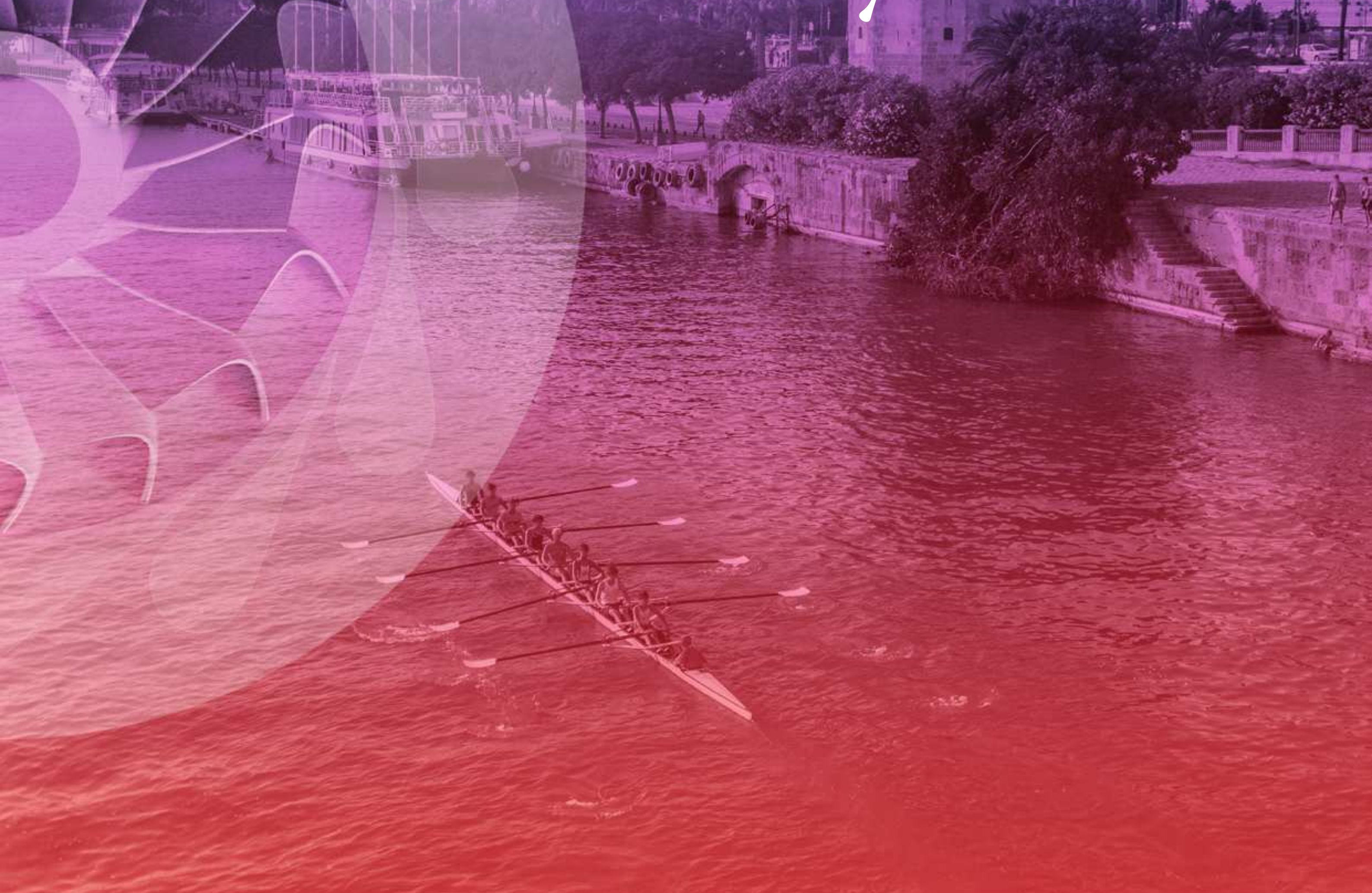
David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems



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POTENTIAL OF TRIGENERATIVE WASTE HEAT RECOVERY CO_2-MIXTURE TRANSCRITICAL POWER PLANTS FOR INCREASING THE SUSTAINABILITY OF DISTRICT HEATING AND COOLING NETWORKS

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ABSTRACT

The waste heat released by high-temperature processes can be exploited by power cycle designed for full electric or combined heat and power applications, with the potential to cover even the cooling demand in a trigenerative perspective. The use of CO_2-based mixtures as working fluids for power cycles can be a promising solution for power production. These systems present a rejected heat in a temperature range (50–180 °C) that allows, depending on the needs, an effective coupling with a district heating and cooling network. This work investigates the potential of trigenerative system adopting CO_2-based power cycles which exploit the residual thermal power of the exhaust gases of a small-scale gas turbine. First, the performances of lithium bromide absorption chiller are investigated for various heat source levels, adopting different configuration. Then, various designs of CO_2-based power cycles are simulated focusing on the coupling with both the district heating and the chiller. A sensitivity analysis on the cycle minimum temperature is presented, evidencing that CO_2 mixtures can achieve remarkable net electric efficiency values even at high cycle minimum temperatures, marking a significant difference with respect to CO_2 cycles. Considering the yearly demand of district heating and cooling, keeping the electric output at design value, the economic profitability of the investment is characterized presenting the LCOE of the retrofitted solution, considered comparable with actual selling prices.

1 INTRODUCTION

The demand and production of thermal power for space heating and cooling applications rose significantly in the last decades: the International Energy Agency (IEA) reports that from 1990 to 2016 the space cooling demand has tripled, and it is mainly provided by air conditioning systems, consuming 2000 TWh of electricity per years worldwide (IEA, 2018). On the other hand, the agency highlights that fossil fuels account for the 64% of the primary energy used for space heating, with natural gas covering 42% of the share (at around 760 billion Nm^3 in 2021) (IEA, 2022). In this scenario polygeneration plants are crucial not only to achieve high overall efficiency, but also to save primary energy and provide flexible operations. Organic Rankine Cycles (ORC) are nowadays the most used technology for waste heat recovery from high temperature industrial process and from small gas turbine (below 500°C). (Macchi & Astolfi, 2017). In recent years, pure CO_2 (carbon dioxide) and CO_2 mixtures power cycles have been proposed as an alternative technology to ORC (and steam cycle in large size applications) thanks to the higher fluid thermal stability and the compactness of turbomachinery. These systems have been recently investigated in several European H2020 funded projects like SCARABEUS, DESOLINATION, CO_2OLHEAT, SCO_2FLEX and SOLARSCO_2OL. A previous work (Morosini et al., 2023) underlined the advantages of coupling CO_2-based power cycles with a high (180°C) and a low temperature (100°C) thermal user, in a cogenerative perspective. These results indicate a potentially

exploitable source to produce chilled water too, thus adopting both district heating and cooling networks.
The concept of trigeneration and polygeneration, using a pure CO_2 based power block, is well known in the literature (Bellos et al., 2022). These systems typically consider electricity, hot and cold water and hydrogen as useful outputs providing an economic analysis on the specific case study. This work aims at investigating the trigeneration potential of pure and mixture CO_2-based power cycles for heat recovery applications. For this purpose, the absorption chiller (AC) is modelled to estimate the coefficient of performance (COP) of the component at various conditions. Then, different configurations of CO_2-based power cycles are investigated, aiming at an effective coupling with both the DHC networks. In particular, the revamping of a plant operating with small-scale gas turbines is explored. The heat rejected from the CO_2 power block together with the heat recovered from low temperature exhaust gases is used to produce pressurized hot water for the district heating (DH) during winter season and chilled water for the district cooling (DC) during summer season. Finally a techno – economic analysis is carried-out comparing the original and proposed system layouts in order to minimize the differential electricity production cost. Aspen Plus® V12 (*Aspen Plus* n.d.) is selected as modeling software for all the analyses.

2 ABSORPTION CHILLER

An absorption chiller produces cold thermal power (typically chilled water) exploiting a low-medium temperature heat source, adopting a thermodynamic cycle with a mixture of a refrigerant and an adsorbent (Somers et al., 2011). In this work both single effect AC (SEAC) and double effect AC (DEAC) are investigated using H_2O-LiBr (lithium bromide) mixture as working fluid.

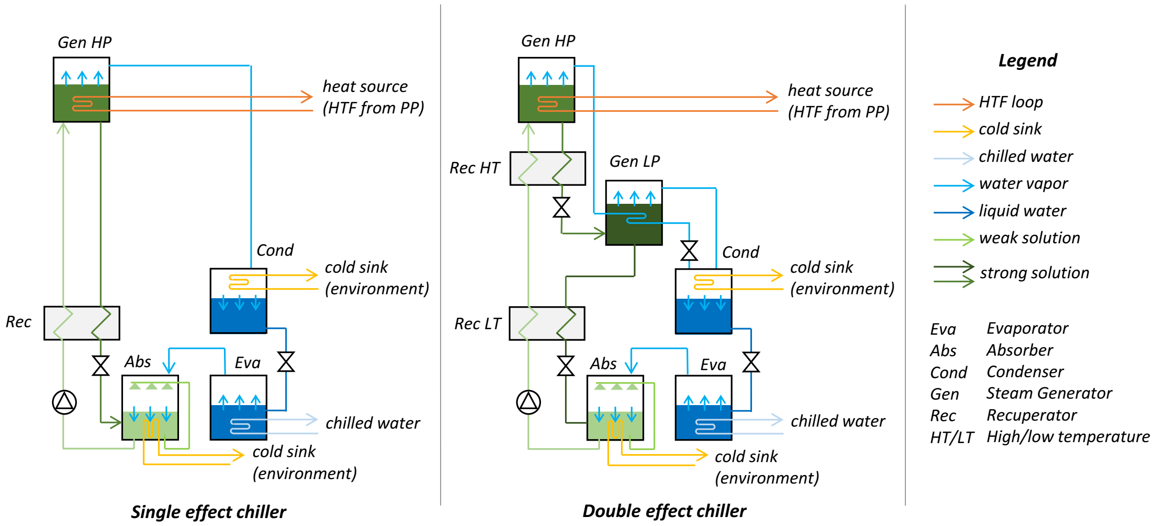

Figure 1: Absorption chiller layout: single effect (left) and double effect (right).

283

Typically, SEAC works at lower pressures (up to 10 kPa) in the vapor generator and exploits heat source temperatures lower than 100°C, leading to COP values of around 0.7. Meanwhile, DEAC and multi effect AC work at higher pressure levels (up to 100 kPa), heat source temperature up to 160°C–180°C, generally resulting in COP above 1 (Somers et al., 2011). In a SEAC (Figure 2 left), the pumped mixture is pre-heated in a recuperator and then heated by the hot source in the generator, up to a partial evaporation, depending on temperature and pressure. The LiBr-rich flow (strong solution) is conveyed to the recuperator, depressurized and fed to the absorber. The generated vapour is first condensed using ambient air as cold sink and expanded down to the evaporation pressure. The pressure levels are set by saturation conditions at the temperature T_{cond} and T_{eva}. In the evaporator, cold thermal power is produced typically by cooling down chilled water. The refrigerant enters the adsorber where is mixed with the strong solution, cooled down and send to the pump. In the DEAC solution (Figure 2, right), a layout with two generators in series at different pressure levels is adopted since this layout allowed the employment of only one pump is adopted (Farshi et al., 2012). As shown in Figure 2 the strong solution exits from the high-pressure generator and it is expanded to an intermediate pressure, corresponding to condensation pressure. Then the evaporated water and strong solution flow into the components previously presented. The advantage of the double effect is to separate more water from the solution, thus increasing the cooling capacity and COP. The H_2O-LiBr mixture is modelled with the ELECNRTL pre-defined package in ASPEN Plus, as suggested in literature (Somers et al., 2011). The boundary conditions and assumptions, which are consisted with other works (Somers et al., 2011) (Farshi et al., 2012), for the simulation of the AC are reported in Table 1.

Table 1: Simulation boundary conditions and assumptions for the absorption chiller

Chiller Parameter	
Condensation conditions, T_{cond}/P_{cond}	40 °C / 7.38 kPa
Evaporation conditions, T_{eva}/P_{eva}	3 °C / 0.75 kPa
LiBr concentration (massic)	55%
Minimum approach temperature difference in the generators	5 °C
Pitch point temperature difference in the generators	5 °C
Temperature approach for DC-AC	3 °C

The thermal power is provided by pressurized water that works as heat transfer fluid (HTF). The analysis consists in identify the COP, defined in Eq. (1) as the ratio between the cooling capacity $\dot{Q}_{CC}$ and the thermal input $\dot{Q}_{HTF}$, for various conditions of the HTF, neglecting the electric consumption of the pump (Osta-Omar & Micallef, 2016).

$$COP = \frac{\dot{Q}_{CC}}{\dot{Q}_{HTF}} \tag{1}$$

An optimization analysis is carried out on the maximum pressure of the double effect cycle to find the best value of COP for each HTF maximum temperature ($T_{HTF,in}$) and HTF temperature difference (ΔT_{HTF}), compatible with system constraints. In fact, according to the Dühring chart, crystallization phenomena occurs when the mass concentration of LiBr is above 65% (Salehi et al., 2019). For this reason, the maximum pressure level must be compatible with this threshold to prevent pipes and components clogging. The resulting trends of the COP for both configurations are reported in Figure 2. 284

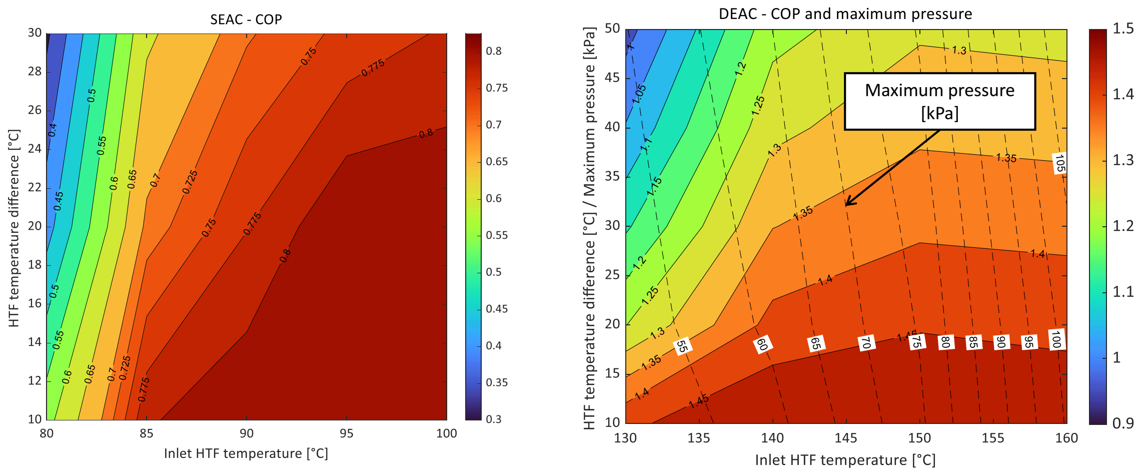

Figure 2: Absorption chiller COP: single effect (left) and double effect (right)

For both categories of AC, higher COP are computed at high $T_{HTF,in}$ and low ΔT_{HTF} (close to isothermal hot source), with a flat response over a certain value of $T_{HTF,in}$, equal to 95°C and 150°C for the SEAC and DEAC, respectively. As in the SEAC the pressure level is fixed by the condensation condition, increasing the $T_{HTF,in}$ corresponds to a higher vapor fraction leading to a higher cooling capacity. Situations in which the ΔT_{HTF} is low also correspond to conditions in which the internal regeneration of the AC is greatest. Therefore, at high $T_{HTF,in}$ the COP is favoured by greater steam production and, at the same time, by less $\dot{Q}_{HTF}$ introduction at low ΔT_{HTF}. In the DEAC, every case is optimised maximising the COP and varying the maximum pressure, so this effect is not evident and it is possible to achieve a COP in the range between 1.25 and 1.45 even with relatively low $T_{HTF,in}$ and large ΔT_{HTF}. In the DEAC the optimum high pressure varies from 50 kPa to 105 kPa and for each case, the limit is set due the reaching of the crystallization constraint.

3 CASE STUDY

An existing trigeneration power plant, located in the city of Milan (A2A, 2023), is selected as case study. In Table 2 the components and the power balance of the system are listed. The electric power is produced by two Taurus 60S-7801 5MWe gas turbines by Solar Turbine (Taurus 60, n.d) each one powered by natural gas with an input thermal power of 17 MW. The expander outlet temperature is 510°C and the flue gas mass flow rate is 43.22 kg/s. In this study, the heat recovery units and the compressors chiller are replaced by a trigenerative CO_2 based power cycle for additional electricity production.

Table 2: Trigeneration power plant components

Components	**Power**
Gas turbines	10 MWe
Heat recovery steam generator	16 MWth
Compressor chillers	7.5 MWth

3.1 CO_2 power cycles

For the power cycles analysis, the stack temperature is set to 120°C leading to an available thermal power ($\dot{Q}_{fg}$) of 18.33 MW from both turbines at full load. The flue gas directly releases heat to the working fluid with a minimum temperature approach of 30°C in the primary heat exchanger (PHE) thus avoiding the use of a high temperature HTF loop. The cascade cycle is selected for this application as suggested by a previous work (Morosini et al., 2023) and it is here proposed in two different architectures (Figure 3) differing in one recuperator with the aim to consider three different design criteria: maximization of thermal recovery (MTR) namely ensure a complete exploitation of the heat source from the power plant, maximization of the power production (MPP) and maximization electric efficiency (MEE). Each cycle is investigated with both pure CO_2 in supercritical cycle and mixture of CO2 and dopant in transcritical cycle.

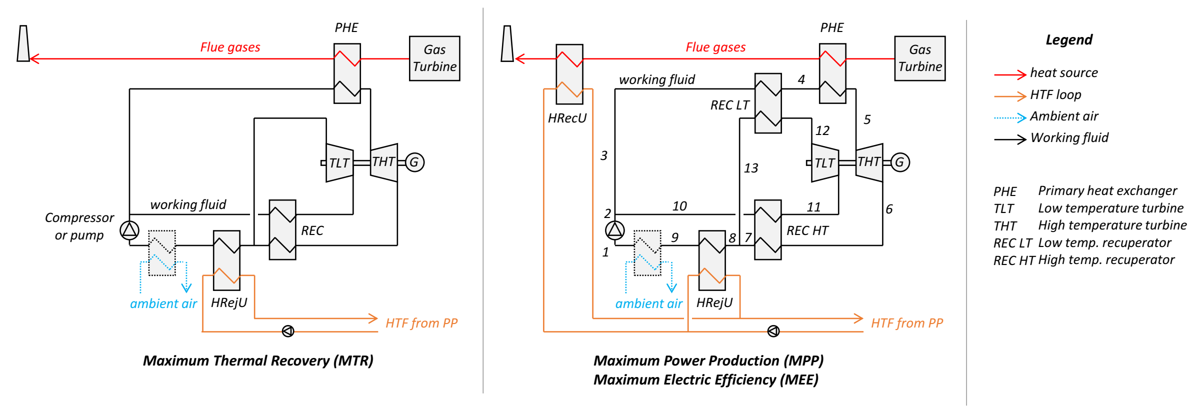

Figure 3: Cascade Cycle. MTR configuration (left), MPP/MEE (right)

MTR cycle adopts only one recuperator to exploit the heat available from high temperature turbine exhaust for heating the fluid expanded by the low temperature turbine. As result the heat introduction process starts right after the compressor (pure CO_2 cycle) or the pump (CO_2 mixtures), allowing for a complete cooling of the heat source in the power cycle heat introduction process and leading to a heat source recovery factor ($\chi = \dot{Q}_{PHE}/\dot{Q}_{fg}$) equal to 1. The heat rejected by the cycle is recovered in the HRejU unit by heating up a HTF loop (Figure 3). On the contrary MPP and MEE cycles adopt an additional recuperator that allows to preheat the high temperature working fluid loop fluid before the heat introduction by cooling down the hot working fluid released by the low temperature turbine. In this case, the internal heat recovery is improved allowing to increase the cycle thermodynamic efficiency at the expense of the heat source recovery factor. As a result, the heat source is not totally exploited and thus after the power plant utilization it can still release heat to the HTF loop in the HrecU unit (Figure 3). A sensitivity analysis is carried out on the minimum cycle temperature ($T_{min,cy}$) between 50°C and 70°C considering ambient air as cold sink when the plant is not working in cogenerative mode or when a fraction of the rejected thermal power cannot be used for HTF heating. For each design criteria (MTR, MPP, MEE), for each minimum temperature and for both pure CO_2 and blended CO_2, the power plant (for a total number of 30 cases) is optimized with the aim to maximized the electric power for MTR (while guaranteeing $\chi = 1$) and MPP architectures and to maximize electric efficiency for MEE one. The following design parameters are varied to identify the optimal design: i) minimum pressure which impacts on the dopant molar fraction for working fluid blends, ii) split fraction after the pump and iii) maximum cycle temperature. Table 3 reports the assumed values for the analysis: turbomachinery efficiencies refer to small-scale applications, while a maximum pressure (250 bar) is necessary in order to do not penalize the performance (Alfani et al., 2021) (Morosini et al., 2023) .

Table 3: Power block parameters

Cycle Parameter	
Cycle maximum pressure	250 bar
PHE minimum temperature approach	30 °C
Recuperators pinch point (MITA)	10 °C
Pressure drops (PHE/HRecU and HRejU)	3/1 bar
Pressure drops recuperator (HP/LP)	1/2 bar
Isentropic efficiency (expander/compression)	85/80 %
Generator/Motor efficiency, $\eta_{g/m}$	97/97 %

286

Table 4 provides the properties for the chosen working fluids. Hexafluorobenzene (C_6F_6) is the selected dopant because its high thermal stability (up to 600°C) and the mixture is modelled with the Peng Robinson EoS (Di Marcoberardino et al., 2022). Differently, CO_2 is modelled with the Span and Wagner EoS (Span & Wagner, 1996).

Table 4: Pure fluids and mixture properties

Fluids	**Molar weight [kg/kmol]**	**T_{cr} [°C]**	**P_{cr} [bar]**	**Binary interaction parameter [-]**
CO_2	44.01	31.06	73.83	$k_{ij} = 0.16297 - 0.0003951 \cdot T\ [K]$
C_6F_6	186.06	243.58	32.73	

3.2 District heating and cooling system

The water for DH is supplied at 95°C and has a return temperature of 60°C (Mise, 2020) and the provided temperature range for the chilled water is from 6°C to 12°C (AIRU - Annual Report, 2022). An intermediate loop (IL) of heat transfer fluid is implemented to avoid direct heat transfer between the power plant working fluid and both the district hating water or the absorption chiller mixture, as proposed in Figure 3. The temperature limits of the IL must be compatible with the source and utilization range; therefore, the selected IL range is defined once the temperature range of working fluid and gases are computed. A minimum temperature approach of 10°C is always considered in each heat exchanger.

3.3 Key Performance Indicators

The performance of the power plant is evaluated trough the electrical efficiency η_{el} and the differential levelized cost of electricity (LCOE). The first is defined in Eq. (2).

$$\eta_{el} = \frac{\sum \dot{W}_{TURB} \cdot \eta_{g/m} - \dot{W}_{pump/comp}/\eta_{g/m}}{\dot{Q}_{PHE}} = \frac{\dot{W}_{el}}{\dot{Q}_{PHE}} \tag{2}$$

From the economical point of view, the goal is finding the layout and the design parameters that minimize the differential LCOE related to the installation of the CO_2 based power plant with respect to a reference case where the electric power is produced only with the gas turbines and the hot and cold thermal power are produced from a total exploitation of the exhaust gases sensible content ($\dot{Q}_{fg}$), and the second, representing the solution proposed in this work. Therefore, the yearly differential cost considers the annualized investment cost of the power block ($Capex$), the operating costs ($O\&M$) and the profit loss related to the reduced fraction of hot and cold thermal power with respect to the reference scenario because of electrical power production. LCOE is expressed as follow:

$$LCOE = \frac{Capex \cdot CRF + O\&M + \Delta E_{hot} \cdot LCOH + \Delta E_{cold} \cdot LCOC}{8760 \cdot u_{el} \cdot \dot{W}_{el}} \tag{3}$$

The $Capex$, corresponding to the sum of the equipment and Balance of Plant (BoP) costs, is calculated according to Weiland et al (Weiland et al., 2019), and the correlation of Wright et al (Wright et al., 2016) only for the gas-CO_2 PHE. The specific $O\&M$ is 30 \$/kW and the BoP cost are the 30% of the total equipment cost (Marchionni et al., 2018). $LCOH$ is the Levelized Cost Of Heating and it is equal to 80 \$/MWh (Næss-Schmidt et al., 2021), while the Levelized Cost Of Cooling (LCOC) is calculated by using the correlations and assumptions proposed by Correa et al (Correa-Jullian et al., 2019): assuming a COP of 1.3, LCOC is equal to 40 \$/MWh. The utilization factor u is 91% (corresponding to 8000h/year) while heat for DH is required for 41% of the running hours (15 weeks/year) and heat for the absorption chiller 24% of the running hours (9 weeks/year). The capital recovery factor (CRF) is 7.8%. ΔE_{hot} and ΔE_{cold} are the reduction of thermal energy and cooling energy with respect to the reference case. 287

4 RESULTS AND DISCUSSION

4.1 Considerations on Cycle optimization

For the MTR and MPP conditions (pure CO_2), optimal cycle minimum pressure is rather constant while reducing the cycle maximum temperature has two opposing effects: from one hand it allows to increase the heat input to the cycle by reducing the recuperator outlet temperature, while, on the other hand, it implies a reduction of cycle efficiency. However, the two effects are nearly balanced and the power output varies by less than 1.5% in the cycle maximum temperature optimal range between 440°C and 480°C. For MEE the highest cycle efficiency corresponds to a turbine inlet temperature of 480°C (compatible with the minimum temperature approach in PHE). Results reported in this paper refer only to the optimal cases and nominal conditions.

4.2 Power Cycle Performance

Figure 4 depicts the trend of electric efficiency and power output for all the cycle combination varying the cycle minimum temperature.

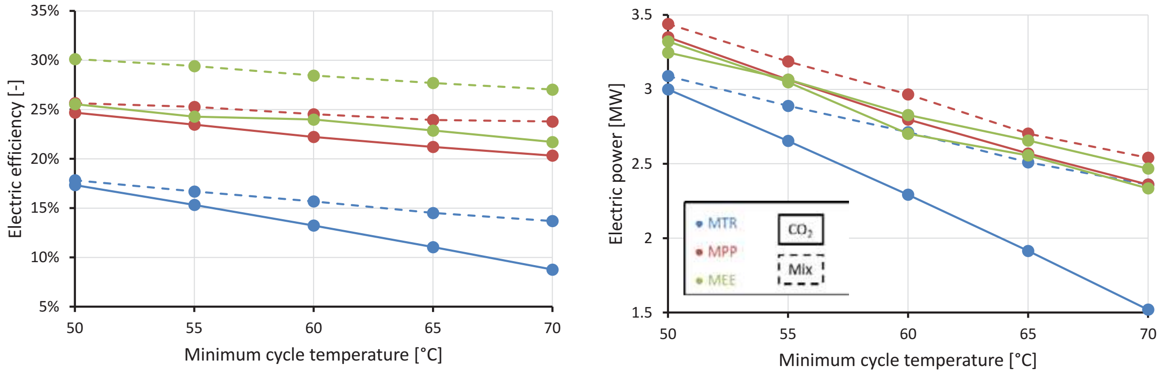

Figure 4: Electric efficiency (left) and electric generation (right)

Increasing the $T_{min,cy}$ implies a penalization of the cycle performance due to the increase of fluid specific volume (for pure CO_2 cycles) and reduction of the pressure ratio (blended CO_2) but the effect is more marked for the pure CO_2 cases since transcritical cycles always benefit by a fluid pressurization in liquid state that is less affected by change in the working fluid volumetric behavior. Among the different cycles configuration, the MTR cycle with CO_2 is the most penalized one since the need of ensuring a complete heat source cooling in the PHE which results in a limitation of the cycle pressure ratio and leads to cycle minimum pressures higher than the optimal one: all these effects are exacerbated when the cycle minimum temperature increases. While the use of pure CO_2 always leads to the lowest efficiency and power production it is interesting to note that the MTR cycle with CO_2 mixture for high cycle minimum temperature can get a power output close to the other cycles because the lower efficiency is compensated by and higher recovery factor with respect to both MPP and MEE configurations. The complete cascade cycle featuring also the second recuperator in both MPP and MEE cases outperform the MTR thanks to a more effective regeneration. In the MEE case, the TIT (Turbine Inlet Temperature) is set to the maximum value of 480°C, compatible with the constraint on the minimum pinch temperature at the PHE, justifying the highest electric efficiency. At the same TIT, the mixture has higher thermodynamic efficiency (around +3%) thanks to more balanced heat capacities within the recuperators. However pure CO_2 takes advantage of the lower PHE inlet temperature due to an unbalanced recuperator, experiencing higher electrical output. This effect is true only at low cycle minimum temperature, while at high temperatures an evident efficiency drops of pure CO_2 compared to the mixture can be noticed due to the increase of the compressibility factor. As regards the MPP strategy, which represents the trade-off between the heat source cooling grade and the cycle efficiency, the mixture outperforms pure CO_2 even if the optimal TIT is around 10°C lower to provide a more effective heat source cooling, without involving a substantial cycle efficiency penalization. In fact, pure CO_2 suffers both the distance from the critical point and the unbalanced recuperators, while the mixture keeps the benefits of liquid compression and balanced heat capacities by varying the composition. 288

4.3 Hot and cold power nominal availabilities

Thermal power is released by the system to the IL through the HRecU and HRejU units that completes the flue gas cooling down to stack temperature and collects the useful heat released by the cycle respectively. The amount of heat collected by each unit depends on the specific cycle layout, the optimization strategy, and minimum cycle temperature, as shown in Figure 5 for 50°C and 70°C cycle minimum temperature cases. Moreover Figure 6 summarizes the resulting annual produced energy. In the MTR configuration, the entire thermal duty of the sensible heat source is provided to the cycle through the PHE and exploited with a relatively-low cycle efficiency compared to the other layouts considered (Figure 3), but the amount of rejected thermal power from the HRejU unit is considerable as represented in Figure 5. The good temperature level (above 180°C of the rejected heat) provided by the MTR configuration allows the exploitation of the total duty for DH and DC purposes, depending on the seasonal need. Above 100°C the HRejU is used in a DEAC, while the fraction below 100°C in a SEAC, with COP evaluated according to the maps in Figure 2.

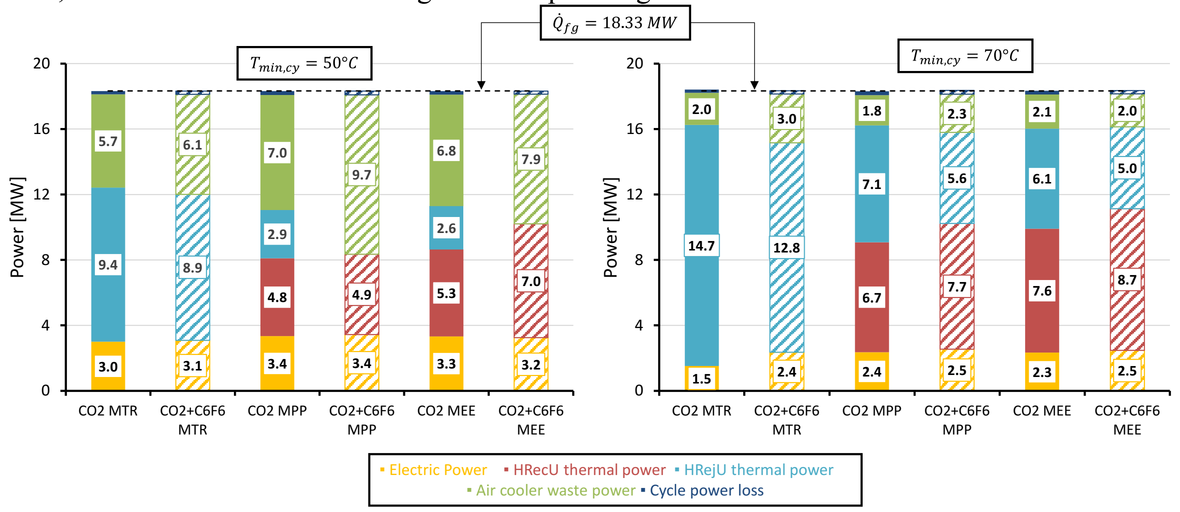

Figure 5: Power balance for $T_{min,cy} = 50°C$ (left) and $T_{min,cy} = 70°C$ (right) in winter season.

On the other hand, the MEE and MPP configurations present a good fraction of thermal power available also from the HRecU at high temperatures (above 220°C), that can be entirely used for DH and DC through a DEAC. As regards the HRejU unit of these configurations, the thermal duty available for DH purposes strongly depends on the cycle minimum temperature since the required temperature range for the intermediate HTF is 70-100°C. Due to this limitation, an air cooler is always necessary to dissipate the residual thermal power. Instead, in summer season, the terminal temperatures of the HTF loop can be adjusted to exploit the entire rejected duty in a SEAC, with an associated COP depending on the specific temperature range and the air cooler is not required. As highlighted in Figure 6, the best annual energy production is reached with the mixture in MEE strategy at 70°C minimum temperature, mainly due to better thermodynamic efficiency compared to CO_2 and higher valuable heat at the HrecU for DC and DH.

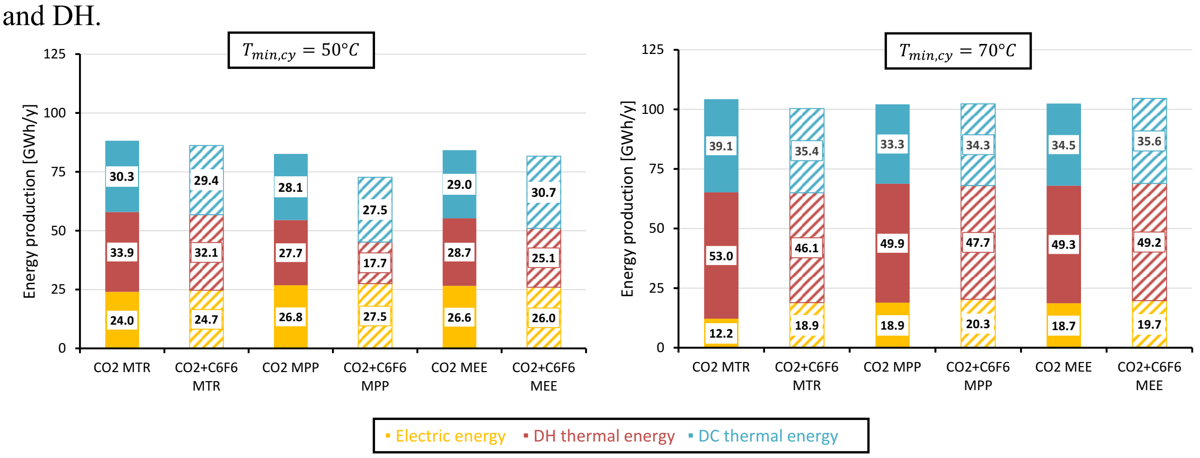

Figure 6: Annual energy production for $T_{min,cy} = 50°C$ (left) and $T_{min,cy} = 70°C$ (right).

4.4 Differential LCOE

Previous results suggests that the cycles operating with a minimum cycle temperature of 70°C are reasonably the most promising ones since they allow to heat rejection to the environment during both summer and winter season, thus only this case is considered in this preliminary economic analysis. Figure 7 shows the resulting differential LCOEof the different cases, as expressed in Eq. (3), highlighting the $Capex$ and $O\&M$ costs and the costs relating to the missed selling of heat and cold. Considering the annual energy balance depicted in Figure 6 (right), the higher electricity output produced by the mixture based cycles, with respect to the cycle operating with pure fluid, lead to a reduction of the LCOE. On the other hand, the CO_2 MTR case of is characterized by the minimum ΔE_{hot} and ΔE_{cold} because of the highest annual thermal production. Nevertheless, the annual electric production mainly affects the differential LCOE. Considering only electricity as output, all the analyzed cases are economically viable compared to different Italian electric gross prices (PUN – Prezzo Unico Nazionale) (GME, 2023), however the cases that has a lower electric energy generation are the most penalized.

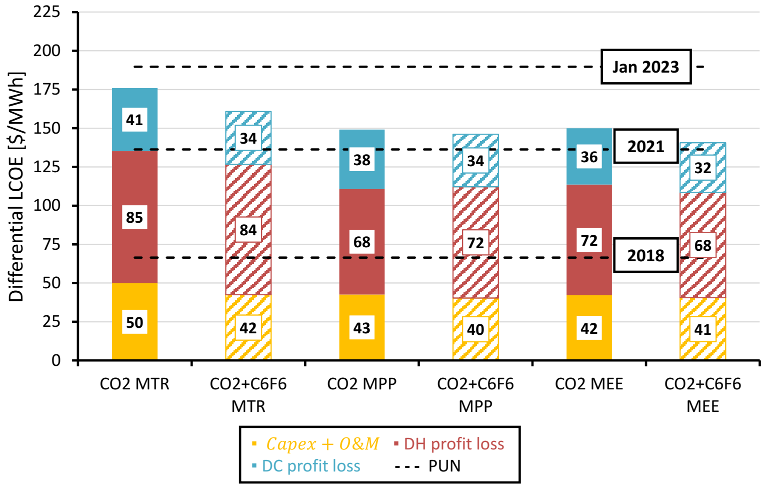

Figure 7: Differential LCOE for $T_{min,cy} = 70°C$ cases compared to the Italian PUN.

MPP configuration has the lowest specific power cycle cost ranging from 3736 $/kW for the mixture to 4723 $/kW for CO_2 since the larger power production is not obtained with a substantial increase of equipment cost with respect to the MEE case. MTR case is on the contrary the most penalized one since it does not maximize power production and uses large heat exchangers to provide complete heat source coolingleading to a cost of 176 $/MWh. Also accounting for profit losses due to the missed selling of heating and cooling due to power production the LCOE strongly increases leading to final values that are up to three times higher than the LCOE due to only *Capex* and *O&M* costs. MEE case with mixture has the lowest differential LCOE with values around 140 $/MWh. The studied trigeneration plant is economically feasible for PUN related to the recent years where the geopolitical circumstances lead to a very variable natural gas price and the relative PUN. On the contrary, adopting lower PUN prices related to a more stable global situation the installation of a cogenerative power plant would require incentives or the application of carbon tax to fossil fuel use to be economically competitive as generally true for any cogenerative power system.

5 CONCLUSIONS

In this paper, a theoretical revamp of a trigeneration plant is presented. By first analyzing the H_2O-LiBr absorber, it is possible to quantify the COP at different heat source condition. Then, simulations were carried out on a heat recuperative plant with different designs using CO_2 and a CO_2 mixture as working fluids. The maximum thermal recovery configuration in the PHE has higher thermal energy production comparing MPP and MEE, however on the economic aspect, the MTR presents a high differential LCOE due to the lowest electric energy production. The MPP strategy ensures high electric production and it proves to be the ideal approach when only energy production is considered as useful outcomes. However, for trigenerative scopes, the MEE configuration presents a good compromise between outputs and investment cost. The mixture of CO_2 and C_6F_6 is preferrable with respect to pure CO_2.

NOMENCLATURE

Acronyms

AC	Absorption Chiller
BoP	Balance of Plant
Capex	Capital Expenditure
COP	Coefficient Of Performance
CRF	Capital Recovery Factor
DC	District Cooling
DEAC	Double Effect Absorption Chiller
DH	District Heating
ELECNRTL	Electrolyte-NRTL
HRecU	Heat Recovery Unit
HRejU	Heat Rejection Unit
HTF	Heat Transfer Fluid
IL	Intermediate Loop
IEA	International Energy Agency
MITA	Minimum Internal Temperature Approach
MPP	Maximum Power Production
MEE	Maximum Electric Efficiency
MTR	Maximum Thermal Recovery
O&M	Operation & Maintenance
PHE	Primary Heat Exchanger
PUN	Prezzo Unico Nazionale
SEAC	Single Effect Absorption Chiller
TIT	Turbine Inlet Temperature

Roman and Greek letter

P	Pressure	bar/kPa
$\dot{Q}$	Thermal power	MW
T	Temperature	°C
$\dot{W}$	Mechanical Power	MW
ΔT	Temperature difference	°C
η	Efficiency	(-)
χ	Recovery factor	(-)

Chemical formula

CO_2	Carbon Dioxide
C_6F_6	Hexafluorobenzene
H_2O	Water
$LiBr$	Lithium Bromide

Subscript

comp	compressor
cond	condensation
cy	cycle
el	electric
eva	evaporation
fg	flue gases
g	generator
m	motor
min	minimum
pump	pump
turb	turbine

APPENDIX A

Table 5: Streams thermodynamic properties for the best economic case for pure fluid and mixture. For the mixture case the dopant fraction is 19%.

	$CO_2 + C_6F_6$ MEE Case				CO_2 MPP Case			
Stream	**Flow [kg/s]**	**T [°C]**	**P [bar]**	**Vapor Frac. [-]**	**Flow [kg/s]**	**T [°C]**	**P [bar]**	**Vapor Frac. [-]**
1	66.1	70	93.5	0	72.7	70	118	1
2	66.1	93.9	254	0	72.7	129.8	254	1
3	35.7	93.9	254	0	38.5	129.8	254	1
4	35.7	278.5	253	1	38.5	236.8	253	1
5	35.7	480	250	1	38.5	470	250	1
6	35.7	412.8	97.9	1	38.5	390.3	122	1
7	35.7	118.6	96.9	0.71	38.5	139.8	120	1
8	66.1	114.3	95.9	0.67	72.7	139.8	120	1
9	66.1	80	94.9	0.28	72.7	80	119	1
10	30.4	93.2	254	0	34.2	129.8	254	1
11	30.4	400.8	253	1	34.2	379.3	253	1
12	30.4	334.4	96.9	1	34.2	303.3	122	1
13	30.4	109.6	95.5	0.64	34.2	139.8	120	1

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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