cycle	options		Optimization variables			
	Δp_{SF}^{Tot}	η_{turb}	T_{eva}	T_{sh}	T_{cond}	T_{flash}
Indirect superheated	Computed /fixed		yes	yes	yes	no
Indirect saturated			yes	no	yes	no
Direct total flash			no	no	yes	yes
Direct partial flash			no	no	yes	yes

Table 3: optimization options and optimization variables for the different cycle configurations investigated in this work

4. RESULTS

First a detailed analysis is carried out for a fixed solar field maximum temperature and comparing different fluids and cycle layouts. Then a couple of sensitivity analyses are presented adopting a different minimum condensation pressure and three different maximum solar field temperatures. Finally, the overall comparison among direct and indirect cycles is presented in terms of both system efficiency and EEED parameter.

3.1 Results for 250°C SF maximum temperature

The first analysis is carried out for a fixed SF maximum temperature equal to 250°C, including calculation of turbine efficiency and pressure drops into the solar field. Moreover, a lower condensation pressure limit equal to 0.8 bar is considered allowing to operate the plant with minor issues related to air leakage. Figure 2.a shows the location of each investigated combination of cycle layout/working fluid on the efficiency/EEED diagram. Each plant is optimized in order to maximize the global efficiency. Six systems are selected for a further discussion: plants A, B, C and D are indirect cycles using working fluids with increasing critical temperature while plants E and F are the best plant for total flash and partial flash configurations respectively. It is possible to highlight the following considerations:

- With a proper choice of the working fluid the indirect cycles can achieve a higher efficiency and a higher EEED with respect to both direct cycles configurations. Moreover, for any direct cycle it is possible to design an indirect system with the same efficiency but with a much higher EEED and thus a cheaper thermal energy storage.
- Trend of superheated and saturated indirect cycles is pretty similar, but the second one can reach higher efficiency but lower EEED using the same working fluid. This is caused by the higher working fluid temperature at the heat exchanger inlet that leads to a lower temperature increase of the HTF and eventually to a larger storage system.
- For indirect cycles, the effect of working fluid choice on efficiency can be explained by comparing different cases. For low critical temperature fluids like R32 (T-s diagram in Figure 2.b) the efficiency is penalized by the small turbine expansion ratio, the limited maximum temperature of the fluid and the large temperature difference in the heat introduction process. On the contrary, for very high critical temperature fluids like md2m (T-s diagram in Figure 2.e) a higher evaporation temperature can be achieved but condensation temperature is remarkably higher than the ambient one because of the limit on condensation minimum pressure. Figure 2.c shows the T-s diagram for the optimal fluid, pentane, which represents the best trade-off between the need of increasing the evaporation temperature adopting high critical temperatures fluids and the difficulties in designing high efficiency turbines for very large volume ratios.
- Similar considerations can be outlined also for the direct cycles. A comparison among them shows that total flash cycles can attain lower efficiency. This is due to the large pressure drop in the throttling process further penalized by the limited availability of working fluid for medium-low solar field maximum temperatures. However, total flash cycles can attain higher EEED parameter thanks to the larger temperature rises in the solar field. Figure 2.f and Figure 2.g show the T-s diagrams for the optimal cycles for both direct configurations.

Main quantities for the six selected systems are reported in Table 4.

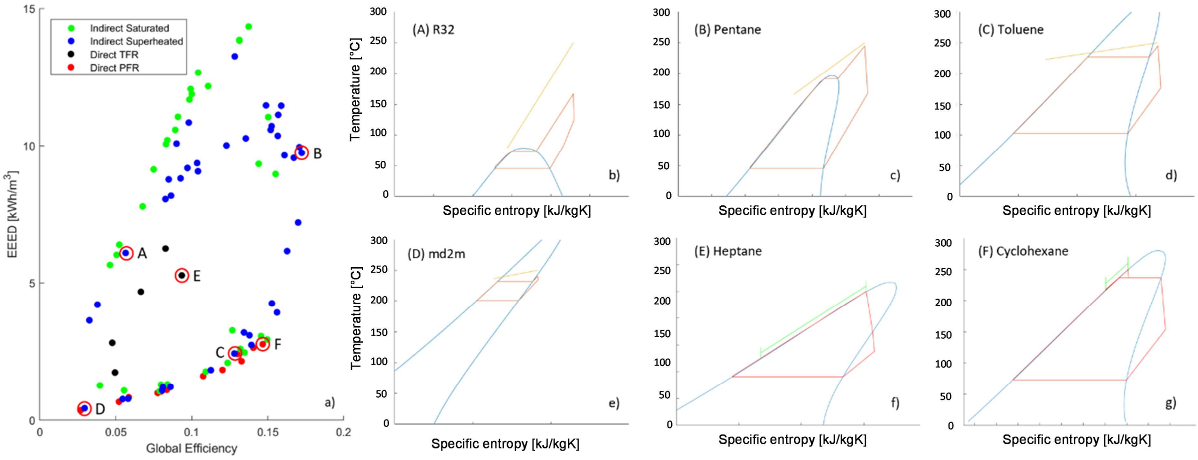

Figure 2: a) maximum efficiency and EEED parameter for all the cycle configurations and all the working fluids investigated, b)-g) T-s diagrams of some selected configuration-working fluid

	(A)	(B)	(C)	(D)	(E)	(F)
Cycle configuration	Indirect SH	Indirect SH	Indirect SH	Indirect SH	Direct TFR	Direct PFR
Working fluid	R32	Pentane	Toluene	Md2m	Heptane	c-hexane
T_{eva}, °C	73	191	227	229	-	-
T_{sh}, °C	167	245	245	235	-	-
T_{cond}, °C	45	45	102	185	90	73
T_{flash}, °C	-	-	-	-	174	237
Δp_{SF}^{Tot}, bar	11,44	6,25	3,97	6,28	7,35	3,83
η_{cycle}	7,53 %	23,23 %	17,37 %	4,54 %	12,52 %	19,91 %
η_{global}	5,65 %	17,24 %	12,80 %	3,34 %	9,34 %	14,67 %
EEED, kWhel/m^3	6,0886	9,7356	2,4259	0,4948	5,2723	2,7637

Table 4: main result for each cycle configuration presented in Figure 2

3.2 Effect of SF maximum temperature and condensing pressure

The analysis is extended to a lower (150°C) and a higher (350°C) solar field maximum temperature and it is repeated adopting a minimum cycle pressure (0.03 bar) and fixing turbine efficiency (90%). Results on the efficiency/EEED diagram are reported in Figure 3. The following outcomes can be listed:

- Considering a minimum pressure of 0.8 bar (Figures 3.a-c) similar results are obtained for both extreme solar field maximum temperatures. For 150°C (Figure 3.a) solar field maximum temperature the performance of direct and indirect cycles are comparable while a marked difference in EEED parameter is still present. For 350°C (Figure 3.c) instead the advantage in adopting indirect cycles is evident and it is mainly related to the possibility to reach a higher turbine inlet temperature while in direct cycles the critical temperature limits the maximum temperature of the working fluid.
- With less conservative assumptions on turbine efficiency and condensation pressure (Figures 3.d-f) the difference between direct and indirect cycles is reduced in terms of efficiency thanks to the possibility to exploit a larger pressure ratio that allows adopting a higher number of candidate working fluids for direct cycles. EEED parameter still benefits the indirect cycles.

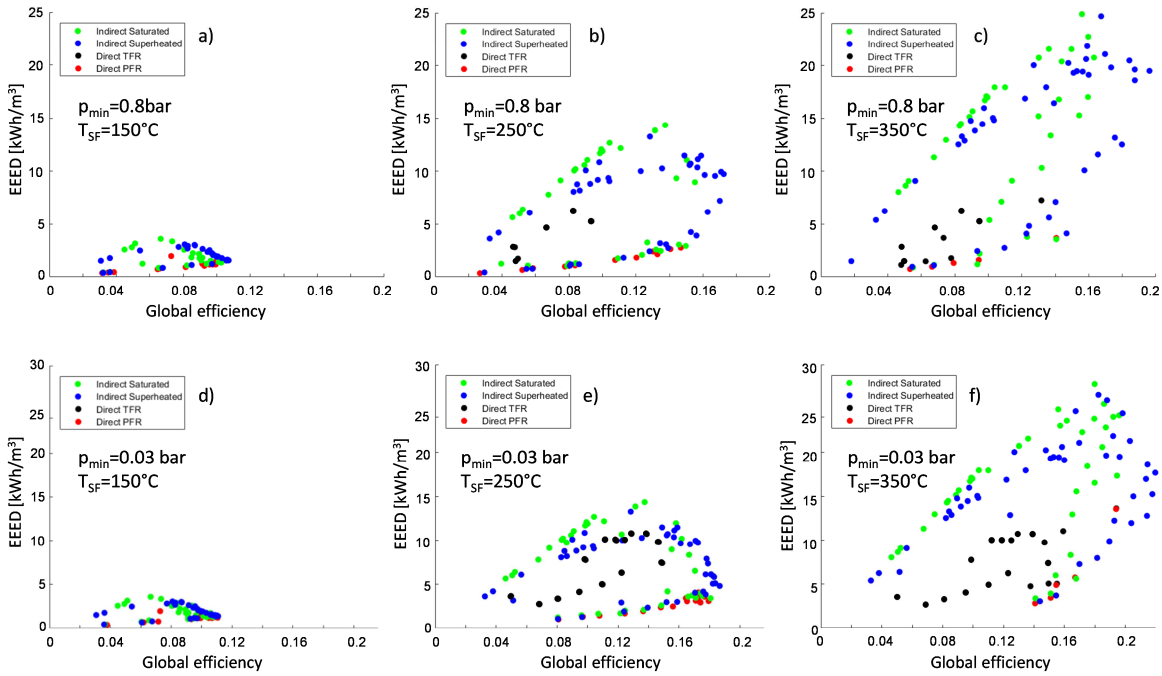

Figure 3: maximum efficiency and EEED parameter for all the cycle configurations and all the working fluids for different SF maximum temperature and minimum condensation pressures

3.3 Final Results

Results are presented only for the case with calculated turbine efficiency and minimum condensing pressure equal to 0.8 bar. Figure 4.a depicts the maximum efficiency achievable with subcritical indirect cycles and a select number of promising working fluids. For each single fluid, the efficiency first increases with solar field maximum temperature until it reach a nearly asymptotic trend when the evaporation temperature results limited by the critical point. The two fluids mostly recommended are cyclopentane and pentane but the latter can reach higher efficiency for SF maximum temperature higher than 200°C. These two fluids have an EEED index (Figure 4.c) that is competitive with respect to the other candidate fluids with a positive trend for increasing maximum HTF temperature. Direct cycles show a similar trend and the point of discontinuity is related to the limitation due to the critical point: maximum efficiency (Figure 4.b) is 2-3% points lower than for the indirect cycles meaning a consistent loss of around 15% of net power output for a fixed solar field size. EEED parameter (Figure 4.d) is much more penalized with values around one fourth of those achievable with indirect cycles

5. CONCLUSIONS

The comparison between direct and indirect ORC cycles carried out in this study suggests that the adoption of direct cycles, in both total and partial flash configurations, is not competitive against indirect cycles in terms of maximum attainable efficiency and size of the storage. However, in some cases (low condensing pressure and high efficiency turbines) the use of direct cycles in partial flash configuration may approach the efficiency of the best indirect cycles with similar EEED values. Conclusions outlined in this work are valid only from a thermodynamic point of view and more accurate results would be obtained by performing techno-economic analysis of the different solutions also including the cost of the fluid inventory and equipment cost.

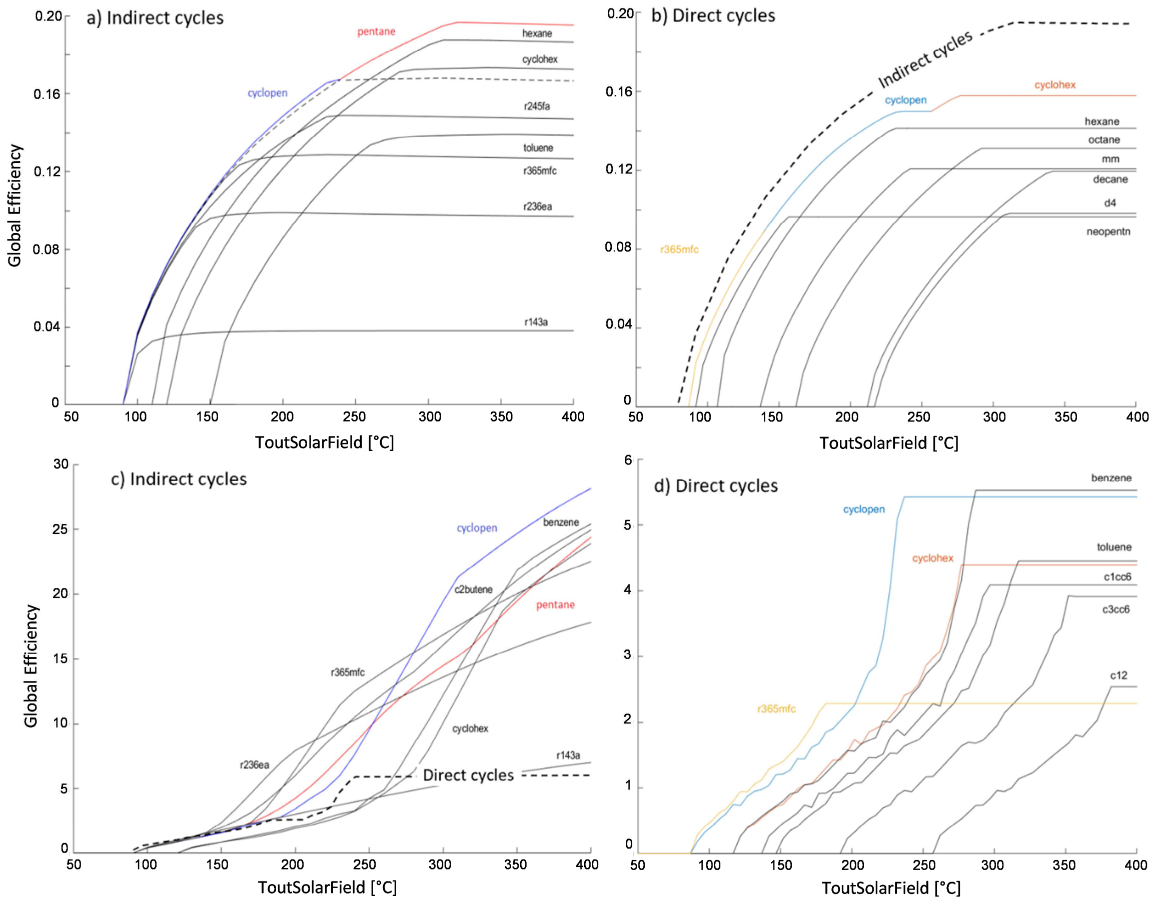

Figure 4: maximum efficiency and EEED parameter for different working fluids against solar field outlet temperature for both direct and indirect cycles

NOMENCLATURE

Subscripts

ap	approach point
aux	auxiliaries
coll	collector
cond	condenser
de-sh	desuperheating
eco	economizer
el	electric
eva	evaporation
liq	liquid
mec-el	mechanical-electrical
pp	pinch point
rec	recuperator
sc	sub cooling
sh	superheater
vap	vapour
turb	turbine

Variables

η	efficiency
p	pressure
ρ	density
s	specific entropy
T	temperature
W	power

Acronyms

CSP	concentrating solar power
HTF	heat transfer fluid
ORC	organic Rankine cycle
PV	photovoltaic
SF	solar field
TES	thermal energy storage
EEED	Equivalent Electricity Energy Density

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