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COMPARISON OF SCO_2, CO_2 MIXTURE, AND STEAM RANKINE CYCLES FOR CSP WITH COMPONENTS SIZING: IMPLICATIONS FOR COST AND FLEXIBILITY

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ABSTRACT

While all commercial concentrating solar power (CSP) plants with central receivers are equipped with the steam Rankine power cycle, supercritical CO_2 (sCO_2) cycles are seen as a promising advancement. They offer higher efficiencies and more compact turbomachinery at high temperatures compared to the steam Rankine cycle, however, sCO_2 cycles face challenges related to high pressures and temperatures, which increase material demands and the mass and cost of heat exchangers. As the need for flexible electricity generation grows, the thermal inertia of the power block, linked to its metal mass, becomes increasingly important. This study examines and compares the component sizes, material usage, and economic and flexibility impacts of sCO_2, CO_2 mixtures, and steam Rankine cycles for commercial-scale CSP applications. All cycles are designed with the same thermal energy input, keeping the solar field and thermal storage constant. Heat exchanger designs, including primary heaters, recuperators, and air-cooled units, follow the ASME Code and are modeled using Aspen Exchanger Design & Rating. Since no commercial sCO_2 turbines exist at this scale, turbine design parameters are estimated using a developed simplified approach. Results are compared with existing studies to assess component-level and system-level implications in terms of cost, performance, and flexibility.

INTRODUCTION

Even though the steam Rankine cycle which represents the state-of-art (SoA) technology is the only power cycle used in commercial concentrating solar power plants with central receivers so far [1], sCO_2 cycles have been considered as a promising advancement solution in this field. sCO_2 cycles can reach higher efficiencies and will have more compact turbomachinery designs compared to the steam Rankine cycle at high temperatures [2]. On the other hand, the high pressures and temperatures of the sCO_2 cycles present challenges for the materials used and together with high mass flows lead to higher mass and cost of heat exchangers [3]. One approach to enhancing their performance involves transitioning to CO_2 mixture cycles, which offer higher critical temperatures and enable condensation at temperatures as high as 50°C [4]. In this paper, a $SiCl_4$ dopant, stable up to at least 650°C [5], [6] is used.

Furthermore, with the increasing need for flexible electricity generation, the thermal inertia of the power block, associated with the metal mass, gains importance. Therefore, to explore the potential, use, and challenges of sCO_2, CO_2 mixtures, and steam Rankine cycle for CSP applications, this work aims to analyze and compare the component sizes, material use, and resulting economic and flexibility implications of the cycles at the component and system level.

METHODOLOGY

This study investigates sCO_2, CO_2 mixtures, and steam Rankine cycles for a commercial scale central receiver CSP plant with molten salt as the heat transfer and storage media, and an air-cooled unit as the heat rejection system [7]. The sCO_2 and CO_2 mixture cycles are designed based on assumptions from [4], [8]. The steam Rankine cycle is designed as the state-of-the-art cycle for commercial central receiver plants with reheat and regenerative feedwater heating, according to Kolb [9]. The cycle is simulated through Ebsilon® Professional. The main characteristics of the cycles are provided in Table 1.

For sCO_2 simple (SC), recompression (RC), and partial cooling cycles (PCC) are chosen while for CO_2+$SiCl_4$ mixture, simple, precompression (PreC), and recompression ones since they are found to be most promising based on previous literature findings [8], [10]. The layout of the CO_2+$SiCl_4$ mixture recompression cycle and its T-s diagram are shown in Figure 1. The thermodynamic properties of CO_2 are modeled using Span-

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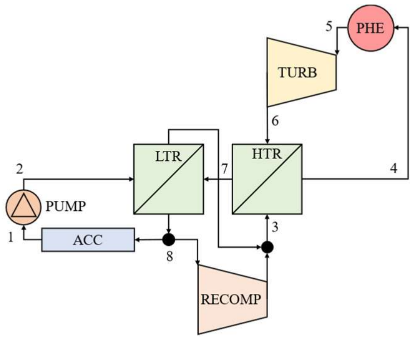
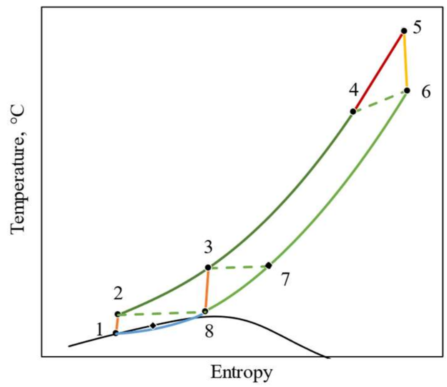
Wagner EoS [11], while for CO_2+$SiCl_4$ mixture properties Peng-Robinson EoS [12] and TRAPP model [6], [13] with binary interaction parameter equal to 0.06098 [14] is used.

Table 1: sCO_2 and CO_2 mixtures power cycles design parameters

Parameter	Value	
	sCO_2 and CO_2 mixtures	**Steam**
Turbine inlet temperature, °C	550	
Max pressure in the cycle / turbine inlet pressure, bar	250	125
Cycle minimum temperature, °C	50	
Recuperators MITA, °C	5	-
Feedwater heat exchangers TTD, °C	-	5
PHE Δp, %	2	-
Condenser/cooler Δp, %	2	-
Recuperators hot side Δp, %	1.5	-
Recuperators cold side Δp, %	0.3	-
Turbine isentropic efficiency, %	92	88
Pump/compressor isentropic efficiency, %	88	80
Motor/generator electrical efficiency, %	97	
Mechanical efficiency, %	99	



(a)



(b)

Figure 1: Layout (a) and T-s diagram (b) of CO_2+$SiCl_4$ mixture recompression cycle.

The cycles are designed with the same thermal energy input, assuming the same solar field and receiver design for both cycles. The mechanical design of equipment for all cases is performed according to ASME standards with assumptions presented in Table 2.

Table 2: Power cycles equipment general design parameters

Parameter	Value	Source
PHE power, MWt	280	[7]
Ambient air design temperature, °C	35	-
Equipment design temperature above maximum, °C	+5	[15]
Design pressure above maximum, %	+10	[15]
Thickness allowance, mm	1	-

MATERIALS

Materials used for steam power cycles are widely known and in the case of subcritical cycles comprise 1-2% Cr steels for piping and turbine high-temperature sections with carbon steels for the low-temperature parts [16], [17], [18].

Regarding the sCO_2 and CO_2 mixtures cycles it is assumed that the same materials could be used in both cases due to the limited data on the corrosion activity of $SiCl_4$ and positive preliminary results in 316L stainless steel [6].

Subcritical steam material results are generally applicable also for sCO_2, while supercritical water is usually more corrosive than sCO_2 [19]. Multiple sources recommend using stainless steel for temperatures above 400-450°C and even lower [20], [21]. But the same criteria (parabolic rate constant) also show that the use of 9-12% Cr steels should not be allowed for temperatures above 500°C in supercritical water. However, in practice, lower alloyed 1-2% Cr steels are used for piping and rotor with 11% Cr stainless steel used for the inner casing [17], [22]. In terms of strength, no significant degradation can be seen for 9-12% Cr steels at 550°C [21]. Therefore, it is assumed that 9% Cr steels up to 550°C (pipes, turbine), 2% Cr steels up to 460°C (HTR, SC recuperator), and carbon steels up to 250°C (LTR and ACC) can be applied for both sCO_2 and CO_2 mixtures cycles [19], [23].

Special attention should be paid to materials used in the primary heat exchanger, which are in contact with the power cycle working fluid and solar salt. Among the materials for steam generators used and considered before are: Inconel Alloy 800, stainless steels 304 and 304H, 316 and 316H, and various ferritic and carbon steels [24], [25]. However since 304, 304H, and 316 stainless steels showed stress corrosion cracking in Solar Two the materials for different sections of the steam generator are chosen as follows [25]: carbon steel for the preheater, 9% Cr steel for the evaporator, and 347 stainless steel for superheater and reheater. For turbine cost estimation the material costs are assumed based on EDR cost database [26] and [27]: 12 USD/kg for 9% Cr steel and 7 USD/kg for 1-2% Cr steel.

PRIMARY HEAT EXCHANGERS

The data for modeling the PHE are shown in Table 3. Since the PHEs for sCO_2 and CO_2 mixtures have operating temperatures close to the superheater, 347 stainless steel is used for the whole heat exchanger. Since both sCO_2 and CO_2 mixtures

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PHEs are limited by the cold side heat transfer coefficient a minimum 16 mm diameter tube is chosen. For the steam Rankine cycle, the PHEs are designed as two trains, based on the parametric study with the following variables:

- tube outer diameter: 16, 19.05 and 25.4 mm
- tube pitch ratio: 1.25 and 1.5
- tube layout: 30° (45° when 30° is not feasible)

It should be noted that no limitation is imposed regarding shell number. Aspen Exchanger Design & Rating (EDR) V14 [26] is used for all the heat exchanger designs according to the ASME Code [28] and cost estimations, including the primary heaters, regenerative feedwater heaters (FWH), recuperators, and the air-cooled heat rejection unit. Shell and tube heat exchanger design is chosen since it allows for high-power applications and is commercially available with clearly established standards.

Table 3: Primary heat exchangers design parameters

Parameter		**Value**	**Source**
Material	Preheater	Carbon steel	[25]
	Evaporator	9% Cr steel	
	Superheater, sCO_2 and CO_2 mixtures PHE	347 steel	
PHE hot end ΔT, °C		20	-
Tube diameter, mm		16/19.05/25.4	-
Maximum tube length, m		15	-
Fouling for hot side, $m^2 \cdot K/W$		$1.76 \cdot 10^{-4}$	[24]
Fouling for cold side, $m^2 \cdot K/W$		$8.8 \cdot 10^{-5}$	[24]

TURBINES

Turbine design is a complex iterative procedure with many thermal, mechanical, vibrational, and other analyses. In this paper, a preliminary design is assumed to estimate the dimensions and mass based on the previous experience in the literature. A two casings design is adopted with balancing piston and dry gas seals.

The repeated stage flow path is designed with a constant mean diameter, flow coefficient (φ), and loading coefficient ($\psi = \Delta H / U^2$). The main parameters for calculating the turbine geometry are shown in Table 4. Blade stagger angles are calculated according to [29]. Since sCO_2 cycles have high mass flows and pressures the bending stress (σ_b) is assumed to be the limiting factor for blade design [30]. Blade height to chord ratio is varied to get the minimum chord length which would satisfy the $\sigma_b \leq 130$ MPa for nickel alloys and $\sigma_b \leq 100$ MPa for lower-grade materials. Bending stress is calculated based on the correlations proposed by Saravanamuttoo et al.[31].

Table 4: sCO_2 and CO_2 mixtures turbines design parameters

Parameter	**Value**	**Source**
Rotational speed, Hz	50	-
Reaction degree	0.5	[32]
Pitch-to-chord ratio	0.85	[32]
Axial spacing relative to the chord	0.3	[29]
Blade pitch to mean radius of curvature ratio	0.5	[29]
Balancing piston diameter	$1.1 \cdot D_{hub}$	-
Balancing piston, seals length, mm	500	-
Inlet/ Outlet length, mm	300	-
Rotor diameter under seals, mm	350	[33]
Rotor diameter under bearings, mm	320	-
Bearing length, mm	400	-

For the sCO_2 and CO_2 mixtures axial turbines there is a limited number of complete designs. The comparison of the developed model with the results in the literature is presented in Table 5 (hub diameter shown for the first stage).

In [33] the turbine has 58 and 53 rotor and stator blades in the first stage and 47 and 42 in the last stage, while the presented model has 55 and 56 in the first and 36 in the last stage, which shows good accuracy of the simplified approach proposed.

For turbine design circumferential velocity (U) is assumed to be 105 m/s, while φ = 0.75. Since the increase in the number of stages both increases the turbine isentropic efficiency and reduces the turbine cost [30], [33] the single stage loading coefficient should be as low as possible. However, the maximum length of the turbine is limited by rotordynamic behavior, which is out of the scope of this paper. Therefore a slenderness ratio criteria (SR<9), which is defined as a ratio between bearing span and hub diameter, is adopted from [30]. The inner casing is assumed to be made out of 9% Cr steel casting, while the rotor and outer casing out of 2% Cr steel. The thickness of the inner and outer casings is preliminarily calculated as a pipe [34]. The algorithm for turbine design is shown in Figure 2. The stage loading coefficient is varied from from $\psi_{min} = 0.65$ to $\psi_{max} = 1.1$ until either the SR criteria is satisfied or the maximum stage loading coefficient or number of stages is reached.

Table 5: sCO_2 and CO_2 mixtures turbine model comparison with literature

Reference	**[16]**				**[22]**		**[33]**	
Power, MW	2		50		50		130	
Rotational speed, Hz	175		50		100		50	
Circumferential velocity U, m/s	88		88		132.5		102	
Loading coefficient ψ	1.32		0.66		0.87		1.1	
Flow coefficient φ	0.38		0.76		0.32		1	
Geometry data	Ref.	Calc.	Ref.	Calc.	Ref.	Calc.	Ref.	Calc.
Flow path length, mm	220	172	800	1506	700	740	1800	1515
D_{hub}, mm	160	142	500	498	358	383	620	618
H_{1st} stage, mm	16	17.6	60	62.2	43	41.1	28.7	30
n_{stages}	6	5	11	11	7	7	14	14

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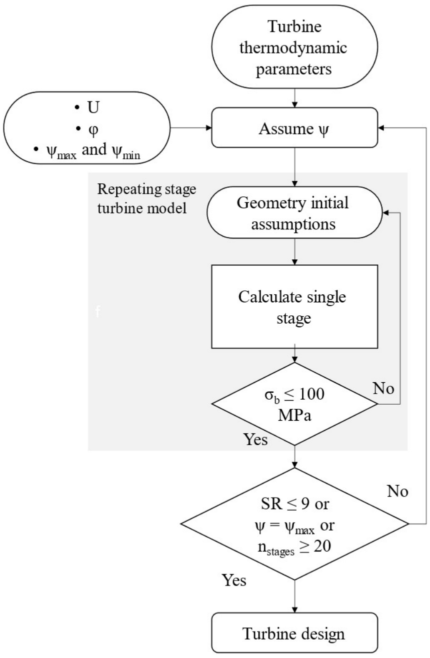

Figure 2: Turbine design algorithm.

The preliminary design of the steam turbine is performed by MAN Energy Solutions for the steam Rankine cycle analyzed in this study.

RECUPERATIVE SYSTEM

The data for designing the recuperators for sCO_2 and CO_2 mixtures cycles are shown in Table 6.

Table 6: sCO_2 and CO_2 mixtures recuperators design parameters

Parameter		Value	Source
Material	T > 250°C	2% Cr steel	[19], [23]
	T < 250°C	carbon steel	
Tube diameter, mm		16	-
Maximum tube length, m		15	-
Fouling for both sides, $m^2 \cdot K/W$		$8.8 \cdot 10^{-5}$	[24]

The closed feedwater heaters (FWH) of the steam cycle are classified into two groups: high pressure (HP) and low pressure (LP). The materials used for the steam cycle can be found in Table 7 a as well as the other main design inputs determined according to parametric studies. The materials are selected based on [35].

Table 7: Regenerative feedwater heat exchanger design parameters

Parameter		Value	
		HP	LP
Material	Shell	SA-204 Grd C	SA-516 Grd 70
	Tube	SA-213 T22	SA-210 Grd C
Tube diameter, mm		16	25.4
Tube Layout		30	45
Pitch Ratio		1.5	1.25
Fouling for both sides, $m^2 \cdot K/W$		$8.8 \cdot 10^{-5}$	

As Aspen EDR is not capable of designing the closed feedwater heaters with multiple zones and multiple shell inlets, only one HP and LP HEX are designed with a simplified approach. Both HP and LP FWHs have TEMA Type B-E-U.

CONDENSERS AND COOLERS

An A-frame air-cooled condenser or cooler (ACC) design is chosen as the configuration for all power cycles since it is the default choice for existing high power CSP plants along with the V-frame.

For sCO_2 and CO_2 mixtures, tube diameters are varied between 13, 16, and 25.4 mm, while air heat up is varied between 15, 20, and 25°C. For the steam cycle, a tube diameter of 25.4 mm emerged as the most economical design. The data for designing the condensers and coolers are shown in Table 8. In the case of condensers, there is only one pass, while for CO_2 coolers multiple passes are used.

Table 8: Condensers and coolers design parameters

Parameter	Value	Source
Material	Carbon steel	-
Fan efficiency, %	75	[36]
Cold end ΔT (ITD for the steam condenser), °C	15	-
Hot end ΔT for the steam condenser, °C	5	-
Maximum tube length, m	15	-
Fouling for hot side, $m^2 \cdot K/W$	$8.8 \cdot 10^{-5}$	[24]
Fouling for cold side, $m^2 \cdot K/W$	$8.8 \cdot 10^{-4}$	[37]

PIPING

In fossil fuel steam Rankine power plants piping can cost almost as much as a turbine unit [38], while in sCO_2 bigger mass flows will lead to even bigger diameter main piping. Therefore, it is important to estimate the piping for both cycles as well. However, the piping layout is highly dependent on the primary heat exchanger configurations, site limitations, and plant requirements.

The pipe cross sections are determined based on the ASME Code for process piping [34]. The design conditions adapted for both steam Rankine and sCO_2 and CO_2 mixtures cycles are shown in Table 9. The working fluid design velocity is based on the value present in both subcritical and supercritical main steam piping [17], [22]. For sCO_2 and CO_2 mixtures, 30 m/s is assumed to reach the ρw^2 similar to steam piping and is in line with other literature works [22].

Table 9: Piping design parameters

Parameter	Value		Source
	Steam	**sCO_2 and CO_2 mixtures**	
Material	2% and/or 9% Cr steel	9% Cr steel	-
Design velocity	50	30	[17], [22]
Pipes in parallel	2		-

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RESULTS

The results of the equipment modeling are compared with the cost estimates available in the literature. All literature cost estimates and specific costs used are updated to 2023 using CEPCI [39]. The results of the cycles calculation are shown in Table 10. Steam Rankine and CO_2 mixture recompression cycles have the highest electrical efficiency (including ACC fan power), followed by mixture and then sCO_2 cycles.

Table 10: Cycles results

Working fluid	H_2O	sCO_2			CO_2+$SiCl_4$		
Cycle	**Rank.**	**SC**	**RC**	**PCC**	**SC**	**PreC**	**RC**
Cycle minimum pressure, bar	0.124	100	110	80	72.4	72.4	83.6
PHE inlet temperature, °C	238	376	421	369	361	360	394
Cycle electrical power, MWe	116.1	96.7	108.1	105.4	108.3	109.3	112.5
Electrical efficiency, %	41.5	34.6	38.6	37.7	38.7	39.0	40.2
Same including ACC fan, %	39.7	34.0	37.9	37.0	38.1	38.4	39.7

The design results for steam Rankine cycle PHE with the lowest cost are presented in Table 11. The resulting cost is 8.48 MUSD or 72 USD/kWe, which is validated against the design study of a 110 MWe solar tower power plant from [40] by direct scaling due to similar size. Despite the differences in the geometries and fouling resistances, current design has only 6.2% higher specific cost compared to literature (67.8 USD/kWe). It should be noted that up to this point, the steam drum is excluded from the analysis. Assuming the cost of the steam drum to be the same as the superheater as in [41], the overall PHE cost for the steam cycle results in 94.2 USD/kWe specific cost. Earlier studies estimate the cost of HEs alone as 159.9 USD/kWe for 1000 MWth (65.5% of whole PHE cost), 311.1 USD/kWe for 260 MWth (73.8% of whole PHE cost), and 302 USD/kWe for 280 MWth (assuming the same portion of HEs in whole PHE cost) [41], [42], [43]. Whereas in the newer study [44] updated to 2023 the cost of HEs is 122.3 USD/kWe for 240 MWth, assuming the HEs cost 70% of the whole PHE. Therefore, the calculations results are slightly under the estimates in the literature, but there is no consensus on the PHEs cost across the sources.

Table 11: Primary heat exchangers design parameters for steam cycle

Parameter	Value			
	Preheater	Evaporator	Superheater	Reheater
TEMA Type	B-F-U	B-E-U	B-E-U	B-E-U
Tube outer diameter, mm	25.4	25.4	16	25.4
Tube pitch ratio	1.5	1.25	1.5	1.25

The cost of the sCO_2 and CO_2 mixtures PHE varies between 22 and 26 MUSD making it 51.6-61.5% of the correlation from [8]. This can be explained by the use of Inconel 617 in the correlation [8] for all temperature ranges, while in the present analysis a much cheaper stainless steel 347 is used. Therefore, the updated correlation for PHE cost with power above 100MWth without installation (which is assumed to be 20% for PHE [45] in the original correlation) is as follows:

$$C_{PHE}^{2023}\left(UA\left[\frac{MW}{K}\right], \Delta P_{tube}[bar]\right) = k \cdot UA \cdot \Delta P_{tube}{}^{-0.2705}$$
$$k = \begin{cases} 2.610, if\ T_{max} \leq 575°C \\ 4.737, if\ T_{max} > 575°C \end{cases} \quad (1)$$

The overall metal mass of the steam turbine designed for this study is concluded to be 165 tons, 125 and 40 tons for casing and rotor, respectively. Subsequently, the material cost of the steam turbine is 1.16 MUSD. According to Akar et al. [46] the material costs contribute only to 5.75% of the overall manufacturing costs of steam turbines with standard design and high manufacturing volume for geothermal application. Assuming that the breakdown applies also to steam turbines for CSP plants, the resulting cost is 20.07 MUSD leading to 172.9 USD/kWe. For validation, a subcritical 325 MW plant [38] is scaled down according to [47], and the cost calculated is 198.6 USD/kWe. Based on the study [43], the bid cost of the 50 MW turbine is scaled up according to [47] to 160.0 USD/kWe, while the estimate is 193.3 USD/kWe assuming 20% installation cost and generator cost from [45]. On the other hand, the cost function given in [48] estimates the cost of the steam turbine as 236.7 USD/kWe or 36.9% higher than this study. Therefore, this study's results are on the lower end of the literature data which gives a cost of 160-195 USD/kWe.

The results of sCO_2 and CO_2 mixtures turbine model are shown in Table 12. Assuming the same material cost contribution as in the steam turbine the resulting cost of the turbines is 3.52-4.07 MUSD, which is 10.9-14.3% lower than the correlation results [45] showing a good agreement.

Table 12: sCO_2 and CO_2 mixtures turbines parameters

Working fluid	sCO_2			CO_2+$SiCl_4$		
Cycle	**SC**	**RC**	**PCC**	**SC**	**PreC**	**RC**
ψ	0.65	0.75	0.74	0.65	0.65	0.65
D_{hub}, mm	615	596	617	629	629	614
n_{stages}	16	12	17	14	15	15
Flow path length, m	2.4	2.5	2.7	2.1	2.3	2.7
SR	8.3	8.8	8.8	7.6	8.0	8.8
Mass rotor, tons	9.2	8.9	9.6	8.7	9.0	9.5
Mass casings, tons	17.7	20.0	18.7	16.5	18.4	18.6

Due to the simplified design of FWHs for the steam Rankine cycle; the FWH costs are expected to be underestimated. For comparison, the cost function provided in [49] is calibrated with the CSP plant inventory listed in [41], which includes vendor data, and in-house cost data for power block. The updated FWH cost function is presented in Eqn. 2.

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$$C_{FWH}^{2023}(Q[kW], TTD[K]) = 28 \cdot Q \cdot \left(\frac{1}{TTD + a}\right)^{0.1}$$
$$where\ a_{LP} = 3, a_{HP} = 6 \quad (2)$$

The costs of designed heat exchangers are found to be 46.6% and 51.2% of the costs calculated by Eqn. 2 for HP FWH and LP FWH, respectively. This difference in the cost estimations is regarded acceptable due to the simplified design of FWHs with a single-zone approach where actual FWHs have multiple zones such as condensing and drain cooling zones [35], and uncertainty of the materials used in the reference FWHs. For improved reliability and accuracy, the costs are calculated by Eqn. (2) for the FWHs of this study for further analysis and comparison. Finally, a similar approach with inventory data [41] calibrated cost correlation [50] is employed for the deaerator.

For sCO_2 and CO_2 mixtures recuperators, the costs obtained are above the ones predicted from the correlation [45]. In Figure 3 the optimized costs from Aspen EDR are compared with cost correlations based on heat exchanger efficiency. One can see a general increase in the cost with further efficiency increase above 95% as can be seen also in the PCHE literature before [51]. It should be noted that special baffle designs could improve the economics of sCO_2 and CO_2 mixtures recuperative heat exchangers [52].

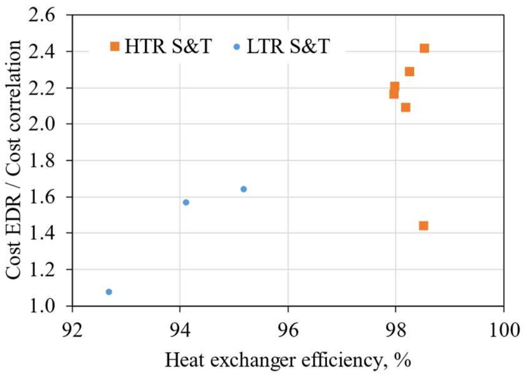

Figure 3: Recuperative system cost.

The cost of the air-cooled heat rejection unit for the steam cycle is calculated as 15.8 MUSD, resulting in the specific cost of 136.0 USD/kWe for 15°C ITD. In the study [53] the costs vary significantly based on ITD used from 236.7 USD/kWe for 16°C ITD to 157.3 USD/kWe for 24°C ITD. However, these numbers include also air removal equipment and support for start-up, training, and testing. The paper [54] focused on the 20 MW power plant gives a specific cost of 154.9 USD/kWe for ITD of around 30°C for ACC frame and fan. Therefore, current study results are within the literature range, but with current ITD higher costs of ACC are expected.

In the case of sCO_2 and CO_2 mixtures cycles the air temperature increase in the ACC is assumed to be 25°C [55] for cost correlation [45], while the fan's electric power is estimated to be 1% of thermal power [8]. A comparison of results between the Aspen EDR and correlation is presented in Figure 4. One can see that either the cost correlations or power consumption is underrated. Since the cost of the ACC is not significant in the final cycle cost, it is better to assume a 20-30% higher cost and lower fan power [56]. It should be noted that while for sCO_2 the resulting pressure drop is on average 75% of design, for CO_2+$SiCl_4$ mixture ACCs it is within 16-26%. This means that in the case of condensing cycle pressure drop in the air-cooled condenser can be assumed to be 0.5% of inlet pressure instead of 2% currently.

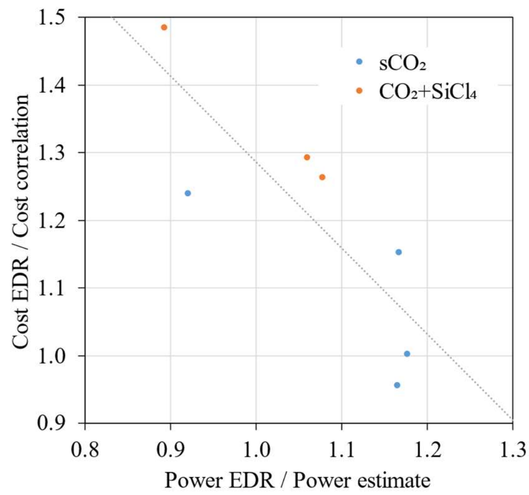

Figure 4: Condensers and coolers Cost vs Power.

The results of the main piping are presented in Table 11 for the cycles with both 2% Cr and 9% Cr Steel, A335 P22 and P91, respectively. One can see a significant increase in piping dimensions including outer diameter (OD) and thickness (δ) moving from steam to sCO_2 and CO_2 mixtures cycles due to both higher volume flow (V) and higher pressure in the latter case. The mass per meter for steam cycles main and reheat piping (accounting for 2 parallel pipes used) is 859 kg/m for P22 and 356 kg/m for P91, while for sCO_2 and CO_2 mixtures cycles it is between 1322 and 2445 kg/m.

Table 13: Main piping parameters

Fluid	Cycle	V, m³/s	Steel	OD, mm	δ, mm	Mass, kg/m
H_2O	Main	2.7	2% Cr	267	42	462
			9% Cr	217	16	162
	To RH	9.6	2% Cr	358	4	73
			9% Cr	355	3	53
	From RH	5.6	2% Cr	460	15	323
			9% Cr	444	7	141
sCO_2	SC	8.6		574	74	1824
	RC	11.5		666	86	2445
	PC	8.2		563	72	1750
CO_2+ $SiCl_4$	SC	6.2		490	63	1334
	PC	6.2		489	63	1322
	RC	8.6		576	74	1832

For the steam Rankine cycle, the pump costs are computed according to [57] and equal 10.47 USD/kWe, while FWHs are

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10.5 USD/kWe. Adding all the calculated costs the steam power cycle main equipment cost is 425.8 USD/kWe with 40.6% being the turbine cost, followed by 31.9% for ACC and 22.1% for PHE. However, the costs of PHE and ACC from the literature are both higher and are highly variable. If conservative assumptions from the literature are used the steam cycle main equipment cost reaches up to 800 USD/kWe. The whole power block with PHE and auxiliary systems cost is estimated as 1286-1350 USD/kWe in [44] and 1196-1270 USD/kWe in [43]. Therefore, taking into account that the additional costs are expected for installation (20-30%) and auxiliary equipment the cost of main equipment is considered reasonable.

In the case of sCO_2 and CO_2 mixtures, the compressors and pumps costs are calculated as compressors from [45]. For CO_2 and CO_2 mixture SC they are 82 and 63 USD/kWe respectively, while for the rest of the cycles they vary between 103 and 124 USD/KWe. The cost of PHE is consistent between cycles and is 210-243 USD/kWe. The same can be seen in the case of the turbine which costs 32.5-38.3 USD/kWe and ACC which costs 63-80 USD/kWe for most cycles, except the 117 USD/kWe for PCC. The main portion of the cost of the cycle is due to heat exchangers and specifically the recuperative system, the high costs of which are due to high efficiency. The cost of the recuperator is 288 and 639 USD/kWe for the sCO_2 and CO_2 mixture cycle respectively, HTR cost varies between 334 and 628 USD/kWe, while for LTR it is between 191 and 504 USD/kWe. The overall cost of cycles equipment for sCO_2 SC is 707 USD/kWe, 1592 USD/kWe for RC, 1056 USD/kWe for PCC, and 1056 USD/kWe for PCC. While for CO_2 mixture SC it is 1034 USD/kWe, 944 USD/kWe for PreC, and 1392 USD/kWe for RC. Compared to literature correlations the cost of all equipment is almost the same for sCO_2 PCC and CO_2 mixture PreC, while recompression cycles are 384 and 291 USD/kWe above. The sCO_2 has a 119 USD/kWe lower cost, while the mixture one has 213 USD/kWe.

The masses of equipment at high temperatures are shown in Figure 5. The mass of steam PHE is 1.7-2.2 times lighter than in the case of sCO_2 and CO_2 mixtures, while the steam turbine mass is 6 times higher. In general, sCO_2 and CO_2 mixtures cycles have a significantly higher mass (4-14 times) mainly due to recuperators. One can see that PHE mass is almost constant across the sCO_2 and CO_2 mixtures cycles, while turbine mass is insignificant compared to heat exchangers. However, the main concern is the recuperator especially the high-temperature part, which has both high mass and is subjected to high temperatures. It should be noted that in current work recuperators masses are overestimated due to high efficiency and adherence to shell and tube design. In the design done according to [58] lower mass and cost by a factor of 2 for HTR and 1.5 for LTR for shell and tube heat exchangers is expected. Another option is to move towards a different heat exchanger design to reduce the mass and increase flexibility, such as PCHE which is widely applied for sCO_2 cycles in literature. In that case, an order of magnitude reduction in mass is expected [59] which will make both cycles masses on par. However, currently, PCHEs can only reach tens of MWth and face scalability problems, which limit their application for the proposed hundreds of MWth range design [60].

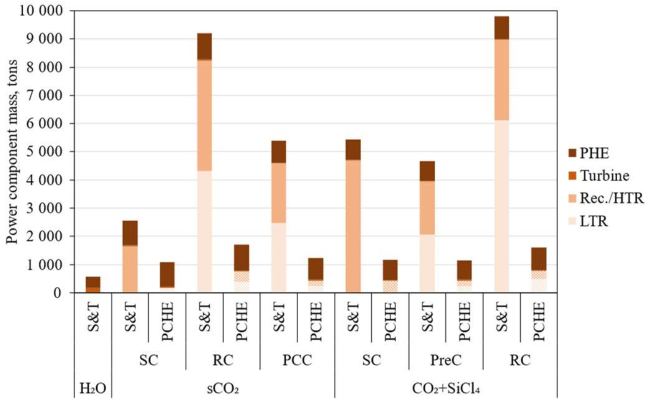

Figure 5: sCO_2 and CO_2 mixtures cycles mass distribution

CONCLUSIONS

Preliminary design of equipment for steam Rankine cycle and sCO_2 and CO_2 mixtures cycles were carried out with results compared to correlations and data available in the literature. Heat exchangers were designed with commercial software and high-temperature piping was sized according to ASME standards. Steam turbine preliminary design was obtained from the industrial partner, while for sCO_2 and CO_2 mixtures cycles a simplified approach was developed.

In the case of the steam Rankine cycle, the primary heat exchanger cost is found to be in good agreement with other research work but below most literature estimates. However, within the literature, the costs for only HEs vary by a factor of 2, which makes this piece of equipment both a significant part of the cycle cost and very uncertain. The air-cooled condenser designed in this study was found to be within the literature cost results, but on the lower side accounting for ITD influence. The turbine cost calculated is equal to lower values found in literature, while most of the literature costs are around 10-15% higher. On the other hand, it was concluded that the detailed feedwater heater design is crucial to determine their cost since there is a wide range of FWHs estimates in literature with significant deviation from the simplified design obtained in current work. Most of the steam power cycle cost is due to PHE, turbine, and ACC, as expected, among which PHE has the highest costs in literature and the highest uncertainty.

In the case of sCO_2 and CO_2 mixtures cycles the primary heat exchanger for 550°C TIT application was designed with a 347H stainless steel which reduces the overall cost by around a factor of 2 compared to literature results. Turbines preliminary design and cost estimation give results 10.9-14.3% lower than literature correlations. Designed recuperators have a significantly higher cost compared to the literature. Analysis shows that this is due to very high exchangers efficiencies and

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similar trends are found in the literature for PCHEs. Therefore, simplified correlations based on UA can be used only for heat exchangers with efficiency below 92-93%. Based on ACC design results it was found that either the cost or fan power has to be increased by 20-30% with cost being more preferable since it is not significant for sCO_2 and CO_2 mixtures cycles. On the other hand, for condensing CO_2 mixtures cycles the working fluid pressure drop can be decreased from 2% of inlet pressure to 0.5%. The rest of the equipment, however, doesn't show significant sensitivity to the transition from sCO_2 to CO_2 mixtures.

Regarding the cost, for both cycles, only the main equipment was considered. Calculation results of the steam cycle give values of around 425 USD/kWe, however, based on conservative literature values the cost can be up to 800 USD/kWe. The sCO_2 simple cycle has equipment cost of 707 USD/kWe which is between average and the conservative steam Rankine power cycle values, followed by precompression cycle with a cost of 944 USD/kWe, sCO_2 partial cooling and CO_2 mixture simple cycles with a cost of 1057 and 1034 USD/kWe, and sCO_2 and CO_2 mixture recompression cycles with 1593 and 1392 USD/kWe. The difference in cycles working principles leads to different cost distributions. For the steam Rankine cycle, most of the power unit cost is due to the PHE, turbine, and air-cooled condenser. On the other hand, for supercritical and transcritical cycles, most of the cost is associated with the recuperative system and primary heat exchanger, which is due to a significant portion of heat being recuperated and large mass flows. It should be noted that for the steam cycles, there are commercial estimates available based on vendors, who already produced equipment on a similar scale, while for sCO_2 the best data available are correlations based on vendor quotes, cost estimates, and published literature, while the equipment wasn't yet produced at that scale.

Regarding flexibility, there are two main considerations: the amount of heat needed to be supplied to the cycle equipment to reach the operating temperature and the limiting factor of thermal stress in high-temperature parts of the cycle. In the first case, the steam Rankine PHE mass is around two times smaller than sCO_2 and CO_2 mixtures cycles PHEs, while the average operating temperature is lower for the steam cycle. The turbine mass is around 6 times higher in the steam case, while the Rankine cycle regenerative system is around 100 tons and much lighter than sCO_2 and CO_2 mixtures recuperative systems. However, it should be noted that a different recuperator heat exchanger technology could reduce the weight by an order of magnitude. Regarding the thermal stresses, sCO_2 and CO_2 mixtures cycles have higher pipe and turbine casing thickness due to around two times higher maximum pressure and large volume flows. On the other hand, the length of pipes may be expected to be shorter in the case of sCO_2 and CO_2 mixtures due to the lack of reheat. Finally, the thick drum walls of the steam cycle should be taken into consideration.

From the operation and maintenance point of view steam Rankine cycle will require water treatment facility, while for sCO_2 and CO_2 mixture cycles the working fluid management system is needed to fill, empty and regulate the amount of fluid in the cycle and cycle minimum pressure in the case of sCO_2. For CO_2 mixture there is also a need of a hot well similar to the Steam Rankine cycle one.

Further studies will focus on the reduction of sCO_2 and CO_2 mixture recuperators cost and weight along with the thermal inertia estimation and techno-economic optimization of the most promising cycles.

NOMENCLATURE

ACC	air-cooled condenser or cooler
CSP	concentrating solar power
EDR	Aspen Exchanger Design & Rating
FWH	feedwater heater
HE	heat exchanger
HP	high pressure
HTR	high temperature recuperator
ITD	initial temperature difference (K)
LP	low pressure
LTR	low temperature recuperator
MITA	minimum internal temperature approach
OD	outer diameter, mm
PCC	partial cooling cycle
PCHE	printed circuit heat exchanger
PHE	primary heat exchanger
PreC	precompression cycle
RC	recompression cycle
RH	steam cycle reheat
SC	simple cycle
sCO_2	supercritical CO_2
SoA	state-of-art
SR	slenderness ratio
TTD	terminal temperature difference (K)

C	cost
D_{hub}	hub diameter (mm)
H	blade height
n_{stages}	number of turbine stages
Q	thermal power (kW)
V	volumetric flow rate (m^3/s)
U	circumferential velocity (m/s)
UA	the overall heat transfer coefficient multiplied by the heat exchange area (MW/K)
δ	pipe thickness, mm
σ_b	bending stress (MPa)
φ	flow coefficient
ψ	loading coefficient
ΔT	temperature difference (K)
ΔP	pressure drop (bar)

AUTHOR CONTRIBUTIONS

V.N.- Conceptualization, Methodology, Validation,

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Formal Analysis, Writing-Original Draft, Visualization; **E.G-** Conceptualization, Methodology, Validation, Formal analysis, Writing - Original Draft, Visualization, Project administration; **N.P.** – Conceptualization, Methodology, Writing - Review & Editing, Project administration, **G.D.M**. – Supervision, Funding acquisition; **P.I**.- Supervision, Funding acquisition, Writing - Review & Editing; **M.W.** – Conceptualization, Supervision, Funding acquisition.

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