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1 Thermodynamic Performance Evaluation of Supercritical 2 CO₂ Closed Brayton Cycles for Coal-fired Power 3 Generation with Solvent-based CO₂ Capture 4

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11 **Abstract**

12 Power generation from coal-fired power plants represents a major source of CO₂ emission into
13 the atmosphere. Efficiency improvement and integration of carbon capture and storage (CCS)
14 facilities have been recommended for reducing the amount of CO₂ emissions. The focus of
15 this work was to evaluate the thermodynamic performance of s-CO₂ Brayton cycles coupled
16 to coal-fired furnace and integrated with 90% post-combustion CO₂ capture. The modification
17 of the s-CO₂ power plant for effective utilisation of the sensible heat in the flue gas was
18 examined. Three bottoming s-CO₂ cycle layouts were investigated, which included a newly
19 proposed single recuperator recompression cycle. The performances of the coal-fired s-CO₂
20 power plant with and without carbon capture were compared. Results for a 290 bar and 593
21 °C power cycle without CO₂ capture showed that the configuration with single recuperator
22 recompression cycle as bottoming cycle has the highest plant net efficiency of 42.96% (Higher
23 Heating Value). Without CO₂ capture, the efficiencies of the coal-fired s-CO₂ cycle plants
24 were about 3.34-3.86% higher than the steam plant and about 0.68-1.31% higher with CO₂
25 capture. The findings so far underscored the promising potential of cascaded s-CO₂ power
26 cycles for coal-fired power plant application.

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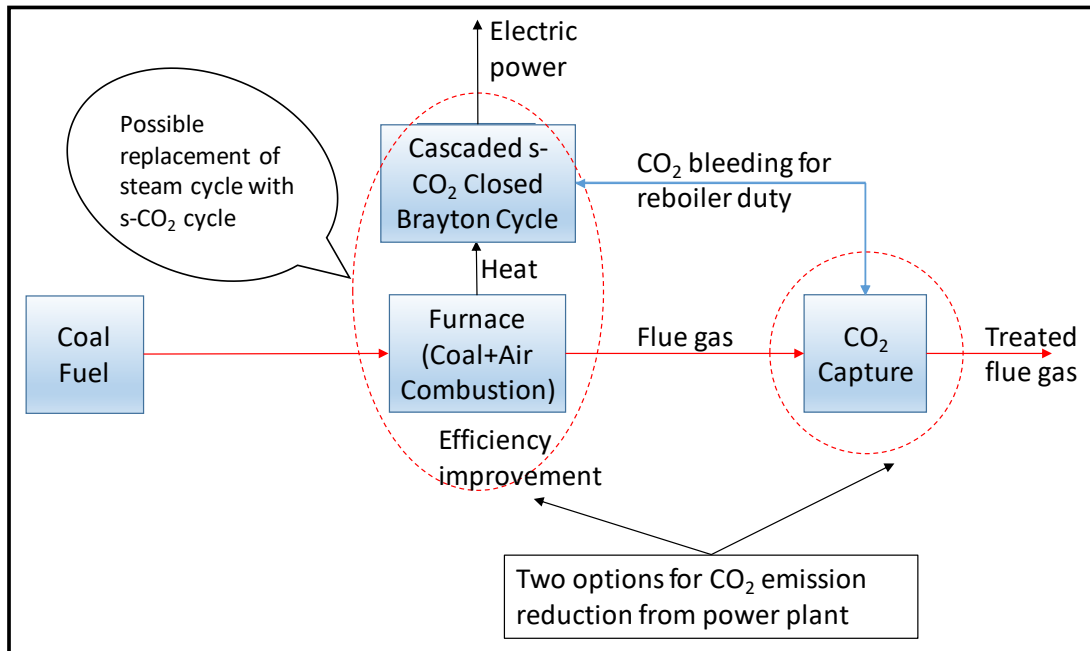
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37 Graphical abstract



38

39 Keywords

40 Coal-fired power plant;
41 Supercritical CO₂ Brayton cycle;
42 Carbon capture;
43 Chemical absorption;
44 Process modelling/simulation;

45

46 Highlights

- 47 • Supercritical CO₂ cycle was investigated for coal-fired power plant application
- 48 • Three s-CO₂ cycles were investigated as possible bottoming cycle options
- 49 • Integration of coal-fired plant with post-combustion CO₂ capture was studied
- 50 • Thermodynamic analysis and performance comparison were performed

51

52 Nomenclature and Units

53 Abbreviations

CCS	carbon capture and storage
ESP	electrostatic precipitator
FD	forced draft
FGD	flue gas desulfurization

HP	high-pressure
HHV	higher heating value
HTR	high temperature recuperator
ID	Induced draft
LP	low-pressure
LTR	low temperature recuperator
MC	main compressor
MEA	monoethanolamine
PA	primary air
PCC	post-combustion CO ₂ capture
PSD	particle size distribution
RC	recompression compressor
s-CO ₂	supercritical carbon dioxide
TTD	terminal temperature difference
USC	ultra-supercritical

54

55 **Symbols**

C_i	Concentration of the <i>i</i> th component (mol/m ³)
E	Activation energy (J/mol)
HHV	Higher heating value (J/kg)
k	Pre-exponential factor
\dot{m}	Mass flow rate (kg/s)
n	Temperature exponent
N	Number of components
P	Power (watt or J/s)
Q	Heat transferred (watt or J/s)
r	Rate of reaction
R	Universal gas constant
T	Temperature (K)
α_i	Exponent of the <i>i</i> th component
η	Efficiency
π	Product operator
Σ	Sum operator

56

57 **Subscripts**

<i>aux</i>	auxiliary
<i>C</i>	Compressor
<i>elec</i>	Electrical
<i>gen</i>	Generator
<i>HP</i>	High pressure
<i>i</i>	Component index
LP	Low pressure
MC	Main compressor
RC	Recompression compressor
<i>T</i>	Turbine

58

59 1 Introduction

60 1.1 Background

61 Coal-fired power plants are still playing a significant role in meeting world energy demands
62 and it is expected to remain a key component of the global energy mix into the future due to
63 its reliability, security of fuel supply, cheap fuel and competitive cost of electricity [1, 2].
64 However, one prime concern about continued use of fossil fuels like coal is the emission of
65 CO₂ to the atmosphere. Therefore, reducing CO₂ emissions from coal-fired power has become
66 a policy focus in many countries. Two options that have been identified for mitigating CO₂
67 emissions from fossil fuel power plants are CCS (carbon capture and storage) and efficiency
68 improvement. Post-combustion CO₂ capture (PCC) by chemical absorption with solvent is
69 currently the most preferred CCS option [3, 4]. Efficiency improvement usually requires
70 increased main steam temperature and pressure. Hence, the state-of-the-art technology for
71 coal-fired power generation, the ultra-supercritical (USC) steam plant, now operates at a steam
72 pressure up to 300 bar and temperature up to 600 °C with reheat [5]. However, CCS systems
73 and efficiency improvement have their limitations. Integration of PCC system with fossil fuel
74 power plants leads to significant efficiency penalty and increased cost of electricity generation.
75 Also, lack of advanced materials to withstand harsh operating conditions limits further
76 improvement in efficiency.

77 In this paper, to improve the efficiency of coal-fired power plants, supercritical carbon dioxide
78 (s-CO₂) Brayton cycle is considered as an alternative to the conventional steam Rankine cycle.
79 Additionally, CO₂ capture is facilitated by integrating an aqueous monoethanolamine (MEA)-
80 based PCC system with the s-CO₂ cycle power plant. S-CO₂ Brayton cycle has been found to
81 have higher cycle efficiency than steam Rankine cycle and other gas Brayton cycles in the
82 temperature range typically encountered in pulverised coal-fired power plant (450 °C to 650
83 °C) [6-10]. Other potential benefits of s-CO₂ cycle compared to steam cycle include [6, 11-
84 19]:

- 85 • Smaller size of the components
- 86 • Less complex system layout
- 87 • Less risk of corrosion and scaling and no formation of water droplets that could
88 damage the turbine blades [10, 20]
- 89 • Reduced water consumption [14, 19]

90 1.2 Review of s-CO₂ power cycle

91 A CO₂ closed Brayton cycle was originally patented by Sulzer in 1950 [21]. Later in the 1960s,
92 Feher [10, 22], Angelino [12, 23] and Dekhtiarev [24] all investigated s-CO₂ power cycle.
93 Feher identified CO₂ as a suitable working fluid due to its unique properties such as low critical
94 pressure, good thermal stability at temperature of interest, inertness, availability of property
95 data, and abundant, non-toxic and inexpensive [10]. Angelino concluded that s-CO₂ power
96 cycle has the potential to perform better than reheat steam cycle on account of efficiency,
97 simplicity and compactness [12]. Dekhtiarev [24] studied condensing reheated s-CO₂ cycles
98 as a good alternative to steam cycle for fossil fuel plant [25]. According to recent
99 comprehensive reviews by Olumayegun et al. [26], Ahn et al. [19], Crespi et al. [27] and Li et
100 al. [28], s-CO₂ power cycles are currently being widely investigated as power conversion
101 system for application in nuclear, fossil, concentrated solar power, biomass, and waste heat
102 recovery systems because of its advantages [8, 11, 15, 17, 20, 29-31].

103 Dostal [11] contributed to renewed interest in s-CO₂ power cycle for nuclear reactor
104 application by providing a detailed analysis based on thermodynamic performance and cost.
105 The study showed that s-CO₂ cycle achieved higher thermal efficiency and reduced cost of
106 power plant compared to steam cycle at 550 °C turbine inlet temperature. Studies by Pharm et

107 al.[8] concluded that the s-CO₂ recompression cycle in condensing mode is the most fitting
108 configuration for pressurised water reactor (PWR) and sodium-cooled fast reactor (SFR)
109 application. The use of mixture of CO₂ with additive gases to improve the performance of s-
110 CO₂ cycle of a nuclear reactor was investigated by Hu et al. [17]. Even though s-CO₂ cycle is
111 usually viewed to provide superior thermodynamic performance than steam cycle only in the
112 medium to high-temperature range (greater than 450 °C), Santini et al. [25] investigated the
113 adoption of s-CO₂ cycle for a far lower temperature (about 260 °C) of an existing PWR. The
114 results indicated that a reheated recompression s-CO₂ cycle achieved a net cycle efficiency of
115 about 34% compared to 33.5% of the existing steam cycle and the plant footprint was 10 times
116 smaller than the steam cycle plant.

117 For concentrated solar power (CSP), Chacartegui et al. [32] investigated two stand-alone s-
118 CO₂ cycle configurations and a combined cycle (comprising a topping s-CO₂ cycle and a
119 bottoming Organic Rankine Cycle (ORC)) as an alternative to the conventional steam cycle.
120 Preliminary results from the study showed that the s-CO₂ cycles could provide both efficiency
121 and cost benefits. Al-Sulaiman and Atif [33] compared the performance of five different s-
122 CO₂ Brayton cycle configurations for CSP application and the recompression cycle was found
123 to give the best efficiency. Recompression and partial cooling cycles were compared by Neises
124 and Turchi [29] for CSP, highlighting the potential reduction in cost and improvement of CSP
125 receiver efficiency with the partial cooling cycle. Recently, Wang et al. [34] reviewed and
126 compared the main s-CO₂ cycle configurations integrated with molten salt solar power towers
127 having both the main heater and a reheater.

128 S-CO₂ cycles and the various configurations have also been investigated as bottoming cycles
129 for fuel cell [20] and gas turbine system [15] as well as an alternative power conversion system
130 for other waste heat recovery processes [35, 36] and biomass plants [7]. Bae et al. [20]
131 investigated s-CO₂ cycle configurations comprising an s-CO₂ Brayton-steam Rankine cycle
132 cascade, a recompression cycle and two simple recuperated cycle (a supercritical and a trans-
133 critical cycle) as bottoming cycles for molten carbonate fuel cell. Kim et al. [15] compared the
134 performance of nine s-CO₂ cycle layouts together with three newly developed concept as
135 bottoming cycles for gas turbine plant. It was concluded that although the recompression cycle
136 has a good cycle efficiency, it is not suitable as a bottoming cycle due to its poor heat recovery
137 factor. Small to medium-scale biomass power plant employing either a simple recuperated or
138 a recompression s-CO₂ Brayton cycle as topping cycle and a simple recuperated s-CO₂ Brayton
139 cycle as bottoming cycle was studied by Manente and Lazzaretto [7]. Results of performance
140 optimisation showed that the cascaded s-CO₂ Brayton cycles plant could achieve about 10%
141 higher efficiency than existing biomass plant.

142 Various researchers have also proposed adaptation of s-CO₂ Brayton cycle as the main/topping
143 cycle for coal-based power plant. However, one problem of such application is the inefficient
144 utilisation of the heat content of the flue gas [16, 30, 31]. Mecheri and Le Moullec [30]
145 investigated the performance of coal-fired s-CO₂ Brayton cycle by comparing the effects of
146 number of reheat and number of recompression, and the effects of advanced flue gas
147 economiser configurations. Heat utilisation was improved by transferring flue gas heat to a
148 fraction of cold CO₂ working fluid taken from the main compressor outlet as well as preheating
149 of combustion air. Results showed that the plant net efficiency was higher than that of
150 supercritical and USC steam plant by 5.3% and 2.4% respectively. Le Moullec [31] presented
151 a conceptual study of coal-fired s-CO₂ Brayton cycle integrated with 90% post-combustion
152 amine-based CO₂ capture unit. Performance improvement entailed the use of double reheat
153 configuration, cold CO₂ bleeding from two locations and two stages of combustion air
154 preheating. Technical and economic evaluation of the plant showed that 15% reduction in
155 levelised cost of electricity and 45% reduction in the cost of avoided CO₂ emission could be
156 achieved. Hanak and Manovic [16] proposed s-CO₂ cycle instead of the conventional steam
157 cycle for electricity generation from the high-grade heat of calcium looping process. Results

158 of retrofitting the calcium looping process with s-CO₂ recompression cycle indicated that a
159 gain in efficiency of about 1-2% over that of the steam cycle could be obtained.

160 1.3 Aim of this study and its novelties

161 The aim of this paper is to evaluate the thermodynamic performance of coal-fired s-CO₂
162 Brayton cycle power plant that has been adapted for efficient utilisation of flue gas heat by
163 using a bottoming s-CO₂ Brayton cycle in conjunction with a main/topping s-CO₂ Brayton
164 cycle. So far, the use of s-CO₂ Brayton cycles as both topping cycle and bottoming cycle of a
165 coal-fired power plant has not been explored in the literature. In this study, a single reheat s-
166 CO₂ recompression cycle was considered as the topping cycle while three simpler s-CO₂ cycle
167 were investigated as possible bottoming cycle for recovering the excess heat in the flue gas
168 exiting the furnace. The investigated bottoming cycle options are simple recuperated cycle,
169 partial heating cycle and a newly proposed concept referred to as single recuperator
170 recompression cycle. Performance evaluation was performed both for s-CO₂ cycle plants
171 without CO₂ capture and for plants with CO₂ capture unit integrated. The performances of the
172 different coal-fired s-CO₂ cycle configurations were compared with reference to a supercritical
173 steam cycle that was chosen as the benchmark. The most promising of the layouts was
174 determined and the effects of cycle parameters such as turbine inlet temperature, precooler
175 outlet temperature/pressure and recuperator's minimum terminal temperature difference (TTD)
176 on the plant performance were investigated. The whole system comprising the coal-fired
177 furnace, the s-CO₂ cycles and the MEA-based PCC plant were modelled and simulated with
178 Aspen Plus software.

179 2 Process configurations and description

180 2.1 Supercritical CO₂ closed Brayton cycle

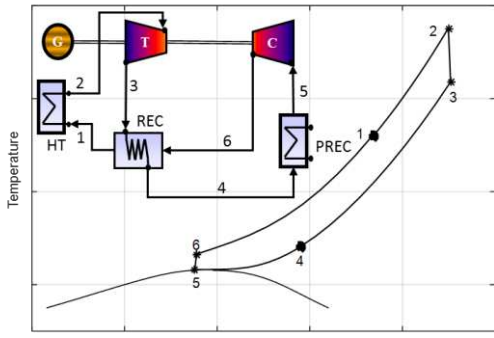
181 A unique feature of CO₂ as working fluid is that its critical pressure (7.3773 MPa) and
182 temperature (30.978 °C) are easily achievable. The properties of CO₂ vary rapidly around the
183 critical point and the density is greatly increased. Hence, s-CO₂ cycles take advantage of the
184 increased density by operating the compressor inlet close to the critical point so that the
185 compression work is significantly reduced. The reduced compression work thus enables the
186 achievement of high thermodynamic efficiency.

187 The baseline closed Brayton cycle is the simple recuperated cycle. The layout and T-S diagram
188 of a simple recuperated s-CO₂ Brayton cycle are shown in Figure 1a. It consists of a heat
189 source (1 – 2), a turbine (2 – 3), a recuperator (3 – 4 & 6 – 1), a precooler (4 – 5) and a
190 compressor (5 – 6). Though the rapidly varying fluid properties around the critical point is a
191 feature that facilitates the reduced compression work of s-CO₂ cycle, it also prevents effective
192 heat transfer in the recuperator of simple recuperated s-CO₂ cycle. This is due to mismatch of
193 specific heat capacity between the high-pressure CO₂ in the cold side and the low-pressure
194 CO₂ in the hot side of the recuperator. This could lead to temperature cross over in the
195 recuperator (the so-called “pinch point problem”) and consequently, the cold stream cannot be
196 preheated high enough to achieve good recuperator effectiveness. Hence, it is difficult to
197 achieve high efficiency in simple recuperated s-CO₂ cycle even with the conventional methods
198 of enhancing efficiency such as reheating and intercooling because of the excessively low
199 effectiveness of the recuperator [37].

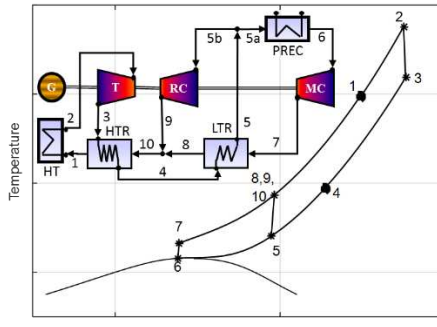
200 Other complex layouts have been suggested in the literature to minimise the detrimental effects
201 of the differences in heat capacities [10-12, 15, 19, 38]. Of all the layouts, the recompression
202 s-CO₂ cycle (Figure 1b) is generally considered the most promising with the highest
203 thermodynamic efficiency and a relatively simpler configuration than most others [11]. A
204 component count of the different layouts by Kim et al. [15] showed that only the simple
205 recuperated (Figure 1a) and the partial heating cycle (Figure 1c) is simpler (fewer components)

206 than the recompression cycle. Hence, this study considered only the simple recuperated cycle,
 207 the recompression cycle and the partial heating cycle. However, an additional new cycle
 208 concept referred to as single recuperator recompression s-CO₂ cycle (Figure 1d) was proposed.
 209 The newly proposed layout has one component less than the recompression cycle and just one
 210 component more than the simple cycle. This configuration has been investigated previously in
 211 an initial study by the authors [39]. Conboy et al. [40] suggested a similar but more complex
 212 cycle layout for geothermal heat application.

213 In the recompression cycle (Figure 1b), the recuperator is separated into two: the high-
 214 temperature recuperator (HTR) and the low-temperature recuperator (LTR). The problem of
 215 heat capacity mismatch is resolved by splitting the flow into two streams at point 5. The main
 216 stream is cooled in the precooler (point 5 to point 6) to the main compressor (MC) inlet
 217 temperature. The second stream is compressed directly in the recompressing compressor (point
 218 5 to point 9) and mixed with the main flow at the exit of the LTR cold stream (point 8) before
 219 entering the cold side of the HTR (point 10). The flow split fraction can be adjusted to make
 220 the heat capacity (i.e. the product of mass flow rates and specific heat capacity) of CO₂ on the
 221 high-pressure side of the LTR the same as that of the low-pressure side CO₂. Hence, with an
 222 optimal selection of flow split fraction, high recuperator effectiveness and consequently high
 223 thermodynamic efficiency can be achieved. The layout, as well as T-S diagram of partial
 224 heating cycle, is shown in Figure 1c. Matching of the heat capacities of the recuperator streams
 225 is achieved by splitting the flow at the compressor outlet (point 6) after compressing the fluid
 226 to the maximum cycle pressure in the compressor (point 5 to 6). The new concept, the single
 227 recuperator recompression cycle is shown in Figure 1d. It is similar to the recompression cycle
 228 except that the HTR was eliminated leaving only one recuperator. The flow is split into two
 229 streams at point 4, just like the recompression cycle. This permits the advantage associated
 230 with splitting the flow, that is, a balance of the heat capacity between the cold stream and the
 231 hot stream of the recuperator.



(a) Simple recuperative closed Brayton cycle



232 Figure 1 Layout and T-S diagrams of simple recuperative, recompression, partial heating and
 233 single recuperator recompression s-CO₂ closed Brayton cycles [18-20, 22]

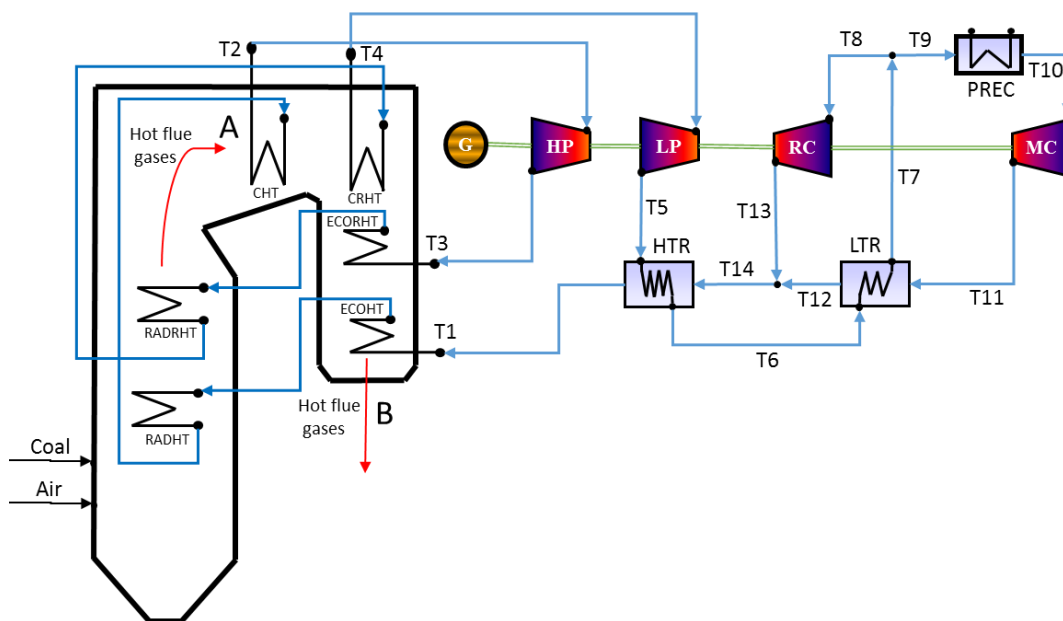
234 **2.2 Supercritical CO₂ Brayton cycles for pulverised coal-fired**
 235 **application**

236 **2.2.1 Coal-fired furnace and the main s-CO₂ closed Brayton cycle**

237 Integration of the main/topping s-CO₂ cycle with the coal-fired furnace is shown in Figure 2.
 238 A recompression s-CO₂ cycle was adopted due to its superior performance when compared to
 239 other s-CO₂ cycle layouts. The performance is further improved with a single stage of reheat.
 240 Preheated CO₂ coming from the HTR entered the furnace at point T1 and exit at T2 after being
 241 heated to the maximum cycle temperature. The hot working fluid is expanded in the high-
 242 pressure (HP) turbine and returned to the furnace at point T3 for reheating. The reheated CO₂
 243 exiting the furnace at T4 is finally expanded in the LP turbine. During each pass through the
 244 furnace, the CO₂ working fluid is heated in three steps: convective economiser
 245 (ECOHT/ECORHT), radiant heater (RADHT/RADRHT) and final convective heater or
 246 reheater (CHT/CRHT).

247 Radiant section of the furnace contains the two radiant heaters while the convective section
 248 contains the four convective heaters. Approximately half of the heat transferred to the CO₂ is
 249 through radiation from the flame to the radiant heaters. Combustion products rise to the top of
 250 the furnace and entered the convection zone at point A. The temperature of the hot flue gases
 251 at A was maintained at 1010 °C so that it was below ash softening temperature [41]. As the
 252 flue gases flow through the convective section, they are first used for final heating of CO₂ to
 253 turbine inlet temperature in the convective heater and reheater. Then CO₂ leaving the HTR and
 254 HP turbine are heated in the economisers to the radiant heaters inlet temperature. The flue
 255 gases leave the furnace at point B. The CO₂ entering the furnace at T1 is at a higher temperature
 256 (about 465 °C) than the usual feedwater temperature in conventional coal-fired steam boiler
 257 (about 260 °C) [42]. This is due to the high level of recuperation in recompression cycle.
 258 Consequently, the flue gases leave the furnace at relatively high temperature (about 495 °C) in
 259 the coal-fired s-CO₂ cycle power plant.

260



261

262 Figure 2 Main single reheat recompression cycle integration with coal-fired furnace

263 2.2.2 Utilisation of flue gases residual heat

264 A major drawback of coupling closed Brayton cycle to coal-fired furnace is the significant loss
265 of heat through the hot flue gases leaving the furnace. If this exiting flue gases are not utilised,
266 it will represent the main cause of inefficiency in the power plant [41]. Several options exist
267 for utilising waste heat of flue gases. The first option is to use the flue gases to produce steam
268 or hot water for industrial use or district heating in a combined heat and power (CHP) system.
269 In fact, some of the early-operated coal-fired closed Brayton cycle plants such as the
270 Oberhausen and Kashira plants were used to generate electricity as well as to produce heat for
271 district heating [43]. Secondly, the hot flue gases can be used to preheat part or all of the cycle
272 working fluid prior to the main heat addition in the furnace. Mecheri and Le Moullec [30]
273 employed this option by transferring the flue gases heat to a fraction of CO₂ flow that is
274 extracted from the MC outlet. A third option is to add a bottoming cycle that uses the flue
275 gases high-grade heat to generate additional electrical power [7, 15]. For instance, Echogen
276 (USA) is in the process of commercialising s-CO₂ bottoming power cycle utilising waste heat
277 [44]. The final option is to use the flue gases to preheat the incoming combustion air. This is
278 a common practice in conventional coal-fired power plants.

279 In this study, the use of bottoming cycle in conjunction with combustion air preheating was
280 selected. In bottoming cycles, the net electric efficiency is a function of not just cycle
281 efficiency (ratio of net electric power produced to heat transferred to the cycle) but also of the
282 heat recovery factor (ratio of recovered heat to available heat in the flue gas) [15]. Closed
283 Brayton s-CO₂ cycle has favourable cycle efficiency. However, when used as a bottoming
284 cycle, the heat recovery in the heater is limited by the high temperature of CO₂ leaving the
285 recuperator [15]. However, the addition of air preheater downstream of the bottoming cycle
286 will help to improve the plant's overall heat recovery factor. Recompression cycle was not
287 used as bottoming cycle in this study. Cycles with simpler layouts and better heat recovery
288 factor were favoured. Hence, the simple recuperated cycle, the partial heating cycle and the
289 newly proposed single recuperator recompression cycle were considered as bottoming cycles
290 in cascade with the main/topping single reheat recompression s-CO₂ cycle.

291 2.2.3 Overall plant configurations and its integration with PCC

292 In this study, three coal-fired s-CO₂ cycle configurations (Figure 3, Figure 4 and Figure 5)
293 representing three different bottoming cycle choices were investigated:

- 294 • Case A: the simple recuperated s-CO₂ cycle was selected as bottoming cycle as shown
295 in Figure 3
- 296 • Case B: shown in Figure 4, the bottoming cycle is the partial heating s-CO₂ cycle
- 297 • Case C: the new concept, the single recuperator recompression s-CO₂ cycle was used
298 as the bottoming cycle (Figure 5)

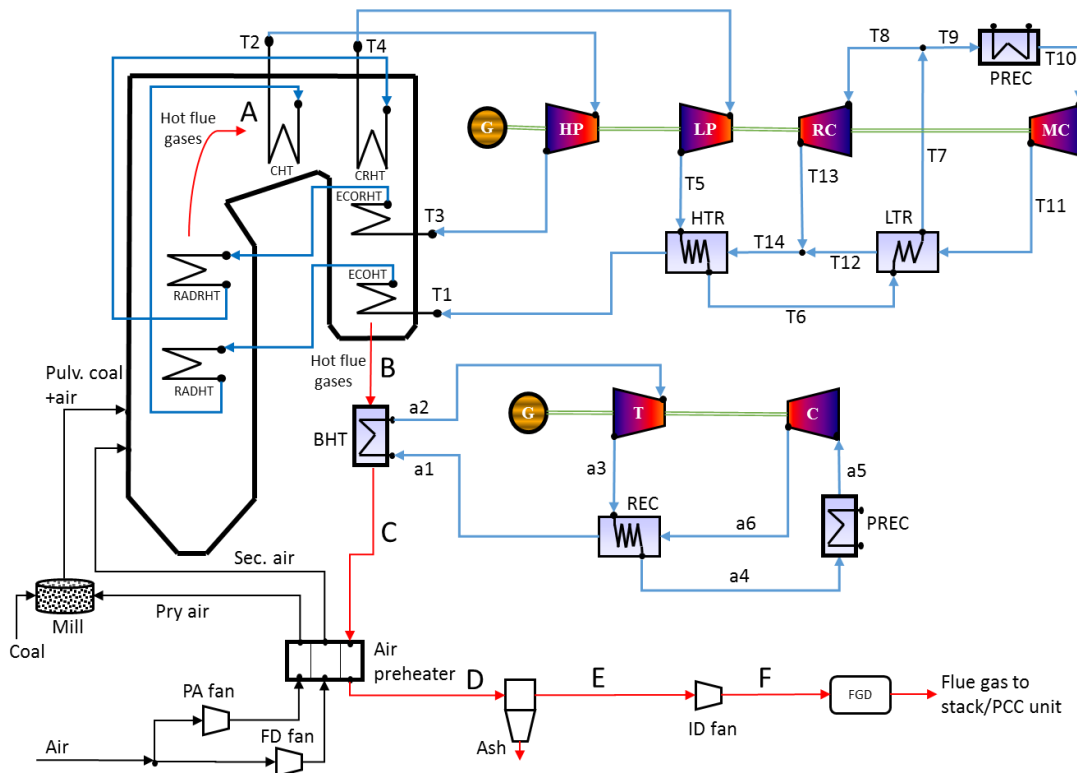
299 In all the cases, the topping cycle remains the single reheat recompression s-CO₂ cycle
300 integrated with coal-fired furnace. Coal is pulverised to fine powder in the mill. Secondary air,
301 which is a large proportion of the incoming air, is sent to the forced draft (FD) fan while the
302 remaining incoming air goes to the primary air (PA) fan. Air from the PA fan and FD fan is
303 heated in the air preheater thereby recovering part of the remaining heat content of the flue gas
304 exiting the bottoming cycle heater at point C. The heated primary air goes to the mill/pulveriser
305 for drying and conveying the pulverised coal to the burners in the furnace. The heated
306 secondary air is also introduced into the burners, where the coal and the air are mixed and
307 combustion takes place. Heat released from the combustion is transferred to the CO₂ working
308 fluid in the radiant and convective heaters.

309 The cooled flue gas leaving the air preheater passes through fabric filters or electrostatic
310 precipitator (ESP) for particulate matters (majorly ash) removal. An induced draft (ID) fan
311 increases the flue gas pressure to provide suction to the flue gas in the furnace and for the flue

312 gas to pass through the flue gas desulfurization (FGD) unit. The cleaned flue gas leaving the
 313 FGD unit is finally sent either to the PCC unit to remove the CO₂ in the flue gases or directly
 314 to the stack.

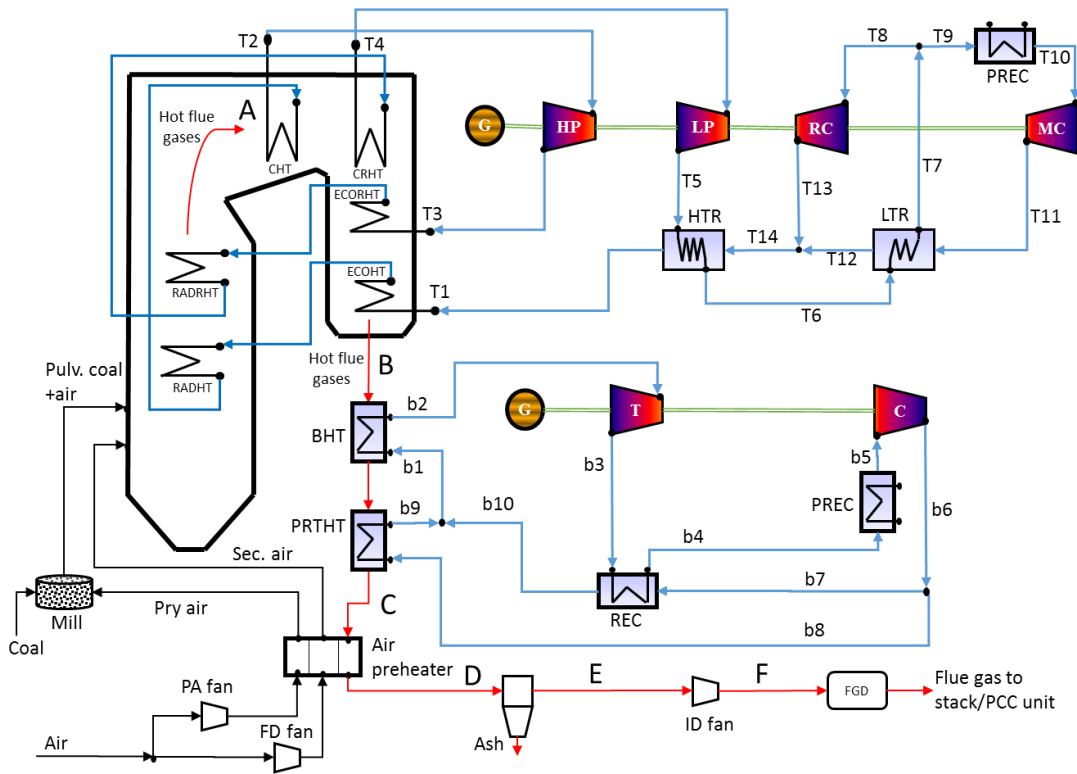
315 The s-CO₂ Brayton cycle will need to be altered when a PCC unit is added. In the conventional
 316 coal-fired plant, low pressure saturated steam from steam turbine is used for solvent
 317 regeneration in the PCC unit. However, in the coal-fired s-CO₂ Brayton cycle plant, sensible
 318 heat of the CO₂ working fluid is used for solvent regeneration. Hence, each of the three cases
 319 is integrated with the PCC unit as shown in Figure 6. Hot CO₂ from the HTR hot stream outlet
 320 is conveyed to the reboiler of the PCC unit. The CO₂ is then returned to the s-CO₂ cycle at the
 321 LTR hot stream outlet after supplying the required reboiler duty. The flue gas from the power
 322 plant is stripped of its CO₂ before being sent to the stack. A detailed description of the PCC
 323 unit is provided in Section 4.

324



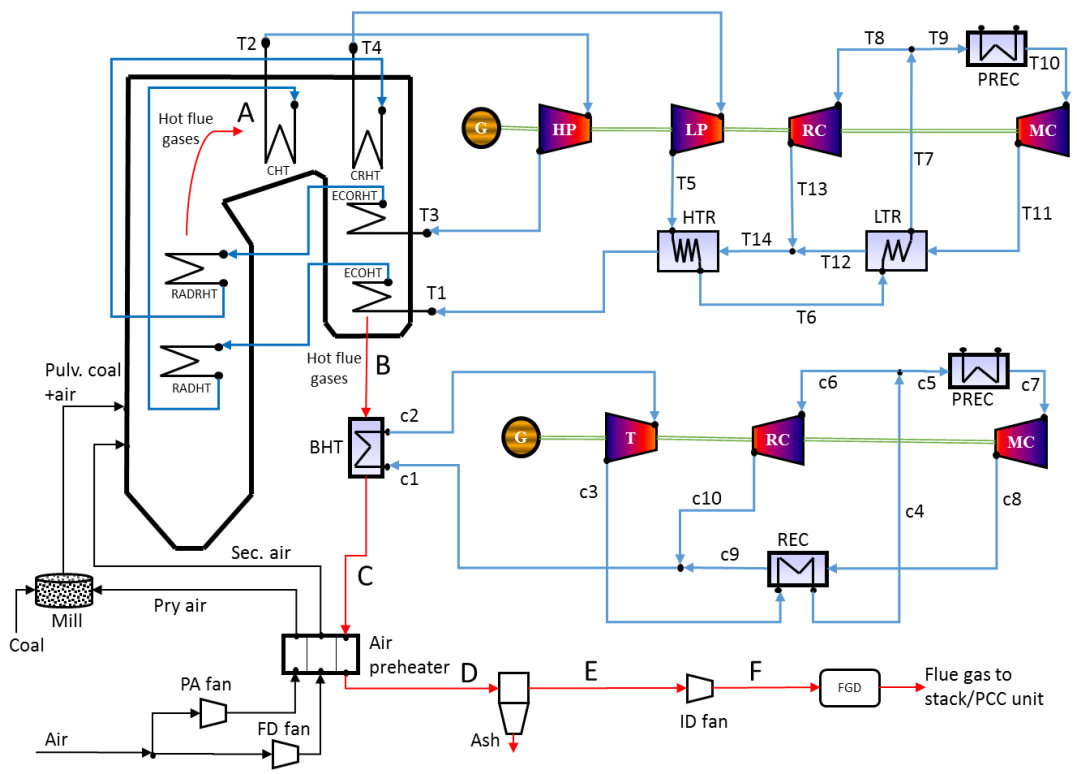
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326 Figure 3 Case A - Simple recuperative bottoming cycle



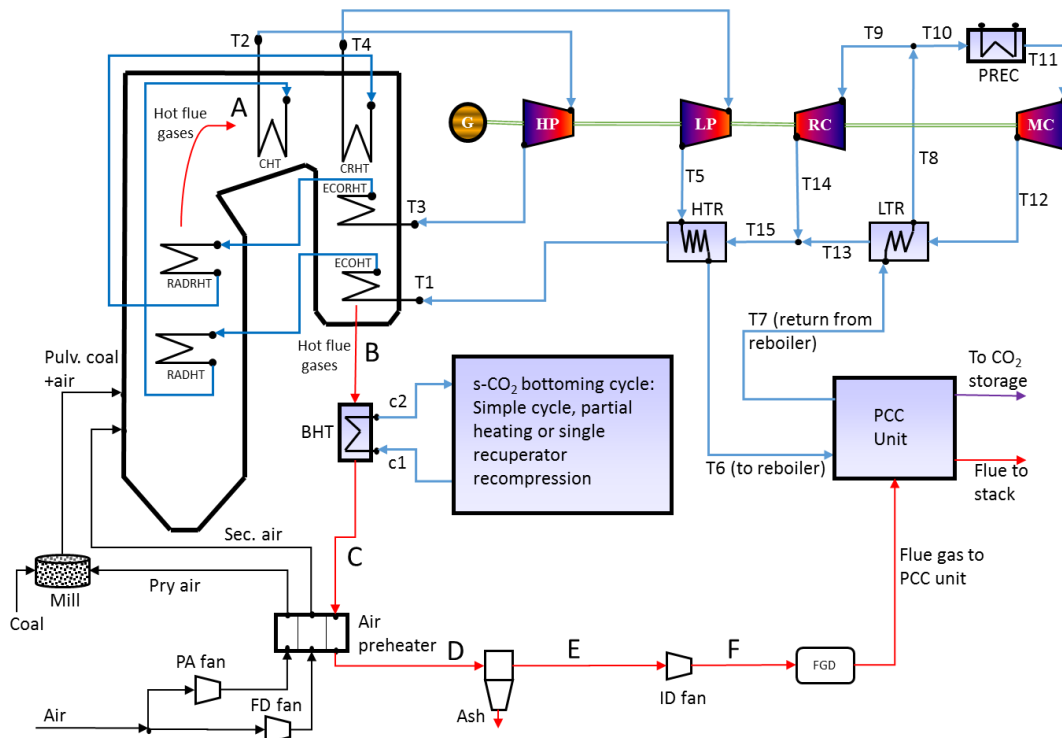
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329 Figure 4 Case B - Partial heating bottoming cycle



330

331 Figure 5 Case C - Single recuperator recompression bottoming cycle



332

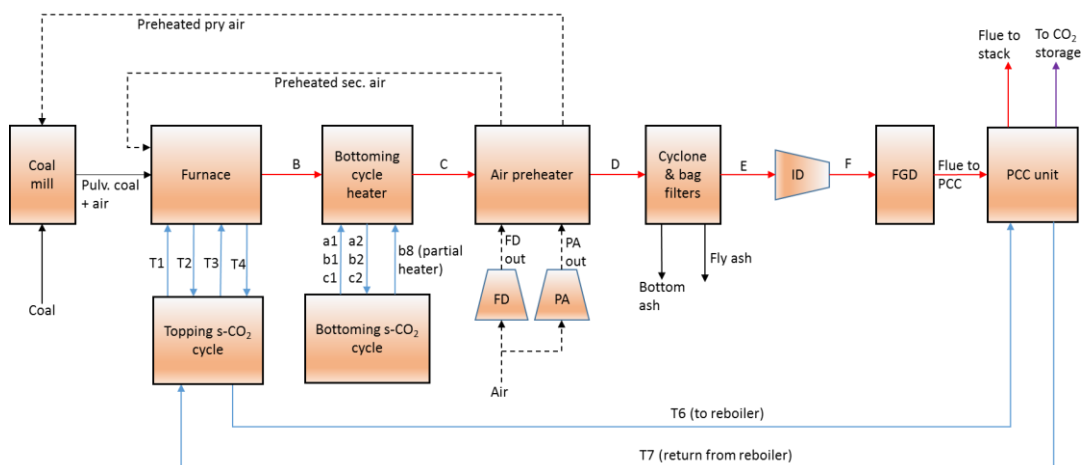
333 Figure 6 Integration of coal-fired s-CO₂ Brayton cycle with PCC unit

334

335 3 Steady state modelling in Aspen Plus®

336 A model of the three cases of coal-fired s-CO₂ cycle power plant with PCC was developed for
 337 performance comparison among the cases as well as comparison with a benchmark coal-fired
 338 supercritical power plant with 90% CO₂ capture. The benchmark plant was not modelled in
 339 this study but the performance results were obtained from Olaleye et al. [45]. A simplified
 340 block diagram of the modelled coal-fired s-CO₂ cycle power plant is shown in Figure 7

341



342

343 Figure 7 Simplified block diagram of the coal-fired s-CO₂ cycle power plant

3.1 Aspen Plus[®] software and thermo-physical property methods

The steady state models were implemented with Aspen Plus[®] V8.4 software to simulate the performance of the coal-fired s-CO₂ cycles power plants. The simulation environment is very flexible for describing the power plant components and connections. The plant components modelled include coal mill, fans, preheaters, pulverised coal-fired furnace, ash removal components, flue gas desulfurization and s-CO₂ cycle components like the external heat sources, turbine, compressor, recuperator and precooler. Description of the PCC structure and its modelling are left until the next section.

Concomitant with process simulation is the need for accurate physical property data and models [46]. Aspen Plus[®] contains extensive property calculation methods for the physical, chemical and thermodynamic properties of different solid, liquid and gaseous substances. In Aspen Plus[®], coal and ash were modelled as nonconventional solids. The HCOALGEN and the DCOALIGT physical property models were used to calculate the enthalpy and density of coal and ash [47]. Peng-Robinson equation of state with Boston Mathias modification (PR-BM) was used to estimate the properties of air and combustion products. For the s-CO₂ properties, REFPROP property package in Aspen Plus[®] was used. REFPROP has been reported to be accurate and widely applicable to a variety of pure fluid and mixtures [48, 49].

3.2 Coal combustion and furnace modelling

The coal type fired is the Illinois No 6 bituminous coal. Details of the ultimate and proximate analysis of the coal are given in Table 1. The higher heating value (HHV) of the coal was calculated from the ultimate analysis by using the Dulong and Petit formula [50]:

$$HHV \left(\frac{MJ}{kg} \right) = 33.83C + 144.45 \left(H - \frac{O}{8} \right) + 9.38S \quad (3.1)$$

Where C, H, O and S are mass fractions of carbon, hydrogen, oxygen and sulphur in coal respectively.

Incoming air was assumed to consist of nitrogen (76.8 wt.%) and oxygen (23.2 wt.%) at 15 °C and 1.01 bar. Percent excess air supplied was specified to be 20%. A user-defined Fortran subroutine calculator was implemented to calculate the flow rate of air required for combustion based on the specified percent excess air, the coal flow rate and the coal characteristic. About 23.5% of the incoming air was sent to the PA fan while the rest was sent to the FD fan as secondary air. By specifying the isentropic efficiencies of the fan, the inlet conditions and the discharge pressure, Aspen Plus[®] determined the power required by the fans. Coal is dried with preheated PA and grounded to fine powder in the coal mill. Volatile matter may be distilled off from the coal in addition to moisture if the temperature of the PA is too high, which may lead to fire hazard [50]. Therefore, the primary air was only preheated to about 215 °C so that after drying the coal the temperature at pulveriser outlet was within the allowable pulveriser outlet temperature of 75 °C.

The drying process was modelled with RStoic block. Wet coal and hot PA streams were fed to the RStoic block. The block was used to model drying by converting a portion of the coal to form water. The outlet, which is a stream of dried coal and moist air, is fed to the pulverising mill. The milling process was modelled with a combination of crusher and screen. The crusher was modelled by specifying the outlet particle size distribution (PSD) of coal. The screen block was used to separate the coarse material from the fine material. The coarse portion was returned to the crusher for further grinding. The PSD of the pulverised coal was specified such that about 70% of coal will pass through a 200-mesh screen and less than 1.3% retained on the 50 mesh. The pulverised coal is then conveyed with the PA to the furnace.

In the furnace, the pulverised coal and PA are mixed with the heated secondary air for combustion. A sequence of RYield and RGibbs Aspen Plus[®] built-in reactor models were used

390 to simulate combustion of coal. RGibbs models chemical equilibrium and phase equilibrium
 391 by minimising the Gibbs free energy of the system. Therefore, there was no need to specify
 392 the reaction stoichiometry, only a list of possible products may be specified. However, Gibbs
 393 free energy can only be calculated for conventional components. Since coal was specified as
 394 a nonconventional component, it was first decomposed into its constituent elements by the
 395 RYield block. A calculator block was used to determine the actual yield distribution based on
 396 the inlet coal attributes. The products of the decomposition together with the heat of reaction
 397 associated with the decomposition was then passed to the RGibbs block.

398 During combustion, the chemical energy in the coal is converted to heat energy, which is
 399 transferred to the CO₂ working fluid. Heat radiation from the centre of the flame and absorption
 400 of the radiant heat by the working fluid were modelled with HEATER blocks. The radiant heat
 401 was divided in the ratio 0.65/0.35 between the main radiant heater and the reheat radiant heater.
 402 The exit of the radiant heat source corresponds to the top of the furnace and entrance to the
 403 convective zone where the flue gases temperature was maintained at 1010 °C. Convective
 404 heaters in this zone comprising of two final CO₂ heaters and two economisers were modelled
 405 with HEATX blocks with flue gases as the hot stream and CO₂ as the cold stream. For a given
 406 coal flow rate, a design specification was defined in Aspen Plus® to determine the topping
 407 cycle CO₂ flow rate required to cool the flue gases such that a 30 °C minimum temperature
 408 difference was maintained between the flue gases leaving the furnace at point B and CO₂
 409 entering the furnace at point T1.

410

411 Table 1 Proximate and ultimate analysis of Illinois No 6 coal [42]

Parameter	Weight %
Proximate Analysis (as received)	
Moisture	11.12
Ash	9.70
Volatile matter	34.99
Fixed carbon	44.19
Total	100
Ultimate Analysis (as received)	
Moisture	11.12
Carbon	63.75
Hydrogen	4.50
Nitrogen	1.25
Chlorine	0.29
Sulphur	2.51
Ash	9.70
Oxygen	6.88
Total	100

412

413 3.3 Modelling of s-CO₂ closed Brayton cycles

414 The topping and bottoming s-CO₂ cycles have the same maximum cycle pressure of 290 bar
 415 corresponding to the maximum cycle pressure of the benchmark supercritical steam turbine
 416 cycle [45]. Similarly, the topping cycle HP and LP turbines inlet temperature were fixed at
 417 593 °C. Both topping and bottoming cycles' compressor inlet temperature and pressure were
 418 fixed just above the critical point at 31 °C and 76 bar. The bottoming cycle' turbine inlet
 419 temperature was fixed at 465 °C, which is 30 °C below the flue gas temperature entering the
 420 bottoming cycle heater. The values of recuperator's minimum TTD, compressor and turbine

421 isentropic efficiencies, and heat exchanger pressure losses were selected based on values
 422 reported in literature. Hence, a minimum TTD of 10 °C was specified for the recuperators [30].
 423 Main compressor, recompression compressor and turbine isentropic efficiencies were 90%,
 424 89% and 93% respectively [30]. Heat exchanger relative pressure losses were fixed at 0.5%
 425 [15]. For cycles with split flows, the split fractions could be independently adjusted to obtain
 426 optimum cycle efficiency.

427 Compressors and turbines were simulated in Aspen Plus® with COMPR block. Aspen Plus®
 428 calculates the power required (or delivered) based on the inlet conditions, discharge pressure
 429 and efficiency. Recuperators were modelled with HEATX block while precoolers were
 430 modelled with HEATER blocks. In the bottoming cycle, design specification was used to
 431 determine the needed CO₂ flow rate based on a minimum temperature difference of 30 °C
 432 between the flue gas leaving the bottoming cycle heater and the CO₂ entering the heater.

433 3.4 Preheater, ash removal and flue gas desulfurization

434 Air preheater was modelled with MHeatX block, which represents heat transfer between the
 435 hot flue gases leaving the bottoming cycle heater and two cold streams (i.e. PA and SA). Outlet
 436 specifications must be given for two of the three streams. PA and flue gas outlet temperatures
 437 were specified. Flue gas outlet temperature of 116 °C specified for the benchmark steam plant
 438 was assumed. Then, an overall energy balance determines the unspecified outlet temperature
 439 of the secondary air.

440 Ash removal from the flue gas was modelled with cyclone and bag filter blocks. 20% of ash
 441 was removed as bottom ash by the cyclone while the remaining 80% was removed as fly ash
 442 by bag filters. The ash-free flue gas is pushed through the FGD unit by ID fan. The power
 443 required by the fan was determined based on its discharge pressure and isentropic efficiency.
 444 The FGD removed sulphur oxide in the flue gas before entering the PCC unit.

445 3.5 Performance calculation

446 MS Excel™ spreadsheets were used to carry out the performance calculations. Therefore, the
 447 MS Excel™ was linked with Aspen Plus® to access simulation results.

448 Two important performance indicators are the furnace (or heat recovery) efficiency and the
 449 cycle efficiency. The furnace efficiency is an indication of the ability of the power cycle to
 450 receive the heat available in the heat source while cycle efficiency indicates the ability to
 451 convert the received heat into electrical power [7]. The furnace efficiency, $\eta_{furnace}$ is
 452 calculated by taking the total amount of heat transferred to the s-CO₂ cycles and dividing it by
 453 the coal fuel power supplied to the plant.

$$\eta_{furnace} = \frac{(Q_{cycle})_{top} + (Q_{cycle})_{bottom}}{\dot{m}_{coal}(HHV)} \quad (3.2)$$

454 Where $(Q_{cycle})_{top}$ is the sum of the heat transferred to the topping s-CO₂ cycle through the
 455 economisers, radiant heaters and final convective heater/reheater, $(Q_{cycle})_{bottom}$ is the heat
 456 input from flue gases to the bottoming s-CO₂ cycle, \dot{m}_{coal} is the mass flow rate of coal and
 457 HHV is the higher heating value of the supplied coal.

458 Cycle efficiency, η_{cycle} , is calculated by taking the electrical power output of the cycle and
 459 dividing by the heat transferred to the cycle. Hence, cycle efficiency for the topping
 460 cycle, $(\eta_{cycle})_{top}$, is

$$(\eta_{cycle})_{top} = \frac{(P_{elec})_{top}}{(Q_{cycle})_{top}} \quad (3.3)$$

461 Where $(P_{elec})_{top}$ is the topping cycle electrical power output given as:

$$(P_{elec})_{top} = \left[\left(\sum P_T \right)_{top} - \left(\sum P_C \right)_{top} \right] \eta_{gen} \quad (3.4)$$

$$= (P_{HP} + P_{LP} - P_{MC} - P_{RC}) \eta_{gen}$$

462 $(\sum P_T)_{top}$ is the sum of topping cycle turbine power, $(\sum P_C)_{top}$ is the sum of topping cycle
 463 compressor power, P_{HP} is the HP turbine power, P_{LP} is the LP turbine power, P_{MC} is the main
 464 compressor power, P_{RC} is the RC power and η_{gen} is the electrical generator efficiency.

465 Cycle efficiency for the bottoming cycle is

$$(\eta_{cycle})_{bottom} = \frac{(P_{elec})_{bottom}}{(Q_{cycle})_{bottom}} = \frac{[(\sum P_T)_{bottom} - (\sum P_C)_{bottom}] \eta_{gen}}{(Q_{cycle})_{bottom}} \quad (3.5)$$

466 Where $(P_{elec})_{bottom}$ is the bottoming cycle electrical power output, $(\sum P_T)_{bottom}$ is the sum
 467 of bottoming cycle turbine power and $(\sum P_C)_{bottom}$ is the sum of bottoming cycle compressor
 468 power.

469 The overall cycle efficiency, $\eta_{overall\ cycle}$ is the ratio of the total electrical power output from
 470 the cycles, $(P_{elec})_{total}$ to the total heat transferred to the cycles, $(Q_{cycle})_{total}$.

$$\eta_{overall\ cycle} = \frac{(P_{elec})_{total}}{(Q_{cycle})_{total}} = \frac{(P_{elec})_{top} + (P_{elec})_{bottom}}{(Q_{cycle})_{top} + (Q_{cycle})_{bottom}} \quad (3.6)$$

471

472 The net power output of the plant, P_{net} is the total or gross power output from the topping and
 473 bottoming cycles, $(P_{elec})_{total}$ minus the auxiliary power consumption, P_{aux} in pumps, fans,
 474 coal mill etc.:

$$P_{net} = (P_{elec})_{total} - P_{aux} \quad (3.7)$$

475

476 The plant net efficiency, η_{net} is defined as the ratio of the net power output to the coal fuel
 477 energy input to the plant:

$$\eta_{net} = \frac{P_{net}}{\dot{m}_{coal}(HHV)} \quad (3.8)$$

478

479 The three cases in this study with different bottoming cycle options will present different cycle
 480 efficiencies and furnace efficiencies. Therefore, the overall impact of the choice of power plant
 481 configurations on the plant net efficiency can only be determined through performance
 482 calculations and comparison among the cases.

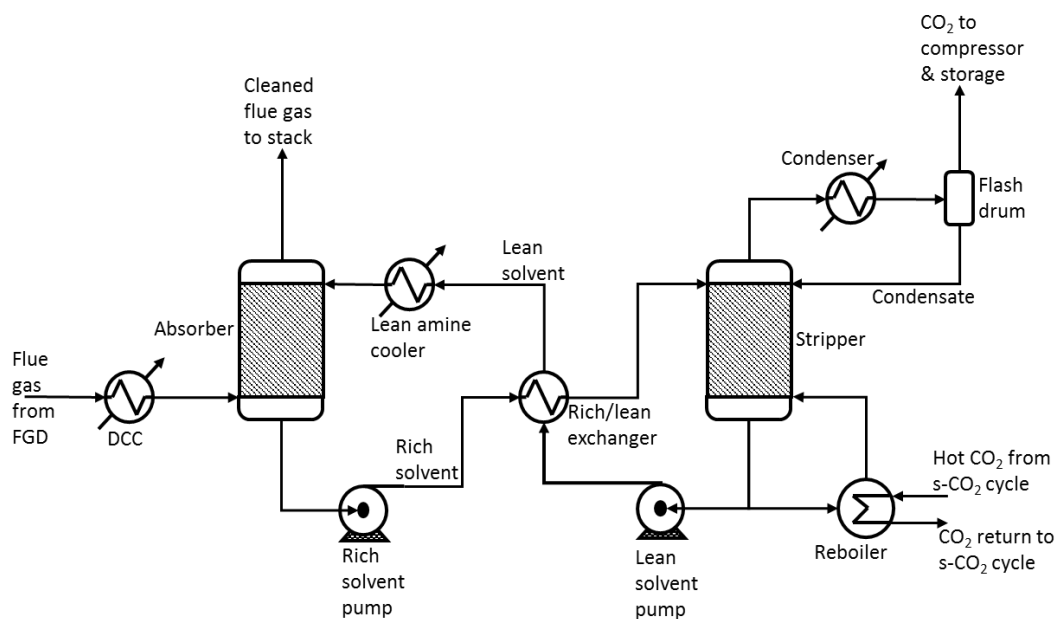
483 **4 Solvent-based post-combustion CO₂ capture**

484 This section discusses the PCC, which is based on chemical absorption through MEA solvent.
 485 Benefits of MEA-based PCC include (1) high separation selectivity; (2) It operates at
 486 atmospheric conditions; (3) Experimental/pilot plant data are available.

487 **4.1 Description of MEA-based CO₂ capture process**

488 Figure 8 shows a simplified process flow diagram for a typical chemical absorption CO₂
 489 capture process. The main components are absorber, stripper with a reboiler and a condenser
 490 attached, direct contact cooler (DCC), rich MEA pump, lean MEA pump, lean/rich cross heat
 491 exchanger and lean MEA cooler.

492 Flue gas from the power plant's FGD unit is first cooled in the DCC to a suitable temperature
 493 for absorption (about 40 °C). The cooled flue gases are introduced into the absorber at the
 494 bottom while the lean MEA solvent solution enters the absorber at the top. The flue gases flow
 495 upward while the MEA solvent solution flows down under gravity through the absorber
 496 (packed column). Chemical absorption of CO₂ in the flue gases by the MEA solvent takes
 497 place during the counter-current flow in the absorber. Treated flue gases leave the absorber
 498 at the top. Rich MEA solvent (i.e. with higher loading of CO₂) leaves the absorber at the
 499 bottom. Its pressure is then increased by the rich MEA pump and heated in the lean/rich cross heat
 500 exchanger before entering the stripper at the top. In the stripper column, the rich MEA solvent
 501 is stripped of the CO₂ by the application of heat energy in the reboiler. The water vapour and
 502 CO₂ mixture released in the stripper is sent to the stripper condenser, which cools the mixture
 503 thereby turning most of the water vapour to liquid water. The condensed water and CO₂ are
 504 separated in the flash drum. The condensed water is returned back to the stripper while the
 505 separated CO₂ leaves the stripper at the top. The resulting lean MEA solvent (i.e. with lower
 506 loading of CO₂) exits the stripper at the bottom. The lean MEA solvent leaving the stripper
 507 is used to heat the rich MEA solvent in the cross heat exchanger and the temperature is further
 508 reduced in the lean MEA cooler before being returned to the absorber column at the top.



509
 510 Figure 8 Simplified process flow diagram for MEA-based post-combustion CO₂ capture unit
 511 [3]

512 4.2 Rate-based simulation of the CO₂ capture system in Aspen 513 Plus®

514 The MEA-based PCC was simulated in Aspen Plus® to determine the performance. The
 515 simulation was based on the parameters reported for the benchmark supercritical steam plant's
 516 PCC unit, which was validated with data from University of Kaiserslautern pilot plant by
 517 Olaleye et al. [45]. The PCC used a 30 wt.% MEA solution as solvent. The temperature of the
 518 flue gas and the lean MEA entering the absorber was 40 °C. Absorber operating pressure was
 519 1.013 bar. The rich MEA solution was heated up to 106 °C in the cross heat exchanger. The
 520 stripper was operating at a pressure of 1.9 bar and the reboiler temperature was maintained at
 521 about 120 °C to avoid thermal degradation of the amine solvent. In Aspen Plus®, RadFrac
 522 block was used to model the absorber and the stripper. Koch FLEXIPAC® 1Y structured
 523 packing was selected for the absorber and stripper. Previously, the MEA-based PCC model

524 has been validated and scaled up to match the flue gas flow rate of the supercritical power
 525 plant by Olaleye et al. [45]. For the plant with 1402 MW of heat input, the design of the
 526 absorber and stripper arrived at four absorber columns with a diameter of 5.41m each and three
 527 stripper column with a diameter of 4.62m each in order to maintain the columns diameters
 528 within the structural limit. Fifteen equilibrium stages were required for each of the absorber
 529 and the stripper column.

530 Modelling of the absorber and stripper in Aspen Plus[®] was through the use of rate-based
 531 models. Rate-based model provides a rigorous and good prediction of the simulation over a
 532 wide range of operating conditions unlike the traditional equilibrium-stage modelling
 533 approach [51]. The Electrolyte Non-Random-Two-Liquid (ElecNRTL) activity coefficient
 534 property package was selected to accurately predict the ionisation equilibrium and the heats of
 535 solution of the MEA-CO₂-H₂O system. The solution chemistry of the MEA-based chemical
 536 absorption process can be represented by the following equilibrium reactions (R1-R5) [52]:

537



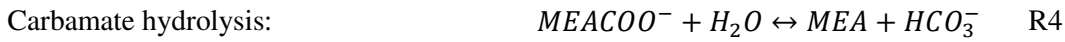
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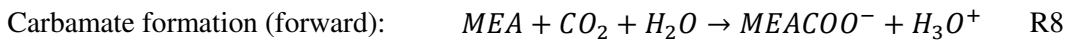
543 Reaction models for the absorber and stripper consist of three equilibrium rate-based
 544 controlled reactions, R1, R3 and R5, in conjunction with the following kinetic rate-based
 545 controlled reactions (R6-R9) [52]:



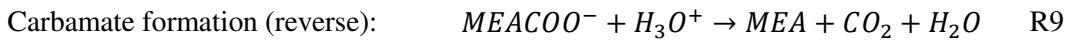
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548



549

550 The kinetic reaction rates, *r*, are described in Aspen Plus[®] by the power law expression:

$$r = kT^n \exp\left(-\frac{E}{RT}\right) \prod_{i=1}^N C_i^{a_i} \quad (4.1)$$

551

5 Results and discussion

5.1 Verification of the s-CO₂ Brayton cycle model

The suitability of the Aspen Plus[®] model for simulating the performances of supercritical CO₂ Brayton cycles was investigated. An s-CO₂ recompression Brayton cycle (Figure 1b) was modelled for verifying the calculation. Independent results of numerical model reported by Dostal et al. [53] were compared with the Aspen Plus[®] simulation results. The input parameters were:

- Maximum cycle pressure - 200 bar
- Turbine inlet temperature - 550 °C
- Precooler outlet temperature - 32 °C
- Precooler outlet pressure - 76.92 bar
- Mass flow rate - 3176.4 kg/s
- MC pressure ratio - 2.6
- Split flow fraction - 0.41
- Turbine isentropic efficiency - 90 %
- Main and recompression compressors efficiency - 89 %

Comparison of the main simulation results against literature value is presented in Table 2. The maximum relative deviation is about 2.51%. The small differences in the result can be attributed to uncertainties in the pressure loss specifications and the round-off error in the input parameters. Otherwise, the simulation results agreed well with the literature values.

Table 2 Validation of s-CO₂ Brayton cycle model against literature value

Parameters	Literature value [54]	Simulation value	Relative difference
Turbine outlet temperature	440.29 °C	440.29 °C	0%
MC outlet temperature	61.1 °C	61.11 °C	0.02%
RC inlet temperature	69.59 °C	71.34 °C	2.51%
RC outlet temperature	157.99 °C	160.25 °C	1.43%
Heater inlet temperature	396.54 °C	397.38 °C	0.21%
Thermal power	600 MWt	596.76 MWt	0.54%
Turbine work	383.71 MW	383.72 MW	0.003%
MC work	38.59 MW	38.57 MW	0.05%
RC work	74.84 MW	75.84 MW	1.34%
Net work output	270.28 MW	269.31 MW	0.36%
HTR duty	985.51 MW	977.49 MW	0.81%
LTR duty	398.8 MW	398.0 MW	0.2%
Precooler duty	328.38 MW	328.11 MW	0.08%
Cycle efficiency	45.05%	45.13 %	0.08%

573

5.2 Baseline boundary conditions and design point parameters

The boundary conditions and parameters such as coal mass flow rate, combustion air conditions, percent excess air, flue gas stack temperature, maximum cycle pressure and turbines inlet temperature were selected based on the information published for the supercritical reheat steam cycle [45]. This will ensure a fair comparison between the performances of the s-CO₂ cycle plants and the conventional supercritical steam plant. Other conditions and parameters like pressure losses and specifications of heat exchangers were selected based on similar studies of s-CO₂ power cycle reported in the literature [15, 30]. A summary of the baseline boundary conditions and design point parameters is given in Table 3.

582

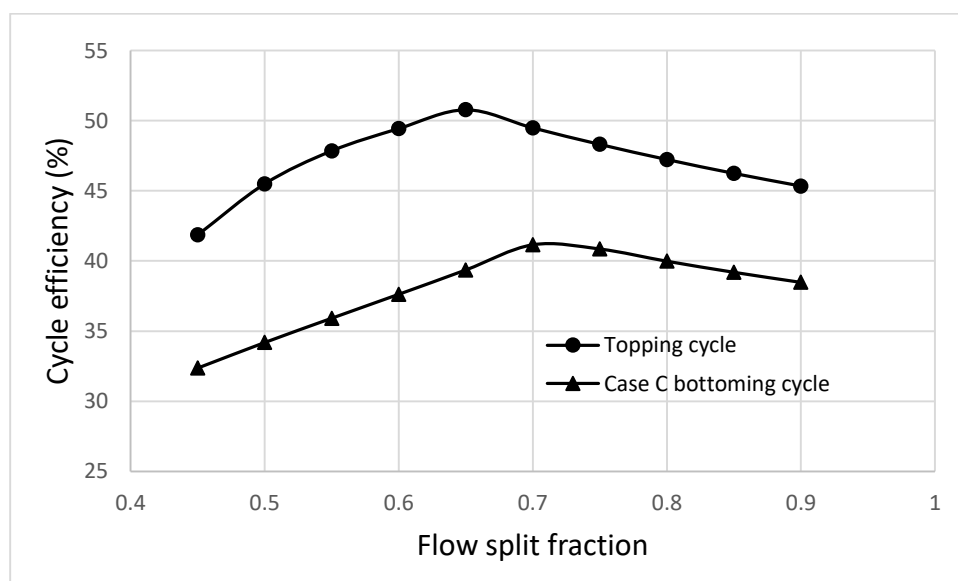
583 Table 3 Boundary conditions and design parameters

Parameter/variable	Value
Coal feed ($^{\circ}\text{C}/\text{bar}/(\text{kg}/\text{s})$)	15/1.01/51.82
Air ($^{\circ}\text{C}/\text{bar}$)	15/1.01
Excess air (%)	20
Maximum cycle pressure (bar)	290
HP & LP turbines inlet temperature ($^{\circ}\text{C}$)	593
Compressor inlet pressure (bar)	76
Compressor inlet temperature ($^{\circ}\text{C}$)	31
Gas-CO ₂ TTD ($^{\circ}\text{C}$)	30
Preheater hot outlet temperature ($^{\circ}\text{C}$)	116
Recuperator TTD ($^{\circ}\text{C}$)	10
Turbine isentropic efficiency (%)	93
MC isentropic efficiency (%)	90
Recompression compressor isentropic efficiency (%)	89
Fan isentropic efficiency (%)	80
Generator efficiency (%)	98.4
Ash distribution, fly/bottom ash (%)	80/20

584

585 **5.3 Performance comparisons among Cases A, B and C of the**
 586 **coal-fired s-CO₂ Brayton cycle power plants**

587 The flow split fraction (i.e. the fraction of the total flow that goes through the pre-cooler/main
 588 compressor) should be adjusted such that the differences in the heat capacities between the hot
 589 streams and the cold streams in recuperators are minimised. This will improve heat transfer in
 590 the recuperators and thereby maximised cycle efficiency. Figure 9 shows the cycle efficiencies
 591 as a function of the flow split fractions. The optimum flow split fraction was found to be about
 592 0.65 for the topping cycle while it was about 0.71 for the single recuperator recompression
 593 bottoming cycle.



594

595 Figure 9 Cycle efficiencies of the topping cycle and Case C bottoming cycle as a function of
 596 the flow split fractions

597 In order to highlight the impact of integrating the coal-fired s-CO₂ power plants with the PCC
598 unit, the performances of the power plants without carbon capture were first determined based
599 the optimum flow split fractions, and the baseline boundary conditions and design parameters
600 presented above. Table 4 shows the pressure, temperature and mass flow for the plants' main
601 points. The stream nomenclature is based on Figure 3, Figure 4 and Figure 5. This was then
602 followed by simulation and performance evaluation of the whole power plants, incorporating
603 the PCC unit. The distribution of the fuel combustion heat energy among the different s-CO₂
604 heaters is shown in Figure 10. About 50% of the input heat energy was transferred by radiation
605 to the s-CO₂ working fluid in the radiant heaters. The Case A and Case B bottoming cycles
606 were able to recover about 12% of the total heat input, which otherwise would have been lost
607 through the exhaust flue gas. In Case C, only about 9% was recovered but the unrecovered
608 heat was utilised for preheating the secondary air to higher temperature level (258 °C) than
609 Case A (177 °C) and Case B (165 °C). This then leads to higher heat transfer in the furnace for
610 Case C. For the three cases, the heat losses were about 12%, that is, a furnace efficiency of
611 approximately 88%. This value of furnace efficiency is comparable to the boiler efficiency
612 obtainable in coal-fired steam power plants. Hence, the addition of the bottoming cycles and
613 the combustion air preheaters enables efficient utilisation of the furnace heat.

614

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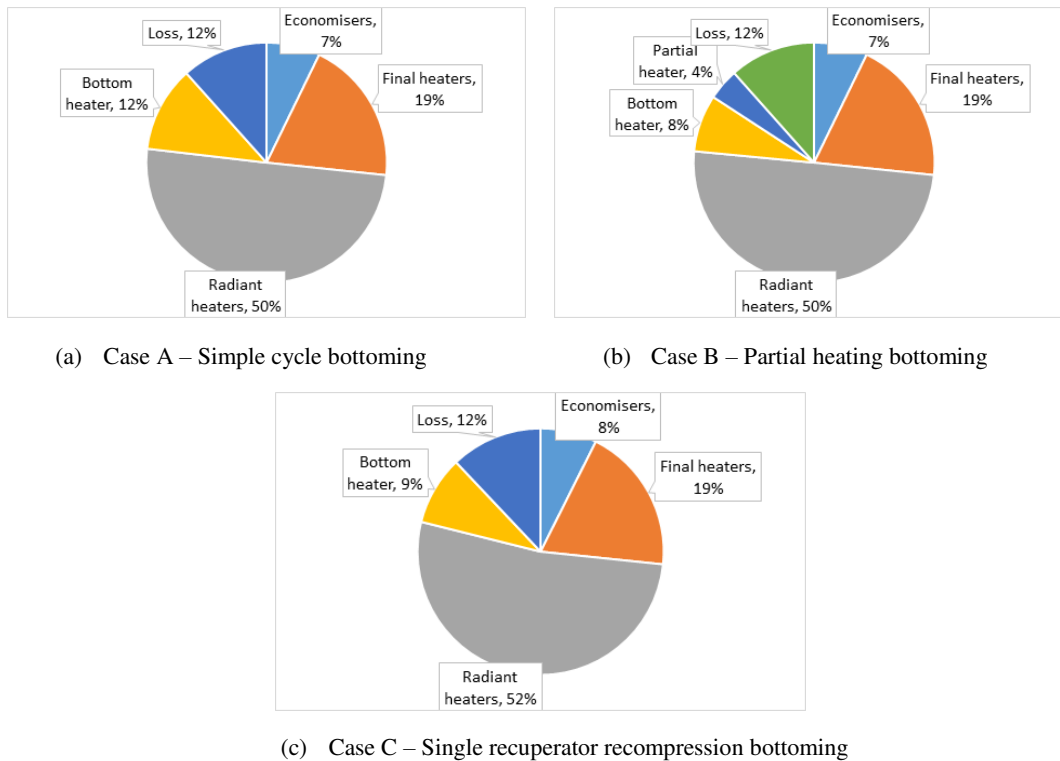
616 Table 4 Summary of the main stream values for the three cases calculated with baseline
617 boundary conditions and design parameters

Stream	Case A			Case B			Case C		
	P (bar)	T (°C)	m (kg/s)	P (bar)	T (°C)	m (kg/s)	P (bar)	T (°C)	m (kg/s)
Coal	1.01	15	51.82	1.01	15	51.82	1.01	15	51.82
Air	1.01	15	540.88	1.01	15	540.88	1.01	15	540.88
Pry air	1.1	215	127.11	1.1	215	127.11	1.1	215	127.11
Sec. air	1.1	177.23	413.77	1.1	164.59	413.77	1.1	257.82	413.77
Pulv.Coal+air	1.09	75.28	178.93	1.09	75.28	178.93	1.09	75.28	178.93
A	1.09	1010	592.7	1.09	1010	592.7	1.09	1010	592.7
B	1.01	496	592.7	1.01	496	592.7	1.01	496	592.7
C	1.01	253.26	592.7	1.01	244.86	592.7	1.01	306.70	592.7
D	1.01	116	592.7	1.01	116	592.7	1.01	116	592.7
Flue to stack	1.01	56.67	585.08	1.01	56.67	585.08	1.01	56.67	585.08
T1	287.12	466	4052.52	287.12	466	4038.78	287.12	466	4163.13
T2	282.82	593	4052.52	282.82	593	4038.78	282.82	593	4163.13
T3	147.72	507.64	4052.52	147.72	507.64	4038.78	147.72	507.64	4163.13
T4	145.51	593	4052.52	145.51	593	4038.78	145.51	593	4163.13
a1,b1,c1	288.55	223.26	511.12	288.70	305.71	526.35	288.55	276.70	523.38
a2,b2,c2	287.25	466	511.12	287.25	466	526.35	287.25	466	523.38
b8	-	-	-	290	69.70	152.64	-	-	-

618

619

620



621 Figure 10 Distribution of the input heat value among the different heaters

622 Table 5 shows the performance result of the PCC unit that was integrated with the coal-fired
 623 s-CO₂ Brayton cycle power plants. Integration of the PCC to the plants penalised the net
 624 efficiency through (1) bleeding of CO₂ for solvent regeneration in reboiler, which resulted in
 625 lower cycle efficiency (2) additional auxiliary loads associated with the PCC units. Table 6 is
 626 a summary of the performance results for the three cases both without the PCC unit and with
 627 the PCC unit integrated. Interestingly, Case C (i.e. the single recuperator recompression
 628 bottoming cycle layout) gave the best overall plant net efficiency with or without PCC even
 629 though the bottoming cycle recovered the least amount of heat and thus produced the least
 630 power. The superior performance of Case C is due to better efficiency of the bottoming cycle.
 631 In contrast, Kim et al. [15] concluded that power produced by bottoming cycle is a more
 632 important factor than the efficiency of bottoming cycle in determining the overall plant
 633 performance and therefore, did not recommend recompression cycle for bottoming cycle
 634 application despite having the best cycle efficiency. However, unlike our study, Kim et al.
 635 compared the performances of various s-CO₂ bottoming cycles without a downstream air
 636 preheater.

637 For a fixed coal fuel input, the plant overall performance depends on auxiliary loads, cycle
 638 efficiency and furnace efficiency. The cycle efficiency is majorly determined by the choice of
 639 cycle layout/configuration. Furnace efficiency, on the other hand, can be improved by heat
 640 recovery in the bottoming cycle and preheating of combustion air. In summary, the cycle
 641 layouts, the bottoming cycle heat recovery, the level of air preheating and the auxiliary loads
 642 will determine the plant net efficiency. Hence, for plants with similar auxiliary loads, plant net
 643 efficiency will be maximised by configurations with high cycle efficiency, good heat recovery
 644 in bottoming cycle and high level of air preheating. Unfortunately, good heat recovery in the
 645 bottoming cycle cannot be achieved simultaneously with a high level of air preheating. For
 646 instance, good heat recovery in the bottoming cycles of Case A and Case B meant that the
 647 temperature of the flue gas entering the air preheater was relatively low, limiting the amount
 648 of air preheating possible. On the other hand, Case C with the least heat recovery (or produced

649 power) in bottoming cycle gave the highest air preheating duty (Table 6). Therefore, the poor
 650 heat recovery was somewhat compensated for by the added air preheater.

651

652

653 Table 5 Parameters and performance results of the PCC unit

Parameter	Value
CO ₂ removal percentage	90%
Flue gas absorber inlet temperature	40 °C
Lean solvent absorber inlet temperature	40 °C
MEA concentration	30 wt. %
Absorber operating pressure	1.013 bar
Stripper operating pressure	1.9 bar
Lean solvent loading	0.29 mol CO ₂ /mol MEA
Rich solvent loading	0.53 mol CO ₂ /mol MEA
Reboiler temperature	120 °C
Condenser temperature	31.98 °C
Condenser duty	1.13 GJ/ton CO ₂
Solvent circulation rate	18 m ³ /ton CO ₂
Reboiler duty	3.4 GJ/ton CO ₂

654

655

656

657 Table 6 Comparison of plant performances with and without post-combustion CO₂ capture
 658 (PCC) for Case A (simple recuperative cycle as bottoming cycle), Case B (partial heating cycle
 659 as bottoming cycle) and Case C (single recuperator recompression cycle as bottoming cycle)

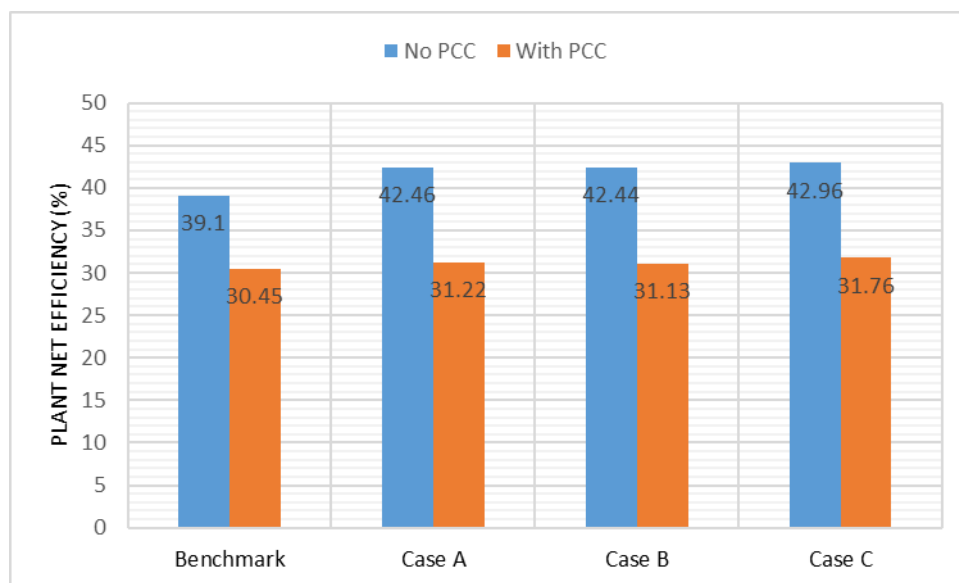
Parameter	Case A		Case B		Case C	
	No PCC	With PCC	No PCC	With PCC	No PCC	With PCC
HHV, MJ/kg	27.05	27.05	27.05	27.05	27.05	27.05
Input heat value, MJ	1401.87	1401.87	1401.87	1401.87	1401.87	1401.87
Heat transferred to top cycle, MW	1077.49	1103.16	1072.8	1095.63	1106.01	1131.81
Heat transferred to bottom cycle, MW	161.46	161.46	167.03	149.57	126.75	106.56
Furnace efficiency, %	88.38	88.74	88.44	88.82	87.94	88.34
Preheater duty, MW	92.61	59.80	87.18	51.15	127.43	96.14
Top gross electric power, MWe	545.40	401.98	543.31	398.08	560	416.32
Bottom gross electric power, MWe	60.17	46.39	61.96	48.95	52.61	39.58
Top cycle efficiency, %	50.62	36.44	50.64	36.33	50.63	36.78
Bottom cycle efficiency, %	37.27	32.94	37.10	32.73	41.51	37.14
Overall cycle efficiency, %	48.88	36.04	48.82	35.90	49.69	36.81
Auxiliaries power, MW	10.38	10.7	10.38	10.7	10.39	10.7
Net electric power, MWe	595.19	437.67	594.90	436.33	602.22	445.19
CO ₂ specific emission, kg CO ₂ /MWh	714.69	98.05	715.04	98.35	706.35	96.39
Specific work output, kWh/m ³	5.28	5.24	5.26	5.23	5.16	5.10
Overall plant net efficiency, %	42.46	31.22	42.44	31.13	42.96	31.76

660

661 In Figure 11, the performances of the coal-fired s-CO₂ Brayton cycle power plants were
 662 compared with the state-of-the-art supercritical reheat steam power plant [51]. The s-CO₂
 663 Brayton cycle power plants, without CO₂ capture, was found to be about 3.34 - 3.86% more
 664 efficient than the steam power plant. When the power plants were integrated with the PCC
 665 unit, the plant net efficiencies of the s-CO₂ power plants were about 0.68 – 1.31% above the
 666 steam plant's efficiency. Although the s-CO₂ Brayton cycle plants with CO₂ capture gave
 667 higher efficiency than steam cycle plant, the s-CO₂ cycle suffered more efficiency penalty
 668 (about 11.2%) than the steam plant (about 8.65%). This is probably due to the use of sensible
 669 heat of s-CO₂ working fluid to meet reboiler thermal requirement instead of low pressure
 670 condensing steam, as is usually the case in steam turbine power plant.

671 A comparison of the specific work output (i.e. the ratio of the generated power to the
 672 volumetric flow rate of the working fluid) of each cycle can give an indication of the relative
 673 size of plants and by extension the relative capital cost [20, 54]. Table 6 shows that the specific
 674 work outputs in all the three cases were comparable (approximately 5 kWh/m³). Case C shows
 675 a slightly lower specific work output but the difference is not considered significant. The
 676 specific work output of the s-CO₂ cycle is over 30 times more than that of the steam cycle.
 677 Therefore, the s-CO₂ cycle plant has the potential to be significantly smaller than the steam
 678 cycle plant. This is in good agreement with previous findings in the literature on the
 679 compactness of s-CO₂ cycle in comparison with steam cycle [11-13, 43].

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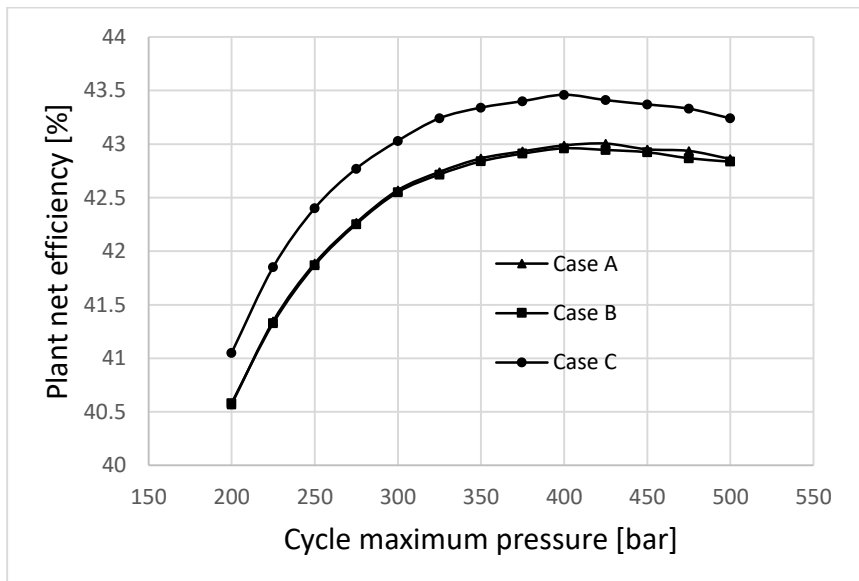
681

682 Figure 11 Comparison of the overall plant net efficiency of Case A (simple recuperative cycle
 683 as bottoming cycle), Case B (partial heating cycle as bottoming cycle) and Case C (single
 684 recuperator recompression cycle as bottoming cycle) with the supercritical steam plant (from
 685 Olaley et al. [45]) as the benchmark

686 In this study, the cycle maximum pressure has been selected to match the maximum pressure
 687 in the steam cycle. However, a common feature of Brayton cycle is that there is an optimum
 688 pressure ratio (or cycle maximum pressure in our case) at which the efficiency has a peak
 689 value. Hence, the effect of cycle maximum pressure on plant performance was investigated by
 690 varying the pressure from 200 bar to 500 bar while the compressor inlet pressure was kept
 691 constant. Figure 12 shows the plant net efficiency as a function of cycle maximum pressure
 692 for the three configurations. Case C was found to maintain the best efficiency over the whole
 693 pressure range. Maximum efficiency occurred at an optimum pressure of about 400 bar.

694 Currently, the choice of such a high pressure might not be feasible due to mechanical design
 695 considerations such as the maximum pressure limit of heat exchangers, turbomachinery seal
 696 solutions to prevent leakage and the need to avoid excessively small compressor blades.
 697 However, the USC steam plant with a maximum pressure of 350 bar and a turbine inlet
 698 temperature of 700 °C is expected to come into operation between 2020 and 2030 [31]. If the
 699 s-CO₂ cycle is operated at such maximum pressure (i.e. 350 bar), a net efficiency gain up to
 700 4.24% above the current efficiency of steam turbine plant can be achieved without a
 701 corresponding increase in turbine inlet temperature to 700 °C as planned. Hence, the s-CO₂
 702 plant has the advantage of increased efficiency at a lower temperature.

703



704

705 Figure 12 Plant net efficiency as a function cycle maximum pressure from 200 bar to 500 bar
 706 for the three configurations (Case A – simple recuperative cycle as bottoming cycle, Case B –
 707 partial heating cycle as bottoming cycle and Case C – single recuperator recompression cycle
 708 as bottoming cycle)

709 5.4 Choice of configuration and parametric study

710 In this study, the performance comparison was carried out for three potential s-CO₂ cycle
 711 configurations. The cycles were adapted for efficient utilisation of furnace heat similar to
 712 boiler heat utilisation in conventional steam turbine plant, albeit with bottoming cycles added.
 713 Operating conditions (290 bar, 593 °C and single reheat) were chosen to match the current
 714 supercritical steam cycle conditions. Hence, current experience with material technology for
 715 pulverised coal-fired boiler and steam turbine could be applied to the development of the coal-
 716 fired s-CO₂ Brayton cycle power plant. The overall net efficiency of Case C option without
 717 CO₂ capture was 0.5% and 0.52% over the efficiency of Case A and Case B respectively. With
 718 CO₂ capture, the efficiency gains were 0.54% and 0.63% above the efficiency of Case A and
 719 Case B respectively. Therefore, of the three alternative configurations considered, Case C
 720 (with single recuperator recompression cycle as the bottoming cycle) is more attractive due to
 721 its better performance. It is also expected to be of similar size as the other two configurations
 722 considering the relative value of the specific work output and the component count. When
 723 compared with steam cycle plant, the net efficiency of Case C was higher than the efficiency
 724 of steam cycle plant by about 3.86% and 1.31% without CO₂ capture and with CO₂ capture
 725 respectively.

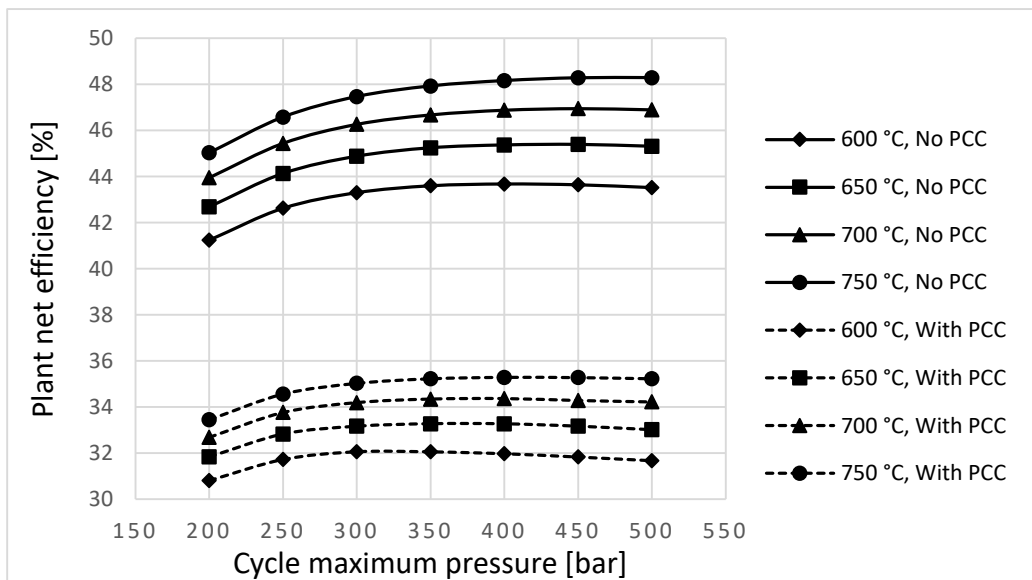
726 Cycle efficiency is known to depend on the turbine inlet temperature, precooler outlet/main
 727 compressor inlet temperature and the recuperator minimum TTD. Hence, a parametric study
 728 was performed to investigate the effects of these parameters on the net efficiency of the chosen
 729 coal-fired s-CO₂ cycle power plant.

730 **5.4.1 Effect of turbine inlet operation conditions**

731 Figure 13 shows the effect of changes in turbine inlet temperature on the cycle performance
 732 for the single recuperator recompression bottoming cycle configuration without PCC unit and
 733 with PCC unit integrated. The figure was produced by varying the cycle maximum pressure
 734 from 200 bar to 500 bar for four different selection of turbine inlet temperatures (600 °C,
 735 650 °C, 700 °C and 750 °C). The cycle performance was calculated with the flow split fraction that
 736 gave the maximum efficiency for each data point while other cycle parameters were
 737 maintained at the baseline condition.

738 The results showed that the plant net efficiency increased with the rise in turbine inlet
 739 temperature. Also for each selection of turbine inlet temperature, there is an optimum cycle
 740 maximum pressure. The optimum cycle maximum pressure increase with an increase of
 741 turbine inlet temperature. With no PCC and at a turbine inlet temperature of 600 °C, the
 742 optimum cycle maximum pressure was about 400 bar, while at 650 °C, the optimum cycle
 743 maximum cycle pressure increased to about 450 bar and the trend continued with increase in
 744 turbine inlet temperature. At the operating conditions of the next USC steam turbine power
 745 plant (700 °C and 350 bar), the efficiency of the s-CO₂ cycle power plant is about 46.67%.
 746 This corresponds to about 7.57% above the efficiency of the conventional supercritical steam
 747 plant.

748



749

750 Figure 13 Plant net efficiency as a function of cycle maximum pressure at different turbine
 751 inlet temperature for the single recuperator recompression bottoming cycle configuration (i.e.
 752 Case C) with no carbon capture and with carbon capture integrated

753 From the foregoing, the adoption of the s-CO₂ cycle for coal-fired power plant application is
 754 promising. The s-CO₂ cycle achieved higher efficiency than steam cycle plant at similar
 755 operating conditions. Even for the advanced USC steam plant that is expected to achieve
 756 efficiency around 47%, this will be done with two or more reheat stages, three or more turbine
 757 modules and series of feedwater heaters. However, with potentially smaller footprint and less

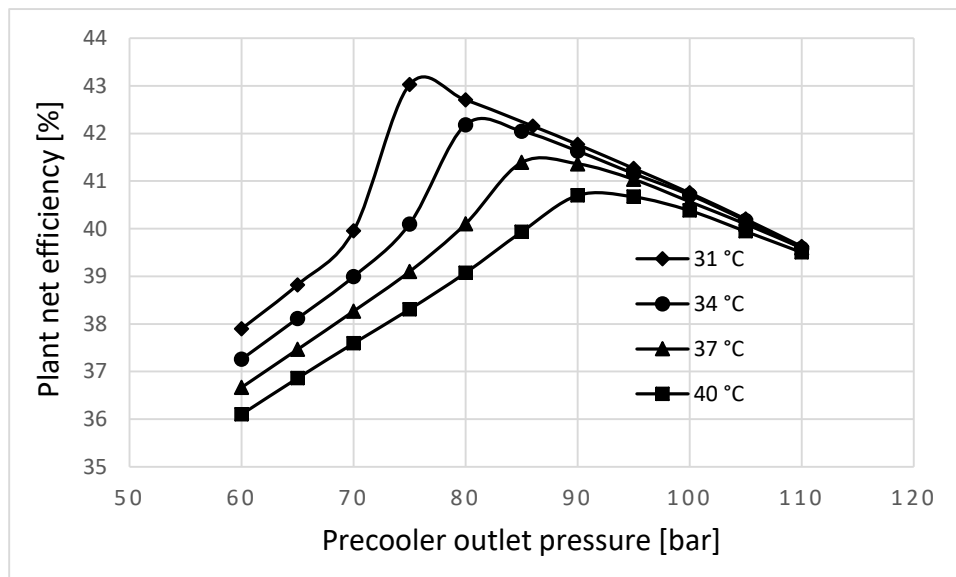
758 complex configuration, similar efficiency can be achieved with coal-fired s-CO₂ Brayton cycle
759 power plant investigated in this study.

760 5.4.2 Effect of precooler outlet/main compressor inlet operating conditions

761 The selection of precooler outlet temperature (or main compressor inlet temperature) is based
762 on the ambient or heat sink temperature, which depends on location as well as the type of
763 cooling (wet cooling or dry cooling). The effect of precooler outlet operating conditions on
764 cycle performance was investigated by varying the precooler outlet pressure from 60 bar to
765 110 bar for four selections of precooler outlet temperature (31, 34, 37 and 40 °C). In order to
766 keep the cycle supercritical at all times, only values of precooler outlet temperature above CO₂
767 critical temperature was considered. The cycle efficiency was optimised with the flow split
768 fraction while other parameters were fixed at the baseline value. Figure 14 shows the plant net
769 efficiency as a function of precooler outlet temperature and pressure.

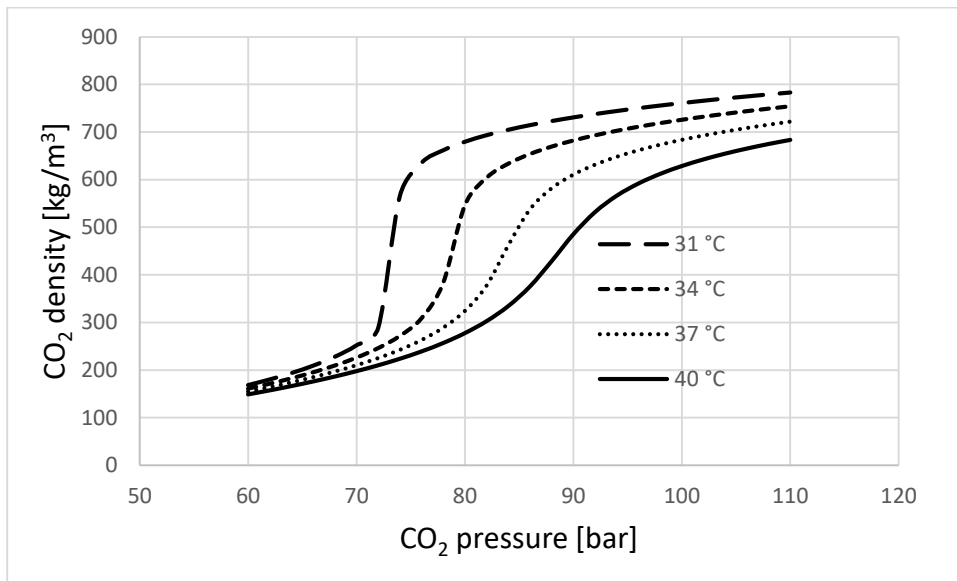
770 The plant net efficiency decreases with rise in precooler outlet temperature. However, for each
771 precooler outlet temperature, there is a corresponding pseudo-critical pressure at which the
772 plant efficiency is maximum. For instance, the highest plant net efficiency for a precooler outlet
773 temperature of 31 °C was achieved at a precooler outlet pressure of 76 bar. However,
774 when the precooler outlet temperature was increased to 34 °C, the optimum precooler outlet
775 pressure also increased to 81 bar. This trend continued with increase in precooler outlet
776 temperature. This is due to rapid rise of the density of the CO₂ working around the pseudo-
777 critical pressures associated with the selected temperatures as shown in Figure 15. The
778 increased density results in reduced compressor work and hence increased net work output or
779 efficiency.

780



781

782 Figure 14 Effect of precooler outlet temperature on plant net efficiency of the single
783 recuperator recompression bottoming cycle configuration with no carbon capture with
784 precooler outlet pressure varying from 60 bar to 110 bar



785

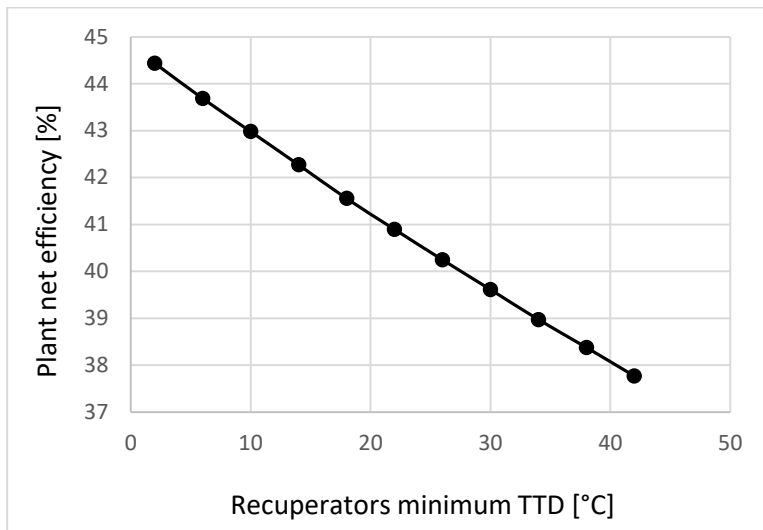
786 Figure 15 Plot of CO₂ pressure against density in the critical region showing the rapid rise in
 787 density at pseudo-critical pressures corresponding to different CO₂ temperatures

788 5.4.3 Effect of minimum terminal temperature difference of the recuperators

789 The minimum TTD of the recuperators is considered to be the smallest temperature difference
 790 between the hot and the cold stream at either hot inlet/cold outlet end or cold inlet/hot outlet
 791 end of the heat exchanger. Supercritical CO₂ recuperator is known to have pinch-point problem
 792 in which the smallest temperature difference occurs somewhere along the heat exchanger and
 793 not at the terminals [30]. The occurrence of pinch-point along the recuperator can be avoided
 794 by using recompression cycle and adjusting the flow split fractions to balance the heat
 795 capacities of the hot and cold stream. Therefore, minimum TTD will be the same as pinch-
 796 point temperature difference if the pinch-point is located at the terminal of the recuperators.

797 The selection of recuperator TTD or pinch-point temperature difference will influence the
 798 cycle efficiency and size of the recuperator [30]. Previous studies showed that the recuperator
 799 constituted the largest percentage of the size of closed Brayton cycle plant [17, 55]. For the
 800 coal-fired s-CO₂ cycle plant with single recuperator recompression bottoming cycle, the effect
 801 of the recuperators' minimum TTD on the plant net efficiency is shown in Figure 16. The plant
 802 net efficiency decreased with increasing minimum TTD of the recuperators. For every 1°C
 803 increase in minimum TTD, the net efficiency was reduced by approximately 0.17%. Hence,
 804 improved plant performance can be achieved by reducing the TTD between the hot and cold
 805 stream. This is because reducing the TTD will improve the effectiveness of the recuperator,
 806 and thus the plant performance. However, this will be at the cost of increased size of
 807 recuperator because more heat transfer area will be required.

808



809

810 Figure 16 Plant net efficiency as a function of recuperators minimum TTD for the single
811 recuperator recompression bottoming cycle s-CO₂ plant

812 6 Conclusions

813 In this paper, s-CO₂ Brayton cycle has been proposed as a potential replacement for steam
814 Rankine cycle of coal-fired power plant with solvent-based post-combustion CO₂ capture.
815 Performance evaluation shows that the s-CO₂ Brayton cycle can be adapted for efficient
816 utilisation of furnace and flue gases heat by using a topping s-CO₂ cycle and a bottoming s-
817 CO₂ cycle in addition to combustion air preheating. The coal-fired s-CO₂ cycle is able to
818 achieve furnace efficiency of about 88% in the three cases, which is comparable to the boiler
819 efficiency of the conventional supercritical steam plant. The plant net efficiency of the s-CO₂
820 Brayton cycle plant without CO₂ capture is about 3.34-3.86% more than that of the
821 supercritical steam plant. With CO₂ capture, the coal-fired s-CO₂ cycle suffers an efficiency
822 penalty of about 11.2%, which is more than the efficiency penalty of the reference supercritical
823 steam cycle plant (8.65%). Nevertheless, the plant net efficiency of the s-CO₂ cycle plant is
824 still about 0.68-1.31% more than that of the supercritical steam cycle with PCC. For the three
825 investigated cases, Case C (newly proposed bottoming cycle) is the most attractive
826 configuration as it gives the highest plant net efficiency either without or with CO₂ capture.
827 Also, comparison of the specific work outputs indicates that the size of the new concept is not
828 expected to be significantly larger than those of Case A and Case B.

829 Taken together, these findings suggest that cascaded s-CO₂ Brayton cycle is a promising power
830 conversion system for coal-fired power plant application. The current study is conceptual in
831 nature. Nevertheless, it provides considerable insight into the thermodynamic performance of
832 s-CO₂ Brayton cycle adapted for coal-fired power plant, employing a topping reheat
833 recompression s-CO₂ cycle and different options of bottoming s-CO₂ cycles. Operating
834 conditions have been chosen to be similar to conditions obtainable in the current supercritical
835 steam boiler so that the current experience with boiler material technology can be applied to
836 the s-CO₂ furnace. Therefore, future development efforts can be focused on the s-CO₂ Brayton
837 cycles.

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